Development of a 1MWe RCG-unit

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Development of a 1MWe RCG-unit

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# Contents

Summary vii  
1 Introduction 1  
1.1 Combined gas turbine systems ...................... 1  
1.2 Problem definition .................................. 5  
1.3 Outline of the thesis ............................... 6  
2 The Rankine Compression Gas turbine (RCG) 9  
2.1 Principle of the RCG ............................... 9  
2.2 Design considerations .............................. 10  
2.3 Preliminary choice of components .................. 10  
2.4 Conclusion ......................................... 13  
3 Thermodynamical analysis 15  
3.1 Thermodynamical model ............................ 15  
3.2 Efficiency comparison .............................. 17  
3.3 Off-design performance ............................ 21  
3.4 Combined heat and power performance .............. 24  
3.5 Economical payback-time .......................... 25  
4 Proof of principle 29  
4.1 Design of experimental set up ...................... 29  
4.2 Sensors ........................................... 31  
4.3 Experiments ....................................... 31  
4.4 Discussion: technical feasibility of the RCG ....... 33  
5 Transient analysis 35  
5.1 Transient behavior of gas turbine systems ....... 35  
5.2 Transient equations ............................... 36  
5.2.1 Flow elements ................................ 36  
5.2.2 Volumes ..................................... 38  
5.2.3 Rotating equipment ............................. 41  
5.3 Model ............................................ 41  
5.3.1 Steam generator ................................ 41
CONTENTS

5.3.2 Compressor and restriction valve . . . . . . . . . . . . . . 43
5.3.3 Steam turbine . . . . . . . . . . . . . . . . . . . . . . . . 43
5.3.4 Combustion chamber . . . . . . . . . . . . . . . . . . . . . 45
5.3.5 Auxiliary burner and mixing point . . . . . . . . . . . . . 45
5.4 Basic model layout . . . . . . . . . . . . . . . . . . . . . . 45
5.5 Basic system transient response . . . . . . . . . . . . . . . . 46
5.6 Simulation results versus experiments with the set-up . . . . . 50
5.6.1 Advanced experimental set-up . . . . . . . . . . . . . . . . 50
5.6.2 Feedwater flow transient response . . . . . . . . . . . . . 54
5.6.3 Auxiliary burner transient response . . . . . . . . . . . . 57
5.7 Strategy for transients of the RCG . . . . . . . . . . . . . . . 58
5.8 Adding auxiliary burner overdrive . . . . . . . . . . . . . . . 59
5.9 Adding steam temperature limiting . . . . . . . . . . . . . . 61
5.9.1 Afterspray . . . . . . . . . . . . . . . . . . . . . . . . . . 61
5.9.2 Aircooling . . . . . . . . . . . . . . . . . . . . . . . . . . 63
5.10 Conclusions . . . . . . . . . . . . . . . . . . . . . . . . . . . . 67

6 Start-up experiments . . . . . . . . . . . . . . . . . . . . . . 69
6.1 Steam generator start-up experiments . . . . . . . . . . . . . . 69
6.2 Steam temperature limiting during start-up . . . . . . . . . . . 72
6.2.1 Burner interval . . . . . . . . . . . . . . . . . . . . . . . . 72
6.2.2 Forced cooling . . . . . . . . . . . . . . . . . . . . . . . . 73
6.3 Conclusions . . . . . . . . . . . . . . . . . . . . . . . . . . . . 74

7 Design of a 1MW e pilot installation . . . . . . . . . . . . . . . 75
7.1 Field of application and demands . . . . . . . . . . . . . . . . 75
7.2 Preliminary skid design . . . . . . . . . . . . . . . . . . . . . 76
7.3 Skid components . . . . . . . . . . . . . . . . . . . . . . . . . 77
7.4 Design and off-design point performance of the pilot-RCG . . 86
7.5 Final skid design . . . . . . . . . . . . . . . . . . . . . . . . . 90
7.6 Economical discussion . . . . . . . . . . . . . . . . . . . . . . 92
7.6.1 Market . . . . . . . . . . . . . . . . . . . . . . . . . . . . 92
7.6.2 Payback time estimation . . . . . . . . . . . . . . . . . . . 92
7.6.3 Conclusions . . . . . . . . . . . . . . . . . . . . . . . . . . 93

8 Concluding discussion . . . . . . . . . . . . . . . . . . . . . . 95
8.1 Conclusions . . . . . . . . . . . . . . . . . . . . . . . . . . . . 95
8.2 Recommendations . . . . . . . . . . . . . . . . . . . . . . . . 97

A Design of a 500kW subcritical once-through steam generator . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . 101
A.1 Thermodynamics . . . . . . . . . . . . . . . . . . . . . . . . . . 101
A.2 Construction . . . . . . . . . . . . . . . . . . . . . . . . . . . . 103
A.3 Heat transfer . . . . . . . . . . . . . . . . . . . . . . . . . . . . 104
A.4 Final design . . . . . . . . . . . . . . . . . . . . . . . . . . . . 106

B Steam generator states . . . . . . . . . . . . . . . . . . . . . . 107
CONTENTS

C  Steam temperature limiting ........................................ 111
  C.1  After spray ..................................................... 111
       C.1.1  Demands ................................................. 111
       C.1.2  Analysis ............................................... 111
       C.1.3  Design ............................................... 115
  C.2  Aircooler ...................................................... 118
       C.2.1  Demands ................................................. 118
       C.2.2  Design ............................................... 119

D  Development of a multi-fuel RCG-combustor ....................... 121
  D.1  Introduction .................................................. 121
  D.2  Fuel .......................................................... 121
  D.3  Fuel line ...................................................... 122
  D.4  Combustion chamber design .................................... 124
  D.5  Experiments .................................................. 124

Curriculum Vitae ...................................................... 127
Acknowledgements ...................................................... 129
Summary

Development of a 1MWe RCG-unit

The Rankine Compression Gas turbine (RCG) is a new type of combined steam and gas turbine (STaG). The novelty of the RCG is that all shaft power is delivered by a free power turbine that is driven by the expansion of the combustion gases. The compressed air used for combustion is fully produced by the steam turbine driven compressor. Therefore, the RCG offers flexible load characteristics that make it applicable in decentralized power generation, mechanical drives and ship propulsion which are typically in the 1-10MW range. This thesis presents the three step plan that is executed to develop the RCG. First, a feasibility study is done, then, an experimental set-up is tested and, finally, a real-scale pilot installation of 1MWe is designed for a combined heat and power application.

The feasibility is studied with a thermodynamical model. The results show that the RCG is technologically and economically feasible with robust radial turbo machinery components, such as a centrifugal turbo compressor, radial expansion turbine and an impulse steam turbine (zero-reaction turbine). To ensure a short start-up time, the RCG is designed with a subcritical once-through steam generator, which is not commercially available in the MW-range. Therefore, a subcritical once-through steam generator was especially designed. An experimental set-up at the Technische Universiteit Eindhoven(TU/e) shows that the chosen key components of the RCG-layout result in stable operation and a start-up time of about 15 minutes.

In its intended fields of application, the RCG must have the ability to go from part-load to full-load within minutes. With a one-dimensional transient model, the transient behaviour of the system and the effect of possible actuators is simulated. The transient model may be used to develop an operating strategy of a real-scale RCG, since the behavior of the set-up and the model proved to be similar. An improved operating strategy is developed where the auxiliary burner is fired during transients (auxiliary burner overdrive) and an aircooler controls the steam temperature. This operating strategy reduces the response time to approximately 250 seconds.
Next, a more advanced set-up was realized. The first aim was to gain experience with operating the RCG with industrial soft- and hardware. The second aim was to develop a start-up procedure, since the transient model can not simulate a start-up. The experiments show that the start-up procedure of the once-through steam generator needs special attention; without precautions, the steam temperature rises to a dangerous peak level before settling at a safe working point. This is solved with the aircooler, and experiments show that this ensures safe steam temperatures during a cold start.

Finally, a 1MWe pilot installation was designed for a pilot project, a combined heat and power (CHP) application. The pilot was designed with the ability to run on crude (bio) fuels, more specifically glycerol, the waste product of biodiesel production. The pilot installation has an electrical efficiency of $\eta_e = 0.20$ and will be available for investment costs of about 1 mln euro. This combination of performance and investment costs results in an estimated payback time of 2.6 years. It is therefore expected that it has the potential to become commercially successful.
Chapter 1

Introduction

1.1 Combined gas turbine systems

Stationary gas turbine systems are available in the shaft power range of 25kW up to order of magnitude 1000MW. Due to ever increasing costs of fossil fuels and the awareness of the impact on the environment of burning fossil fuels, it is desired to decrease fossil fuel consumption.

The most basic gas turbine system is the simple cycle gas turbine (see figure 1.1). Simple cycle gas turbines cover a shaft power range of typically 1MW-1000MW.

![Figure 1.1: Schematic of the simple cycle gas turbine (l) and schematic of the simple cycle gas turbine with free power turbine (r)](image)

In the lower region of 1-10MW an alternative layout with a free power turbine is sometimes employed, which is shown in figure 1.1(r). The free power turbine offers more flexible load characteristics which are often required in situations where variable speed, high torque or black-start capability is required (e.g. mechanical drives or decentralized power generation). Examples of this type of turbine are the Centaur, Taurus and Mars turbine series of industrial gasturbine manufacturer Solar (US). In the >50MWe range for power generation, aero-derivatives are integrated into the principle of figure 1.1(r); the aero gas
turbine (e.g. GE, Rolls Royce) is then placed in front of a large free power turbine that drives an electrical generator.

Various technological developments are employed to lower the fuel consumption and emissions of simple cycle gas turbines, such as high temperature materials and advanced combustion technologies. Figure 1.2 shows the typical cycle efficiency ($\eta$) and specific power of the simple cycle for various compressor pressure ratios ($r$) and turbine inlet temperatures (TIT). The cycle efficiency and specific power are defined as:

$$\eta_{cycle} = \frac{\text{Work (kW) delivered by turbine (T)}}{\text{Fuel input (kW) in combustor}}$$

$$\text{specific power} = \frac{\text{Work (kW) delivered by turbine (T)}}{\text{massflow through turbine (kg/s)}}$$

From figure 1.2 it can be seen that with rising pressure ratio the cycle efficiency increases up to a maximum, and then decreases again. This is because, as

the compressor delivery temperature rises for increasing pressure ratio, less fuel needs to be injected to obtain the fixed TIT. At a certain pressure ratio, this effect is outweighed by the increasing power necessary to drive the compressor. With rising pressure ratio the specific work also rises to a maximum and then decreases again. At pressure ratio $r = 1$ the specific power is obviously zero, with increasing pressure ratio the work will increase. However, again because

![Figure 1.2: Cycle efficiency and specific power of the simple cycle, where it is assumed that the isentropic compressor efficiency $\eta_c = 0.87$ and isentropic turbine efficiency $\eta_t = 0.85$, adapted from [1]](image-url)
the compressor delivery temperature rises, the injected fuel that needs to be injected to obtain the fixed TIT reduces. At the pressure ratio where the compressor delivery temperature equals the TIT, no fuel can be injected anymore, and the work will be even less than zero due to component losses.

From the previous it can be seen that both the maximum obtainable efficiency and the specific power of a simple cycle gas turbine can be elevated by raising the maximum allowable TIT and choosing a pressure ratio that results in an acceptable efficiency and specific power. For gas turbines with a shaft power of order of magnitude 10MW and higher, the costs of high-end materials and components are economically feasible; for these gas turbines a TIT of 1500K and efficiencies in the range of 30-40% are quite common. For gas turbines with shaft powers of order of magnitude 1MW it is not economically feasible to implement technology that results in a high TIT; their typical cycle efficiency is around 20%.

Besides raising the TIT, the efficiency of a gas turbine can also be elevated by implementing a heat exchanger to recuperate the energy of the exhaust gases [1], this is usually referred to as the recuperative cycle (see figure 1.3). Figure 1.4 shows the typical cycle efficiency of the recuperative cycle for various compressor pressure ratios and turbine inlet temperatures (TIT). The specific power of gas turbine with a heat exchanger is the same as that of a simple cycle gas turbine, but only if friction losses in the heat exchanger are assumed to be zero. More importantly, figure 1.4 shows that the efficiency at equal TIT is considerably elevated by implementing a heat exchanger. For micro gas turbines (shaft power up to 100kW) the recuperative cycle is already successfully employed [12],[13], however for larger shaft powers constructional difficulties of the gas-to-gas heat exchanger arise. Also, to prevent fouling of the heat exchanger, the recuperative cycle can only be fired with very clean fuels such as natural gas.

Another way of raising the efficiency is adding a Rankine cycle (steam turbine
CHAPTER 1. INTRODUCTION

Figure 1.4: Cycle efficiency and specific power of the recuperative cycle

cycle) behind the gas turbine. The combined gas turbine and Rankine cycle is most often referred to as a combined cycle. Many analyses have been made of various combined cycles such as [9],[10],[11]. In power stations, the combined cycle is successfully employed to generate electricity at high efficiencies. The typical $\eta_{\text{cycle}}$ of a combined cycle is 50-55%.

The two main combined cycle layouts are shown in figure 1.5, the multi-shaft combined cycle (left) and the single-shaft combined cycle (right). Both make use of a compressor (C), combustion chamber, turbine (T), steam generator (waste-heat boiler), steam turbine (ST), condenser, water pump and generators.

Figure 1.5: The multi-shaft combined cycle (l) and the single-shaft combined cycle (r)

efficiency of the combined cycle can be elevated both by increasing the efficiency
of the gas turbine as well as the Rankine cycle. As was mentioned in the above, the efficiency of the gas turbine cycle can be improved by raising the pressure ratio and TIT. The efficiency of the Rankine cycle can be improved by raising the expansion ratio over the steam turbine; higher steam generator pressure and lower condensor pressure. The latter is constrained by the temperature of the available medium for cooling the condensor (e.g. river water or ambient air).

In modern combined cycle power plants supercritical steam generation is employed, with once-through steam tubes. The steam generator pressure and temperature is constrained by the steam tube strength and steam turbine blade strength. Also, the higher the expansion ratio from steam generator to condensor, the more stages are necessary in the steam turbine to expand the steam.

Typically combined cycles take several hours to start and their ability to be operated at part-load is limited. Because of their load characteristics and (expensive) high-end design combined cycles (STaG) can only be employed in base-load power generation with a typical shaft power of 1000MW.

### 1.2 Problem definition

The alternative gas turbine systems that raise the efficiency of the gas turbine cycle are at both ends of the shaft power range; the recuperative cycle is available for shaft powers up to order of magnitude 100kW, the combined cycle is available for the typical shaft power of 1000MW. In the 1-1000MW range the efficiency is mostly elevated by implementing components with high pressure ratio and high temperature capability. However, for economic reasons this is difficult for the range 1-10MW. Moreover, gas turbine systems in this shaft power range are often required to have the flexibility that is offered by a free power turbine (see figure 1.1(r)). Theoretically, the cycle efficiency of the gas turbine with free power turbine could be elevated by making it a recuperative cycle or by implementing high-end turbo machinery components. However this is not feasible for the range 1-10MW, for the same reasons as for the ordinary simple cycle.

A possible solution could be to create a 1-10MW combined cycle (figure 1.5). The demands to such a combined cycle are:

- the turbo machinery should be low-end to ensure economic feasibility
- the combined cycle efficiencies higher then a typical simple cycle in the same shaft power range
- all the work should be delivered on a free power turbine
1.3 Outline of the thesis

In this thesis a new combined cycle is proposed and developed: the Rankine Compression Gas turbine (RCG), European patent granted [15]. The RCG (see figure 1.6) is a combined cycle where the steam turbine is not coupled to a load, but to the compressor of the gas turbine cycle. Consequently, the expansion turbine of the gas turbine cycle can deliver all its power to the load and functions as a free power turbine.

![Figure 1.6: Schematic of the Rankine Compression Gas turbine Cycle (RCG)](image_url)

With its free power turbine the RCG is fast responsive and offers very flexible load characteristics with only one output shaft. The RCG, is therefore, suitable for applications such as decentralized power generation, and mechanical drives. However, a system like the RCG has never been built before. The technical and economical feasibility should be assessed with theoretical studies and experiments. Furthermore, because the steam turbine cycle drives the compressor of the gas turbine cycle, a kind of feedback-loop occurs that does not occur in conventional combined cycles. Both start-up and transient behaviour of an RCG system should be studied and improved with simulations and experiments. When feasibility and dynamical behavior are satisfactory, a real-scale pilot system can be designed. This thesis describes the three-step plan, executed to bring the RCG to full realization:

- Proof of principle [16],[17],[18]
- Optimisation of transient behaviour [19]
- Design of a real-scale pilot-system
1.3. OUTLINE OF THE THESIS

The first step, proof of principle, is treated in chapters 2, 3 and 4. The choice of components for the RCG that are suitable for the 1-10MW range, is discussed in chapter 2. In chapter 3, a thermodynamical model is used to investigate what performance can be expected following from the choice of components of chapter 2. Chapter 4 discusses the results of the experimental set-up that was realised to prove that the RCG can be started quickly and can be operated stable.

Chapters 5 and 6 treat the second step; optimisation of the transient behaviour of the RCG. In chapter 5, the transient behaviour is analysed with a model and an operating strategy is developed. In chapter 6, the experimental set-up is enhanced to gain experience with starting the RCG with industrial soft- and hardware.

Finally, in chapter 7, step three is initiated; a pilot installation was designed for a pilot customer.
CHAPTER 1. INTRODUCTION
Chapter 2

The Rankine Compression Gas turbine (RCG)

2.1 Principle of the RCG

The novelty of the RCG compared to existing combined cycles is, that the steam turbine (ST) drives the compressor (C) of the Gas turbine cycle (Brayton cycle). This explains the name of the Rankine Compression Gas turbine: the compres-
The Rankine Compression Gas Turbine (RCG) is intended for mechanical drives and decentralized CHP applications. It will be applied in a much smaller shaft power range than existing combined cycles, and will consist of a different type of turbo machinery. Furthermore, the start-up and transient behavior is a very important feature of industrial installations. The demands to the design of the RCG are:

- the general design of the RCG should be able to cover the shaft power range 1 MW up to 10 MW
- the components of the RCG should make it a robust, compact, and economic installation
- the RCG should have a short start-up and transient time

In the next section, the choice of components is briefly discussed.

### 2.3 Preliminary choice of components

#### Compressor

As will be shown in chapter 3, the gas turbine cycle of the RCG will typically have a pressure ratio of about 4. Because of this low pressure ratio, a single stage centrifugal compressor can be employed. This is advantageous because a
centrifugal compressor is more robust and economic than an axial compressor. For shaft powers above 3MW an axial compressor may be employed to benefit from their higher isentropic efficiency, although a centrifugal compressor will still be the most robust and economic solution.

Steam turbine

In factory plants with central boilers, radial steam turbines are used to drive a variety of equipment in the shaft power range up to 6MW (e.g. Siemens KK&K). These radial steam turbines are in fact impulse steam turbines (zero-reaction). The degree of reaction expresses to which extend the medium is expanded in the rotor blades as opposed to expansion in the stator blades [1]. Impulse steam turbines expand the steam completely before it hits the turbine rotor blades. Therefore, impulse steam turbines are able to handle expansion ratios of up to 70 with just one turbine stage, while axial steam turbines (reaction \( \neq 0 \)) need several stages. Axial steam turbines have a higher isentropic efficiency, however, radial steam turbines are much more compact, economic and robust than axial steam turbines. Recently, a new generation of radial impulse steam turbines has become available [7] with a turbine efficiency of up to 80\%. These impulse steam turbines are very suitable for employment in an RCG-installation and can be considered as "proven technology".

Steam generator (boiler)

An RCG will need to have a steam generator that is compact and economic. So, even though it will be at the cost of thermal efficiency, it is favorable to operate the boiler at relatively low pressure. This results in high temperature differences between exhaust gas and steam, so that a compact boiler can be employed. Also, the steam generator of an RCG will need to be able to respond very fast to load changes. Typical industrial size boilers use an abundance of water that is heated to boiling temperature, after which steam bubbles are separated. This principle ensures stability, but it results in a long start-up time and transient time (typically hour(s)). A once-through boiler basically consists of a tube in which all water is converted into superheated steam by bringing it in counter-flow heat-exchange contact with hot (exhaust) gasses. Combined with the earlier demands of compactness and economic feasibility, a once-through boiler is very appealing. Because the boiler pressure should be relatively low, this implies a sub-critical once-through boiler.

Super-critical once-through boilers are quite common in large-scale power generation, but sub-critical once-through boilers are definitely not widespread. Furthermore, special attention should be paid to the control of a sub-critical once-through boiler. Nevertheless, a sub-critical once-through boiler is selected for the RCG, because it meets all the demands to the RCG design: compact, economic and fast responding.
CHAPTER 2. THE RANKINE COMPRESSION GAS TURBINE (RCG)

Condenser

Robust industrial steam condensers are widely available in all sizes. Depending on the location and purpose of the RCG installation, one will have to decide whether an air-cooled or water-cooled condenser is favorable. If enough cooling water is available, water-cooled condensers have the advantage of being more compact and economic.

Feed water pump

The water from the condenser is pumped (and pressurized) into the steam generator by a feed water pump. Like the condenser, industrial feed water pumps are widely available. However, because it is chosen to implement a sub-critical once-through boiler, it is expected that the control of the feed water pump will need specific attention.

Auxiliary burner

To start an RCG, first the steam cycle has to be started. This way, the steam turbine will be powered up and will start to drive the compressor of the gas turbine cycle. Then the combustion chamber will be supplied with air and can be fired up. The steam cycle can be started with an auxiliary burner. The auxiliary burner can be of the type that is used in industrial small-scale boilers (natural gas or oil fired).

