



**THERMODYNAMIC CYCLE DESIGN OF A BRAYTON -
RANKINE COMBINED CYCLE FOR A PEBBLE BED
MODULAR REACTOR**

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THERMODYNAMIC CYCLE DESIGN OF A BRAYTON - RANKINE COMBINED CYCLE FOR A PEBBLE BED MODULAR REACTOR

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ABSTRACT

The rapid development in nuclear technology worldwide has created the need for an efficient power conversion unit to extract the energy from the new generation IV reactors. The generation IV reactor currently under investigation in South Africa is the PBMR-DPP (Pebble Bed Modular Reactor Demonstration Power Plant) based on the High temperature Reactor Modul. This reactor produces 200 MW of thermal energy at inlet/outlet temperatures of 250°C/700°C. Due to the reactor layout and accompanying thermal fluid path design outlet temperatures in the order of 900°C would be possible.

This dissertation is aimed at the design and optimisation of a Brayton-Rankine combined cycle for use with a PBMR-DPP. The combination of these two cycles improves the thermal efficiency due to the large difference between the maximum and minimum temperatures.

The Brayton and Rankine cycles will be developed independently and optimised to ensure that the best possible efficiency is gained from the combined cycle. The heat energy available in the reactor is the input parameter for the Brayton cycle, After the Brayton cycle's pressure ratio has been optimised the heat rejected to the Rankine cycle will be known. The aim of the design is to determine if 50% combined cycle thermal efficiency is achievable.

The initial sizing calculation of the cycle parameters has been done in a software package that has been developed for use in the thermo-hydraulics field. Engineering Equation Solver (EES) makes use of an iterative process to simultaneously solve the set of equations. The results obtained from EES were verified by Microsoft Excel with a specialised macro installed for thermo-hydraulics.

A very specific methodology was used to solve the Brayton cycle. Traditionally the Brayton cycle is optimised for maximum cycle efficiency to ultimately obtain the best combined cycle efficiency. Very complex cycles such as reheat and multi-shaft Brayton cycles were used. The solution methodology used in this dissertation is to optimise the simple Brayton cycle for the maximum specific work produced in the



cycle. The large amount of heat at the turbine outlet is then transferred to the Rankine cycle.

The results obtained from the calculations performed were that a combined cycle efficiency of 52.914% has been achieved at optimum operating conditions. The combined cycle has been shown to operate above 50% efficiency in a wide variety of load-following conditions.



OPSOMMING

Die snelle ontwikkeling van kernenergie wêreldwyd het 'n behoefte laat ontstaan vir 'n doeltreffende kragomsettingseenheid om die energie van die nuwegenerasie- IV-reaktors te onttrek. Die generasie IV-reaktor wat tans in Suid-Afrika ondersoek word, is die PBMR-DPP (Korrelbed Modulêre Reaktor Demonstrasie-kragaanleg) wat gebaseer is op hoëtemperatuur-reaktor Modul. Dié reaktor produseer 200 MW termiese energie by inlaat- en uitlaattemperature van 250°C/700°C.

Die verhandeling is gemik op die ontwerp en optimalisering van 'n Brayton-Rankine gekombineerde siklus vir gebruik in die PBMR-DPP. Die kombinasie van hierdie twee siklusse verbeter die termiese doeltreffendheid vanweë die groot verskille tussen die maksimum- en minimumtemperatuur.

Die Brayton- en Rankine-siklusse sal onafhanklik ontwikkel word en geoptimaliseer word sodat die bes moontlike doeltreffendheid deur die gekombineerde siklus behaal kan word. Die hitte-energie wat beskikbaar is in die reaktor is die insetparameter vir die Brayton-siklus. Nadat die Brayton-siklus se drukverhouding geoptimaliseer is, sal die hitte wat afgekeur word na die Rankine-siklus bekend wees. Die doel van die ontwerp is om vas te stel of 50% termiese doeltreffendheid vir die gekombineerde siklus moontlik is.

Die aanvanklike grootteberekening van die siklusparameters is gedoen in 'n sagtewarepakket wat ontwikkel is vir gebruik op termies-hidrouliese gebied. Engineering Equation Solver (EES) maak gebruik van 'n iteratieweproses om gelyktydig die stel vergelykings op te los. Die resultate wat verkry is van EES is geverifieer met Microsoft Excel met 'n gespesialiseerde makro geïnstalleer vir termo-hidroulika.

'n Baie spesifieke metodologie is gebruik om die Brayton-siklus op te los. Tradisioneel is die Brayton-siklus geoptimaliseer vir die maksimum siklusdoeltreffendheid ten einde uiteindelik die beste gekombineerde siklusdoeltreffendheid te bekom. Baie ingewikkelde siklusse soos herverhitting en multiskag- Braytonsiklusse is gebruik. Die oplossingsmetodologie wat in hierdie



verhandeling gebruik is, is om die eenvoudige Brayton-siklus te optimaliseer vir die maksimum spesifieke werk wat gelewer word in die siklus.

Die groot hoeveelheid energie by die turbine-uitlaat word dan oorgedra na die Rankine-siklus.

Die resultate behaal van die berekeninge wat uitgevoer is, is dat 'n gekombineerde siklusdoeltreffendheid van 52.914% behaal is by optimum werkstoestande. Die gekombineerde siklus het ook bewys dat hy by hoër as 50% doeltreffendheid kan werk in 'n wye verskeidenheid kragaanvraagtoestande.



DECLARATION

I Cornelius Petrus Kloppers (Identity Number: 860518 5060 082) hereby declare that the work contained in this dissertation is my own work. Some of the information contained in this dissertation has been gained from various journal articles and text books has been referenced accordingly.

C.P. Kloppers

Witness



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NOMENCLATURE

$^{\circ}\text{C}$	Degrees Celsius
AVR	<i>Arbeitsgemeinschaft Versuchsreaktor</i>
BFP	Boiler feed pump
CBC	Closed Brayton Cycle
CC	Combined Cycle
DPP	Demonstration Power Plant
EES	Engineering Equation Solver
ELS	European Lead-cooled System
ESV	Emergency stop valve
GFR	Gas-cooled Fast Reactor
GIF	Generation IV International Forum
HPC	High-pressure compressor
HPLWR	High-performance Light-water Reactor
HRSG	Heat Recovery Steam Generator
HRSG	Heat recovery steam generator
HTR	High-temperature Reactor
LFR	Lead cooled fast reactor
LMFBR	Liquid metal fast breeder reactor
LOCA	Loss of coolant accident
LPC	Low pressure compressor
LWR	Light-water Reactors
MSBR	Molten Salt Breeder Reactor
MSR	Molten Salt Reactors
PBMR	Pebble Bed Modular Reactor
PCU	Power Conversion Unit
r_p	Pressure Ratio
SCWR	Super Critical Water-cooled Reactor
SFR	Sodium-cooled Fast Reactor
SL	Saturated liquid
SSTAR	Small Secure Transportable Autonomous Reactor
THTR	Thorium high-temperature nuclear reactor
TIV	Turbine inlet valve
VHTR	Very High-temperature Reactor





1 CHAPTER 1: INTRODUCTION

1.1 BACKGROUND

The energy that is available to mankind is present in mainly two forms; namely renewable and non-renewable energy. Renewable energy is any form of energy that is sustainable for a prolonged period of time without depletion. The commercially viable types of renewable energy are:

- Hydro energy
- Wave energy
- Solar energy
- Wind energy

The aim of this dissertation is to look at the non-renewable energy source. This non-renewable energy market makes use of fossil fuels to create usable power for industries and residential use. Examples of these types of energy are:

- Coal energy
- Nuclear energy
- Oil energy
- Natural gas energy

The use of coal fired power stations releases high amounts of carbon dioxide gas into the atmosphere. By limiting the carbon dioxide pollution into the atmosphere the advance of global warming may be reduced.



1.2 OBJECTIVE OF STUDY

With the development of the new generation IV Nuclear Reactors a suitable power conversion unit (PCU) is needed to ensure that the high gas outlet temperature is utilised. The new high temperature reactor (HTR) plants are able to deliver gas outlet temperatures as high as 900°C continuously.

The objective of this dissertation is to investigate the possibility of the use of a combined cycle (CC) with the 200 MWth Pebble Bed Modular Reactor. If it were found to be feasible to use a CC with a HTR, a cycle design must be done to determine the performance of the power conversion unit (PCU).