The auxiliary burner might also be used for supplementary firing. With supplementary firing it will be possible to generate extra power. Also, it will be possible to shorten the response-time from part-load to full-load, because the steam generator can be fired up quickly.

Combustion chamber

Gas turbine manufacturers have developed a large range of combustion chambers. For an RCG it would of course be favorable to employ an existing combustion chamber of a gas turbine manufacturer, but since the RCG will typically have a pressure ratio of about 4, this has proven to be difficult; the average industrial gas turbine has a typical pressure ratio of around 10. The combustion chamber of the RCG has to be a kind of a cross-over between an industrial duct burner (atmospheric pressure) and an industrial gas turbine combustion chamber. This type of combustion chamber will need to be developed especially for the RCG.

Power turbine

For the power turbine, considerations similar to that of the compressor apply. Because of the relatively low pressure ratio of about 4, a single stage radial turbine can be employed. Like centrifugal compressors, also radial expansion
2.4. CONCLUSION

Turbines are more robust and economic than axial turbines. Again for shaft powers above 3MW, axial equipment may be employed to benefit from its higher isentropic efficiency, although a radial turbine will still be the most robust and economic solution. Note that the modular build-up of the RCG also allows multiple parallel radial turbines to be implemented.

2.4 Conclusion

With the preliminary choice of components as described above, the RCG will be able to meet the preset requirements of being robust, compact and economic. All the required turbo machinery components can be considered proven technology. Two non-rotating components will need to be developed especially for the RCG; the steam generator and the combustion chamber.

It is hoped that the RCG can obtain fuel efficiencies that are competing with internal combustion engines, but this still has to be determined. This will be done with a thermodynamical model in chapter 3.
3.1 Thermodynamical model

The major thermodynamical difference between a conventional combined cycle (Figure 1.5) and the RCG cycle (Figure 2.1) is that, for the RCG cycle, in steady-state the power of the steam turbine has to be equal to the power consumed by the compressor of the Brayton cycle. This is because the essence of the RCG cycle is that the compressor (in Figure 2.1: C) is driven by the steam turbine (in Figure 2.1: ST), and that the steam turbine drives nothing else but the compressor [16]. Therefore, in steady-state, the power of the compressor and the steam turbine are equal:

\[ P_c[kW] = P_{st}[kW] \]  

(3.1)

Where \( P_c \) is the compressor power and \( P_{st} \) is the steam turbine power. The power consumed by the compressor is determined [8] by:

\[ P_c = \dot{m}_a c_{p,a} \frac{T_a}{\eta_c} (r^{\frac{\gamma-1}{\gamma}} - 1) \]  

(3.2)

In equation (3.2), \( \dot{m}_a \) is the air mass flow, \( c_{p,a} \) is the specific heat at constant pressure of the ambient air, \( T_a \) is the ambient air temperature, \( \eta_c \) is the isentropic compressor efficiency, \( r \) is the pressure ratio realized by the compressor and \( \gamma \) is the specific heat ratio (the specific heat constant pressure divided by the specific heat at constant volume).

The compressor delivers compressed air to the combustion chamber where it is heated up to a certain temperature by burning fuel. The combustion chamber is assumed to be ideal, which means that it obtains complete combustion and has zero pressure drop. For industrial gas turbines this assumption is nearly
true. Also, the gas turbine systems that are compared in this chapter are all being modeled with an ideal combustion chamber, therefore it will not affect the comparison of various system as such.

The heated compressed mixture that comes from the combustion chamber expands in the power turbine (in Figure 2.1: PT). The amount of shaft power generated by the power turbine is calculated [8] with:

\[ P_{pt} = \dot{m}_g c_{p,g} \eta_t T_{IT} \left( 1 - \left( \frac{1}{r} \right)^\frac{\gamma-1}{\gamma} \right) \] (3.3)

In which \( \dot{m}_g \) is the mass flow of the compressed hot gas mixture entering the turbine, \( c_{p,g} \) is the specific heat at constant pressure of the hot gas mixture, \( \eta_t \) is the isentropic turbine efficiency, \( T_{IT} \) is the temperature of the compressed hot gas mixture entering the turbine (turbine inlet temperature), \( r \) is the expansion ratio of the turbine and \( \gamma \) is the specific heat ratio. Since the pressure loss of the combustion chamber and the back-pressure due to the steam generator are neglected, the compression ratio and expansion ratio are considered equal.

In the steam generator, steam is produced using the waste-heat of the hot exhaust gasses that leave the turbine. The amount of steam that can be generated by the steam generator follows from the energy balance between the temperature drop of the turbine exhaust gas and the enthalpy rise of the water to steam in the steam generator [2]:

\[ \dot{m}_g c_{p,ex} (T_{ex} - T_{stack}) = \dot{m}_{st} (h_{st,in} - h_l) \] (3.4)

The left hand side of equation (3.4) describes the exhaust-gas energy transfer, where \( c_{p,ex} \) is the specific heat of the exhaust gas, \( T_{ex} \) is the exhaust gas temperature between the turbine and the steam generator, \( T_{stack} \) is the exhaust gas temperature leaving the steam generator. The right hand side of equation (3.4) describes the energy that is absorbed for steam production, in which \( \dot{m}_{st} \) is the steam flow, \( h_l \) is the enthalpy of the feedwater entering the steam generator and \( h_{st,in} \) is the enthalpy of the steam exiting the steam generator and entering the steam turbine. The power delivered by the steam turbine can be calculated with [5]:

\[ P_{st} = \eta_{st} \dot{m}_{st} (h_{st,in} - h_{st,out}) \] (3.5)

Where \( \eta_{st} \) is the steam turbine isentropic efficiency, \( \dot{m}_{st} \) is the steam mass flow, \( h_{st,in} \) is the enthalpy of the steam entering the steam turbine, \( h_{st,out} \) is the enthalpy of the steam leaving the steam turbine.

Note that, for a given turbine inlet temperature \( T_{IT} \), the exhaust gas temperature after the power turbine \( T_{ex} \) depends on the isentropic efficiency of the power turbine and on the pressure ratio \( r \): a higher pressure ratio gives a lower exhaust gas temperature. Therefore, at steady-state, an RCG installation will operate at a pressure ratio \( r \) for which both equations (3.1) and (3.4) are valid.
Finally, the thermal efficiency of an RCG installation is determined by the ratio of the amount of fuel that is injected into the combustion chamber, and the power that is delivered to the load by the power turbine [16],

$$\eta_{th} = \frac{P_{\text{powerturbine}}}{Q_{\text{fuel}}} \quad (3.6)$$

### 3.2 Efficiency comparison

With the thermodynamical model, comparative calculations [17] were made of the simple cycle (figure 1.1), the recuperative cycle (figure 1.3) and the RCG. For the recuperative cycle the pressure ratio ($r$) was optimized for maximum thermal efficiency. This is not possible for the simple cycle at turbine inlet temperatures higher than 1200[K], because the efficiency is ever increasing with increasing pressure ratio (see figure 1.2). Therefore, it was chosen to compare the simple cycle at pressure ratios with a good balance between thermal efficiency and specific power. For the RCG, the choice of components as discussed in Chapter 2 were assumed. Table 3.1 shows the properties that were assumed for the simple cycle, the recuperative cycle and the RCG.

<table>
<thead>
<tr>
<th>Simple cycle</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>288[K]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{compressor}}$</td>
<td>0.87</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{turbine}}$</td>
<td>0.85</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Recuperative cycle</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>288[K]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{compressor}}$</td>
<td>0.80</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{turbine}}$</td>
<td>0.85</td>
<td></td>
<td></td>
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<tr>
<td>Recuperator efficiency</td>
<td>80%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Rankine Compression Gasturbine (RCG)</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>288[K]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{compressor}}$</td>
<td>0.80</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{turbine}}$</td>
<td>0.85</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\eta_{\text{steamturbine}}$</td>
<td>0.80</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Boiler pressure</td>
<td>30[bar]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steam temperature</td>
<td>773[K]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Condensor pressure/temperature</td>
<td>10[kPa]/318[K]</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: Properties assumed for the thermodynamical model

Note that, for the simple, cycle higher isentropic compressor efficiency is assumed. This is an advantage for the simple cycle, but as the comparison will
show, is also realistic: because the simple cycle has higher compression ratios an axial compressor is likely to be employed, while the RCG and recuperative cycle are likely to have a centrifugal compressor with their lower compression ratio.

For both the steam generator and the recuperator, it is assumed that they do not result in back pressure for the power turbine. This gives a small advantage for the recuperative cycle, since a recuperator normally results in more back pressure than a steam generator.

Figure 3.1 shows the results regarding the obtained thermal efficiency of the simple cycle, recuperative cycle and the RCG at varying Turbine Inlet Temperature (TIT). The efficiency shown at a certain TIT is the efficiency of a certain installation with a turbine with that TIT as its maximum. As expected, the efficiencies of the simple cycle are much lower than those of the recuperative cycle and the RCG.

The results show that both the RCG and the recuperative cycle can obtain efficiencies of about 30% up to about 45% in a range of realistic TIT’s. For industrial gas turbines, the maximum TIT for an uncooled turbine with current state-of-the-art materials and manufacturing technology is around 1300[K]. At this TIT both the RCG and the recuperative cycle have a thermal efficiency of
about 40%. At TIT’s higher than 1300[K] the recuperative cycle has somewhat higher efficiencies than the RCG, but at TIT’s below 1300[K] it is the RCG that has the highest efficiencies. It must be noted that the differences between the RCG and the recuperative cycle are small, and that the assumptions that were made are a little bit in favour of the recuperative cycle. These efficiencies were calculated, assuming modest component efficiencies, and without intercooling. So it can be concluded that the RCG will be an appealing alternative next to the recuperative cycle, when a higher efficiency than that of the simple cycle is preferred.

Figure 3.2 shows the pressure ratios for the RCG, recuperative cycle and simple cycle. It can be seen that the pressure ratios of the simple cycle are much higher than those of the RCG and the recuperative cycle. This follows from the earlier discussed assumption to compare the simple cycle at pressure ratios with a good balance between thermal efficiency and specific power. Since it is not possible to optimize for maximum thermal efficiency, as for the simple cycle at maximum thermal efficiency, the specific power is equal to zero. The pressure ratio of the recuperative cycle is optimized for maximum thermal efficiency, and the pressure ratio of the RCG follows from the balance between the power production of the steam turbine and the power consumption of the compressor.
Most striking is, that the equilibrium pressure ratios of the RCG and the pressure ratios at optimum efficiency for the recuperative cycle are of the same magnitude. Furthermore, the pressure ratio of the gas turbine cycle of the RCG can be realized with a centrifugal compressor. So, the efficiencies of the RCG shown in figure 3.1, are those of a very robust RCG-installation: centrifugal compressor in the gas turbine cycle, impulse steam turbine and low-pressure boiler in the steam cycle.

Figure 3.3 shows that the recuperative cycle and simple cycle have almost the same specific power, while the RCG has up to twice as much specific power at the same turbine inlet temperature. The reason for the large increase in specific power of the RCG, compared to the recuperative and simple cycle, is that the waste heat in the exhaust gasses is converted into extra shaft power. In the recuperative cycle, the waste heat is also put to use, but it is employed to reduce the amount of fuel that is burned to reach the same turbine inlet temperature. Therefore the rise in specific power of the RCG, compared to the recuperative and simple cycle is typical for all combined cycles, not just the RCG.

It is well known that existing combined cycles can achieve efficiencies of up to 54%. One could therefore conclude that it is no use to introduce the RCG.

Figure 3.3: Specific power of the RCG, recuperative cycle and simple cycle at variable inlet temperature
would be a false conclusion: the RCG is not meant to be a competitor of the existing combined cycles. The purpose of the RCG is to make it possible to employ a combined cycle installation, where until now this was not possible: mechanical drive and decentralized combined heat and power (CHP) applications.

3.3 Off-design performance

The flexibility of the RCG lies not only in the free power turbine characteristics itself, but also with the fact that the compressor is independently driven from the gas turbine cycle. This makes it possible to maintain the compressor power, and thus keep the airflow stable, in a part-load situation. To study this, the part-load characteristics of the turbo machinery components, such as the compressor map, turbine map, and steam turbine curve, are added to the model of section 3.1.

The characteristics of the turbo machinery components of the RCG are derived from the characteristics of the turbo machinery of the experimental set-up (see chapter 4). The compressor map, turbine map, and steam turbine curve are scaled to the appropriate airflow and steam flow of a 2MW e RCG. Furthermore, it is assumed that the turbo compressor, power turbine and steam turbine have maximum isentropic efficiencies of $\eta_{\text{compressor}} = 0.80$, $\eta_{\text{turbine}} = 0.78$ and $\eta_{\text{steam turbine}} = 0.75$, which is considered realistic. From this it follows that the modeled RCG reaches its full power of 2MW when the combustion chamber is fired at 5500kW thermal power and the auxiliary burner is fired at 1500kW thermal power, both at the same time. It can be seen that part-load can be obtained with a range of possible combinations of combustion chamber power and auxiliary burner power. With these characteristics incorporated in the model, the range of possible combinations of thermal power and shaft power is investigated.

Figure 3.4 shows the range of possible combinations of combustion chamber power, auxiliary burner power and shaft power (electrical power) of a typical 2MW e RCG. It has to be noted that all combinations of figure 3.4 are valid working points, only if for all working points the maximum turbine inlet temperature ($T_{\text{IT}}$) of the turbine is not exceeded. A typical uncooled radial turbine has a maximum $T_{\text{IT}}$ of around 1300K. The $T_{\text{IT}}$s that correspond to the working points of figure 3.4 are shown in figure 3.5.

It can be seen that the maximum $T_{\text{IT}}$ is not exceeded and is only reached in the working point where the combustion chamber power is at 5500kW and where the auxiliary burner is at 0kW; this latter working point corresponds to a shaft power of 1750kW (87.5% part-load). It can also be seen that firing the auxiliary burner towards 1500kW (moving towards 2MW shaft power in figure 3.4), while maintaining the combustion chamber at 5500kW, lowers the $T_{\text{IT}}$. This can be explained from the fact that the auxiliary burner is behind the power turbine (see figure 2.1). When the auxiliary burner is fired, more energy will be supplied to the steam generator, resulting in a higher steam flow and
CHAPTER 3. THERMODYNAMICAL ANALYSIS

Figure 3.4: Obtainable shaft power range of a 2MW e RCG in off-design operation

steam turbine power. Consequently more power is supplied to the compressor, the pressure ratio is increased and more air will be supplied to the combustion chamber. If the combustion chamber power is kept constant, the $TIT$ will drop. Figure 3.6 shows the thermal efficiency $\eta_{th}$ corresponding to the various combinations of combustion chamber and auxiliary burner power. The maximum $\eta_{th}$ is reached at a combustion chamber power of 5500kW and an auxiliary burner power 0kW. This situation coincides with the $TIT$ at the maximum allowable value of 1300K. From figure 3.4 it was already seen that firing the auxiliary burner elevates the shaft power, however, figure 3.6 shows that it also lowers the $\eta_{th}$. This is mainly caused by the fact that the thermal power of the auxiliary burner only participates in the steam turbine cycle and does not (or at least only partly because the steam turbine drives the turbo compressor) participate in the gas turbine cycle. The maximum $\eta_{th}$ of any STaG is reached when all thermal power is added in the gas turbine cycle only, this is also true for the RCG.

However, the drop in thermal efficiency that is seen for rising thermal power of the auxiliary burner from 0 to 1500kW, is not only caused by this effect; the drop in thermal efficiency is also caused by the fact that the $TIT$ decreases for rising auxiliary burner power. The relation between $TIT$ and $\eta_{th}$ was already described in section 3.2 and can also be seen from figure 3.6 and figure 3.5 by
3.3. OFF-DESIGN PERFORMANCE

Figure 3.5: TIT range of the RCG

Figure 3.6: Obtainable $\eta_{th}$ range of the RCG
CHAPTER 3. THERMODYNAMICAL ANALYSIS

following a constant auxiliary burner power curve (e.g. 1500kW) from 5000kW to 5500kW combustion chamber power. This means, as was mentioned earlier, that in a real RCG installation the combustion chamber power will probably be controlled in such a way that the TIT will be kept constant just under the maximum TIT. This will result in a $\eta_{th}$-curve that will drop less steep for increasing auxiliary burner power than in figure 3.6.

From the preceding, it follows that the RCG will in general have a load characteristic with a $\eta_{th}$-curve that shows a maximum at part-load and drops slightly towards full-load. This characteristic is similar to that of a diesel engine and is new for a gas turbine based system. This load characteristic is an asset that will make the RCG appealing for its intended fields of application; mechanical drive, ship propulsion and decentralized CHP.

3.4 Combined heat and power performance

For CHP-applications not only the shaft power and thermal efficiency is of interest, but also the rejected heat of the system.

![Figure 3.7: Obtainable $P_{condensor}$ range of the RCG in CHP operation](image)

Figure 3.7 shows the rejected heat by the condensor of the 2MW e RCG, corresponding to the various combinations of combustion chamber and auxiliary burner power. By looking at figure 3.7 and figure 3.4 together, it can be seen that one and the same RCG installation can give numerous combinations of
3.5 Economical payback-time

To assess whether the RCG is economically appealing, a comparison is made with simple cycle gas turbines. Of course, a lot of issues play a role in the economical feasibility, and they differ per application, user and country. In this thesis, it is not possible to take all these matters into account and, at this stage, it is impossible to make exact calculations. The goal is to roughly assess whether it is economically attractive to further develop the RCG. Because the assumptions are not very exact, the outcome of the calculations will also not be exact. However, the systems are compared with similar assumptions. So, all together, the calculations will give a good idea whether the RCG is economically attrac-

This can be illustrated with the following example; if the 2MW RCG is operated at 5500kW combustion chamber power and 0kW auxiliary burner power, it will run at a shaft power of 1750kW and will supply a heat flow of 3000kW. Suppose the process or plant which this RCG were dedicated to, would temporarily demand more heat and equal shaft (electrical) power, say 3400kW thermal power (13.3% extra) and 1750kW shaft power. The demanded combination of shaft power and thermal power can be realized if the auxiliary burner is fired up to 1100kW, and the combustion chamber is powered down to 5000kW.

Figure 3.8: Obtainable $\eta_{total}$ range of the RCG in CHP operation

shaft power and thermal power (heat rejected by the condensor).
Because the RCG is meant for the power range 1-10MW, it is chosen to compare simple cycle and RCG installations with a shaft power of 2.5MW and 10MW (Table 3.2): The numbers shown in Table 3.2 for the two simple cycle installations are estimated for industrial gas turbines of manufacturer Solar, world market leader in the 1-10MW range. For the two RCG installations, the numbers are the result of current market prices of the components that the RCG consists of, and, of realistic estimates of the costs to assemble and realize the installation. The condenser of the RCG is assumed to be aircooled.

For 2.5MW shaft power, the extra investment costs of an RCG compared to the simple cycle are relatively higher than at 10MW shaft power. This is mainly caused by the (impulse) steam turbine of the RCG. Although, an impulse steam turbine is the most cost-effective solution at these relatively low shaft powers, at 2.5MW it is still more costly per MW shaft power than at 10MW shaft power.

When comparing the thermal efficiencies of the RCG and simple cycle, it shows that the gain in efficiency of an RCG at 2.5 MW is relatively higher than at 10MW. The reason for this is that in general, gas turbines can be designed more efficient with increasing shaft power. Since, with increasing shaft power it becomes more cost efficient to employ a higher compression ratio (more stages) and better thermal resistant materials. Due to the lower thermal efficiency of the 2.5MW simple cycle gas turbine, the exhaust gases have a higher temperature, thus contain more energy to be utilized by a waste-heat steam cycle. Therefore, the gain in thermal efficiency of an RCG compared to the simple cycle is the highest at the lower shaft power of 2.5MW.

Calculations were made assuming an average of € 5.80/GJ for natural gas in Europe [14]. Furthermore, an availability of 90% was assumed. This is not for maintenance reasons, it is just assumed that mechanical drives are not running full load all the time. With Table 3.2, the natural gas price and an availability of 90% the costs over the years can then be calculated (3.9).

Figure 3.9 shows the total of the investment costs and fuel costs per kW, divided by the number of years that the installation has been in use. Comparing

<table>
<thead>
<tr>
<th>Shaft power</th>
<th>$\eta_{th}$</th>
<th>Investment costs (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.5MW</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Simple cycle</td>
<td>0.30</td>
<td>1.400.000</td>
</tr>
<tr>
<td>RCG</td>
<td>0.35</td>
<td>1.800.000</td>
</tr>
<tr>
<td>10MW</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Simple cycle</td>
<td>0.38</td>
<td>3.700.000</td>
</tr>
<tr>
<td>RCG</td>
<td>0.42</td>
<td>4.500.000</td>
</tr>
</tbody>
</table>

Table 3.2: Assumptions for comparison of the simple cycle and RCG
the two 2.5MW installations to the two 10MW installations, the curves of the 2.5MW are above the curves for 10MW. This is, of course, because the investment costs of the smaller installations are higher per kW. Also their fuel costs per kW are higher because their thermal efficiency in general is lower than that of larger installations. Comparing the simple cycle to the RCG, figure 3.9 shows that both for 2.5MW and 10MW shaft power, the extra investments of an RCG installation are paid back within about two years.

Of course, this two years pay back time is anything but exact. A lot of non-exact assumptions were made to make this calculation. But, it can be stated that the expected payback time of the extra investments of an RCG installation compared to a simple cycle gas turbine is in the order of magnitude of a few years (two up to four years), which is economically very appealing.
It is has to be noted that, it is possible to design an RCG installation smaller than 2.5MW. The results are not shown in figure 3.9, but, down to 500kW, they result in about the same curve as for 2.5MW shaft power. Furthermore, with the present assumptions, it is not possible to design an RCG installation with more than 12MW shaft power. In that case, a high-efficiency impulse steam turbine of over 6MW is needed to drive the compressor, but those devices are not yet available. However, due to the free power turbine principle of the RCG, it is possible to implement multiple parallel impulse steam turbine and compressor units, supplying compressed air to the combustion chamber(s) of one larger power turbine. This type of installation was not studied yet. Furthermore, such an installation should then also be compared to an installation with one axial steam turbine.
Chapter 4

Proof of principle

In the intended fields of application of the RCG, controllability is of utmost importance. To investigate whether the RCG-principle offers stable operation, a small-scale test set up has been built.

4.1 Design of experimental set up

The key issue of the RCG is that the gas generator is steam powered. Regarding the controllability, the focus lies at the steam powered gas generator. The experimental set-up does not comprise a power turbine, but a restriction upstream of the combustion chamber instead (see figure 4.1). This choice of design for the experimental set-up gives maximum flexibility to experiment with the steam powered gas generator of the RCG. Although the experimental set-up
is small-scale, it comprises the same components as a real-scale RCG should have, in accordance with the design choices of Chapter 2. However, instead of a condenser only a feedwater reservoir is employed. This corresponds to a condenser that condenses at atmospheric pressure. The other components (figure 4.1) match the earlier discussed design choices.

The steam turbine is a KK&K impulse steam turbine (type AF3.5), the compressor is a Vortech turbo compressor (type V4XX) and a subcritical once-through boiler is employed.

The steam generator (once-through boiler) is of own design (see Appendix A), because there was not such a system commercially available at the scale of the set-up (500kW thermal power). The once-through boiler consists of a 480m stainless steel tube (ø21.3mm) inside a 0.6x0.6x2.6[m] casing and is designed to deliver superheated steam up to a maximum temperature of 500°C and pressure of 30bar(abs). As was discussed in section 2.3, the sub-critical once-through boiler is preferred for the RCG because it is compact, economic and fast responding, but special attention should be paid to the control. Figure A.2 shows the construction schematic and figure 4.3 shows the working principle of the once-through boiler in the experimental set-up. Basically, the sub-critical once-through boiler is a tube where feedwater enters on the one end, which is then converted into superheated steam and comes out on the other end. The exhaust gas is in counter flow heat exchange contact with the tube. The conversion of feedwater into superheated steam inside the tube takes place in three stages. In the economiser stage the feedwater is heated up to the boiling temperature.

![Figure 4.2: Schematic of the once-through boiler in the experimental set-up](image-url)
4.2 Sensors

The experimental set-up (see figure 4.1) was fitted with several sensors. The steam temperatures and the exhaust gas temperature at the steam generator inlet and outlet (stack temperature) were measured with thermocouples. The airflow was measured between the turbocompressor and the restriction valve with an orifice meter which makes use of a differential pressure transducer. The pressure in the steam generator was measured with a water pressure gauge at the outlet of the feedwater pump. The water pressure gauge measures the pressure in the steam tube at the steam generator inlet, this equals the pressure at the economiser inlet (see figure 4.3). The steam turbine was (standard issue) fitted with a steam pressure gauge. All the sensors were connected to a data-acquisition computer, except for the pressure gauges.

4.3 Experiments

During the experiments, the pressure at the steam generator inlet was about 1 bar higher than the pressure at the steam generator outlet. This is caused by two reasons. Firstly, because the steam generator is 2.4m high and the steam tube inlet is at the bottom and the steam tube outlet is at the top. Gravity will cause a negative pressure gradient from the inlet to the outlet of the steam tube resulting in a total pressure difference of about 0.12bar (assuming that half of the steam tube is filled with water) between the steam tube inlet and outlet. Secondly, due to friction resulting from the high velocity of the steam traveling through the steam tube, there is a negative pressure gradient from the inlet to the outlet of the steam tube. Figure 4.4 shows the airflow, pressure ratio of the turbo compressor, exhaust gas temperature, steam temperature and the stack temperature during an experiment. At t=0s air is supplied and at t=400s the combustion chamber is ignited and thus supplies the steam generator with hot
CHAPTER 4. PROOF OF PRINCIPLE

Figure 4.4: Results of experiment: from top to bottom, airflow, pressure ratio of the turbo compressor, and the exhaust gas, steam and stack temperatures

exhaust gasses. First, the combustion chamber is warmed up, for combustion stability.
At \( t=600 \) s the airflow is increased to 0.45\( \text{kg/s} \) and the exhaust gas temperature is increased to 560\( ^\circ \text{C} \). Then at \( t=1200 \) s the steam turbine starts to deliver power to the turbo compressor, thus rising the compression ratio. It can be seen that the temperature of the steam at the exit of the boiler gives an overshoot of up to 500\( ^\circ \text{C} \). This is compensated for by increasing the feedwater flow. The steam temperature lowers again, but again trips and becomes to low (200\( ^\circ \text{C} \)). Once the once-through boiler steam turbine and turbo compressor are steadily operational at \( t=1800 \) s, they respond very quickly to a change in fuel flow in the combustion chamber. This can be seen at \( t=1800 \) s and \( t=3200 \) s. The pressure ratio increases about a minute after the rise in temperature of the exhaust gas. At \( t=4200 \) s the fuel flow to the combustion chamber is cut of, and the installation powers down.

Given the exhaust gas mass flow and temperature, the temperature of the generated superheated steam can only be controlled with the amount of feedwater flow. The steam pressure follows from equilibrium with the steam flow through the steam turbine. In steady-state this does not pose a problem. However, in transients from part-load to full-load the feedwater flow will need to be increased to go from one equilibrium to another. The feedwater is incompressible and the steam is compressible, so when the feedwater flow is increased, the economiser will act as a "piston" that compresses the evaporator and superheater of the boiler. So in transients the lengths of economiser, evaporator and superheater will start to vary, affecting the transient system response. This effect has to be anticipated by a feedwater flow controller, to be able to properly control the steam temperature at the exit of the boiler.

The results show that the once-through boiler is of major influence on the transient behaviour of the RCG. Assuming that for a larger (real-scale) installation a once-through boiler would be employed that can be considered as several parallel boilers like the one in the set-up, a real RCG-system is expected to start within 10 minutes. Also it will be controllable from part-load to full-load. However, both the start-up procedure and transient control strategy need extensive attention and further study.

### 4.4 Discussion: technical feasibility of the RCG

The RCG will be controllable, but the operation strategy needs specific attention. Since for a real-scale RCG all the required components are commercially available and since the requirements regarding start-up and transient behaviour are fulfilled, the RCG is considered technologically feasible. The RCG start-up time of about 10 minutes is very fast for a combined cycle, and is acceptable for decentral and maritime applications.
CHAPTER 4. PROOF OF PRINCIPLE
Chapter 5

Transient analysis

For industrial, maritime and decentralized power applications, a short response time is vital. To become successful in these intended fields of application, the RCG is required to be able to go from part-load to full-load within minutes. Therefore, the transient behavior of the gasgenerator of the RCG, powered by the steam cycle, has to be optimized by the right choice of system design. A transient model of the RCG gasgenerator is developed and its basic responses are compared to basic responses of the experimental set-up. Next, with the transient model the effect of several possible actuators is studied and a system design for improved transient response is developed.

5.1 Transient behavior of gas turbine systems

Simple Cycle gas turbine with single shaft

For a Simple Cycle gas turbine (figure 1.1), the fuel flow is the main actuator to realize a certain output power. If a simple cycle gas turbine is to perform a transient from part-load to full-load, the fuel flow is elevated until the maximum allowable turbine inlet temperature (TIT) is reached. As soon as the turbine ("T" in figure 1.1, left) speeds up, because it is delivering more power due to the elevated TIT, the compressor will deliver more air, and more fuel can be injected without exceeding the maximum TIT. For a gas turbine without free power turbine this "increasing airflow effect" only occurs if the turbine speed can vary. If a single shaft gas turbine is implemented in generating electricity, the shaft of the turbine and compressor will be directly connected to the generator at constant speed, therefore, the air mass flow will be approximately constant.

Simple Cycle gas turbine with free power turbine

For a simple cycle gas turbine with a free power turbine ("PT" in figure 1.1, right), the increasing airflow effect always occurs during a power increase, regardless whether the free power turbine is coupled to a generator at constant
speed or not. This is because the turbine (“T” in figure 1.1, right) that is driving
the compressor, can vary in speed. The combination of compressor, combustor
and turbine that drives the compressor are often referred to as the gas generator
of the power turbine, since they provide the power turbine with the amount of
pressurized hot gasses that it needs to drive its load.

The RCG
Just like the Simple Cycle gas turbine with free power turbine, the RCG can
be regarded as a combination of a gas generator and a free power turbine: in
the RCG-system the gas generator consists of the compressor, combustor and
the steam turbine cycle that drives the compressor. As in all turbine systems,
transients in the RCG are induced by changes in the fuel-flow to the combustion
chamber. This leads to a direct increase in the power production. The next step
in the transient behavior is due to the change in the exhaust temperature of
the turbine, which leads to a change in the heat exchange in the boiler. In its
turn, this gives a transient in the steam production and, thus, in the power
production in the steam turbine. The closing of the transient cycle is the power
balance in the steam turbine and the air compressor, which leads to a change
of the compressor speed and, therefore, to a change in the air delivered to the
combustion chamber.

5.2 Transient equations
The transient model needs to ascribe all the interactions which take place dur-
ing the RCG transient. To describe these interactions, a model of the RCG
gasgenerator was made using the filling-and-emptying method (similar to [6]).
This model will first be applied on the experimental set-up (figure 5.1). In the
filling-and-emptying method (figure 5.2), the system is split in boundaries and
volumes. In the boundaries, the unsteady mass flow follows momentum con-
servation, using the characteristic of the described element combined with the
pressure in the surrounding volumes. On the other hand, the unsteady pressure
and temperature in a volume are described using energy and mass conservation
in the volumes using the in- and outlet conditions that follow from each of its
boundaries. Along with the flow properties, a model for the rotating equipment
is necessary to describe the transient state of the combined compressor and
steam turbine.

5.2.1 Flow elements
The unsteady equations describing boundaries are deduced for the integral mo-
momtum equations. For the modeling of surge/stall phenomena, this approach
was proposed by Greitser [22] and, since then, has been well-established for
unsteady modeling of turbine equipment. The (one-dimensional) integral momentum equations are written as:

\[
\frac{dm}{dt} = - \int_{L} A(x) \frac{dp}{dx} \, dx + F_{\text{external}} \tag{5.1}
\]

where \(L\) is the length of the element, \(A(x)\) the local cross-section and \(dp/dx\) the pressure gradient. The forcing term \(F_{\text{external}}\) describes the loss/gain of momentum in the specific system element. For example, in valves, the forcing term induces the viscous dissipation, while for a compressor it stands for both the blade forcing and the added dissipation.
The integral over the pressure gradient is approximated using an estimated surface and length scale:

$$\int_{L} \frac{A(x) \, dp}{dx} = \frac{A}{L} \Delta p$$

(5.2)

The forcing term is modeled using the steady flow (steady-state) characteristics of the element:

$$F_{\text{external}} = \frac{A}{L} \Delta p_{ss}(\dot{m})$$

(5.3)

Ergo,

$$\frac{d\dot{m}}{dt} = \frac{A}{L} (\Delta p_{ss}(\dot{m}) - \Delta p)$$

(5.4)

For typical system properties ($\Delta p \approx 10^5 \text{N/m}^2$, $A \approx 0.01 \text{m}^2$, $L \approx 0.1 \text{m}$, $\dot{m} \approx 1.0 \text{kg/s}$), the timescale ($\tau_m$) for these balances can be estimated:

$$\tau_m \approx \frac{L \dot{m}}{A \Delta p} = 10^{-4} \text{s}$$

(5.5)

This means that, for all boundary elements, these time-scales are much smaller than those of the thermodynamic balances (see equations (5.13) and (5.17)). Therefore, in our modeling the boundary elements will all be described from their steady characteristics:

$$\dot{m} = f(\Delta p, ...)$$

(5.6)

5.2.2 Volumes

For the volumes, integral mass and energy conservation are used to follow the mass ($m$) and temperature ($T$) in the volume ($V$):

$$\frac{dm}{dt} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}}$$

(5.7)

$$\frac{dU}{dt} = \dot{H}_{\text{in}} - \dot{H}_{\text{out}} + \dot{Q} = c_p(\dot{m}_{\text{in}}T_{\text{in}} - \dot{m}_{\text{out}}T_{\text{out}}) + \dot{Q}$$

(5.8)

where $\dot{m}$ is the mass flux and $\dot{H}$ is the enthalpy flux ($= \dot{m}h$), the indices $\text{in}$ and $\text{out}$ refer to incoming and outgoing boundaries respectively and $\dot{Q}$ represents any added heat flux (e.g. in the combustion chamber).

The change of internal energy $dU_{CV}$ of a control volume with mass $m$:

$$dU_{CV} = c_v dmT = c_v T dm + c_v m dT$$

(5.9)

Equations (5.7) and (5.9) can be combined into:

$$\frac{dU_{CV}}{dt} = c_v T \frac{dm}{dt} + c_v m \frac{dT}{dt} = c_v T(\dot{m}_{\text{in}} - \dot{m}_{\text{out}}) + c_v m \frac{dT}{dt}$$

(5.10)

With equation (5.8):

$$c_v m \frac{dT}{dt} = \dot{m}_{\text{in}}(c_pT_{\text{in}} - c_vT) - \dot{m}_{\text{out}}(c_pT_{\text{out}} - c_vT) + \dot{Q}$$

(5.11)
5.2. TRANSIENT EQUATIONS

For each of the (perfectly-stirred) volumes, the out-going temperature equals the temperature in the volume \((T_{\text{out}} = T)\). While \(m\) and \(T\) can be solved from the balance equations, the pressure inside the volume follows from the equation of state:

\[ pV = mR_sT \]  

(5.12)

The time-scale \(\tau_V\) connected to these thermodynamic balance-equations can be estimated by using typical system dimensions: \(m \approx 1\) kg, \(\dot{m} \approx 1\) kg/s, \(c_p/c_v = \gamma \approx 1.4\):

\[ \tau_V \approx \frac{m}{\gamma \dot{m}} \approx 0.5 - 1\text{s} \]  

(5.13)

For \(t >> \tau_V\), \(\frac{dm}{dt} \approx 0\) and \(\frac{dU}{dt} \approx 0\), therefore with equations (5.7) and (5.8) it follows:

\[ \dot{m}_{\text{in}} = \dot{m}_{\text{out}} = \dot{m} \]  

(5.14)

\[ c_p \dot{m}_{\text{out}} T_{\text{out}} = c_p \dot{m}_{\text{in}} T_{\text{in}} + \dot{Q} \]  

(5.15)

Combining equations (5.14) and (5.15) gives:

\[ c_p \dot{m}_{\text{out}} T_{\text{out}} = c_p \dot{m}_{\text{in}} T_{\text{in}} + \dot{Q} \]  

(5.16)

This means that for timescales much larger than 1 second the thermodynamic properties in the volumes respond steady, i.e.:

\[ T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{c_p \dot{m}} \]  

(5.17)

**Steam generator**

The steam generator is a special case of a thermodynamic volume in which the balance of the heat exchange from the hot gas and the water/steam is the basis of the transient response. The state of the steam generator is determined by separating it into three separate sections (economizer, evaporator and super heater) describing the lengths of these sections depending on the state of the boiler, the amount of water and the amount of heat from the exhaust gas that is supplied to it. For the evaporator and economizer, the unsteady energy conservation is used:

\[
\frac{dU_{\text{CV}}}{dt} = \dot{H}_{\text{in}} - \dot{H}_{\text{out}} + \dot{Q} - p \frac{dV}{dt}
\]  

(5.18)

where \(\dot{Q}\) is the heat supplied by the hot-gas flow. The internal energy \(U_{\text{CV}}\) is now written as \(mu\), where \(u\) is the specific energy of the steam/water in the specific steam-generator section. We now assume that this mean specific energy is constant in time (assuming that the temperature distribution in the economizer and evaporator are constant, ergo, the change in steam-pressure is fast as compared to the change of the lengths).

This leads to:

\[
\frac{dU_{\text{CV}}}{dt} = \frac{d(mu)}{dt} = m \frac{du}{dt} + u \frac{dm}{dt} \approx u \frac{dm}{dt} = u \rho \frac{dV}{dt}
\]  

(5.19)
Where $dV$ is the change of volume of a steam generator section. With the enthalpy definition $h = u + pv$ it follows:

$$\frac{dU_{CV}}{dt} = (h - pv) \rho \frac{dV}{dt} = (\rho h - p) \frac{dV}{dt}$$

Combining equation 5.18 and 5.20 gives:

$$\frac{dV}{dt} = \frac{(\rho h - p) \frac{dV}{dt}}{(\rho h - p)} = \dot{H}_{in} - \dot{H}_{out} + \dot{Q} - p \frac{dV}{dt}$$

$$\frac{\rho h}{dt} = \dot{H}_{in} - \dot{H}_{out} + \dot{Q}$$

Therefore, the state of the boiler is described by:

$$\rho_{water} \pi r_{pipe}^2 h_{water} \frac{dL_{EC}}{dt} = \dot{H}_{in,Ec} - \dot{H}_{out,Ec} + \dot{Q}_{lm,Ec}$$

$$\pi r_{pipe}^2 \frac{\rho_{water} Y + h_{satst} \rho_{sat}}{1 + Y} \frac{dL_{EV}}{dt} = \dot{H}_{in,Ev} - \dot{H}_{out,Ev} + \dot{Q}_{lm,Ev}$$

where $\dot{Q}_{lm}$ is the heat transferred from the hot gas-flow, and $Y$ is the density-ratio of water and steam. The length of the super-heater follows from the remaining length of the fixed total tube length. The timescale of the steam-generator transient length ($\tau_L$) can be estimated using the properties of the once-through boiler (see chapter 4) and equation (5.24). Assuming $L \approx 100m$, $\dot{Q} \approx 10^9W$, $\rho_{water} = 10^3kg/m^3$, $r_{pipe} = 0.01m$ and $u_{water} \approx 10^6 J/kg$ for the economizer this leads to:

$$\tau_L \approx \frac{L \rho_{water} r_{pipe}^2 u_{water}}{\dot{Q}} = 100s$$

In essence, the pressure in the boiler follows from balance between ingoing water, outgoing steam through the steam turbine nozzle and supplied heat by the exhaust gases. But the boiler pressure is also influenced by the changes of length of the three sections. The boiler pressure is determined by determining $\frac{\partial \rho}{\partial t}$ which is derived from the state of the evaporator:

$$\frac{d\rho_{EV}}{dt} = \frac{dL_{EV}}{dt} \frac{\rho_{EV} A}{\rho_{EV}} + A L_{EV} \frac{d\rho_{EV}}{dt}$$

From appendix B it follows that:

$$\frac{d\rho_{sat}}{dt} = \frac{\frac{n_{m} - n_{i}}{A} + \rho_{i} \frac{dL_{EC}}{dt} + 2 \frac{\rho_{sat} \rho_{w}}{\rho_{sat} + \rho_{w}} \frac{dL_{EV}}{dt} - \rho_{sat} \frac{T_{sat}}{T_{out} - T_{sat}} \ln \left( \frac{T_{out}}{T_{sat}} \right) (\frac{dL_{EC}}{dt} + \frac{dL_{EV}}{dt})}{L_{EV} \frac{2 \rho_{s}}{(\rho_{sat} + \rho_{w})^2} + L_{SH} \frac{T_{sat}}{T_{out} - T_{sat}} \ln \left( \frac{T_{out}}{T_{sat}} \right) \frac{T_{sat}}{T_{out} - T_{sat}} \frac{dL_{EV}}{dt}}$$

The time scale for the boiler pressure can then be estimated with:

$$\tau_p \approx \frac{\rho_{sat}}{\rho_{water}} \tau_L \approx 1s$$
5.2.3 Rotating equipment

The coupling of the compressor and steam turbine is determined by the power balance of both components. The speed $N$ describes the state of the compressor. The acceleration is determined by the difference between the power supplied by the steam turbine ($P_{st}$) and the power required by the compressor ($P_c$) at a certain time, divided by their combined inertia $I_{st+c}$.

$$\frac{dN}{dt} = \frac{P_{st} - P_c}{NI_{st+c}} \quad (5.29)$$

For characteristic properties of the proposed RCG ($P \approx 10^5 \text{W}$, $N \approx 500 \text{s}^{-1}$ and $I \approx 10^{-2}$), this leads to a characteristic time scale:

$$\tau_N \frac{N^2 I}{P} = 3 \times 10^{-2} \text{s} \quad (5.30)$$

For all transients more slower that $\tau_N$, eq.(5.29) can be replaced by:

$$P_{st} = P_c \quad (5.31)$$

The details of this balance (which steam-turbine power and restriction valve combination leads to which operating point in the compressor characteristic) are difficult to calculate in advance. Therefore, even though this time scale may be small as compared to the other system components, we have chosen to include this transient-scale within the model. Note, detailed calculation of $I$ is unnecessary, since the corresponding transients are much faster than those in the steam generator.

5.3 Model

Based on the timescales derived from the unsteady state equations, it can be concluded that the transient behavior of the RCG is determined by the dynamics inside the steam-generator, since:

$$\tau_m, \tau_N, \tau_V \ll \tau_{\text{steam}} \quad (5.32)$$

Therefore, most components can be described by only their steady-state characteristics.