The performance parameters gained from this design will determine if it will be profitable to construct a combined cycle power conversion unit for the pebble bed modular reactor PBMR-DPP. If the CC is compared with a traditional coal fired power station the main comparison will be the cycle efficiency. The aim of the CC is to obtain a cycle efficiency higher or equal to 50%, which is 10 - 18% higher than the equivalent coal fired power station.

Many studies have been completed focussing on comparing different cycles to be used with the HTR. Learning from the results of these studies this dissertation will be focused more on the optimisation of a PCU. The off-design conditions have to be incorporated due to the load-following behaviour of this PCU.

1.3 LIMITS OF STUDY

As described in the title this dissertation is focused on cycle design in particular, thus the design of turbo machines as well as heat exchangers are not included. Although these designs are not included, the turbo-machine efficiencies (politic and isentropic) as well as heat exchanger effectiveness have been incorporated.

This study is limited by the practical implications. Because generation IV nuclear reactors are still in the design phase, it is not possible to conduct tests on these reactors. This dissertation is focused on the design of the PCU so that it may be implemented once the HTR has been completed. The design calculations will be



verified by using similar software packages, but due to the reasons mentioned it will not be possible to validate the results.

This dissertation is focused on a macroscopic analysis of a possible PCU. Thus fluid flows within the cycle which may be solved with computational fluid dynamics (CFD) have not been considered. It will be possible to apply the CFD technology to the cycle once the design has been completed to improve the cycle efficiency, but due to the complexity of CFD this is not the aim of this design.

1.4 DISSERTATION STRUCTURE

The dissertation consists of a number of chapters each focusing on a specific area of the study. This section aims to summarise the chapters briefly.

Chapter 1

The first chapter serves as an introduction to the dissertation, the basic goal to be achieved as well as the limitations of the study will be mentioned here. This chapter could also serve as a research proposal to motivate the need of the specific research that will be done, as well as clearly define the scope of the project.

The scope has to be clearly identified as efficiently as possible to ensure that every aspect that is mentioned is thoroughly investigated and the correct results are obtained. This chapter also serves as a guide during the research process to ensure that the objective of the study is achieved

Chapters 2 & 3

The first goal of this dissertation is to determine what background information is available. If the background information is thoroughly examined a better understanding of the problem may be gained. It also provides useful information on how the problems were solved in the past as well as a possible solution methodology that may be used in the present.



The next step is to conduct a literature survey to obtain the most current information available on a specific problem under consideration here. This also ensures that work is not duplicated and the scientific community makes the best progress possible.

In this dissertation the background and literature survey of the two main topics were identified. The first area is the nuclear reactors past, present and future; a study from the first time nuclear fission was achieved to the most modern generation IV reactors has to be investigated.

The second research area is the development and operation of combined cycles worldwide. Valuable information may be gained if similar designs or operating plants were found. These two main topics are divided into two chapters focusing on the literature survey.

Chapters 4 & 5

The combined cycle consisting of the Brayton and Rankine cycles will be discussed in three different chapters, the first being chapter 4 where the development of the closed Brayton cycle will be considered.

The different possible cycles will be investigated and then optimisation has to be carried out on these cycles. The cycles are developed with the parameters of the 200MWth PBMR-DPP as the input parameters. The physical limitation of the reactor as well as the Brayton cycle has to be considered.

Optimisation of the cycle for the CC has to be done after the main design parameter of the cycle has been determined. The result of this chapter is a fully optimised Brayton cycle that may be used with the PBMR-DPP and the Rankine cycle to form a combined cycle power conversion unit.

Chapter 6

The Rankine cycle is commonly used in South Africa and throughout Africa for power generation. This makes performance parameters for the Rankine cycle easily



available. The Rankine cycle used for power generation has to be adapted for the use in the combined cycle.

As the Brayton cycle has been developed in the previous chapter, the Rankine cycle may now be developed with the Brayton cycle outlet conditions acting as the inlet conditions of the Rankine cycle.

The deliverable of this chapter is a fully developed Rankine cycle that is adapted to receive heat input via a heat recovery steam generator and operated as efficiently as possible with the given inlet conditions.

Chapter 7

The focus of this chapter is combining the cycles from a fully developed Brayton and Rankine cycle. To ensure that the cycles match perfectly an energy balance has to be done, this is done using the First Law of Thermodynamics.

As the combined cycle will operate at load-following conditions the effect of a change in different parameters has to be determined. All the different parameters that change due to atmospheric conditions, breakdowns or an increase/decrease in load have to be investigated to ensure that there are no dramatic differences in off-design conditions. The deliverable for this chapter is a completely optimised combined cycle.

Chapter 8

The last chapter of the dissertation gives a summary of the results obtained in the design process. The design calculation results have to be verified and the difference in results has to be accounted for. The possible future work that may be done on the cycle has to be determined and explained



2 CHAPTER 2: LITERATURE SURVEY NUCLEAR REACTORS

2.1 INTRODUCTION

Because of the nature of the project there will be two main topics of research. The first area of research will be nuclear reactors and the development of these reactors. The second area of research will be combined cycles and their application in industry.

2.2 HISTORY OF NUCLEAR REACTORS

Looking at the history of nuclear reactors is of the utmost importance. The development of these reactors and design changes is the key in understanding the current reactors. This will also be valuable in anticipating the reactor designs of the future so that the best power conversion unit possible may be designed.

2.2.1 Generation I nuclear reactors:

The first nuclear reaction was achieved by Enrico Fermi (1901 - 1954) by bombarding uranium atoms with neutrons. These neutrons that collided with the uranium atoms caused the atoms to split and then released large amounts of energy. (Allison, 1957). From there on many radio chemists and physicists participated in the development and refinement of the nuclear reaction.

The first nuclear reactor to reach criticality, Chicago Pile 1, was completed on 2 of December 1942. This reactor was mainly used to produce fuel for nuclear bombs (U.S. Department of Energy Office of Nuclear Energy, 1994).

Nine years after the first production of nuclear power the first electricity was generated on 20 December 1951. This had taken place in Idaho in the United States of America (USA). The reactor power was around 100kW. The only Generation I nuclear power plant that is still running today was developed between 1950 and 1960



and is operational in the United Kingdom (U.S. Department of Energy Office of Nuclear Energy, 1994).

2.2.2 Generation II Nuclear reactors

Generation II reactors are mainly focused on the generation of electrical power from the nuclear fission reaction. In many power plants that are powered by nuclear fission, heat is used to produce steam to power turbines (U.S. Department of Energy Office of Nuclear Energy, 1994).

The efficiency of the Generation II reactors coupled with the steam generation system has in the past not been optimal. The development of power conversion units has grown with the development of the nuclear reactors.

2.2.3 Generation III reactors:

The biggest difference between the Generation II and III reactors is that the newer reactors make use of passive safety features. These safety features make use of a natural process such as natural convection of resistance to high temperatures

Third generation reactors make use of the following attributes that were absent in the previous generation of nuclear reactors (World Nuclear Association, 2008):

- To expedite licensing, there are standardised designs for each type of reactor.
- To ensure easy operation a more simple and rugged design is used.
- This results in a longer life cycle and higher availability, and a higher burn-up of fuel to reduce fuel consumption and minimise waste.

Generation II reactors were not designed for load following behaviour, where some of the generation III reactors are (World Nuclear Association, 2008). In addition these reactors are larger than the reactors that preceded them.



2.3 GENERATION IV REACTORS

The future of nuclear reactors will present itself in the form of generation IV nuclear reactors. Because of higher security and tighter regulations, each country is not allowed to develop a new generation reactor independent of an International forum (Pavel Hejzlar, 2005).

The Generation IV International Forum (GIF) was initiated and chartered by the USA in 2000/2001. This forum represents countries and their governments that see nuclear power as a viable alternative for future applications (Pavel Hejzlar, 2005).

After two years deliberation by the GIF six types of reactors were identified as viable options. The research and development will commence and the reactors may be operating as soon as 2030. The reactors that they identified will be discussed briefly

2.3.1 Gas-cooled fast reactors

There are three main focus areas of the gas cooled fast reactor (GFR). Every reactor should be as economical, safe and sustainable as possible.