5.3.1 Steam generator

As discussed, in the model the steam generator is separated into three separate sections: economizer, evaporator and super heater (see figure 5.3). The temperatures of the steam generator have subscript 1 at the beginning of the economiser, subscript 2 at the transition from economiser to evaporator, subscript 3 at the transition from evaporator to superheater and subscript 4 at the end of the superheater. The water is converted into steam from $T_{H2O,1}$ to


Figure 5.3: Overview of the once-through steam generator in the transient model

$T_{H20,4}$, while the exhaust gas is cooled down from $T_{g,4}$ to $T_{g,1}$ in counterflow. The temperatures $T_{H20,2}$ and $T_{H20,3}$ represent the boiling temperature of the water at the current pressure in the tube. Since the pressures losses due to friction of the steam flow in the tube are neglected, $T_{H20,2}$ and $T_{H20,3}$ are always equal. The steam pressure in the tube is determined from equation (5.27). The lengths of the sections are calculated with equations (5.23) and (5.24). In each section, heat is added ($\dot{Q}_{lm,section}$) from the exhaust gasses to the steam tube by heat transfer:

$$\dot{Q}_{lm,section} = \pi DL_{section}\overline{h}\Delta T_{lm} \quad (5.33)$$

Where $\overline{h}$ is the average convectional heat transfer of a section, which is determined from the "bank of tubes" approach [23].
5.3.2 Compressor and restriction valve

The power required for the compressor is derived from equation (3.2). The compressor isentropic efficiency is assumed constant, $\eta_c = 0.70$, because the compressor is operated in the area of the compressor map where the compressor efficiency is around 70% (see figure 5.4). The compressor experiences back pressure from the valve that is in front of the combustion chamber:

$$m_{\text{air}} = \frac{C_{\text{valve}} p_{c,\text{out}}}{\sqrt{T_{c,\text{out}}}}$$  \hspace{1cm} (5.34)

The compressor pressure ratio $r_c$ is determined from equation (5.34) and a simplified compressor map: a polynomial fit from a relation between the speed $N$ and the air mass flow.

$$r_c = 1 - 3.26 \cdot 10^{-7} N_c + 4.16^{-10} (N_c)^2$$  \hspace{1cm} (5.35)

5.3.3 Steam turbine

The steam turbine drives the compressor, therefore, the steam turbine speed is directly linked to the compressor speed by the transfer ratio of the integrated gearbox of the steam turbine. An integral part of the RCG design is the choice for a relatively economic impulse steam turbine [17]. The steam turbine power $P_{st}$ is calculated with equation (3.5). Note that an impulse steam turbine is a radial steam turbine of the impulse type (zero-reaction) and has an expansion nozzle (see figure 5.5): the steam expands completely in the expansion nozzle.
CHAPTER 5. TRANSIENT ANALYSIS

Figure 5.5: Principle of the impulse steam turbine

...before it hits the steam turbine blades. The isentropic efficiency $\eta_{st}$ of an impulse steam turbine is calculated from the nozzle spouting velocity $c_0$, the turbine blade speed $u$, and the steam turbine characteristic:

$$c_0 = \sqrt{2(h_{ST,\text{in}} - h_{ST,\text{out}})} \quad (5.36)$$

$$u = \frac{2\pi r_{ST} N_{ST}}{60} \quad (5.37)$$

With $N_{ST}$ the speed (RPM) of the steam turbine wheel. The experimental set-up incorporates a KKK AF3.5 steam turbine, figure 5.6 shows the isentropic efficiency curve of the AF-series. In the transient model $\eta_{st}$ is determined from the following polynomial fit:

$$\eta_{st} = 0.54 - 8.64 \times \left( \frac{u}{c_0} - 0.25 \right)^2 \quad (5.38)$$

Which shows that 0.54 is the maximum steam turbine efficiency. Newer models of the impulse steam turbine can reach turbine efficiencies of up to 0.80. The

Figure 5.6: Performance map of the steam turbine in the experimental set-up
steam mass flow through the steam turbine is determined by the steam temperature and the pressure drop over the steam turbine nozzle. The steam mass flow through the steam turbine nozzle can be calculated with:

$$\dot{m}_{w} = \frac{C_{\text{nozzle}}p_{\text{boiler}}}{\sqrt{T_{\text{ST,in}}}}$$  \hspace{1cm} (5.39)

in which the $C_{\text{nozzle}}$ is a constant.

### 5.3.4 Combustion chamber

The combustion chamber outlet temperature follows from the heat that is added by combusting fuel and mass flow of air that is delivered by the turbo compressor, through the restriction valve.

$$\dot{m}_{g}c_{p,g}(T_{\text{comb,out}} - T_{\text{comb,in}}) = \dot{Q}_{\text{fuel}}$$  \hspace{1cm} (5.40)

In which $\dot{m}_{g}$ is the exhaust gas mass flow and $c_{p,g}$ is the specific heat at constant pressure of the exhaust gas mass flow.

### 5.3.5 Auxiliary burner and mixing point

The hot gasses of the auxiliary burner are mixed with the hot gasses from the combustion chamber at a mixing point. The mixed hot gasses then enter the steam generator. The auxiliary burner is a boiler burner with an air blower and fuel nozzle. Unlike the combustion chamber, the hot gasses of the auxiliary burner are assumed to have a constant temperature of 1073K, because this type of burner is usually operated with a constant fuel-air ratio. The power of the auxiliary burner can, therefore, only be varied within a limited range by varying the mass flow of the hot gasses.

### 5.4 Basic model layout

To give a general overview of the transient model, the basic layout of the calculation processes is shown in figure 5.7. The model is far to extensive to show all calculation processes. Therefore, only the most important calculation processes, that are considered to form the core of the model, are shown. Basically, the steam turbine and compressor equations are being iterated with the restriction valve to find a pressure ratio and gas mass flow. The outcome is forwarded to the steam generator where the steam production is determined with the steam generator state and heat transfer. The steam flow and state is then forwarded to the steam turbine, where the steam turbine power and compressor power balance is calculated again.
5.5 Basic system transient response

Figure 5.8 shows the modeled RCG system without power turbine (similar to figure 4.1) with its basic control strategy. The main actuator of the RCG is the fuel flow of the combustion chamber, which is the same as for simple cycle gas turbines. As discussed, the RCG can be regarded as a combination of a gasgenerator and a free power turbine. If an RCG-installation is to perform a transient from 50% part-load to full-load, this transient will not be completed until the airflow into the combustion chamber is elevated enough to double the fuel flow. Also similar to other gas turbine systems, the fuel flow of the combustion chamber of the RCG is limited by the maximum Turbine Inlet Temperature (TIT). Because the model does not incorporate a power turbine, there is no determined TIT-limit. Instead, an arbitrary turbine outlet (exhaust gas) temperature limit of 1073K is used. The most important control actuator of the steam cycle is the boiler feedwater flow. The feedwater flow is controlled
5.5. BASIC SYSTEM TRANSIENT RESPONSE

such that the amount of air that is produced by the compressor is as high as possible, this means maximum steam turbine power output. The feedwater flow is also controlled such that the steam conditions are within limits at all times so that the steam turbine does not sustain any damage. The feedwater flow controller (standard PID) is set to maintain a boiler outlet steam temperature of 823K, which corresponds to the maximum inlet temperature for the steam turbine in the set-up. First the transient behaviour of the basic RCG system is studied, with only fuel flow control and feed water control.

Figure 5.9 shows the compressor air flow, exhaust gas temperature and combustion chamber power during a simulation of a transient for the combustion chamber power from 250kW to 500kW. At t=400 seconds the transient is started; the fuel flow in the combustion chamber is elevated such that the maximum allowable exhaust gas temperature is reached for the available airflow at that time. This results in a very quick rise in thermal power from 250kW to about 350kW. After that, it rises more slowly, because it is limited by the available airflow. From the moment that the transient is started and the thermal power was raised to 350kW instantly, the exhaust gas temperature rises. The steam generator then starts to generate more steam and the steam turbine supplies more power to the the compressor so that it starts accelerating. While the airflow rises, the thermal power can rise gradually from 350kW at 400s to 500kW at around 850s.

Considering the time it takes for the combustion chamber to reach the level of 500kW as the transient time, it can be seen from figure 5.9 that the modeled
basic system performs a transient from 50% part-load to full-load in about 450 seconds.

Figure 5.10 shows the response of the steam generator to the basic transient of the combustion chamber power from 250kW to 500kW. Shortly after the exhaust gas temperature has risen to its maximum, the boiler outlet steam temperature \( T_{\text{H2O}} \) becomes considerably higher during the transient. Immediately when the steam temperature rises, the feed water control compensates by almost doubling the feed water flow. The feedwater pump can increase the feedwater flow instantly because it is a positive displacement pump. However, the steam temperature continues to rise because the steam turbine flow does not double simultaneously. At first, at 400 seconds the steam turbine flow rises quickly together with the feedwater flow, but, after a few seconds, the evaporator section collapses temporarily causing a shorter evaporator and lower steam pressure for a period of about 10 seconds.

Altogether, the steam turbine flow stays behind for almost 400 seconds after the sudden increase of feedwater flow. This is because the evaporator section can not keep up with the sudden increase in feedwater flow, it takes time before
twice as much water is being evaporated and a higher pressure and boiling temperature is realised. Note that the outlet flow of the steam generator is equal to the steam flow through the steam turbine nozzle and will, therefore, only increase if the steam pressure is increased (equation 5.39). Together with the delayed increase of evaporation and steam pressure, also the increase of outlet flow of the steam generator is delayed. Because of the above mentioned effect the $q_{SH}$, the heat that is transferred to the superheater by the exhaust gasses, rises quicker than the steam flow through the superheater. Therefore, the steam outlet temperature is higher during the power increase of 250kW to 500kW.

The response time of the basic RCG system of about 450 seconds is relatively good for a combined cycle [20],[21] and is mainly due to the employment of a once-through boiler. A conventional drum-type steam boiler would have a response time of three up to five times that of the once-through steam generator. However, in the intended applications the RCG would benefit from a further improved transient response. As was shown in the above, the transient response of the basic RCG is limited by the maximum allowable turbine inlet temperature (TIT) and the maximum steam turbine inlet temperature. The latter is
caused by the delayed response of the steam generator, which makes it still the slowest responding component.

5.6 Simulation results versus experiments with the set-up

The simulations of the RCG cycle were performed with a model of the experimental set-up to make it possible to validate them with experiments at a later stage. The experimental set-up is small-scale (500kW thermal power), however, the simulation results with respect to response time are believed to give the right order of magnitude for future real-scale RCG-installations. The reason for this is that the determining component with regard to the time-scale of the dynamical response of the RCG-cycle is the once-through boiler; the moment of inertia of the turbo machinery is negligible compared to the time-scale of the steam-process in the boiler. The once-through boiler of the experimental set-up comprises one 480m long stainless steel tube (21.3mm x 2.77mm). Future real-scale RCG installations are expected to have once-through boilers that consist of multiple similar tubes in parallel. Assuming that the tubes have similar dimensions, multiple tubes in parallel are expected to have the same order of magnitude response time as one single tube; therefore, real-scale RCG installations are also expected to have the same order of magnitude response time as the experimental set-up that was modeled in this study.

It is, therefore, important to compare boiler transient experiments of the real set-up to the results of the model. Unfortunately, it is impossible to perform experiments and simulations with exactly the same boundary conditions. This is mainly caused by the fact that the set-up suffers from large parasitic losses. For practical and economic reasons the power of the set-up was down-scaled. This results in relatively high heat losses and mechanical losses. It is, however, possible to study whether the behaviour of the set-up and the model is similar and whether the response delays are of the same order of magnitude.

5.6.1 Advanced experimental set-up

Demands

The experimental set-up of is adapted to incorporate all relevant technical aspects that a real-scale installation will incorporate. The experimental set-up had an open Rankine cycle; at the outlet of the steam turbine, the steam was vented to the environment. The advanced set-up should have a closed Rankine cycle with a condenser and feedwater reservoir. This way the response can be compared to that of the transient model. This also gives the opportunity to gain practical experience with the steam management of the closed Rankine cycle of the RCG.
5.6. SIMULATION RESULTS VERSUS EXPERIMENTS WITH THE SET-UP

The experimental set-up was started with the aid of compressed air. A real-scale RCG will be started with an auxiliary burner on the steam boiler (see chapter 2). The advanced set-up, therefore, incorporates an auxiliary burner for experiments with the start-up procedure of the RCG (this will be treated further in chapter 6).

The experimental set-up was operated by hand and had a Labview data-acquisition. The advanced set-up will have an operating system and data acquisition similar to a real-scale RCG. This is an operating system as accepted in the industry, so that the advanced set-up operating system that will be developed and tested can be transferred almost completely to the RCG demo system.

Design

Figure 5.11 shows the schematic of the advanced set-up according to the demands to the design; an RCG prototype with a closed Rankine cycle and an auxiliary burner for start-up and supplementary firing. It is still chosen to implement a restriction instead of a power turbine, just like the experimental set-up. The reason for this is similar as for the experimental set-up; for the prototype the main points of interest lie at developing an operating system, start-up procedure and transient control. Just like with the previous experimental set-up, there are little new insights to be expected from the free power turbine of the RCG. Since the scale is still the same as for the previous set-up, the gas turbine

![Diagram of advanced experimental set-up](image)

Figure 5.11: Schematic of advanced experimental set-up

combustor, the steam generator, the steam turbine and the turbo compressor
can be transferred to the advanced set-up. The most important components that are added, are a condenser, feedwater reservoir, auxiliary burner and operating system.

Since there is only little cooling water available in the lab, it is chosen to implement an air-cooled condenser. The advanced set-up incorporates a condenser of 600kW thermal power from the company Klima. The total internal volume of the steam tube of the steam generator is 90 litres, the feedwater reservoir should, therefore, be able to hold at least 90 liters of water. The auxiliary burner has a fan that takes in air and can be started independently from the rest of the installation. This type of burner is very common on steam boilers and widely available. The RCG prototype features such a burner with a thermal power of 400kW from the manufacturer Riello.

It is chosen to implement a Siemens operating system combined with "In Touch" software for the visual interface. Figure 5.12 shows a picture of the advanced set-up in which the condenser, feedwater reservoir, auxiliary burner and operating system can be recognized clearly. Figure 5.12 shows that the steam turbine is placed above the condenser and feedwater reservoir; this way all condensed water (droplets) that might be in the steam at the steam turbine outlet will be forced towards downwards away from the steam turbine by gravity. This is necessary, because if water could accumulate in the steam turbine casing, the turbine wheel could be severely damaged since it could hit liquid water at high speed. An alternative solution is pumping condensed water to the condenser or feedwater reservoir, but gravity is considered the most failsafe way.
5.6. SIMULATION RESULTS VERSUS EXPERIMENTS WITH THE SET-UP

Operating system

Figure 5.13 shows the P&ID of the advanced set-up. The P&ID shows the main

Figure 5.13: Schematic of the actuators and data acquisition of the advanced set-up
components, such as the steam generator (1-B1), the steam turbine (1-D1), the aircooled condensor (1-A1; with two fans A and B), the compressor (1-V1), a restriction valve (2-HV-1); which emulates the imaginary expansion turbine (2-D2), the combustion chamber 2-D1, and the auxiliary burner (1-D2).

The P&ID also shows the supporting components such as oil pumps, oil coolers, and air fans. These are all part of the RCG operating system together with the data acquisition and actuators such as the feedwater pump and fuel flow controllers. Oil pump (1-P5) supplies the bearings and hydraulic internal operating system of the steam turbine with oil. The bearings of the turbo compressor are supplied with oil by oil pump (1-P4). The feedwater reservoir (1-B2) is placed downstream of the condensor and upstream of centrifugal feedwater pump (1-P1). This centrifugal pump (1-P1) pressurizes the feedwater from atmospheric pressure up to 1 bar to prevent cavitation in the piston feedwater pump (1-P2) that pressurizes the feedwater up to boiler pressure. Figure 5.13 also shows that part of the feedwater that is pressurized by the centrifugal pump (1-P1) is bypassed to the pipeline in between the steam turbine exit and condensor inlet. This is necessary to improve the condensing process in the condensor and stabilize the condensing temperature. To maintain a pressure of 1 bar in the feedwater pipeline to the feedwater pump, an adjustable restriction is placed in the bypass line. In the gas turbine cycle, a blow-off valve is placed in between the steam turbine driven air compressor (1-V1) and the combustion chamber (2-D1) so that the combustion air can be regulated and even fully blown-off if necessary.

In figure 5.13 all temperature sensors are dedicated with TE and all pressure transducers are dedicated with PE. Figure 5.14 shows the interface with the Siemens operating system, a desktop computer and monitor with the software program "In Touch". With the computer, all actuators can be operated except the combustion chamber, the blow-off valve and the restriction valve. The latter have their own separate operating system that originates from the earlier experimental set-up. The actuators that can be operated with the computer are the condensor, the centrifugal and feedwater pump, the oil pumps of the steam turbine and compressor, and the auxiliary burner. All sensor signals and actuator settings are logged continuously by the computer.

5.6.2 Feedwater flow transient response

Figure 5.15 shows an experiment with the laboratory set-up where the feedwater flow was varied with constant power and temperature of the auxiliary burner. The feedwater flow pump was varied between a setting of 9% and 8% which corresponds to 0.09kg/s and 0.08kg/s. It can be seen that if the feedwater flow is reduced at around t=1550 seconds, the steam temperature starts to increase gradually after about 70 seconds and reaches 250°C after 500 seconds. If the feedwater flow is elevated to the original value at around t=1870 seconds, it takes about 200 seconds before the steam temperature starts to decrease. Clearly, the response is not symmetric to for decreasing and increasing the feed-
5.6. SIMULATION RESULTS VERSUS EXPERIMENTS WITH THE SET-UP

Figure 5.14: Visual interface operating system and data acquisition of the RCG prototype at Technische Universiteit Eindhoven

Figure 5.15: Experiment with lab set-up: steam generator response with feed-water variation
water flow: a decrease in steam temperature takes more time than an increase of steam temperature. At around 2290 seconds both the feedwater pump and the auxiliary burner are switched off, the experiment is ended. From this lab experiment it clearly follows that, the steam temperature can be controlled with the feedwater flow, however with a delay of order of magnitude $10^2$ seconds. Also, the fact that the steam temperature increases faster than it decreases is a point of attention for ensuring a safe steam turbine inlet temperature.

For comparison, the same experiment was simulated in the transient model. The result is shown in figure 5.16. In the transient model simulation, the steam temperature increases almost immediately after the feedwater flow is reduced at around $t=1550$ seconds. After 300 seconds, the steam temperature has risen from $150^\circ\text{C}$ to $250^\circ\text{C}$. When the feedwater flow is elevated to the original value at around $t=1870$ seconds the steam temperature starts to decrease almost immediately. After 300 seconds the steam temperature has decreased from $250^\circ\text{C}$ to $200^\circ\text{C}$ and is still decreasing. Like the lab experiment, the transient model also shows an non symmetric response for decreasing and increasing feedwater flow: the decrease of the steam temperature takes longer than the increase of steam temperature. The transient model does not show a delay in the steam temperature response to a change of the feedwater flow. This can be explained from the fact that the model considers an average water/steam flow for every

![Figure 5.16: Simulation with transient model: steam generator response with feedwater variation](image-url)
section (see Appendix B). Furthermore, it does not consider friction from the inner tube wall and consequently finds no pressure drop over the length of the boiler tube. Therefore, the model will find an instant change of flow throughout the steam generator when the feedwater flow is varied. While in reality this will only be valid for the economiser, which is filled with incompressible water. However, the total response time and amplitude of the lab set-up and the transient model can be considered of the same order of magnitude ($10^2$ seconds and $10^2 ^\circ C$ respectively).

### 5.6.3 Auxiliary burner transient response

Figure 5.17 shows the transient response of the steam generator when the auxiliary burner power varies step wise, while keeping the feedwater flow constant. This experiment was started when the steam generator was in a stable work-

![Figure 5.17: Experiment with lab set-up: steam generator response with auxiliary burner variation](image)

point at auxiliary burner setting 40%, which corresponds to thermal power of 170kW. At $t=4000$ seconds, the auxiliary burner setting is changed to 30% which corresponds to 155kW (non-linear scale). After 100 seconds a very strong drop in steam temperature to the saturated value can be observed, which means that the entire superheater section disappeared. The auxiliary burner setting is switched again to the original value of 40% (200kW) at 4400 seconds, and after 100 seconds the superheater reappears very quickly. At $t=4700$ seconds the steam temperature has increased to about $310^\circ C$. After that, it takes 1500 seconds for the steam temperature to reach the original value of $380^\circ C$. At $t=5900$ seconds a smaller decrease of burner power is implemented; from 40% to 35%. Then, a more gradual drop in steam temperature can be observed. At
t=7700 seconds the steam temperature is elevated again by elevating the auxiliary burner setting to 37%. At t=8100 seconds, the burner power is increased further to 38%. The same auxiliary burner response experiment was simulated with the transient model for comparison (see figure 5.18). It can be seen that for auxiliary burner power variation between 40% and 37%, both the response delay and response time of the transient model are very similar to that of the lab set-up. For auxiliary burner power variation between 40% and 35%, the model response time is a longer than of the lab set-up. This might be caused by an inaccurate modeling of the heat capacity of the stainless steel tube. However, from these auxiliary burner experiments it can be seen that the response behavior and delays of the model and the set-up are very similar and of the same order of magnitude. It can also be seen that the auxiliary burner is indeed an excellent actuator for controlling the steam generator and thus the RCG in general.

5.7 Strategy for transients of the RCG

In the previous section it was shown that the response behavior and delays of the model are similar and of the same order of magnitude as of the set-up. The model is therefore considered suitable to develop a strategy for transients of the RCG.

**Auxiliary burner overdrive**

The main purpose of the auxiliary burner is to start the RCG system, but it can also be used for supplementary firing. The response of the RCG is expected to
5.8. ADDING AUXILIARY BURNER OVERDRIVE

improve by firing the auxiliary burner during a transient. The auxiliary burner is behind the power turbine, therefore it can be fired without increasing the TIT and thus without damaging the power turbine. In general this is called supplementary firing, here it will be called auxiliary burner overdrive because the aim is to improve the transient response with the auxiliary burner.

Steam temperature limiting

Because of the steam temperature rise during a transient, some form of steam temperature limiting has to be implemented, otherwise the power of the combustion chamber would even have to be limited more in order to keep the steam turbine inlet temperature under its maximum allowable value. This can be done by after-spray, which means injection (spray) of water directly into the steam after the steam generator. An alternative way of avoiding a possibly dangerous temperature rise of the steam temperature during a transient is restricting the temperature of the exhaust gas of the supplementary burner by mixing it with ambient air, so called aircooling. This will result in a higher exhaust gas flow of a lower temperature, thus increasing the generated steam flow, but also restricting the temperature of the steam flow.

In the following the results of implementing an auxiliary burner overdrive, after-spray and aircooling are discussed.

5.8 Adding auxiliary burner overdrive

Figure 5.19 shows the operating strategy of the auxiliary burner overdrive. The auxiliary burner overdrive fires the supplementary burner when the maximum TIT of the power turbine for steady state is exceeded, indicating a demand for an increase of shaft power and insufficient airflow. The auxiliary burner overdrive is modeled as a relay that switches the supplementary burner on at a power of 125kW or off (0kW). The auxiliary burner is switched off again when the exhaust gas temperature drops 50K below its maximum set value. There is implemented a 50 degree margin to prevent the supplementary burner from keeping switching on and off again when operating near the limit of the supplied airflow. The auxiliary burner overdrive was tested in a transient simulation of the combustion chamber power from 250kW to 500kW (see figure 5.20). Starting at 400 seconds, the requested combustion chamber power was 500kW, but this could not be realized immediately because of insufficient airflow to the combustion chamber. At this point, the auxiliary burner overdrive detects an exhaust gas temperature of 1073K and activates the auxiliary burner. At around t=660 seconds, the airflow is increased such that the exhaust gas temperature drops below the 50K margin and the auxiliary burner is switched off. The combustion chamber power is already at 500kW at t=590 seconds. The response of the system with overdrive is similar to that without overdrive, only it takes fewer time; 260 seconds versus 450 seconds. Figure 5.21 shows the response of the steam
generator to the transient with overdrive. The evaporator length, steam pressure and steam flow are fluctuating much more than during the transient without overdrive (see figure 5.10). It can be seen that during the transient the steam pressure rises to almost 39 bar and settles at 31 bar after the transient. During the transient without auxiliary burner overdrive the steam pressure gradually increased to 31 bar, without overshoot. The temporary high steam pressure with auxiliary burner overdrive is caused by the fact that there is added 125kW extra thermal power to the steam generator during the transient. This causes a temporary extra steam production, resulting in extra pressure, extra steam flow and thus extra steam turbine power. Consequently the steam turbine and compressor speed up faster and sufficient airflow for 500kW thermal power in the combustion chamber is obtained sooner. Because the auxiliary burner is behind the (virtual) expansion turbine, the turbine sustains the same thermal load as without auxiliary burner overdrive. However, during the transient with overdrive, the steam turbine inlet temperature is about 830K, which is higher than during the transient without overdrive. This was to be expected because more energy is added to the steam generator during the transient.

The auxiliary burner overdrive gives a considerable improvement in the response time of the RCG-cycle. It can be seen in figure 5.20 that the auxiliary burner is an excellent way to shorten the response time of the "RCG gas generator", without compromising the maximum allowable temperature of the hot gas mixture (TIT) that is supplied to the expansion turbine. However, with auxiliary
5.9. ADDING STEAM TEMPERATURE LIMITING

5.9.1 Afterspray

The afterspray injects and vaporises water directly into the steam line between the steam generator and the steam temperature (see figure 5.22). In reality the afterspray unit will need to have a certain volume to ensure the residence time that is needed to evaporate all injected water. However, in the model it is represented by a point in the steam line where the water is injected and evaporated instantly. Furthermore, the afterspray unit is powered by a positive displacement pump with PI-controller. When the steam temperature becomes higher than the maximum allowable temperature level for the steam turbine, the steam generator is much more disturbed and the steam temperature rises even more than during the transient without auxiliary burner overdrive. This means that a form of steam temperature limiting needs to be implemented, to ensure that the steam generator is controlled better and that the maximum steam turbine inlet temperature is not exceeded.

Figure 5.20: Transient of combustion chamber power from 250kW to 500kW with exhaust gas temperature limit 1073K and auxiliary burner overdrive with 50K margin
the PI controller detects this and injects water to compensate for this. This results in a larger steam mass flow with a lower temperature. Figure 5.23 shows the transient simulation of auxiliary burner overdrive combined with afterspray. The feedwater control is set for a boiler outlet steam temperature ($T_{H2O4}$) of 808K instead of 798K, while the afterspray is set for a steam turbine inlet temperature ($T_{steam\,turbine}$) of 798K. This means that the afterspray is continuously injecting a small waterflow between the steam generator exit and steam turbine inlet, to cool down the steam 10 degrees. This setting is chosen because simulations showed that the after spray takes some time to become stable when it starts injecting from zero-flow. This is caused by the model itself. However, a similar effect will also occur in reality; if the afterspray nozzle has to start spraying from zero-flow, it will take time before pressure builds up and a small droplet size is realised. Therefore it is expected that in reality it will also be beneficiary to have a continuous minimal afterspray flow, so that the afterspray can quickly kick in.

In figure 5.23 it can be seen that the system is responding the almost the same as with just auxiliary burner overdrive. Except now, the steam turbine inlet temperature is kept almost continuously stable at 798K by the afterspray, while
the steam generator outlet temperature $T_{H2O_4}$ still rises up to almost 850K during the transient. Figure 5.24 shows the response of the steam generator to the transient with both overdrive and afterspray. The response of the evaporator length, steam pressure and steam flow is much calmer than with only overdrive (figure 5.21). The reason for this is that the afterspray injects water before the nozzle of the steam turbine. The condition the steam generator depends on the steam turbine nozzle conditions: steam pressure, flow and temperature. If the temperature of the steam entering the nozzle is kept constant, this has a tempering effect on the steam pressure and steam flow. The results show that afterspray is a good way to regulate the steam temperature and the steam generator as a whole during a transient with auxiliary burner overdrive. It has to be noted that the afterspray is modeled such that all injected water will evaporate instantly. In reality this will require a certain residence time. The requirements to the design of an afterspray unit and practical feasibility will be treated in chapter 6.

### 5.9.2 Aircooling

The aircooler limits the steam temperature by lowering the temperature of the exhaust gasses that enter the steam generator. A lower gas temperature will result in a lower temperature of the steam exiting the steam generator ($T_{H2O_4}$).
The exhaust gas temperature is lowered by mixing it with ambient air, thus resulting in a larger exhaust gas flow with a lower temperature. The ambient air is added to the exhaust gas stream with a blower and is mixed with the exhaust gas flow of the auxiliary burner at mixing point M (figure 5.25). The aircooler is modeled as a relay that switches the blower on or off. The aircooler is switched on when the steam temperature is higher than 820K, the aircooler is switched off when the steam temperature is lower than 815K. This margin of 5 degrees is necessary to prevent the aircooler from continuously switching on and off. The feedwater control is still set for a steam temperature of 798K. The aircooler cannot be set for 798K because then the feedwater control would no longer respond during a transient. The aircooler would then realise a steam temperature of around 798K, and the feedwater control will not elevate the feedwater flow, because it only responds to an elevated steam temperature. Since the feedwater control would then not take action, the aircooler would remain operational until the power was in the combustion chamber was reduced again to the original level. This is undesirable because the aircooler limits the steam temperature by reducing the transferred heat to the steam; the temperature difference between the exhaust gas and the steam is lowered. This reduces the overall efficiency and should therefore not last longer than strictly necessary.
With afterspray this situation can be prevented by sending the signal of the steam temperature in front of the afterspray to the feedwater control. With aircooling however, such a measure is not possible because once the aircooling is operational, the elevated steam temperature does not occur at any point in or after the steam generator. The only way is to allow the steam temperature to rise enough so that the feedwater control will increase the feedwater flow. Then the aircooler can prevent the steam temperature from rising to a dangerous level on the short term, while the feedwater control is controlling the steam temperature for the long term. Figure 5.26 shows the results of a transient simulation with these settings. The same as in the experiment with afterspray, the simulation starts at 400 seconds, the requested combustion chamber power is elevated from 250kW to 500kW and the auxiliary burner overdrive activates the auxiliary burner. The $T_{\text{exhaustgas}}$ that is shown in figure 5.26 is the temperature signal that is used for the auxiliary burner control; the temperature after the combustion chamber and before the auxiliary burner and aircooler. At around 630 seconds, the airflow is increased enough for the combustor to realise the demanded power of 500kW. The demanded power of 500kW is reached at 630s and at 760s the auxiliary burner is switched off, because the exhaust gas temperature drops below the 50K margin. Figure 5.26 shows that during the transient the steam temperature rises to around 820K and then gradually drops.
to 815K, which was to be expected. Between 700s and 800s the steam temperature shows some irregularities. The steam generator response (figure 5.27) shows what causes this. It can be seen that around 700s the aircooler switches off because the 815K boundary has been reached, and then on again because the steam temperature rises back to 820K again. This is repeated two more times between 700s and 800s, causing fluctuations in steam temperature, steam pressure and steam flow. At around 760s the auxiliary burner is switched off which is also responsible for fluctuations in the steam generator. However, the steam generator fluctuations have no significant effect on the compressor speed and as a result there are no oscillations in the combustion chamber airflow and the combustion chamber power.

The response of the system with aircooling takes around a minute longer than that of the system with afterspray. This can be explained from the fact that the aircooler limits the steam temperature by reducing the transferred heat to the steam, thus reducing the product of steam mass flow and steam enthalpy. Afterspray keeps the product of steam mass flow and steam enthalpy constant by injecting water that is all converted into steam. Therefore with aircooling, less energy is fed to the steam turbine during the transient than with afterspray.
This results in a smaller compressor acceleration and a longer transient with aircooling. From this point of view, afterspray is the preferred way of steam temperature limiting. However, also constructional considerations have to be weighed. This will be done in chapter 7.

5.10 Conclusions

The behavior of the set-up and the model are similar, and the response delays are of the same order of magnitude. Therefore, the results and insights that were obtained in this chapter with the transient model will be valid for the operating strategy of a real-scale RCG. However, the model could not yet be validated under the same boundary conditions. Therefore, the operating strategy insights will be implemented in the demo RCG, but there will need to be a lot of attention to the fine-tuning of the parameters. Once a real-scale RCG is operational, the transient model can be validated under equal boundary conditions. With these validation results the model could be fine tuned to make it suitable for more accurate transient predictions of real-scale RCG installations.

The auxiliary burner overdrive improves the transient behaviour of the RCG
cycle considerably; it reduces the response time from 450 seconds to 200 seconds, without compromising the maximum allowable temperature of the hot gas mixture (TIT) that is supplied to the expansion turbine. However, with auxiliary burner overdrive the steam temperature rises to dangerous levels. Therefore a form of steam temperature limiting needs to be implemented: either afterspray or aircooling will ensure that the maximum steam turbine inlet temperature is not exceeded. With aircooling the response time increases again to 250 seconds, with afterspray it remains about 200 seconds. However, also constructional considerations have to be weighed.

Figure 5.27: Steam generator response to transient of combustion chamber power from 250kW to 500kW with aircooling
Chapter 6

Start-up experiments

In chapter 5 the operating strategy for transients of the RCG was optimized with a transient model. The steam generator proved to be the dominant component in the transient behaviour of the RCG.

The transient model cannot simulate a start-up from cold of the RCG. This is mainly caused by the model part of the steam generator; to converge, the model needs a fully developed economiser, evaporator and superheater section in the steam tube. At the beginning of a cold-start, the entire steam tube will be filled with a certain amount of cold water. The water in the steam tube will be heated to boiling temperature. Then the water will start to evaporate and there will be a small steam flow at the steam generator exit. At a certain moment this will evolve into a fully developed economiser and evaporator. And, eventually, also a superheater will appear at the end of the steam tube. It is chosen to develop an operating strategy for starting the RCG with the experimental set-up of chapter 5.

6.1 Steam generator start-up experiments

The start-up procedure is divided into two steps. The first step is starting the RCG-gas generator (the Rankine cycle driven compressor) by firing the auxiliary burner. The second step is starting the gas turbine cycle by firing the combustion chamber. After starting the RCG gas generator, it is not necessary to start the gas turbine cycle if shaft power is not (yet) required. When the gas generator is running, the combustion chamber can be started at any given time when power from the free power turbine is demanded for driving a load. In this section, the first step, starting the RCG gas generator is treated.

In principle the gasgenerator is started by firing the auxiliary burner, but first the gasgenerator has to be made ready for operation. This means starting the oil pumps of the steam turbine and turbo compressor. This way their bear-
ings are provided with oil. Also the steam turbines hydraulic control system is powered up, thus opening the steam turbines steam inlet valve. The centrifugal pump in the feedwater circuit has to be started to provide the feedwater pump with feedwater of 1 bar and to circulate water through the condensor. During start-up of the Rankine cycle, the feedwater pump should provide the once-through boiler with a feedwater flow that corresponds to a steam temperature of about 400°C at the given auxiliary burner power. This gives a safety margin of 70°C since the maximum allowable steam temperature of the steam turbine is 470°C. When a cold-start is performed, first the steel of the tube of the steam generator and the water in the tubes have to be heated from room temperature to a temperature that corresponds to a stable working point. Figure 6.1 shows

![Graph showing steam generator signals](image)

Figure 6.1: Steam generator signals during a cold-start experiment of the gas generator

the relevant steam generator signals of a "cold-start". For the signal of the feedwater pump setting it has to be noted that when it is not operational, the setting is still logged. Unless mentioned otherwise, the feedwater pump is always switched on, at the same time the auxiliary burner is switched on. At t=40s the auxiliary burner is fired. The feedwater pump is set to supply a feedwater flow to the steam generator that should result in a steam temperature of 400°C.
The temperature of the hot gas flow at the inlet of the steam generator rises quickly to 800-900°C. It takes some time before the water starts to heat up, at 140s the water in the steam generator is heated up from room temperature to 100°C, the boiling temperature at 1 bar(abs).

Once the water is at boiling temperature it stays at boiling temperature until t=330s. All this time the pressure remains 1 bar(abs), this means that the steam flow through the steam turbine nozzle is negligible. Because the steam generator is producing superheated steam, it is assumed that there now exists a superheater section at the end of the steam tube (see figure 4.3). At around t=700s the steam pressure starts to elevate, which means that the steam generator starts to produce a relevant steam flow. Because the steam temperature still continues to increase, the feedwater flow is elevated at around 750s and again at around 800s. The feedwater flow increases are futile and at 830s the steam temperature reaches the dangerous level of 500°C, therefore the auxiliary burner is switched off. However, the steam temperature continues to increase and at 900s the steam temperature reaches 550°C, which is the maximum range of the steam temperature sensor. Therefore between 900s and 960s the steam temperature is logged as 550°C, while in reality it is higher. From interpolation of the steam temperature curve between t=900s and t=960s it follows that the steam temperature increased to around 600°C before it started to decrease again. At around t=980s the steam temperature drops very quickly to the saturated level.

This start-up experiment shows that during start-up the steam temperature cannot be limited by supplying feedwater. The reason for this is that the hot gasses of the auxiliary burner with a temperature of 850°C are supplied to the upper steam tube layers of the steam generator. During start-up there is not yet a fully developed flow and section distribution in the steam tube like in figure 4.3. The hot gasses heat up the stainless steel mass of the steam tube at the top of the steam generator. Then the little remaining water in the tube is heated. This water is standing still and is little compared to the auxiliary burner power. Therefore, the water is heated up to a temperature near to the hot gas temperature. It is futile to supply feedwater at the steam generator tube inlet, since there is not yet a developed evaporator section this will not result in a higher steam flow at the end of the steam tube. From 700s and on, an evaporator section is developing, but does not develop quick enough to supply enough steam flow to the superheater section to prevent a dangerous steam temperature level.
6.2 Steam temperature limiting during start-up

6.2.1 Burner interval

Dangerous steam temperature levels during start-up could be prevented by lowering the hot gas temperature. Since it is not possible to lower the temperature of the gas flow of the auxiliary burner, the burner is switched on and then switched off again, to allow the steam temperatures to even out through the steam tube. Figure 6.2 shows such an experiment. The auxiliary burner is fired and the hot gasses quickly go to a temperature of 850°C. The water in the steam tube heats up and starts to boil. Once a superheater section starts to form at around t=450s, the auxiliary burner is switched off, and is switched on again after 100 seconds. The steam temperature at the steam generator outlet continues to rise until it reaches 350°C. Then at around 700s the steam temperature decreases to 160°C, while the boiler pressure drops from 5 bar to 4 bar. Now all the sections of the steam generator are formed, there is a relevant steam flow and the steam temperature is at a safe level. The steam temperature increases slowly towards its original level. From t=4000s until t=4500s the feedwater flow is decreased a little to speed up the formation of a substantial superheater that

![Figure 6.2: Steam generator signals during a cold-start experiment of the gas generator with intermitting auxiliary burner power](image-url)
6.2. STEAM TEMPERATURE LIMITING DURING START-UP

results in a steam temperature of about 350°C. After this was successful, the auxiliary burner is shut down at \( t=4600 \) s and the experiment is ended.

The results show that starting with a burner interval can be considered a safe start-up procedure. Unfortunately, this start-up procedure causes an initial peak and drop of the steam temperature, before it results in a stable working point without the occurrence of dangerous steam temperature levels.

6.2.2 Forced cooling

The start-up problem of the steam generator could be solved more elegantly if either the gas temperature could be limited to 600°C (little more than the maximum allowable steam temperature) or if the steam could be cooled in between the boiler steam outlet and the steam turbine inlet. This can be done by air cooling or after spray respectively (Section 5.9). Appendix C.1 and C.2 show that, from a constructional point of view, the air cooler has the advantage. The after spray unit will consist of a long pipeline or large pressure vessel to allow the water droplets to evaporate, which is expected to cause problems during a cold start-up. The air cooler will consist of an air fan and a gas mixing volume. The advantage for the air cooler is that the mixing volume contains gasses of atmospheric pressure, and is therefore easier to construct. During a cold start the mixing volume will also need time to warm up, but it will cause no steam condensation problems since it is only in contact with the hot gasses. In chapter 5 transient simulations showed that after spray improves transient behavior far more than the air cooler. However, these simulations are related to transients of a fully warmed up and stable working installation. There was not simulated a cold start-up, simply because the transient model cannot simulate a cold start-up. Taking all of this into consideration, it was finally chosen to let constructional simplicity and robustness prevail; the air cooler was selected as the steam temperature limiting device of choice for the RCG. An air cooler was fitted on the advanced set-up in accordance with the design of section C.2.

Figure 6.3 shows the relevant signals of the steam generator during a cold-start with the air cooler set to 500°C. At \( t=200 \) s the auxiliary burner is started, as can be seen from the quick temperature rise from room temperature to 500°C. At \( t=300 \) s the water in the steam generator is at atmospheric boiling point; 100°C. Around \( t=500 \) s a superheater starts to form and the steam temperature starts to increase. Before the air cooler was installed, the steam temperature would increase up to a dangerous level, unless the auxiliary burner was switched off. Now the auxiliary burner remains switched on and the air cooler remains set at 500°C. Therefore, the steam temperature will not be able to reach levels above 500°C, which proves to be true; at \( t=1500 \) s the steam temperature stops increasing at a value of around 480°C, and then remains stable. This proves that the air cooler is working excellent. Experiments without the air cooler showed that once there is a relevant steam flow, steam temperature limiting is no longer necessary. From around \( t=1500 \) s the steam pressure has build up,
and therefore also the steam flow. The air cooler is then set to a higher hot gas temperature, in two steps it is elevated to the maximum temperature 900°C. The feedwater flow is set such that it keeps the steam temperature at around 470°C. It can be seen that by feedwater control the steam temperature can now be kept stable. This is because now the steam generator sections (economiser, evaporator and superheater) are fully developed.

6.3 Conclusions

From the start-up experiments it can be concluded that an RCG with auxiliary burner and aircooling can be started quickly (about 10 minutes) while maintaining a safe steam temperature. An effective start-up strategy was developed and tested. During start-up the aircooler ensures a safe steam temperature. Once the steam generator has a fully developed economiser, evaporator and superheater section, the aircooler can be switched off and the feedwater control can ensure a safe and stable steam temperature. These design choices and start-up strategy are adopted for the RCG demo system.
Chapter 7

Design of a 1MWe pilot installation

7.1 Field of application and demands

The RCG is intended for mechanical drives, decentralized CHP and ship propulsion. The aim of any demo-installation is to prove that it is fit for the market that it was intended for, thus leading to follow-up sales in that particular market. Therefore, it was recognized that the demo RCG should be realized in one of the three intended fields of application. Furthermore, the demo RCG should be realized in a market that is:

- market volume: to ensure profitable follow-up sales
- easy adopting: a market that is willing and daring to try new technologies
- geographically nearby: to ensure 24/7 trouble-shooting at the demo site

Taking all this into account, it was decided to aim for an RCG demo project in the Dutch CHP-market. This market comprises greenhouses and factories that consume a lot of heat in the form of hot water or steam. By generating electricity and utilising the waste-heat, they can realise both energy savings and energy cost reduction. This market is obviously nearby, and very appealing with a sales volume of two hundred conventional CHP-units per year, in the shaft power range of 1-4MWe. By interviewing several potential pilot customers, it was investigated what the demands are to the design of the pilot system:

- electrical power 1000kWe
- CHP application
- dual fuel: natural gas and crude liquid bio-fuel: glycerol
- availability of 6000 hrs/year
- payback time <4 years
7.2 Preliminary skid design

First, a preliminary skid design will be made according to the demands of the pilot customer. From the electrical power demand of 1000kWe, it follows that the skid will incorporate a power turbine that delivers 1000kW plus generator losses. Assuming a typical generator efficiency of 96% results in a required shaft power of about 1040kW. From the thermodynamics (see chapter 3) it follows that the shaft power of the impulse steam turbine in the RCG cycle is typically about half the shaft power of the power turbine. Therefore, for first estimates of the required turbo machinery, the aim will be a power turbine of 1040kW and a steam turbine of 520kW shaft power driving the turbo compressor of the RCG cycle.