Waste minimisation is important in the GFR. To reduce the radio toxicity in the repository below that of natural occurring uranium within a period of 1000 years all transuranics (TRU) with losses of 0.1% or less must be recycled. Thus the GFR makes use of a closed loop fuel cycle to ensure low radio toxicity in the repository. In addition the GFR is loaded with TRUs from spent light water reactor fuel (L. Cinotti, 2009).

The removal of heat in the case of loss of coolant accident (LOCA) is very important. If the decay heat is not removed a reactor meltdown might occur, which gives way to a radioactive release. Passive as well as fully active decay heat removal systems were considered in the preliminary design considerations of the GFR. A study revealed that CO₂ is a very favourable gas in the case of LOCA (L. Cinotti, 2009).



Very important is that if GFR is deployed it has to be economically competitive with the other types of reactors as well as other means of power generation. Studies that have been conducted on direct and indirect cycle design have shown that the indirect cycle design has a number of advantages such as a less likelihood of LOCA because of rotating machinery (L. Cinotti, 2009).

Despite the advantages of the indirect cycle, the loss in efficiency and the cost of the intermediate heat exchangers make the indirect cycle economically unviable.

2.3.2 Lead-cooled fast reactors

In reaction to the goals set by the GIF, the development of the lead-cooled fast reactor (LFR) leads to the currently proposed candidates for international co-operation and joint development. These reactors are the Small Secure Transportable Autonomous Reactor (SSTAR) and the European Lead-cooled System (ELS).

The concept SSTAR is a 20MWe natural circulation small shipable reactor vessel. The lead that is used as a coolant is contained in two pressure vessels - the first is the reactor vessel and the second a guard vessel. Lead is used instead of a Lead-Bismuth coolant to reduce the alpha-emitting ^{210}Po isotope formed in the coolant (L. Cinotti, 2009).

ELS is a 600 MWe reactor, also cooled by pure lead. Compact and simple primary circuits with all removable internal components are the objective of this reactor. ELS has been designed to identify innovative solutions to reduce the primary system volume and to simplify complexity of the reactor internals (L. Cinotti, 2009).

2.3.3 Molten salt reactors

Molten salt reactors (MSR) have not attracted as much attention as the other types of reactors discussed here. The research and development of the MSR has been abandoned in the USA and has been replaced by the liquid metal fast breeder reactor (LMFBR) (Moir, 2008).





There are ample economic motivations and predictions to justify the use of an MSR. There are no serious safety concerns on the MSR as there are three barriers to contain the fission products present (Moir, 2008).

The problem with MSR's is that the material issues experienced 40 years ago, have not yet been resolved. The deficiency in the development of the MSR is because several decades ago rapid expansion took place in the nuclear reactor field, while only limited uranium resources were available. Thus, breeding reactors was of the utmost importance. The race for a breeder was between the LMFBR and the molten salt breeder reactor (MSBR). The LMFBR had more neutrons per fission and less loss of neutrons due to parasitic capture, which is a property of a fast reactor and thus won the race (Moir, 2008).

2.3.4 Sodium-cooled fast reactors

A sodium-cooled fast reactor (SFR) utilises depleted uranium in the fuel and has a coolant temperature of 550°C. The USA, France and Japan are currently in agreement to cooperate in the development of the SFR (Pavel Hejzlar et al, 2005).

The PCU of the SFR has previously been a Rankine cycle but there had been 12 sodium leakages in 15 years of operation of the previous generation reactor. This gave way to a sodium-water chemical reaction which is very undesirable. To combat this problem advanced gas Brayton cycles with a chemical inert gas such as Helium will be able to address this problem (Haihua Zhaoa, 2008).

Although the closed Brayton cycle would solve the chemical problem the outlet temperature of the SFR is not high enough to achieve an efficiency of 42 % - 48% as can be achieved in the high temperature reactor. One solution to improve the thermal efficiency of an SFR is to introduce multiple reheat stages. Different studies are currently examining the possibility of this concept (Haihua Zhaoa, 2008).



2.3.5 Super-critical water-cooled reactors

The super critical water-cooled reactor (SCWR) was investigated as early as the 1950s and 1960s. After being dormant renewed interest was gained by the University of Japan and the Russian, Kurchatov, in the 1990's (T. Reiss et al, 2010).

The USA, Korea and Europe have evaluated many different SCRW concepts. Some of these concepts will be discussed briefly:

European concept:

The work on this reactor is mainly carried out by Forschungszentrum Karlsruhe and the concept is called high performance light water reactor (HPLWR). The original design was of Japanese origin, and extensive mechanical, neutronics and thermal-hydraulics research was done on this reactor. This reactor originally had a one pass design but this resulted in hot spots. The change to a three pass design solved this problem with an outlet temperature of 500°C (T. Reiss, 2010).

American concept:

The USA found that for this type of reactor, ZrH_x as a moderator has the most advantage. They used square assemblies with water rods, which have a constant cross-sectional area in the axial direction. The core, in contrast to the European's, is a one-pass core with an outlet temperature of 500°C (T. Reiss, 2010).

2.3.6 High temperature reactors

The focus of this study is the high temperature reactor (HTR) as well as the very high temperature reactor (VHTR). Thus, special attention will be paid to this section in investigating the research and development of these reactors.



HTR fuel

Reactor fuel varies in shape and method of conveying energy. Light water reactors (LWR) use cylindrical pellets effectively forming fuel rods in the reactor heart. In the HTR coated particles are used. These particles are coated by three layers (pyrolytic carbon, silicon carbon and pyrolytic carbon) that protect the fuel itself in case of a complete loss of cooling. (Kugeler, 2009)

These coated particles are then inserted into a porous buffer layer of graphite between, the particles and fuel pebbles are thus formed. The density of these fuel pebbles if inserted into the reactor could vary widely depending on the arrangement. This has a direct influence on the energy that the reactor is able to produce. The reactor core power density is typically between $3 \text{ MW} / \text{m}^3$ and $6 \text{ MW} / \text{m}^3$ to ensure intrinsic safety measures. (Kugeler, 2009)

The pebbles used in the HTR are able to operate normally up to 1350°C (in extreme cases, where accidents occur, up to 1600°C) without the release of significant quantities of fission products. A depiction of a pebble can be seen in the figure below:

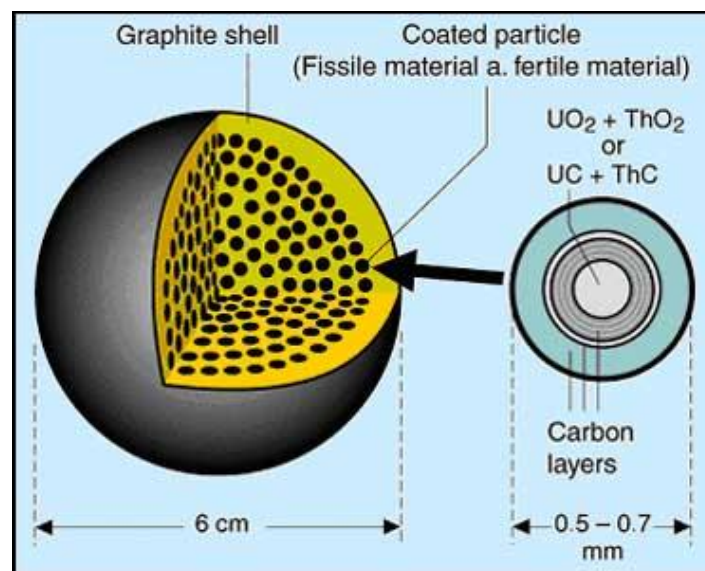


Figure 1: HTR fuel pebble depiction



The loading and utilisation of fuel in an HTR reactor is a continuous process. The fuel pebbles are loaded at the top of the reactor and moves downward as it becomes depleted. The pebbles that are unloaded from the reactor at the bottom is reinserted if it has not reached the predetermined level. These pebbles are added to the centre of the reactor to ensure a more even heat distribution throughout the cylindrical core of the reactor (Kugeler, 2009).

The control of the reactor is carried out via a set of control rods inside the side reflector. These rods comprise of neutron absorbers and effectively kill the nuclear reaction. These control rods are located in the side reflector regions of the nuclear reactor. If a reactor heart diameter is larger than approximately 3 m, control rods have to be introduced in the centre of the reactor.