A power of 520kW is in the lower region of the shaft power range of an impulse steam turbine, which is available from about 50kW up to 6MW. Because the steam turbine has a relatively low shaft power, it is to be expected that it will not be available with an isentropic efficiency of 80% (as was assumed in chapter 3). It will more likely be in the range of 50-65%. The maximum allowable steam temperature will be 530°C.

The steam generator will be of the same design as the one in the prototype, so it will be a subcritical once-through steam generator that produces superheated steam at a pressure of 35 bar.

The turbo compressor and the power turbine will be of the centrifugal and radial type respectively, therefore they are both expected to have an isentropic efficiency of around 80%. Radial turbines are typically uncooled, therefore, a maximum TIT of around 980°C (1253K) is assumed. To select the turbo compressor, the power turbine and the steam turbine, a preliminary calculation (first iteration) of the required gas flow and steam flow has to be made. Besides the properties of figure 7.1, also the compressor pressure ratio is needed for such a preliminary calculation. From figure 3.2 it follows that the typical pressure ratio of the gas turbine cycle at a TIT of 1300K is around 4. However, this followed from an assumed compressor efficiency of 0.80, a power turbine efficiency of 0.85 and a steam turbine efficiency of 0.80. For an 8MW shaft power RCG this could be valid, but for a 1MW shaft power RCG the values of figure 7.1 are considered to be much more realistic. Lower isentropic efficiencies of the turbo machinery will result in a lower pressure ratio, therefore the pressure ratio is estimated at a value of 3. Then the required mass flow of the power turbine can be determined by using equation (3.3):

\[ P_{\text{powerturbine}} = 1040 \cdot 10^3 = \dot{m}_g \cdot 1.148 \cdot 10^3 \cdot 0.80 \cdot 1253 \left(1 - \left(\frac{1}{3}\right)^{0.25}\right) \]  

(7.1)

Which gives \( \dot{m}_g = 3.8 \) kg/s. The power consumed by the compressor can then
be determined from equation (3.2):

\[ P_{\text{compressor}} = 3.8 \cdot 1.005 \cdot \frac{288}{0.80} \left(3^{0.2857} - 1\right) = 507\text{kW} \] (7.2)

Since the compressor is driven by the steam turbine, the required steam turbine should deliver about 510kW shaft power. With the assumptions of figure 7.1 and the results of the calculations of the shaft powers of the turbine, compressor and steam turbine it is possible to determine whether the pressure ratio of 3 could actually be achieved. Assuming a pressure ratio of 3 does not fit exactly, one could iterate until the exact pressure ratio is found. However, the pressure that would follow is only valid for the assumed turbo machinery efficiencies, while these efficiencies will in fact follow from the components that are commercially available. Therefore, it is first investigated which turbo-machinery components are commercially available for the shaft powers with the calculated order of magnitude.

### 7.3 Skid components

#### Power turbine

With a gas flow of 3.8 kg/s, a pressure ratio of 3 and a TIT of 980°C, the power turbine should deliver 1042kW. Unfortunately, it is not possible to select the expansion turbine of an existing simple cycle gas turbine to this purpose. Gas turbines are not modular built and are therefore only available as a complete package. It is not possible to acquire just an expansion turbine, without a
compressor. However, it is possible to acquire the power turbine of a two-shaft simple cycle gas turbine, since it is on a separate shaft with bearings. Power turbines have a typical design pressure ratio of about 3, which makes them applicable in the RCG. However, such a power turbine will have a relatively low allowable maximum TIT because it is designed to be the second stage of the expansion of the gas turbine. Alternatively, a custom made high-end power turbine could be acquired, the costs of which are estimated at around 300k euro, depending on sales volume.

Contrary to gas turbines, turbochargers are constructed in a modular way. The centrifugal compressor and radial turbine can be selected independently, and assembled according to the design parameters of the internal combustion engine that it should turbo charge. This means that it is also possible to select a turbine assembled onto a bearing but without a compressor. Turbocharger turbines for a gas flow of 3.8 kg/s are available, but they cannot handle a TIT of 980°C. This is because they are designed for large internal combustion engines that have relatively low temperature exhaust gasses because they run on a lean mixture and high pressure ratio. The temperatures of the exhaust gasses of industrial gas engines, that run on a stoichiometric fuel mixture, can go up to 1000°C. Therefore, there are turbochargers available for this type of engine that have a maximum allowable TIT of the same temperature. The largest available turbine of this type is the K44 of BorgWarner (Germany). It has a gas flow of about 1 kg/s at $r = 3$ (see figure 7.2) which does not suffice.

Figure 7.2: Characteristics of the Borgwarner K44 turbine at 70,000RPM

However, four of these turbines in parallel would deliver the required power. The modular build-up of the RCG-layout allows the power turbine to consist of multiple turbines in parallel. Since the K44 turbine costs about 4.5kEuro, including turbine housing and bearing, it is very appealing to implement four
K44 turbines in parallel for 18kEuro as opposed to about 300kEuro for one large high-end power turbine. First, because the investment costs of the power turbine section can then be very low. Second, because the pilot-RCG installation is to be operated on crude bio-fuels. One of the most economically appealing crude bio-fuels is crude glycerol: the by-product of bio-diesel production. As a fuel, crude glycerol is technically very challenging, since it contains potassium salts. It is to be expected that the power turbine section is going to take severe wear from the potassium salts in the exhaust gas stream. If the power turbine section can be replaced with a short time interval, combined with low replacement costs, this will be a very interesting maintenance concept. Therefore, the power turbine section will be designed with four K44 turbines in parallel.

All four parallel turbines of the power turbine section should each drive their own generator or should all drive one large generator with a four-in-one-out gearbox. At a mass flow of 1\text{kg/s} and \( r = 3 \), a speed of 70,000 RPM is required to run at the maximum obtainable isentropic efficiency of 0.76 (see figure 7.2). Conventional generators of about 250kWe run at 1500RPM or 3000RPM, so this would result in four gearboxes with a speed ratio of at least 70 to 3, which is not realistic. High-speed generators of 250kWe that can run up to 30,000RPM are available at investment costs of about 200kEuro which is economically unattractive. Generators of about 1000kWe typically run at 1500RPM, therefore the above mentioned four-in-one-out gearbox should have a 70 to 1.5 speed ratio. Although the speed ratio is very high, there is only one gearbox needed, that will incorporate one large wheel, driven by all four turbines (see figure 7.4). This gearbox is available for 100kEuro, which is economically appealing. For the power turbine section, four K44 turbines are chosen, driving one conventional 1500RPM generator (investment costs: 20kEuro) through a four-in-one-out gearbox.
The most important demand to the combustion chamber section is that it should be capable of burning both natural gas and glycerol. As is shown in appendix D, the dual fuel combustor that is developed for the RCG is capable of burning both natural gas and glycerol, while fulfilling exhaust gas legislation requirements. Experiments with the combustor showed that it rates a thermal power of up to 1000 kW. The power of the combustion chamber section of the pilot-RCG is 4500 kW. Therefore, four parallel combustors of the same design as the prototype dual-fuel combustor in parallel suffice. Since the power turbine section consist of four parallel expansion turbines, this would match perfectly with four parallel combustion chambers. It will then be unnecessary to create some form of a collector between the combustion chamber section and the power turbine section, but can simply consist of four parallel sections that do not interfere with one another. It is preferred to incorporate this principle also with the compressor section, which would result in four parallel and non-interfering compression-combustion-expansion processes.

Turbo compressor

Like the power turbine section, the turbo compressor section could either consist of one large, or of several smaller turbo compressors. In the previous paragraph it was stated that 4 parallel turbo compressors would be preferred because this would match perfectly with the four parallel combustion chambers and 4 expansion turbines. The specifications for the pilot-RCG turbo compressor section are that it should be capable of an airflow of 3.8 kg/s and a compression ratio $r = 3$. Industrial turbo compressors that match these specifications are available at an
For the design option with a turbo compressor section that consists of 4 parallel turbo compressors, the specification of these turbo compressors would be an airflow of $1.0\text{kg}/s$ at a compression ratio of $r = 3$. That kind of capacity is too small for high-efficiency industrial turbo compressors, but is available in the automotive industry. In the automotive market these type of turbo compressors are employed as superchargers of internal combustion engines, both for road and race applications. Figure 7.5 and 7.6 show the picture and compressor map of an automotive supercharger that could deliver the demanded airflow of about $1.0\text{kg}/s$. This turbo compressor is available for about 2kEuro investment costs. The compressor map of the Vortech V4XX shows that a very high compressor speed of about 65,000RPM is needed to obtain the desired pressure ratio of about three. According to the specifications of the manufacturer, 65,000RPM is the maximum allowable speed of this turbo compressor, so this compressor could just be operated at the required airflow and pressure ratio. The isentropic efficiency at this working point will be about $\eta_c = 0.65$, which is considerably lower than that of the industrial turbo compressor. On the other hand the investment costs of 4 automotive compressors will amount to about 10kEuro, versus 200kEuro investment costs for the industrial turbo compressor. Since the compressor map of automotive turbo compressor shows that isentropic efficiencies of $\eta_c = 0.74 - 0.78$ are possible at lower pressure ratios, it is appealing to employ this turbo compressor in a two-stage configuration, perhaps with intercooling. When applying the Vortech V4XX in two-stage configuration both
stages should be matched in such a way that they both rate an efficiency that is in the range of \( \eta_c = 0.74 - 0.78 \). This is possible by running the low pressure stage at about \( r = 2.1 \) and the high pressure stage at about \( r = 1.45 \). However, two-stage compression with intercooling makes the system more complicated. It is decided to employ four parallel Vortech compressors in single-stage compression in the pilot-system. In follow-up systems two-stage compression with intercooling will be considered as an upgrade.

Steam turbine

The steam turbine section drives the compressor section of the RCG. The steam turbine should deliver about 600kW. Because impulse steam turbines of about 600kW shaft power are typically applied for driving generators and pumps, they are normally designed to have an output shaft that runs at 1500 or 1800RPM. The steam turbine wheel runs typically at 10,000 up to 20,000RPM, this speed is reduced with an integral gearbox. Such a standard impulse steam turbine with integral gearbox is available for about 225kEuro. The compressor section that should be driven by the steam turbine runs at a speed which is much higher than 1500 or 1800RPM. Therefore the steam turbine manufacturer Siemens-KKK was requested to offer a steam turbine without integral gearbox. When purchasing only one turbine this results in higher investment costs because the entire turbine housing needs to be altered.

The AFA4 steam turbine that was offered by Siemens-KKK has a steam consumption of 4500kg/hr at 600kW, this corresponds to an isentropic steam tur-
bine efficiency of $\eta_{st} = 0.58$. This moderate steam turbine efficiency results in a steam temperature of about $275^\circ C$ at the steam turbine exit, even though the steam expands to 1.05 bar(abs). It would of course benefit the efficiency of the demo-RCG if the energy in the exhaust steam could be recuperated before the steam is condensed to water in the condenser. The temperature of the compressed air leaving the turbo compressor section is about $200^\circ C$, therefore it cannot recuperate any energy of the exhaust steam flow.

**Recuperator**

If intercooling would be applied in between the two stages of compression this will reduce the temperature of the compressed air leaving the turbo compressor. This will make it possible to recuperate some of the waste-heat in the exhaust steam (see figure 7.7). Also, with intercooling a higher pressure ratio can be obtained from the same steam turbine shaft power, which has a positive effect on the efficiency of the RCG, but only if combined with recuperation of the steam waste heat. This will not be employed in the pilot system, but will be considered in follow up systems.

**Condenser**

The purpose of the condenser is to condensate the steam, coming from the recuperator, into water so that this water can be pumped into the boiler again, thus making the Rankine cycle complete. This condenser should operate at 1.05 bar(abs). Furthermore the condenser is the component which delivers the

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Figure 7.7: Schematic of the RCG-cycle with recuperation
heat of the combined heat and power installation; the condenser should heat the water of the buffer tank of the greenhouse central heating system by condensing the steam. Therefore, the condenser will be in fact a water cooled condenser, that receives its "cooling water" from the buffer tank. In case of heat utilisation in a factory, the steam will be led through the plant before going to the condenser.

**Feedwater reservoir**

The feedwater reservoir should ensure feedwater supply to the steam generator and should be able to take in all water that comes from the steam condensor, during start-up, transients and constant full-load. During full-load operation the steam generator holds typically 50% of its volume in feedwater (see figure 4.3). The remaining volume is filled with saturated and superheated steam that has a negligible mass compared to the water and saturated water in the steam generator tubes. The feedwater reservoir of the prototype was designed to be able to hold as much feedwater as the entire volume of the steam generator tubes (90 liters). This means that it is possible to flood the steam generator entirely without running out of feedwater. Experiments with the prototype showed that the feedwater reservoir sometimes runs at low level because a large amount of water was also in the path from steam turbine to the condensor and in the condensor itself. Therefore, an extra 30% to the volume of the feedwater reservoir is added. The steam generator of the pilot installation will incorporate 8 steam generators, each identical to that of the prototype, with a total volume of 720 liters. Adding a safety margin of 30% results in 936 liters; therefore, the pilot system is designed with a feedwater reservoir of 1000 liters (1 m$^3$).

**Steam generator**

The steam generator should deliver about 4500 kg/hr superheated steam of 530°C at 35 bar. The start-up and transient behaviour of the RCG is dominated by the steam generator, as was shown in the previous chapters. If the RCG-demo system has to have a short start-up and transient response time, it should simply have a fast responding steam generator. In chapter 6 it was found that the once-through steam generator of the prototype has a start-up time of 15 minutes and a transient response time of 3 minutes, which is quite satisfactory. If this boiler were to be scaled up linearly by enlarging the steam tube diameter and the cross-section of the path of the exhaust gasses, this would result in longer response times, and most likely in a different transient behaviour. To avoid these disadvantages of upscaling of the design, an assembly of multiple units of exactly the same steam generator as in the prototype is applied. This should result in almost the same start-up and transient response time. Furthermore, the steam generator will be equipped with an aircooler to ensure safe steam temperatures during start-up and transients. Because the RCG-pilot system is running on g-phase, glycerol with alkaline soaps, there are alkaline salts present in the exhaust gas stream in the form of small droplets of liquid alkaline salts.
In the RCG-cycle these droplets will remain liquid while expanding in the radial turbines from 1000°C to 800°C. In the steam generator the temperature drops to a level where the alkaline salts will become solid. To prevent clogging, the salt droplets need to be separated before they enter the steam tube bundles. The steam generator is therefore designed to have an impaction separator that separates the alkaline salts right after the turbines in a collector (see figure 7.8).
7.4 Design and off-design point performance of the pilot-RCG

To predict the design and off-design point performance of the pilot-RCG, the design choices of the previous sections were implemented in the off-design performance model of section 3.3. First it is checked for which combinations of combustion chamber power and auxiliary burner power the maximum allowable TIT is not exceeded, and whether the design point electrical power of 1000kWe can be achieved. The maximum allowable TIT of the K44 turbine is around 1250K. Figure 7.9 shows the TIT for the relevant combinations of combustion chamber power and auxiliary burner power. The dotted line shows the working points for which the TIT is equal to 1250K. This means that all valid working points, with a TIT lower than 1250K, lie in the direction of the arrow. Also in the next figures in this section the "1250K-line" and arrow are shown to make it clear which is the valid area of working points. In section 3.3 the shaft power was calculated with the off-design performance model. Figure 7.10 shows the electrical power output for different working points; the known mechanical efficiency of the gearbox $\eta_m = 0.96$ and the electrical efficiency of the generator $\eta_{\text{gen}} = 0.96$ are implemented in the model. The electrical power of the
pilot-RCG is calculated with:

\[ P_{electrical} = \eta_m \cdot \eta_{gen} \cdot P_{powerturbine} \]  (7.3)

In which \( P_{powerturbine} \) is in fact the power generated by the 4 parallel K44 turbines in total.

It can be seen that the output power target of 1000kWe can be achieved, without exceeding the maximum TIT. The electrical power output shows a sudden drop in the area with low auxiliary burner power and high combustion chamber power. This area is already a no-go area where the TIT becomes too high because the airflow is too low for the combustion chamber power. The sudden drop in electrical output power shows that the turbine shaft power drops; this is because the turbo compressor and power turbine run at low efficiency at this low airflow.

The electrical efficiency range of the pilot RCG is shown in figure 7.11. It can be derived with:

\[ \eta_e = \eta_m \cdot \eta_{gen} \cdot \eta_{th} \]  (7.4)

From figure 7.11 it can be seen that the electrical efficiency of the pilot RCG at the design point of 1000kWe is \( \eta_e = 0.22 \). This is quite reasonable for a
1000kWe turbo machinery installation, constructed with commercially available turbo-machinery parts. With dedicated parts, the RCG will be able to perform much better. However, this electrical efficiency of $\eta_e = 0.22$ is well enough to satisfy the demand of 3-4 years payback time, as will be treated in section 7.6. Figure 7.11 shows that at part-load a higher electrical efficiency can be obtained; the maximum electrical efficiency lies at the entire “1250K-line” ($\eta_e = 0.24$). One would expect an even higher efficiency at TIT’s higher than 1250K, however the large drop in efficiency of the turbo machinery components in this working area totally cancels the efficiency increasing effect of a higher TIT. Also at the design working point of 1000kWe the efficiency could be somewhat increased by implementing a higher TIT than that is shown here. It is however believed that a design working well below the maximum TIT will result in a longer lifetime of the power turbines and thus a more reliable installation. Section 3.4 already showed that an RCG can give many combinations of shaft power (in this case electrical power) and thermal power (see figure 7.12). This, of course, also applies for the pilot-RCG. For example the pilot-RCG can be operated on full-load, 1225kW electrical output and and 3400kW heat output from the condensor, but it can also be operated on 80% part-load, 975kW electrical output and 2400kW heat output. Figure 7.13 shows the total efficiency $\eta_{total}$ of all work points, and can
be derived from:

$$\eta_{\text{total}} = \frac{P_{\text{electrical}} + P_{\text{condensor}}}{Q_{\text{fuel}}}$$ (7.5)

The $\eta_{\text{total}}$ is always between 0.84 and 0.86 as can be seen from figure 7.13. The remaining energy (16-14%) is in the exhaust gasses, so if a $\eta_{\text{total}}$ of up to 100% were demanded, then the remaining heat in the exhaust gasses with a stack temperature of about 120°C should be utilized. This could be done with a so called exhaust gas condensor.
7.5 Final skid design

Finally from the previous sections it follows that the RCG pilot skid will incorporate:

- 1 steam generator, consisting of 8 once-through steam generators, each one equal to that of the prototype
- 1 Siemens-KKK AFA4 impulse steam turbine
- 1 Klima watercooled condensor
- 1 feedwater reservoir
- 4 Vortech V4XX turbo compressors, driven by the steam turbine
- 4 EM-group combustors: dual fuel natural gas and glycerol
- 1 EM-group auxiliary burner: dual fuel natural gas and glycerol
- 4 Borgwarner K44 expansion turbines
- 1 Artec 4-in 1-out reducing gearbox
7.5. FINAL SKID DESIGN

- 1 Leroy-Somer 1000kWe generator
- auxiliaries; feedwater pumps, oilpumps and coolers, airfans etc.

Figure 7.14 shows the layout of the RCG pilot skid. The skid consists of two 20ft sea container frames, on top of each other. It is chosen to place the steam turbine in the top frame, and the steam generator in the bottom frame. Also the condensor and recuperative heat exchangers are placed under the steam turbine. This way, it will be easier to keep the steam turbine free of water because the water will always be forced down to the steam generator or the condensor by gravity. The total investment costs of this installation will be about 1.0 mln euro. This is ex works and does not include set-up at the location.

Figure 7.14: General skid layout of the pilot RCG
7.6 Economical discussion

7.6.1 Market

The RCG is intended for decentral combined heat and power (CHP) applications in the range 1-10MW electrical power, and more specific for the range 1-4MW electrical power. Gas engines (CHP), steam turbine cycles (CHP) or regular boilers are currently employed in this sector. When natural gas is available then the gas engine is usually preferred because of its appealing investment costs of 500 euro per installed kW electrical power and high electrical efficiency ($\eta_e = 0.40$). At locations where natural gas is not available, where bio-fuels are desired, or where cheap crude fuels or waste products are to be utilized, one has to choose between a water- or steam boiler and a steam turbine system. A water or steam boiler is relatively economic but can only provide heat or steam and does not produce electricity. A steam turbine system can provide electricity and heat or steam. However, steam turbine systems in the range 1-4MWe offer relatively low electrical efficiency ($\eta_e = 0.10-0.15$) at relatively high investment costs (2000-4000 euro per installed kWe). The RCG is intended to be a third appealing option: in the 1-4MWe range, the RCG will be available for investment costs of 1000-1500 euro per installed kWe and will offer an electrical efficiency of $\eta_e = 0.20 - 0.30$.

7.6.2 Payback time estimation

The RCG is intended for crude liquid (bio)-fuels and waste products, therefore the comparison will be made between a boiler, steam turbine system and RCG-system. We will consider an industrial location where there is a demand of about 3MW thermal power (hot water or steam). A boiler is the basic solution that fulfils this demand. The user then has to decide whether or not to invest more money in a CHP-system that also fulfils the demand for heat or steam and additionally produces electricity. The extra investment for the CHP-system should then be paid back from electricity revenues. It is assumed that the total efficiency for all CHP-systems is $\eta_{total} = 0.80$, $\eta_e = 0.10$ for a steam turbine system and $\eta_e = 0.20$ for an RCG system. This means that the heat demand of 3MW can be fulfilled by a 430kWe steam turbine system or a 1000kWe RCG system. At 2000 euro per installed kWe the steam turbine system will cost about 860k euro, at 1000 euro per installed kWe the RCG will cost about 1000k euro, just like the 1MWe demo system. The yearly electricity revenue is determined with:

$$ e_{revenue/year} = \text{running hours/year} \cdot P_e \cdot \text{revenue/kWhe} $$  \hspace{1cm} (7.6)

With 6000 hours of operation per year and a realistic revenue of 0.065euro per kWhe, this results in a yearly revenue of 167700 euro for the steam turbine system and 390000 euro for the RCG system. The payback time is estimated with:

$$ \text{Payback time} = \frac{\text{investment costs}}{\text{yearly revenue}} $$  \hspace{1cm} (7.7)
The estimated payback time of the steam turbine system is then 5.1 years, while the RCG has an estimated payback time of 2.6 years. However, these payback time calculations really are estimations. The comparison between a steam turbine system and RCG-system is case dependant: different design considerations and choices will differ from case to case, leading to different performances and investment costs. Furthermore, far more aspects have to be considered e.g. maintenance, depreciation etc.

7.6.3 Conclusions

Although the comparison made in the above is not exact, it can be concluded that the 1MW e demo RCG of 1 mln euro as was designed in this chapter, is economically appealing. The pilot system offers a combination of performance and investment costs that are very competing with the conventional steam turbine system. It is therefore believed that if this pilot RCG can prove to be reliable and user-friendly in a demo-project (2010), it has the potential to become commercially successful.
CHAPTER 7. DESIGN OF A 1MWE PILOT INSTALLATION
Chapter 8

Concluding discussion

8.1 Conclusions

The Rankine Compression Gas turbine (RCG) is a new type of combined steam and gas turbine (combined cycle). In this work the feasibility of the RCG is studied, and a real-scale pilot plant is designed.

The novelty of the RCG is that the steam turbine drives the compressor of the gas turbine cycle. Since the expansion turbine of the gas turbine does not drive the compressor, it can deliver all its power to the load. This way, the expansion turbine will function as a free power turbine. Therefore, the RCG is the first combined cycle to deliver all shaft power with one free power turbine. The flexibility that is offered by the RCG will typically be an asset in mechanical drive applications, ship propulsion and decentralized combined heat and power applications. The typical shaft power range of these applications is 1 to 10MW.

The RCG is intended for combined heat and power (CHP) applications and in future also ship propulsion applications. Therefore it will need to be robust and compact. With a thermodynamical model the implications of possible design choices are studied. Because the compressor of the gas turbine cycle is driven by the steam turbine, their shaft powers have to be equal to each other. Thermodynamical equilibrium is reached at pressure ratios of 3 to 4, which fits perfectly for radial turbo machinery components. The results show that with robust radial turbo machinery components, such as a centrifugal turbo compressor, radial expansion turbine and an impulse steam turbine (zero-reaction turbine), the RCG will obtain thermal efficiencies of 30 to 45%. Furthermore it was concluded that the RCG would benefit from the compactness and short start-up time of a subcritical once-through steam generator.

An experimental set-up was realized to experiment with the chosen key components in RCG-layout, and proof of principle was delivered, also for the subcritical
once-through steam generator. However the experiments showed that the control of the steam generator needed special attention.

An improved operating strategy for optimized transient behaviour of the RCG is developed. A transient model is developed and its basic responses are compared to basic responses of the experimental set-up. The behavior and response delays of the set-up and the model are of the same order of magnitude. With the transient model, the effect of several possible actuators is studied and a system design for improved transient response is developed. The results show that the once-through steam generator, although it is fast responding compared to a conventional steam boiler, is still the slowest responding of all RCG components. Therefore, the response time of the steam generator is determining for the RCG response time in general. To make the RCG to go from part-load to full-load as fast as possible, the thermal power (fuel flow) has to be increased as fast as possible. The combustion chamber fuel flow can only be increased up to the point that the maximum turbine inlet temperature is reached. This results in response time of 450 seconds for a transient from 50% part-load to 100% full-load. The strategy where the auxiliary burner is fired during transients (auxiliary burner overdrive) reduces the response time from 450 seconds to 200 seconds, without exceeding the maximum allowable temperature of the hot gas at the inlet of the expansion turbine. However, with auxiliary burner overdrive the maximum steam temperature at the steam turbine inlet is exceeded. Therefore, either afterspray or aircooling (steam temperature limiting) needs to be implemented. The response time increases again to 250 seconds with aircooling, while afterspray does not diminish the effect of auxiliary burner overdrive. Because it is more straight forward from a constructional point of view, aircooling is chosen as the steam temperature limiter of choice for the demo system.

For a pilot project, a 1MW e pilot installation was designed. The installation will be capable of burning both natural gas and glycerol. The pilot was designed with an impulse steam turbine that is driven by steam from a once-through steam generator. The steam turbine drives 4 parallel centrifugal compressors, that supply 4 combustion chambers with compressed air. The combustion chambers are dual fuel (natural gas and glycerol). The 4 combustion chambers supply 4 radial expansion turbines with hot compressed gas, to expand. The radial turbines are fitted on a 4-in-1-out gearbox that drives one generator. The heat can be supplied in the form of steam from the steam turbine exit or in the form of hot water from the water cooled condensor. The entire installation is fitted in two 20ft containers on top of each other with sound insulation. The glycerol...
is in fact a mixture of glycerol and alkaline-soaps that remain from the biodiesel production process. This results in a hot gas stream that contains potassium carbonate particles. These particles will cause wear in the expansion turbines, however because they are turbocharger based they can be replaced on a regular bases at relatively low costs. The particles are separated in the steam generator, partly in a collector before they encounter the steam tubes and partly by washing them from the steam tubes.

8.2 Recommendations

The model could not yet be validated under the same boundary conditions of a real-scale installation. Therefore, the operating strategy insights that are implemented in the demo RCG, will need validation of the parameters. Once a real-scale RCG is operational, the transient model can be validated under equal boundary conditions. With these validation results the model could be fine tuned to make it suitable for more accurate transient predictions of real-scale installations.

Although the once-through steam generator has a much shorter start-up and response time than a conventional boiler, it is still the slowest component in a transient. Therefore, efforts to optimize the transient behaviour of the RCG, should focus on the steam generator. In the current set-up it is only possible to measure the water and steam conditions at the in- and outlet of the steam generator. It is recommended that a set-up is realized that can measure and/or visualize the movement and condition of the evaporator section. This way the transient model can be further validated and there can be gained many insights on how to control the steam generator better, and moreover, how to control and operate it faster.
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Appendix A

Design of a 500kW subcritical once-through steam generator

A.1 Thermodynamics

The experimental set-up has a subcritical once-through steam generator. Table A.1 shows the design criteria that the steam generator should fulfil. Basically, a sub-critical once-through boiler is a tube where feedwater enters on the one end, which is then converted into superheated steam and comes out on the other end. The exhaust gas is in counter flow heat exchange contact with the tube. Figure A.1 shows the working principle and relevant properties of the subcritical once-through boiler.

<table>
<thead>
<tr>
<th>Steam generator assumptions</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Feedwater temperature</td>
<td>80°C</td>
</tr>
<tr>
<td>Boiling pressure</td>
<td>30 bar (233.9°C)</td>
</tr>
<tr>
<td>Superheated steam temperature</td>
<td>400°C</td>
</tr>
<tr>
<td>Water/steam mass flow</td>
<td>0.124 kg/s</td>
</tr>
<tr>
<td>Exhaust gas temperature</td>
<td>700 °C</td>
</tr>
<tr>
<td>Exhaust gas mass flow</td>
<td>0.6 kg/s</td>
</tr>
<tr>
<td>Stack temperature</td>
<td>180 °C</td>
</tr>
</tbody>
</table>

Table A.1: Design criteria of the steam generator

With the power that is needed to maintain the economiser, evaporator and superheater section, the temperature distribution of the exhaust gas is determined:
Economiser

\[ q_{EC} = \dot{m}_{H2O} c_{p_{water}} (T_{H2O_2} - T_{H2O_1}) = \dot{m}_{exgas} c_{p}(T_{g,2} - T_{g,1}) \quad (A.1) \]

\[ q_{EC} = 0.124 \cdot 4.263(233.9 - 80) = 0.6 \cdot 1.148(T_{g,2} - 180) \quad (A.2) \]

For the economiser it then follows that \( q_{EC} = 81.4\,kW \) and \( T_{g,2} = 281.2^\circ C \). This means that the temperature difference at the pinch-point is about 40 degrees, which suffices.

Figure A.1: Principle of the sub-critical once-through steam generator and its relevant properties
A.2. CONSTRUCTION

Evaporator

\[ q_{EV} = \dot{m}_{H2O} h_{fg} = \dot{m}_{exgas} c_p (T_{g,3} - T_{g,2}) \]  
\[ q_{EV} = 0.124 \cdot 1795.7 = 0.6 \cdot 1.148 (T_{g,3} - 281.2) \]  
(A.3)

For the evaporator it then follows that \( q_{EV} = 222.7 \text{kW} \) and \( T_{g,3} = 604.5 \text{oC} \).

Superheater

\[ q_{SH} = \dot{m}_{H2O} (h_{\text{superh.steam}_{400 \text{oC},30 \text{bar}}} - h_{\text{sat.steam}_{30 \text{bar}}}) = \dot{m}_{exgas} c_p (T_{g,4} - T_{g,3}) \]  
\[ q_{SH} = 0.124 \cdot (426.7) = 0.6 \cdot 1.148 (T_{g,4} - 604.5) \]  
(A.5)

For the superheater it then follows that \( q_{EV} = 52.9 \text{kW} \) and \( T_{g,4} = 681.3 \text{oC} \).

A.2 Construction

It is chosen to construct a steam generator with staggered tube layers. Each tube layer consists of a AISI316L stainless steel tube with diameter 21.3mm, wall thickness 2.77mm and length 6m. The tube layers will be in a rectangular casing, connected in series. The water/steam will be inside the tube layers, and the layers themselves will be placed in the hot gas stream, inside a frame with rectangular casing. The tube layers will be installed into the frame in such a way that they can expand freely. Figure A.2 shows the basis of the construction.

From the pressure equipment directive (PED), a safe wall thickness can be determined for the steam tube. The strength calculation according to the

![Figure A.2: Schematic of the once-through boiler in the experimental set-up](image-url)
Figure A.3: 1 tube layer (left) and 4 staggered tube layers (right) of the once-through boiler in the experimental set-up

The minimal wall thickness of the tube that is still safe can be calculated with:

\[
t = \frac{p_{\text{steam}}D_i}{2f - p_{\text{steam}}}
\]  
\[\text{(A.7)}\]

Where \(p_{\text{steam}}\) is the steam pressure (MPa), \(D_i\) is the internal diameter of the tube (mm) and \(f\) is the allowable stress (N/mm\(^2\)) of the tube material. Although the steam generator will be fitted with a pressure safety valve that releases at 35bar, for the strength calculation, still the maximum pressure that can be generated by the feedwater pump, which is 50 bar, is considered. Furthermore, it is expected that, due to the high density and heat transfer of the steam to the tube, the wall of the tube will hardly be higher than the maximum steam temperature of 400°C. To be on the safe side, it is assumed that the tube wall temperature can go up to 600°C. The maximum allowable stress of stainless steel at 600°C and 100k operational hours is 40N/mm\(^2\):

\[
t = \frac{5 \cdot 15.8}{2 \cdot 40} = 1.053 \text{mm}
\]  
\[\text{(A.8)}\]

This means that the selected tube provides for an extra safety factor of 2.63, on top of the legislatory safety factor. A wall thickness of less than 2mm can be difficult to weld properly. It is therefore decided that the selected tube with a wall thickness of 2.77mm is suitable.

### A.3 Heat transfer

The heat transfer of the economiser, evaporator and superheater section is determined with:

\[
q_{\text{section}} = h_{\text{section}}A_{\text{tube}}\Delta T_{\text{lm,section}} = h_{\text{section}}\pi D_{\text{tube}}L_{\text{section}}\Delta T_{\text{lm,section}}
\]  
\[\text{(A.9)}\]
Nusselt equation. To be able to apply the bank of tubes method, the cross-
section of the steam generator is considered to have the same tube length and
tube diameter in the same cross-section area as the actual steam generator lay-
ers (see figure A.3), only, instead of bended tubes, it is calculated with straight
tubes (see figure A.4). Since the temperature of the exhaust gas gradually
changes within each section, also the gas properties change within a section.
The properties are averaged, thus resulting in an average $h$ for each section:

\[
\bar{h}_{\text{EC}} = 114 \text{W/m}^2\text{K}
\]
\[
\bar{h}_{\text{EV}} = 118 \text{W/m}^2\text{K}
\]
\[
\bar{h}_{\text{SH}} = 144 \text{W/m}^2\text{K}
\]

With equation A.9 the required tube length per section can be determined:

\[
L_{\text{EC}} = 151m
\]
\[
L_{\text{EV}} = 179m
\]
\[
L_{\text{SH}} = 16m
\]

Total = 346m
A.4 Final design

Since each tube layer consists of 6m tube, a total of 58 tube layers is required. However, the calculations where based on an "equivalent" tube layer with straight tubes. Also, it is assumed that the tube temperature is the same as the steam inside. In reality the tube temperature will be somewhat higher than the steam temperature, thus resultin in a lower logmean temperature difference. To ensure sufficient heat transfer, it is chosen to implement about 38% extra tube layers: 80 tube layers in total. Figure A.3 shows a 3D drawing and a picture of the steam generator under construction, partially filled with tube layers. The once-through boiler consists of a 480m 316L stainless steel tube (ø21.3mm) inside a 0.5x0.5x2.6[m] casing.

![3D design tube layers in frame and casing (left) and realisation tube layers in frame and casing (right)](image)

Figure A.5: 3D design tube layers in frame and casing (left) and realisation tube layers in frame and casing (right)
Appendix B

Steam generator states

Mass balance

The change of total mass $m$ which is present in the steam tube is dependant of the in- and out going mass:

$$\frac{dm}{dt} = m_{\text{boiler in}} - m_{\text{boiler out}} = \dot{m}_p - \dot{m}_t$$  \hspace{1cm} (B.1)

Where $\dot{m}_p$ is the feedwater pump mass flow and $\dot{m}_t$ is the steam turbine mass flow. Also, the total mass $m$ which is present in the steam tube is dependant of the mass changes of all three sections together:

$$\frac{dm}{dt} = \frac{dm_{EC}}{dt} + \frac{dm_{EV}}{dt} + \frac{dm_{SH}}{dt}$$  \hspace{1cm} (B.2)

The water content of the economiser is assumed to be of constant density:

$$\frac{dm_{EC}}{dt} = \rho_wA \frac{dL_{EC}}{dt}$$  \hspace{1cm} (B.3)

The water and steam mixture in the evaporator has an average density which is time dependant:

$$\frac{dm_{EV}}{dt} = \rho_{EV}A \frac{dL_{EV}}{dt} + A L_{EV} \frac{d\rho_{EV}}{dt}$$  \hspace{1cm} (B.4)

The steam density in the superheater is also time dependant:

$$\frac{dm_{SH}}{dt} = \rho_{SH}A \frac{dL_{SH}}{dt} + A L_{SH} \frac{d\rho_{SH}}{dt}$$  \hspace{1cm} (B.5)

The steam tube has a fixed length and therefore:

$$\frac{dL_{EC}}{dt} + \frac{dL_{EV}}{dt} + \frac{dL_{SH}}{dt} = 0$$  \hspace{1cm} (B.6)

If equations B.1 - B.6 are to be combined into a time dependant relationship for the saturated steam density, then the average densities of the three sections need to be further specified.
APPENDIX B. STEAM GENERATOR STATES

Economiser average steam densities

The density of water is assumed constant.

\[ \rho_{EC} = \rho_w = \text{constant} \]  

(B.7)

And therefore:

\[ \frac{d \rho_{EC}}{dt} = 0 \]  

(B.8)

Evaporator average steam density

In the evaporator water is converted into steam and therefore holds both water and saturated steam.

\[ m_{EV} = m_{\text{sat}} + m_w \]  

(B.9)

The heat flux is assumed to be constant over the entire evaporator length. This results in an equal mass of saturated water and steam.

\[ m_{\text{sat}} = m_w \]  

(B.10)

\[ V_{\text{sat}} \rho_{\text{sat}} = V_w \rho_w \]  

(B.11)

The average steam density is determined with:

\[ \rho_{EV} = \frac{m_{EV}}{V_{EV}} = \frac{V_{\text{sat}} \rho_{\text{sat}} + V_w \rho_w}{V_{\text{sat}} + V_w} \]  

(B.12)

Equation B.11 can be converted to:

\[ \frac{V_w}{V_{\text{sat}}} = \frac{\rho_{\text{sat}}}{\rho_w} \]  

(B.13)

With equation B.13, equation B.12 can be converted to:

\[ \rho_{EV} = \frac{2 \rho_{\text{sat}}}{\rho_{\text{sat}}/\rho_w + 1} = \frac{2 \rho_{\text{sat}} \rho_w}{\rho_{\text{sat}} + \rho_w} \]  

(B.14)

The water density is assumed constant and therefore:

\[ \frac{d}{dt} \rho_{EV} = \frac{\rho_{\text{sat}}}{(\rho_{\text{sat}} + \rho_w)^2} \]  

(B.15)

Superheater average steam density

The superheated steam in the superheater section is considered an ideal gas. For superheated steam this is valid. However for the steam at the beginning of the superheater section this is a simplification because its condition is still close to the saturated condition.

\[ \rho_{SH} = \frac{\int_{T_{\text{sat}}}^{T_{\text{out}}} \frac{\rho_{\text{sat}} T_{\text{sat}}}{T} dT}{T_{\text{out}} - T_{\text{sat}}} = \frac{\rho_{\text{sat}} T_{\text{sat}}}{T_{\text{out}} - T_{\text{sat}}} \int_{T_{\text{sat}}}^{T_{\text{out}}} \frac{1}{T} dT} \]  

(B.16)
\[
\rho_{SH} = \frac{\rho_{sat} T_{sat}}{T_{out} - T_{sat}} \ln(\frac{T_{out}}{T_{sat}}) 
\]

(B.17)

And therefore:

\[
\frac{d}{dt}\rho_{SH} = \frac{\rho_{sat} T_{sat}}{T_{out} - T_{sat}} \ln(\frac{T_{out}}{T_{sat}}) \frac{\rho_{sat}}{T_{sat}} \frac{dT_{out}}{dt} 
\]

(B.18)

Time dependant saturated steam density

Combining equations B.1-B.18 gives:

\[
\frac{d\rho_{sat}}{dt} = -\left( \frac{\dot{m}_{in} - \dot{m}_{out}}{A} + \rho_{w} \frac{dL_{EC}}{dt} + \frac{2 \rho_{sat} \rho_{w}}{\rho_{sat} + \rho_{w}} \frac{dL_{EV}}{dt} - \frac{\rho_{sat} T_{sat}}{T_{out} - T_{sat}} \ln(\frac{T_{out}}{T_{sat}}) \left( \frac{dL_{EC}}{dt} + \frac{dL_{EV}}{dt} \right) \right) 
\]

\[
L_{EV} \left( \frac{2 \rho_{w}^2}{(\rho_{sat} + \rho_{w})^2} + L_{SH} \frac{\rho_{sat} T_{sat}}{T_{out} - T_{sat}} \ln(\frac{T_{out}}{T_{sat}}) \right) 
\]

(B.19)
Appendix C

Steam temperature limiting

C.1 After spray

C.1.1 Demands

Experiments with the RCG-prototype show that during start-up and transients the steam temperature at the outlet of the steam generator can fluctuate severely (see section 6.1), mainly due to the high-temperature of the gas stream of the supplementary burner. The steam temperature can even rise to values as much as 650°C. The maximum allowable temperature of the steam turbine in the prototype is 470°C. Damage to the steam turbine of the RCG-prototype and of the steam turbine of the RCG in general due to high steam temperature should be prevented. This could be done with a device that is called after spray. An after spray unit injects water into the superheated steam at the steam generator outlet. The injected water should then fully evaporate, thus creating an increased steam flow with a decreased temperature. It was chosen to demand to the design of the after spray-unit that it should be capable of regulating the steam temperature within a safe bandwidth during a transient from 50% part-load to 100% full-load. The preliminary experiments with the prototype show that at 50% part-load a steam flow occurs of 0.075 kg/s with a maximum steam temperature of 650°C while at 100% full-load a steam flow of 0.15 kg/s with a maximum temperature of 530°C occurs. The after spray unit of the prototype should be capable of cooling down both the 50% part-load steam flow and 100% full-load from their maximum occurring values to the safe steam temperature of 450°C.