Coolant

The HTR coolant used is helium. Its use as a coolant is very favourable because it is chemically inert- and does not influence the neutron economy at all (Kugeler, 2009).

The specific heat capacity is the measure of the heat energy required to increase the temperature of a unit quantity of a substance by a unit of temperature (e.g. one degree Celsius or one Kelvin). Compared to air (1.0035 kJ/kg-K), helium (5.1932 kJ/kg-K) has a very high specific heat capacity. The higher specific heat capacity reduces the size of heat exchangers and turbo machinery (less gas is required to transfer the same amount of heat). (Kugeler, 2009).

The coolant gas that passes through the core can be heated up to very high temperatures - in the ranges of 700°C to 950°C . This is achieved with a core inlet temperature of around 250°C to 550°C , depending on the application. The pressure could range between 40 and 80 bar. The power density in the core is between 3 MW/m^3 and 6 MW/m^3 to be able to realise self-acting decay and to avoid overheating of the fuel elements. (Kugeler, 2009)



Operated HTR plants

The first gas cooled HTR operated in Great Britain with the DRAGON-project. The reactor was 20 MWth and used tubular ceramic fuel elements. This reactor was successfully operated for 9 years (1966 – 1975). The project was abandoned because of political and financial reasons (European University Institute, 2006).

At the same time the USA developed a HTR reactor using tubular fuel elements at the peach bottom reactor. This was a 115 MWth reactor and produced 40 MW electrical power. This reactor operated for 23 years (1965 – 1988) and was phased out as the Fort St. Vrain reactor came into operation (Kingrey, 2003).

Fort St. Vrain was operated for 13 years (1976 - 1989) and showed that a medium sized HTR with block type fuel can be operated with reasonably good results. This reactor was an 852MWth and 342 MWe.

Germany commenced their HTR technology with pebble shaped fuel elements for the *Arbeitsgemeinschaft Versuchsreaktor (AVR)* which synchronised to the grid in 1967. This reactor showed that a reactor using pebble shaped fuel could work. Many different safety tests were carried out on this reactor and 22 different fuel types tested (K.J. Kruger, n.d.).

The AVR was stopped after all development and experiments were completed. There were many problems during construction and start-up but the reactor achieved the design data and the availability was good (K.J. Kruger, n.d.).

2.4 REACTOR CHOICE

This choice of nuclear reactor was made with the assistance of experts in the nuclear field. All influences and experience gained by the nuclear community was considered in the choice of a specific reactor. The choice was based on the reactor with the highest possibility of being commissioned in future.



The reactor chosen is the 200MWth HTR MODUL. The acronym MODUL refers to the modular nature of this type of plant (MODUL = *Mehrfach Durchlauf*, multipass movement through the core). This choice was made after extensive testing had been done on the other types of PBMR. (Kugeler, 2009)

2.4.1 Reactor specifications

A proper PCU may only be designed if the correct operating parameters are available. Thus all specifications that are available must be thoroughly examined. The 200 MWth PBMR--DPP is a reactor that is still in the design phase and thus all the specifications are not yet available.

The specifications needed for the design of the PCU have been established within a specific range. Thus within that range the PCU may be operated but the optimum performance parameters have to be specified.

2.4.2 General overview

The PBMR-DPP has a cylindrical core with a height of 9.43m filled with pebble fuel elements (60 mm diameter). The helium coolant flows from the top of the reactor core downwards heating the coolant gas to approximately 950°C. The coolant inlet temperature is also critical for this reactor as the temperature difference from inlet to outlet is limited. The minimum inlet temperature that the reactor is able to operate at continuously is 250°C. (Kugeler, 2009)

Mehrfach Durchlauf literally translated into English is “multiple passage”. Thus the fuel is added continuously and circulated approximately 10 times through the reactor before the fuel is finally burnt up (Ryder, 1997).

2.5 SUMMARY

The different types of reactors have now been examined and a suitable reactor choice has been made. The application of a combined cycle to the PBMR-DPP is possible. Thus, the next step is to investigate the different types of combined cycles



found in the world today. It is also important to have a look at some proposed cycles that are specifically aimed at nuclear reactors.



3 CHAPTER 3: LITERATURE SURVEY COMBINED CYCLES

3.1 INTRODUCTION

A combined cycle consists of two different thermal cycles with the same or different working fluids. The cycle with the higher operating temperature is called the “topping cycle” and the cycle operating on the rejected heat is called the “bottoming cycle”.

The most common combined-cycle is the gas-turbine and water/steam cycle. The development was relatively easy and inexpensive because both power cycles existed and operated relatively efficiently.

3.2 COMBINED CYCLE CONCEPTS

There are different combinations of components used in a combined cycle. These variations may lead to changes in efficiency and cycle performance. These variations in the combined-cycle are mainly classified in different operating pressures. (Kehlhofer et al., 1999)

- Single pressure cycles
- Dual pressure cycles
- Triple Pressure cycles

The different pressure cycles will be examined in Chapter 4 & 5

3.3 *Applications of combined cycles*

3.3.1 Possible PCU designs

A number of studies have been conducted on the PCU for different reactors. A number of different combined cycles have been identified that is promising for high

efficiency on nuclear reactors. These PCU's will be explained in this chapter and used in the following chapters as a guideline for the PCU design.

The most recent study (Laurens, 2009) completed in 2009 has identified the following three cycles as the most promising. These cycles will be discussed in order Cycle X, Cycle Y and then Cycle Z.

The input parameters for all three cycles have to be kept constant to evaluate the three cycles on an equal standing. These parameters are:

Table 1 Input parameters.

Input Parameters for evaluation of Cycle X, Cycle Y and Cycle Z	
Thermal power	500 MWth
Gas turbine inlet temperature	900°C
Minimum helium temperature	23°C
Maximum Brayton cycle pressure	90 Bar
Gas turbine speed	6000 rpm
Maximum steam temperature	540°C
Condenser temperature	35°C
Maximum steam pressure	180 Bar

Results (Laurens, 2009)

The thermal efficiency of Cycle X proved to be the highest for the given input parameters. The other cycles (Y and Z) are only half a percentage point lower than Cycle X. These results are of great importance to this study, but may not be applied directly because of the different input parameters. (Laurens, 2009)

As stated earlier, there is limited information available on combined cycles applied to nuclear reactors. Thus, other combined cycles operated from various heat sources have to be investigated. (Laurens, 2009).



3.4 Combined cycles in operation

As stated earlier the nuclear reactors that are currently operating worldwide is the Generation II and Generation III reactors. There are no Generation IV reactors with combined cycle power generating units that are operational.

Brazil:

Currently Brazil is able to economically justify the use of natural gas to generate power. This is done because of diversification of power-generating methods that is currently underway. They make use of combined cycles to generate electricity by using natural gas turbines as well as the heat recovery steam generator (HRSG) and steam turbines (Felipe R. Ponce Arrieta, 2005).

The simplified diagram that is used in the combined cycle is shown in the figure below. The gas turbines used are the AG 501F Siemens turbines, the power ratings of these turbines as well as the gas turbines will be discussed later. The combined cycle consists of two natural gas turbines that feed the two HRSG's. The steam is passed through the high-pressure steam turbines and then reheated. It then passes through the intermediate pressure turbines and finally through the low pressure turbines where the steam is then condensed and pumped back into the HRSG to start the cycle again (Felipe R. Ponce Arrieta, 2005).

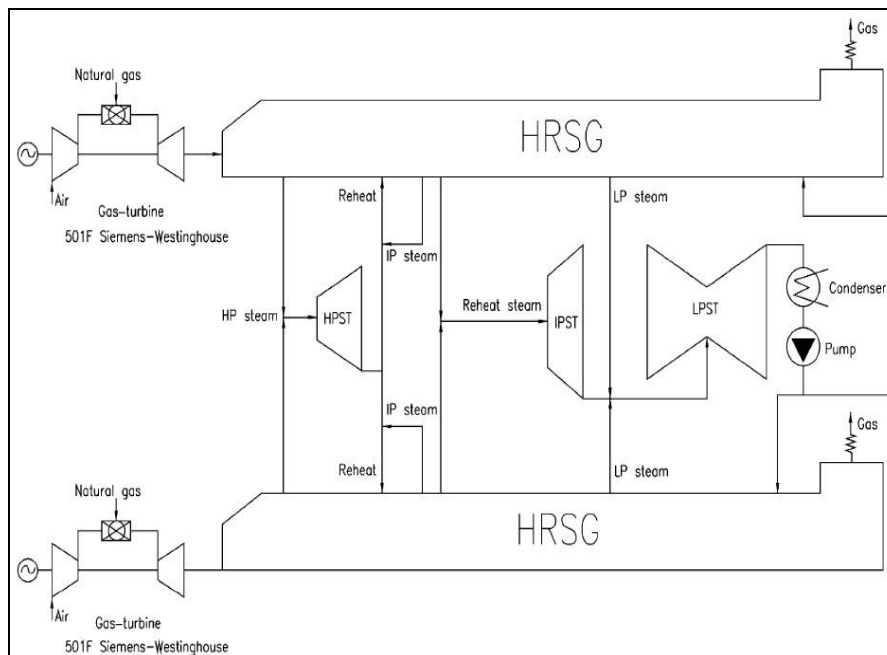


Figure 2 Combined cycle used by Brazil power generation units

The HRSG cannot be assumed to be a simple heat exchanger. Secondary firing takes place to increase the temperature of the exhaust gas. There is a sufficient oxygen supply in the gas because of the air that passes through the turbine blades that is used for cooling. The HRSG can be seen in more detail in the figure below (Felipe R. Ponce Arrieta, 2005).

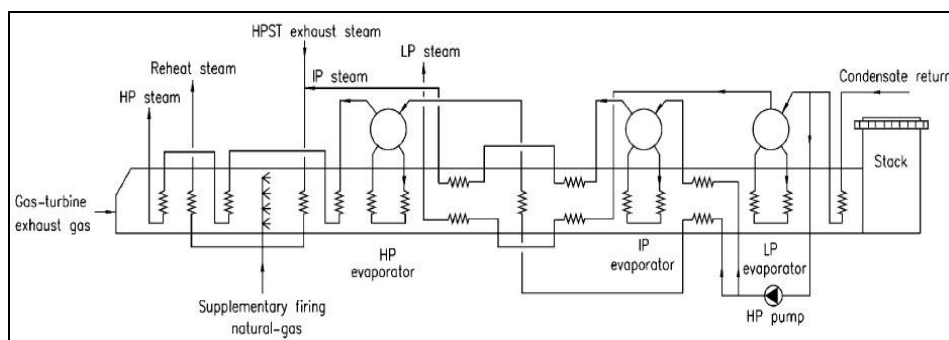


Figure 3 HRSG for Brazil's power plant.

The following table summarises the data for the specific plant being discussed.

Table 2: Specific data of combined cycle

Parameter	Value	Unit
Ambient temperature	15	$^{\circ}\text{C}$
Atmospheric pressure	101.3	kPa
Relative humidity	0.60	
Net total electric power	600	MW
Fuel		
Natural gas	46515	KJ/Kg
Supply conditions	2.758/15	$\text{MPa}/^{\circ}\text{C}$
Gas turbine (Siemens AG 501F)		
Gross power	174.66	MW
Maximum combustion efficiency	90	%
Combustion efficiency	99	%
Turbines isentropic-efficiency	94,31	%
Generator efficiency	98,5	%
Turbine inlet temperature	1382.5	$^{\circ}\text{C}$
Turbine outlet temperature	608	$^{\circ}\text{C}$
Auxiliary power consumption	1.18316	MW
Steam turbine		
High Pressure Steam	15.6/530	$\text{MPa}/^{\circ}\text{C}$
Intermediate Pressure	3.2/530	$\text{MPa}/^{\circ}\text{C}$



Low pressure	0.75/305	$\text{MPa}/^{\circ}\text{C}$
Condensate operating pressure	5.06	kPa
Net power	253.37	MW
HP turbine isentropic efficiency	80,89	%
IP turbine isentropic efficiency	92,59	%
LP turbine isentropic efficiency	88,67	%
Minimum quality of LP steam	85	%
HRSG		
Inlet gas temperature	675	$^{\circ}\text{C}$
Heat transfer coefficient	0.45426	$\text{KJ}/\text{ms}^2\text{K}$

General Electric combined cycles

General Electric (GE) is one of the world's leading combined cycle producers. They have a wide range of single and multi shaft combined cycle configurations. This section will be dedicated to a discussion of the different cycle configurations (D.L. Chase, 2003).

Non-reheat single shaft:

This cycle consists of a single gas turbine with a single HRSG that generates steam for a steam turbine. The steam turbine, gas turbine as well as the generator is mounted on a single shaft (D.L. Chase, 2003).



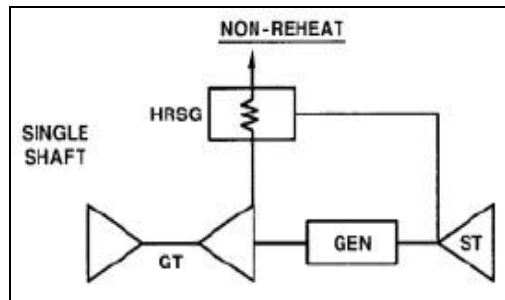


Figure 4: Non reheat single shaft combined cycle

Non-reheat multi shaft:

The following cycle is the Non-reheated multi shaft combined cycle. It consists of two gas turbines that are used to drive the two separate generators to produce electricity. The exhaust gas is then used to boil water in the HRSG to produce the steam needed for the steam turbine to produce enough power for the generator that is coupled to the turbine.

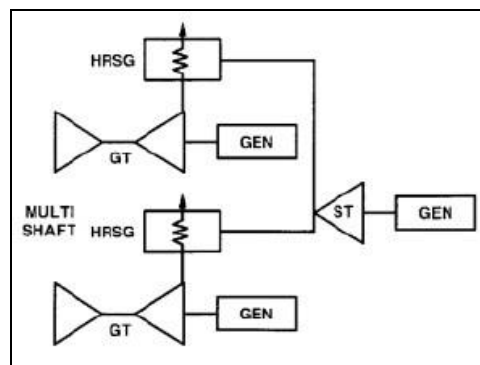


Figure 5: Non-reheat multi shaft combined cycle (Chase 2003)

Reheat single shaft:

In this cycle much like the non-reheat single-shaft cycle the gas and steam turbine as well as the generator are mounted on a single shaft. The only difference is the reheat cycle that is introduced.

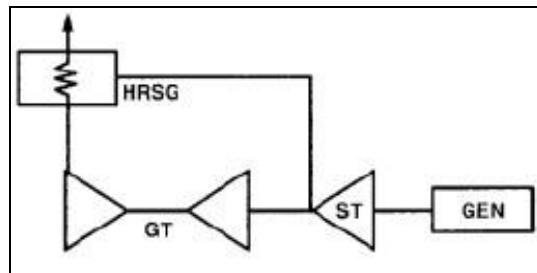


Figure 6: Single shaft cycle (Chase 2003)

Multi shaft:

This cycle has two different generators on two different shafts. The gas turbine drives the first generator and the exhaust gas is then used to generate steam in the HRSG. This steam is then used to power the steam turbine to drive the generator.

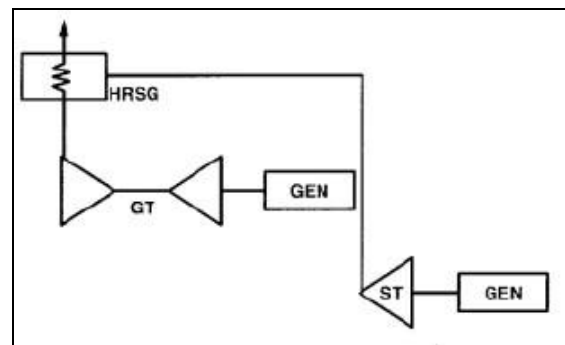


Figure 7: Multi shaft cycle (Chase 2003)

There is another variation on combined cycle; the pressure of the working fluid may be a single or a multi-pressure. The different pressures are used mainly in multi-shaft cycles. The next cycle is a multi-pressure reheat cycle.