C.1.2 Analysis

It is important that all the injected water is evaporated before it reaches the steam turbine, since it would damage the steam turbine blades. Therefore the evaporation time is limited to the time it takes the steam to travel through the steam line from the after spray unit to the steam turbine:
APPENDIX C. STEAM TEMPERATURE LIMITING

\[ \Delta t_{\text{max}} = \frac{L \rho_s \frac{1}{4} \pi D_i^2}{\dot{m}_s} \]  

(C.1)

With \( L \) the length of the steam line, \( \rho_s \) the density of the steam, \( D \) the diameter of the steam line and \( \dot{m}_s \) the steam mass flow. It is important to determine how long it takes a water droplet to evaporate at the steam conditions that are occurring between the steam generator and steam turbine. Then, at a given steam line geometry, the maximum allowable diameter of the droplets injected by the after spray unit can be determined. For the water droplet evaporation time analysis the following assumptions are made:

- the speed of the water droplets is the same as that of the steam flow
- during evaporation the temperature of the droplet stays equal to the saturated water temperature at the given steam conditions in the steam line
- all heat that is adsorbed by the droplet is consumed by the evaporation
- the water droplets are assumed to be perfect spheres

Evaporation is, in this case, a mass flow from the water droplet to the surrounding steam. This mass flow results from the enthalpy transport from the steam to the droplet. The driver of this enthalpy transport is the temperature difference between the steam and the droplet. The enthalpy transport from the steam to the droplets cools down the steam thus giving the desired result. Therefore the mass flow and enthalpy transport between the droplet and the surrounding steam have to be described for the analysis of the evaporation of the droplets[3]. The heat flow from the steam to the droplet is equal to the enthalpy transport by evaporation from the droplet to the steam:

\[ \dot{Q} = H(t) = h_{fg} \frac{dm}{dt} = h_{fg} \frac{d}{dt} \left( \rho_l \frac{4\pi}{3} r_d^3(t) \right) = 4\pi \rho_l h_{fg} r_d^2(t) \frac{dr_d(t)}{dt} \]  

(C.2)

The heat flow is the result of convective heat transfer from the steam to the droplet:

\[ \dot{Q} = N_u h 2\pi \lambda_s [T_s(t) - T_d] r_d(t) \]  

(C.3)

The evaporation speed is determined by combining equations C.2 and C.3:

\[ \frac{dr_d(t)}{dt} = \frac{N_u h 2\pi \lambda_s [T_s(t) - T_d]}{4\pi \rho_l h_{fg} r_d^2(t)} \]  

(C.4)

\[ \frac{dr_d(t)}{dt} = \frac{N_u h 2\pi \lambda_s [T_s(t) - T_d]}{2\rho_l h_{fg} r_d(t)} \]  

(C.5)

\[ 2r_d(t) \frac{dr_d(t)}{dt} = \frac{N_u h 2\pi \lambda_s [T_s(t) - T_d]}{\rho_l h_{fg}} \]  

(C.6)

\[ \frac{dr_d^2(t)}{dt} = \frac{N_u h 2\pi \lambda_s [T_s(t) - T_d]}{\rho_l h_{fg}} \]  

(C.7)
C.1. AFTER SPRAY

Which describes a time derivative relation for the square of the droplet diameter. From equation (C.7) a time-dependant relation for the droplet radius can be derived:

\[ r_d(t) = \sqrt{r_d^2(t = 0) - \int_0^t \left( \frac{dr_d^2(t)}{dt} \right) dt} \]  
\( \text{(C.8)} \)

The evaporation time depends on the temperature of the surrounding steam as can be seen in equation (C.7), while the temperature of the surrounding steam itself is time dependant. To capture the evaporation time completely, it is therefore necessary to describe the steam temperature as a time dependent function:

\[ T_s(t + dt) = T_s(t) - dT_s \]  
\( \text{(C.9)} \)

Where \( dT_s \) can be determined from:

\[ \dot{m}_s c_{ps} dT_s = N_d \int_t^{t+dt} H(t) dt = \frac{\dot{m}_w}{\dot{m}_d(t = 0)} \int_t^{t+dt} H(t) dt \]  
\( \text{(C.10)} \)

Now, with a given droplet diameter and initial steam temperature the evaporation time of a droplet can be determined. Equations (C.2-C.10) were solved in Matlab. The evaporation time as a function of the droplet diameter was solved for \( T_s(t=0) = 530^\circ C \), \( \dot{m}_s = 0.15 \text{kg/s} \) and \( \dot{m}_w = 0.0144 \text{kg/s} \). The result is shown in figure C.1. It can be seen that for an increasing droplet diameter (at \( t = 0 \text{s} \)), the evaporation time is rising more then linear. This was to be expected since the water mass of the droplet that has to be evaporated rises with a \( 3^{rd} \) power, while the heat exchanging surface of the droplet rises with the \( 2^{nd} \) power.

Apart from the droplet evaporation time, also the water flow that has to be injected to obtain a certain temperature drop of the steam flow needs to be determined. This will differ for every operation point (combinations of steam mass flow and steam temperature) and therefore has to be determined for the operating range of the prototype. Figure C.2 shows the steam temperature decrease resulting from varying injected water flow at different steam mass flows. These results were obtained by assuming that the difference of heat capacity for varying steam temperature is negligible, since the steam is superheated. With figure C.2 the water injection mass flow range that needs to be covered by the after spray unit to fulfill the demands that were set in the previous section can be determined.

It was demanded from the after spray unit that it should be capable of cooling down both the 50% part-load steam flow of 0.075 kg/s at 650\(^\circ\)C and the 100% full-load steam flow of 0.15 kg/s at 530\(^\circ\)C occurs down to the safe steam temperature of 450\(^\circ\)C. From figure C.2 it can be seen that from the first demand it follows that the after spray unit should be able to inject 50liter/hr, from the second demand it follows that 40 liter/hr is the required injected water flow. Therefore the required injection water flow range for the after spray unit is 40-50 liter/hr.
Figure C.1: Evaporation time of water droplets for varying initial droplet diameter (at t=0s)

Figure C.2: Steam temperature decrease for varying steam flow and injected water flow

Now that the required injection water flow range and the droplet evaporation time relation have been obtained, these can be used for the design of the after spray unit.
C.1.3 Design

The after spray unit basically consists of a water pump, a spraying nozzle, and a steam pipe line between the steam generator and steam turbine that also functions as evaporation space (figure C.3). Now that the evaporation time for different sizes of water droplets is known, the nozzle that will produce the droplets can be selected and the steam line that will function as evaporation space can be selected. In this section the nozzle and steam pipeline are designed.

![Diagram of the after spray unit](image)

Figure C.3: Schematic of the after spray unit between the steam generator and steam turbine

The droplet size of the droplets produced by a nozzle are dependant on the pressure drop over the nozzle; the higher the pressure drop, the smaller the droplets. Because all droplets should be evaporated before they reach the steam turbine, the droplet size should of course be as small as possible so that they will evaporate quicker. The nozzle should be selected such that it will deliver the maximal required water injection flow at the maximum possible pressure drop over the nozzle. The maximum pressure at the exit of the (positive displacement) feed water pump is 50 bar, the pressure in the steam line at 100% full-load is 35 bar therefore the maximum available nozzle pressure drop is 15 bar. To account for losses a maximum available nozzle pressure drop of 12 bar is assumed. The droplet size is also influenced by the spraying angle and and the nozzle type (e.g. solid cone, hollow cone). Hollow cone nozzles typically produce smaller droplets and a larger spraying angle typically produces smaller droplets. Therefore a hollow cone nozzle with the smallest possible spraying angle and a water flow of 50 liters/hour at 12 bar pressure drop should be selected. Nozzle type WM808 from manufacturer Delavan fits these specifications. To be able to design a steam line in such a way that all the water droplets will evaporate, the droplet size that is produced by Delavan nozzle type WM808 has to be
determined. This can be done by first determining the Sauter Mean Diameter (SMD). The SMD represents the diameter of a sphere with the surface-volume ratio of all produced droplets together. In figure C.4 the SMD of hollow cone nozzles is shown as a function of flow and pressure.

From figure C.4 it follows that a flow of 50 liters/hour and a pressure drop of 12 bar gives an SMD of 100 $\mu$m. The SMD itself is not equal to the maximum droplet diameter, but can be used to determine the maximum droplet diameter that will occur. The relation between the SMD and the maximum droplet diameter is dependant of the spreading of droplet diameters, which is given by value of $q$ [8]. The value of $q$ lies between 2 and 2.8 for most nozzles. With $q$ it is possible to determine $D_{0.999}$ which gives the largest droplet diameter; 99.9% of the droplets is smaller then $D_{0.999}$. However, this results in a maximum diameter that occurs very seldom. It seems inefficient to design the steam line for this droplet size, since it will concern only a few droplets. Single droplets cannot damage the turbine, large numbers of droplets can. It is therefore chosen to design the steam line for $D_{0.9}$, which is given by:

$$D_{0.9} = X (2.3025)^{\frac{1}{q}} \quad (C.11)$$

In equation (C.11) $X$ is another characteristic diameter that will not be
explained further explained here; $X$ can be eliminated by:

$$\frac{SMD}{X} = 0.56418$$  \hfill (C.12)

Substituting equation (C.12) into equation (C.11) gives:

$$D_{0.9} = \frac{SMD}{0.56418} (2.3025)^{\frac{1}{q}}$$  \hfill (C.13)

Assuming $q = 2$ (worst-case) and filling in the known SMD of $100\mu m$ it follows that $D_{0.9} = 270\mu m$. To put this into perspective, also the most occurring droplet diameter $D_{peak}$ is determined:

$$D_{peak} = X \left(1 - \frac{1}{q}\right)^{\frac{1}{q}}$$  \hfill (C.14)

Similar to $D_{0.9}$ it can then be determined that $D_{peak} = 125\mu m$. It is therefore concluded that choosing to design the steam line for $D_{0.9}$ gives enough safe margin.

Figure C.5: Required steam line dimensions to ensure evaporation of 90% of all droplets

With the maximum (considered) droplet size known, the maximum evaporation time can now be determined. With $D_{0.9} = 270\mu m$ it can be seen from figure C.1 that the maximum evaporation time $\Delta t_{max}$ is about 1 second. Filling in $\Delta t_{max} = 1s$ and $\dot{m}_{steam} = 0.15kg/s$ into formula C.1 gives a relation for the minimum required length and diameter of the steam line which is shown in figure C.5. It can be seen from figure C.5 that either very long steam line or a large high pressure vessel with thick walls is needed to ensure evaporation
of 90% of the water droplets. Besides the fact that this will take up a lot of space, problems are foreseen during start-up of the installation: when either the very long pipe line or the high pressure vessel (with thick walls) is cold during start-up, steam will condensate on the walls. This will cause just that what was meant to be prevented: water will reach the steam turbine. Therefore also another way of steam temperature limiting is explored in the next section.

## C.2 Aircooler

### C.2.1 Demands

In this section another way of steam temperature limiting will be treated: air cooling. Section 6.1 showed that the hot gasses from the burner with a temperature of about 850°C can result in dangerous steam temperature levels during start-up of the installation.

![Air cooler schematic](image)

Figure C.6: Schematic of the air cooler unit between the gas turbine and steam generator

A way of limiting the steam temperature can be implementing after spray as was treated in the previous section. An alternative way would be to (temporarily) take away the cause of the high steam temperature, which is the high gas temperature. The auxiliary burner mixes fuel (natural gas) and air in a fixed ratio to ensure clean combustion, therefore the exhaust gas temperature cannot be controlled with the auxiliary burner itself. However, it is possible to mix cold air with the hot exhaust gasses after the combustion process of the burner, resulting in a larger exhaust gas flow with a lower temperature. If there will be added enough cold air, the exhaust gas temperature could be chosen such that the steam temperature could never exceed its maximum allowable value. From now on we will call this way of steam temperature limiting "air cooling". Figure C.6 shows the working principle of air cooling. If would apply the same
demands to the design of the air cooler as for the after spray unit, the air cooler of the prototype should be capable of cooling down both the 50% part-load steam flow and 100% full-load from there maximum occurring values to the safe steam temperature of 450°C. However from the working principle it follows that the air cooler does not cool down the steam to a safe temperature value, but that it prevents the steam from ever reaching an unsafe value. Therefore the demand to the design of the air cooler is that it should prevent the steam from reaching a temperature level higher then 450°C, both at 50% part-load and 100% full-load. Basically, the steam flow is is irrelevant for the design of the air cooler; it is the exhaust gas flow that should be cooled down. Therefore the air cooler should be capable of adding enough cold air to the exhaust gas flow so that the exhaust gas temperature stays 600°C.

C.2.2 Design

The auxiliary burner takes in air with an air fan, mixes it with natural gas in a fixed ratio and then combusts this mixture. The thermal power of the auxiliary burner is equal to:

\[ P_{auxburner} = m_g c_p (T_{hotgas} - T_{inlet}) \]  \hspace{1cm} (C.15)

The maximum auxiliary burner power is 200kW. The hot gasses of the auxiliary burner have a temperature of 900°C. Assuming an inlet air temperature of 15°C, therefore the air mass flow of the burner can be calculated with:

\[ 200 \cdot 10^3 = m_g 1.145 \cdot 10^3 (900 - 15) \]  \hspace{1cm} (C.16)

It then follows that the required airflow \( m_g = 0.2 \text{kg/s} \). An air fan that can deliver this airflow was selected and fitted on the prototype.
APPENDIX C. STEAM TEMPERATURE LIMITING
Appendix D

Development of a multi-fuel RCG-combustor

D.1 Introduction

Since the RCG operates at the relatively low pressure ratio of 3 up to 4 (see chapter 3), it cannot make use of existing combustion chambers of conventional gas turbines, which are all operating at much higher pressure ratios. Therefore, a combustion chamber was developed especially for the RCG, together with the company EM-group (Netherlands).

D.2 Fuel

Because the development of the RCG started at the beginning of the 21st century, it is only logical that the RCG is designed to be capable of running on biofuel. Therefore the RCG should be able to combust almost all types of gaseous and crude liquid biofuels. Also, the RCG should be capable of handling not only high calorific fuels, but also low calorific fuels. To set this high standard it was chosen to develop a prototype of a RCG combustor that runs on natural gas, diesel and unpurified glycerol. Glycerol is the by-product of bio-diesel production, which can be economically appealing even without government support. From a technological point of view glycerin is a very challenging bio fuel; it has a low calorific value (16\text{MJ/kg}) and a high viscosity (617\text{mm}^2/s @23^\circ\text{C}). Also, the most economically attractive form, g-phase (glycerine with alkaline and ashes), is even more challenging. It is believed that once the RCG is suitable for g-phase, it is suitable for any liquid bio fuel, and for any crude liquid fuel in general.
D.3 Fuel line

When operating at any bio fuel, preheating of the fuel is essential. A schematic of the RCG fuel line is shown in figure D.1, from left to right: fuel tank, 1\textsuperscript{st} stage preheating, filter, fuel pump, 2\textsuperscript{nd} stage preheating and fuel nozzle in the combustion chamber. Preheating is an important measure to guarantee easy pumping and good atomization of the fuel spray when it is injected in the combustion chamber. G-phase contains water and methanol. Water of course boils at 100\textdegree{}C and methanol has its boiling point at around 65\textdegree{}C. Even if the water and methanol mass percentage would be very low, the volume occupied by their vapour would still be considerable because of the liquid-to-gas phase change. To prevent cavitations in the fuel pump and interrupted fuel flow through the nozzle, a two stage preheating arrangement is implemented. The first heating stage should not exceed 65\textdegree{}C. The second heating stage can go up to 195\textdegree{}C because it is after the fuel pump which pressurizes the fuel from 1 to 40 bar. However, for bio-fuels in general the temperature of 195\textdegree{}C might be too high and cause deterioration, therefore the preheating temperature will be lowered to the optimal value of 80\textdegree{}C. To be able to achieve the pressure ratio of 40 a positive displacement pump is the most suitable choice (figure D.2). The most efficient preheating temperature seems to be 80\textdegree{}C, above this temperature there can not be obtained much more improvement of the droplet SMD. Also considering that at temperatures well above 80\textdegree{}C the bio-fuel could deteriorate chemically or even start boiling, we will study more closely the atomization of an average bio-fuel at 80\textdegree{}C. The atomization characteristics were mainly determined with the following equations. Lin and Reitz (1998) proposed the following condition that will ensure atomisation (break up of the fuel flow after the nozzle into droplets):

\begin{equation}
We_g \geq 40.3 \quad (D.1)
\end{equation}

With $We_g$ the Weber-number that is determined by:

\begin{equation}
We_g = \frac{U^2 \rho D}{\sigma} \quad (D.2)
\end{equation}
Where $U$ is the speed of the fluid leaving the nozzle, $\rho$ the density of the fluid, $D$ the nozzle hole diameter and $\sigma$ the surface tension. Hiroyasu and Arai (1989) obtained the following empirical relationship for the SMD:

$$\frac{SMD}{D} = 0.38Re^{0.25}We^{-0.32} \left( \frac{\mu_l}{\mu_g} \right)^{0.37} \left( \frac{\rho_l}{\rho_g} \right)^{-0.47}$$  \hspace{1cm} (D.3)

Where $Re$ is the Reynolds number, $\mu_l$ the dynamic viscosity of the fuel, $\mu_g$ the dynamic viscosity of the combustion air, $\rho_l$ the density of the fuel and $\rho_g$ the density of the combustion air.

Figure D.3 shows the minimum required nozzle exit velocity that is necessary to ensure atomisation with varying nozzle diameter, both for diesel and an average bio-oil preheated at 80°C. For the average bio-oil, a density of 925 kg/m$^3$
and a dynamic viscosity of 0.013875 Ns/m³ are assumed.

### D.4 Combustion chamber design

Because the RCG operates at a pressure ratio of 3 up to 4 which is in between the pressure ratio of duct burners (0 – 0.05 bar) and conventional gas turbines combustors (10 – 40 bar), the RCG-combustor is a cross-over of a duct-burner and an industrial gas turbine combustor. Because the RCG should give low NOₓ emissions and be able to combust low calorific fuels, its design combines internal cooling with preheating of the combustion air. Figure D.4 shows the working principle of the combustion chamber of the RCG.

![Figure D.4: Working principle of the combustion chamber of the RCG](image)

The intake air is preheated by the hot inside wall of the burner. Because the air is already warmed up before it takes part in the combustion process, it is possible to combust low calorific fuels and fuels that vaporise more difficult. Because the inside wall heats the intake air, it means that the inside wall is in fact cooled to a lower temperature by the intake air. Because the inside wall is cooled to a lower temperature it will be able to absorb more heat radiation of the combustion process. This means that the temperature of the hot spots in the combustion process will be reduced, thus reducing NOₓ formation.

### D.5 Experiments

The first tests showed that the burner was able to combust both natural gas and glycerol. Running on natural gas the burner obtained full combustion (CO=0PPM), however when running on glycerol there was CO-formation at
the required 200kW thermal power. Because there was $CO$-formation it was concluded there was not enough residence time for all the glycerol to combust fully.

![Diagram of the combustion chamber of the RCG](image)

Figure D.5: Drawing of the combustion chamber of the RCG

To provide for enough residence time, the combustor was prolonged from 1200mm to 2100mm (see figure D.5) which almost doubles the length of the space where the combustion takes place, which therefore also practically doubles the residence time. Furthermore, to improve atomisation of the fuel, the fuel was preheated by leading it through a coil which is integrated in the airpreheating space in the combustor. The fuel is preheated up to a temperature of 80°C, which is considered the ideal preheating temperature.

After the improvements were implemented, the combustor was tested again. The combustor was still able to perform full combustion of natural gas and, this time there also proved to be full combustion of glycerol at the required 200kW thermal power; it was measured that $CO = 0$ PPM. Furthermore, the $NO_x$ emission was measured; $NO_x = 120 \text{mg/Nm}^3$. This means that the combustor already complies with the most stringent current current rules for $NO_x$-emissions of installations that combust bio-fuels ($NO_x < 200 \text{mg/Nm}^3$). It
also shows that with little effort, the combustor can be made to comply with the most stringent NO\textsubscript{x}-limit that will be enforced in 2010: NO\textsubscript{x} < 120mg/Nm\textsuperscript{3}. 
Curriculum Vitae

Henk Ouwerkerk was born on 26 February 1974. He attended secondary school (Atheneum) from 1986 to 1992 at Lorentz Lyceum in Eindhoven (Netherlands). After finishing secondary school he studied Aeronautical Engineering at Delft University from 1992 to 1994. In 1994 he started to study Mechanical Engineering at Eindhoven University. In the year 2000 he came up with an idea for a new type of combined cycle, the “Rankine Compression Gasturbine” (RCG). Between 2000 and 2002 a patent application was filed. He also founded the company Heat Power to develop the RCG from concept to product, together with the section Energy Technology of the faculty Mechanical Engineering. In 2002 he obtained a master’s degree for his work and thesis on a feasibility study of the RCG. As a TU/e-researcher he realised subsequently an experimental set-up and RCG-prototype in the lab of the section Energy Technology between 2002 and 2006. Thanks to a STW Valorisation Grant, in 2006 his position of TU/e-researcher was converted to that of a PhD student. From 2006 to 2008 he developed an operating strategy for the RCG and designed a first commercial 1MWe RCG.
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I was very lucky that, over the years, more than 10 students participated, both in Bachelors and Masters projects. I would like to thank all of them for their valuable contributions.

For this project a prototype was realised, incorporating over two tonnes of steel, I would like to thank Gerard, Geert-Jan and Henri from the workshop, for their heavy construction work, and their precision work.

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