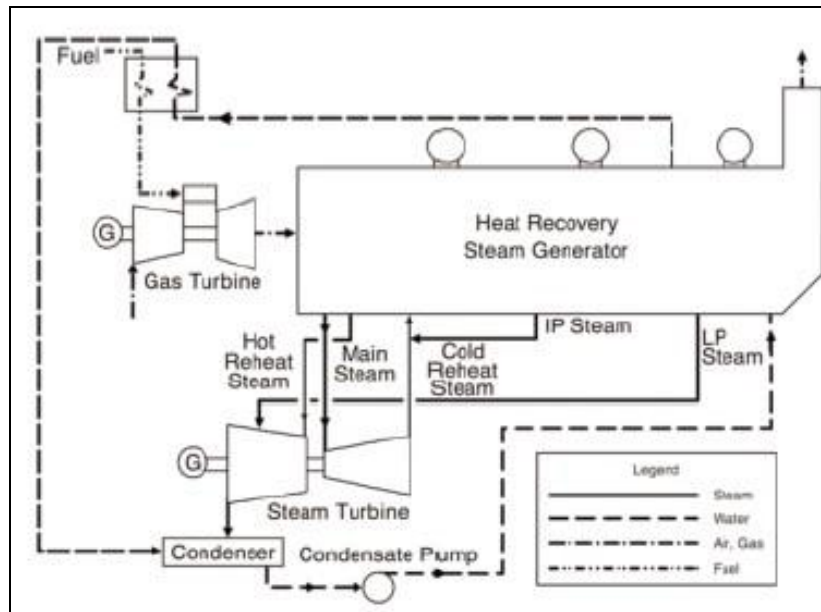


Figure 8: Multi pressure reheat cycle (Chase 2003)

The fuel is pre-heated using the water that is pumped through the system. It is then ignited and the power produced in the gas turbine is used to drive a generator. The waste heat from the gas turbine is then used to heat the HRSG. The steam is generated and expanded through the high pressure steam turbine. The steam is then reheated and expanded through the intermediate pressure turbine. The low pressure steam is then heated again to heat the fuel that enters the gas turbine.

The HRSG is a complex piece of equipment and the flow cycles within the HRSG have to be carefully planned to be as efficient as possible. The flow inside the HRSG of this triple pressure cycle can be seen in the figure below:

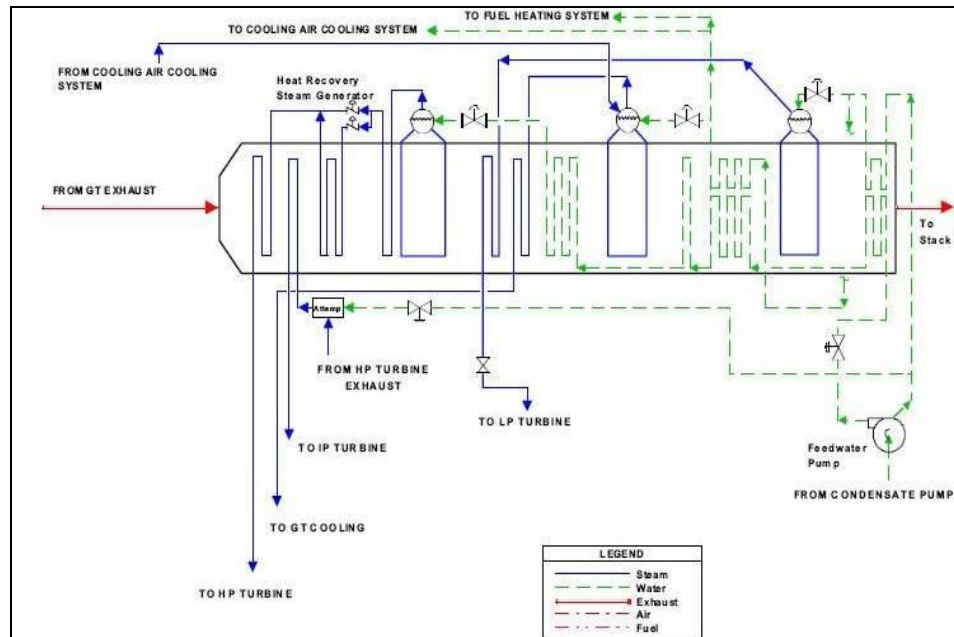


Figure 9: HRSG of a triple pressure cycle (Chase 2003)

As can be seen, the design of the HRSG is critical. The design of the HRSG is a complex task that evolves the design of the basic type of heat exchanger as well as the order of the heat exchanger for super-heating re-heating and such.

The next section will explain the basic design parameters of the HRSG for the new generation PBMR. This differs considerably from HRSG systems above because the helium of the PBMR has to be returned to the reactor at a specific temperature.

4 CHAPTER 4: CLOSED BRAYTON CYCLE

4.1 INTRODUCTION

The focus of this chapter is to investigate the different types of closed Brayton cycles (CBC) for use with the PBMR-DPP. Optimisation techniques for these cycles have to be determined so that they function optimally with the combined cycle (CC).

There are many different permutations of among others turbo machines, and heat exchangers that can be used to form a CBC. Each cycle has advantages and disadvantages. The aim is to find the cycle with the most advantages for the specific application of a CC.

4.2 Closed Brayton cycles

4.2.1 Simple closed Brayton cycle

The simple CBC consists of only one compressor and turbine to deliver work (single pressure cycle). In a simple cycle, permutations are limited to two different cycles. The first of these cycles and the simplest cycle; consists of a compressor, PBMR, turbine and HRSG. The heat energy not used in the turbine is rejected to the Rankine cycle via the HRSG.

The second permutation is one containing a heat exchanger to increase the CBC's efficiency. The heat exchanger uses the heat from the turbine exhaust to pre-heat the gas before it enters the PBMR. This improves the cycle efficiency because less heat has to be added to the gas by the reactor. It also decreases the heat rejected to the Rankine cycle for steam generation, but it would reduce the specific work output (from the theory optimum pressure ratio (r_p))

A cycle layout can be seen below (ideal cycle).

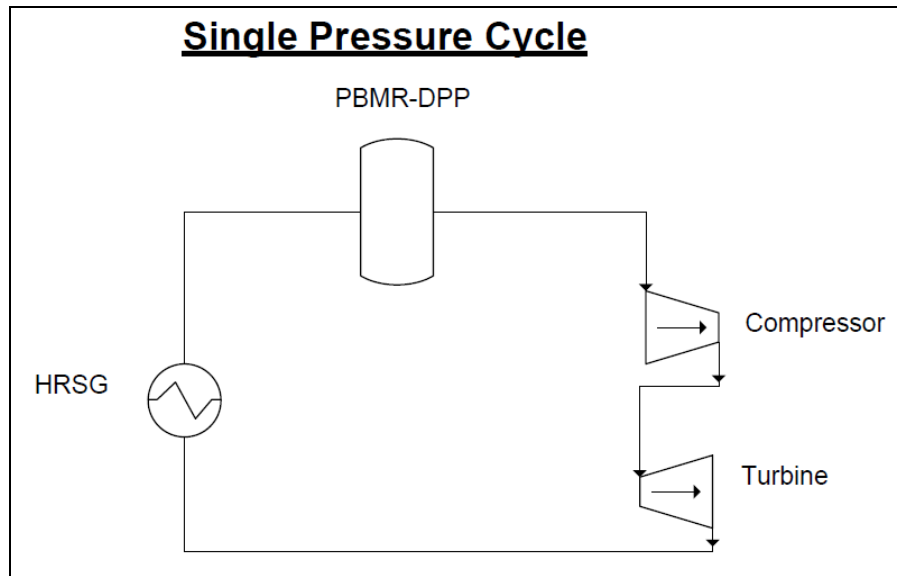


Figure 10: Simple cycle flow diagram

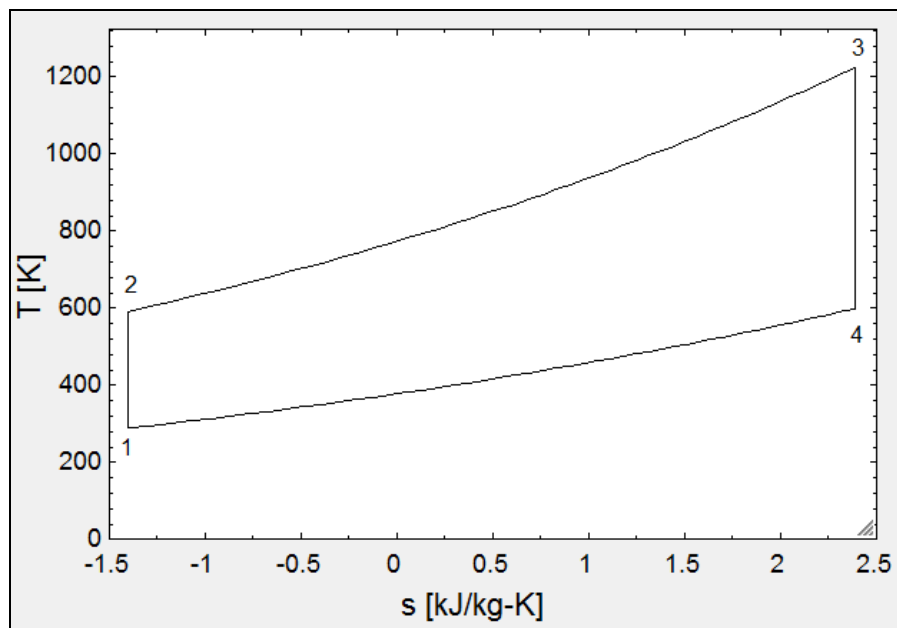


Figure 11 T-s diagram of the simple cycle

The compressor takes suction from point 1 and delivers the gas to point 2 at the maximum cycle pressure. The temperature at point 2 is critical for this ideal cycle. This temperature and the temperature at point 4 have to be identical to allow the



cycle to produce maximum work. Maximum work is produced in the simple cycle if there is no opportunity for heat exchanging.

Point 3 is defined by the maximum possible cycle temperature determined by the maximum temperature that can be delivered by the PBMR – DPP. In an ideal cycle, for more work output there is no opportunity for heat exchange due to the fact that the temperature at points 2 and 4 are the same.

The energy input into the cycle is both heat and mechanical energy. The heat energy is used to heat the gas from point 2 to 3 and, the mechanical energy is used to turn the shaft of the compressor in order to compress the gas. The energy extracted from the CBC is in the first case the work produced by the turbine, secondary the heat rejected to the Rankine cycle via the HRSG. This ensures that the cycle is balanced according to the First Law mass-energy balance.

4.2.2 Dual pressure closed Brayton cycle

The dual pressure cycle has the same two permutations as the simple cycle, the two permutations being one cycle with a heat exchanger and the other without one. The main difference in the two cycles is the mode of compressing the gas.

The dual-pressure CBC has a compression stage that has been split into two sections with a heat exchanger between the compression stages. The dual pressure cycle provides an improvement on specific work output. If the gas is cooled to the inlet temperature of the low-pressure compressor it can be shown that maximum specific work is produced when the pressure ratios of the two compressors are equal (HIH Saravanamutto, 2001). The dual pressure cycle is thus considered due to the high specific work produced.

The flow and T-s diagram of the ideal dual-pressure cycle as well as the flow diagram can be seen in the figure below:

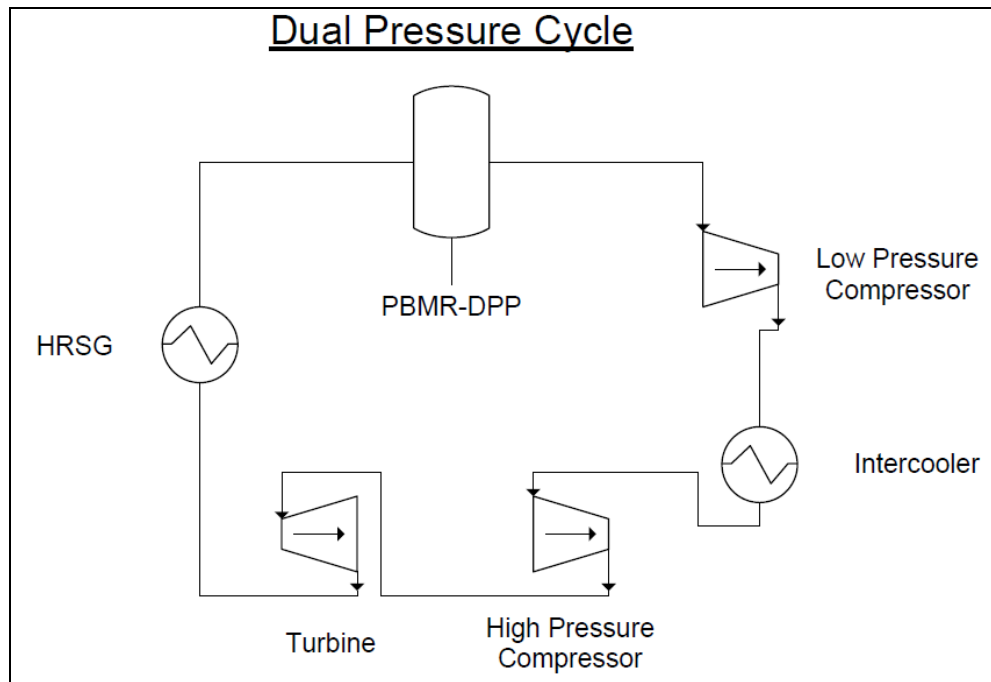


Figure 12: Dual pressure cycle flow diagram

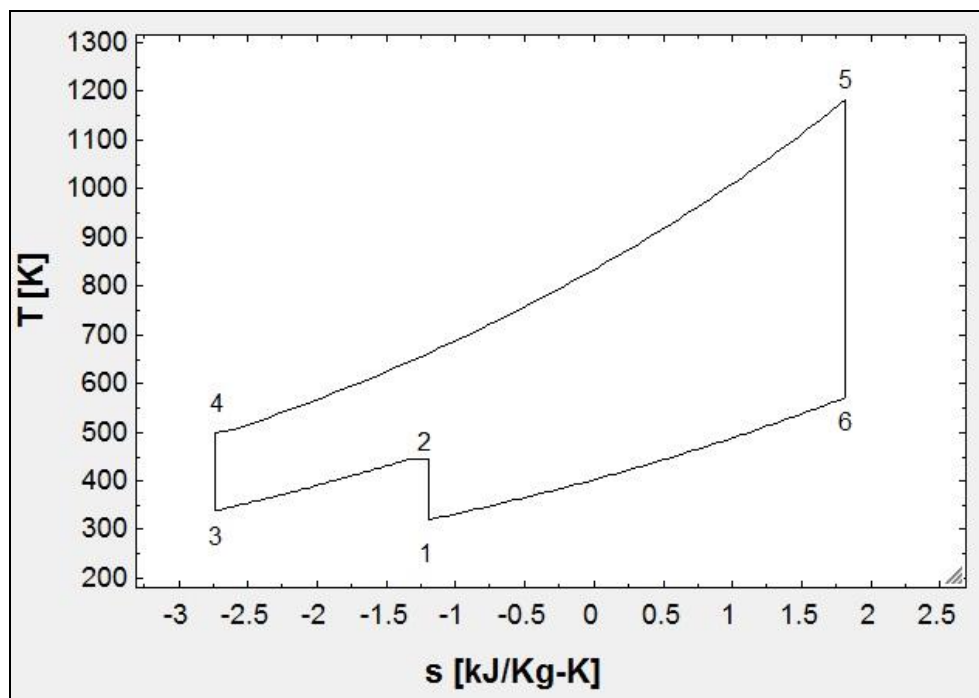


Figure 13: Ideal dual-pressure cycle

The first compression stage takes suction at point 1 and compresses the gas to point 2. The ideal heat exchanger then cools the gas at a constant pressure to point 3 where T_3 is the same as T_1 . The final compression stage then raises the gas to its maximum pressure and a lower temperature than in the simple cycle.

As in the ideal simple CBC there is no opportunity for heat exchange. In this case temperatures T_2 , T_4 and T_6 are all equal. Both compressors have to do an equal amount of work for minimum work and the turbine expands the gas to the inlet pressure of the low pressure compressor.

The heat rejected after the expansion of the gas-turbine (between point 6 and point 1) is rejected to the Rankine cycle via an HRSG. The heat rejected between point 2 and point 3 is usually not rejected to the Rankine cycle but to the heat sink (atmosphere), to keep the HRSG as a single-pressure design. Thus, the HRSG design would result in two parallel heat exchangers, which is usually not economically viable due to the relatively small amount of heat rejected by the intercooler.

4.2.3 Dual-pressure single reheat cycle

This cycle is very similar to the dual-pressure cycle with the only exception that the gas from the turbine exhaust is reheated and expanded through another turbine to increase the work produced. The gas that passes through the reheat and the low-pressure turbine is at a much lower pressure than in the high pressure turbine.

If the high- and low-pressure gas has to be heated in the same reactor, it would result in a double input into the cycle and a single output (two reactors of different size). The pressure leaving the PBMR-DPP would be at a lower mean pressure and temperature. The only way to overcome this problem is to use a separate reactor to reheat the gas that exits in the turbine outlet.

4.2.4 Optimisation



A primary design parameter has to be identified for this CBC. The primary design parameter for any cycle is usually maximum cycle efficiency (the international trend); the reason is due to economical factors behind the CBC. The main operating cost of a Brayton cycle is usually natural gas to fire the turbine, but in the case of the PBMR–MODUL reactor fuel is not the main economic concern. Thus, the higher the efficiency of the CBC the better the cycle makes use of the fuel, and the less fuel is used to produce a specific amount of work.

The primary design parameter of the combined cycle chosen here is not maximum efficiency but maximum specific work, the reason being that the CBC is not the last line of defense for the combined cycle efficiency. The heat that is not used to produce work in the CBC is transferred to the Rankine cycle via an HRSG, where it can be converted to work in the steam turbine.

One of the most important parameters is the maximum operating temperature of the PBMR-DPP. The helium cooling the reactor is limited to 950°C (1223.15 K) due to metallurgical factors. The minimum and maximum cycle temperatures determine the Carnot efficiency of the cycle. The higher the Carnot efficiency of the cycle, the higher the theoretical maximum cycle efficiency will be. Thus the larger the temperature difference in the cycle the more advantageous it will be for the cycle. The total work output will also increase proportionally.

The compressor compresses the gas from this minimum temperature to a specific pressure determined by the maximum allowable cycle pressure. In this compression stage heat is added to the gas because, the compressor outlet temperature has to meet the specific minimum reactor inlet temperature. If the reactor inlet temperature is not met, the gas will not be heated to the maximum temperature that the reactor can achieve. This will result in a lower Carnot efficiency, as well as a lower CBC efficiency. Less work will also be done at a different optimum pressure ratio.

From the turbine outlet the gas is cooled via the HRSG as the heat is rejected to the Rankine cycle. The theoretical optimum is to cool the gas close to the absolute zero point of temperature to extract all the possible energy contained in the gas.



If the gas is cooled to lower than the ambient temperature the atmosphere will start to exchange heat with the helium. Thus, the practical limit for the minimum gas temperature is a few degrees above the ambient temperature.

The work done and work required in the cycle as well as the heat added and rejected from the gas is dependent on the specific heat at constant pressure (C_p) of the gas. The higher the value of the specific heat the easier the gas exchanges heat, although the value of the constant heat for most gas change with temperature. In the case of the PBMR-DPP that is cooled by helium, this is not the case. Because helium is a perfect gas, the variation in specific heat is zero, right through the temperature scale. Thus for a CBC, a constant value of specific heat (constant pressure and constant volume) may be used.

When considering the above, the CBC must be optimised within the limitations set to it by the PBMR-MODUL and the optimising criteria for the combined cycle for maximum work.

4.2.5 Choice of a simple CBC for the CC

The number of permutations has to be limited as far as possible to ensure that the correct cycles are optimised and compared. Some of the above mentioned cycles will prove more efficient, while will produce more work.

The choice of cycle is between whether to use a heat exchanger or not. In both cases the heat rejected to the Rankine cycle is reduced by the use of a heat exchanger. On the other hand the cycle efficiency is increased by the introduction of the heat exchanger. The work produced in the cycle remains unchanged in the case of the dual-pressure cycle.

The choice whether to use a heat exchanger has therefore to be made if the combined cycle were viewed. If the Rankine cycle operates more efficiently than the CBC, it would be better not to use a heat exchanger. The opposite is also true if the CBC operates more efficiently before a heat exchanger is added, then the heat should be utilised in the CBC.



In the case where the Rankine cycle operates more efficiently, it could be argued that that heat has to be rejected to the Rankine cycle and the CBC would effectively be useless. This will not be the case because the PBMR-MODUL has energy available at a high temperature which is utilised by the Brayton cycle that cannot be utilised by the Rankine.

4.2.6 Conclusion

If the preliminary assumption is made that the Rankine cycle operates at a higher efficiency than the CBC, the choice is only between the simple and dual pressure CBC. The simple cycle and the dual pressure cycle will be investigated in the next chapter to determine which would suit the combined cycle the best.



5 CHAPTER 5: CLOSED BRAYTON CYCLE DESIGN RESULTS.

5.1 Introduction

The aim of this chapter is to determine the optimum CBC to be used with the combined cycle. Various aspects of the CBC will have to be investigated such as the different losses in the cycle and work produced compared to the heat rejected from the cycle.

5.2 Simple closed Brayton cycle

5.2.1 Solution methodology

To solve the set of equations that represents the CBC reliability accurately, a programme is necessary that is suited to the problem. The solutions have to be verified to ensure that the equations has been solved correctly.

The primary programme used to solve the CBC is Engineering Equation Solver (EES). This programme produces reliable results of a system of equations and also allows a graphical representation of the CBC in the form of a T-s diagram. The solutions are generated by solving a set of equations simultaneously.

A solution methodology for solving a CBC has to be generated to ensure that the best answer is gained with the combined cycle kept in mind. The traditional sequential solution of this CBC will be adequate in this situation. The sequential solution as well as all the input parameters to the cycle can now be discussed.

5.2.2 Input parameters

The input parameters for the cycle have to be as limited as possible. The more input parameters that can be fixed, the more room there is for optimisation of the cycle. The input parameters will now be mentioned but will be fully explained when the cycle limitations are discussed. The three input parameters are:



$$\begin{aligned}T_{\max} &= 1223.15K \\P_{\max} &= 6000kPa \\Q_{In} &= 200MWt\end{aligned}$$

To implement these input parameters, the maximum temperature and pressure has to be assigned to a specific point in the cycle. The maximum pressure occurs at the compressor outlet and the maximum temperature occurs at the reactor outlet (or turbine inlet). From these parameters the ideal cycle can now be solved sequentially.

The pressure at the first point can be solved by stating that this pressure present is the maximum pressure divided by the pressure ratio of the cycle. The temperature at point 1 is a very important parameter. This temperature in the combined cycle will result where the feed-heating of the Rankine cycle will be applied as well as to where the steam must be cooled for that cycle. For this ideal cycle a guess value is entered for point 1 to start the iterative process.

The pressure at point 2 is already known as the maximum cycle pressure (compressor outlet); the entropy at this point is also known. Since the cycle is ideal there is no change in entropy from point 1 to point 2. Point 3 is solved by knowing that the pressure at point 2 is equal to the pressure at point 3. The temperature at point 3 is known as the maximum cycle temperature. The ideal turbine outlet is calculated from two directions. The entropy at point 4 is the same as the entropy at point 3, while the pressure at point 1 is equal to the pressure at point 4.

Although all the points have thus been solved for the ideal cycle, the solution methodology for the practical cycle is very different. Before this process can be explained, all the component losses have to be accounted for and their effect on the CBC has to be determined.



5.2.3 Component losses

Pressure losses:

There are various pressure losses in a CBC, ranging from vary large to very small. The two main pressure losses in the CBC is the pressure loss through the nuclear reactor and the second is the loss through the HRSG.

The reason for the pressure loss through the reactor is due to the friction of the flow of gas over the pebbles, as well as the dividing and converging of tubes before and after the reactor. The loss through the specific PBMR-DPP has been calculated and verified by the experimental high temperature test unit (HTTU) facility at the North West University's Potchefstroom campus. The pressure loss is best described as a percentage of the pressure at point 2. This ensures that if there is a rise in pressure the losses increase proportionally and are not a constant value. The pressure loss in the PBMR-DPP is approximately 2% of the maximum system pressure (Kugeler, 2009).

The other larger pressure loss in the system is the pressure drop in the HRSG when the heat is being rejected to the Rankine cycle. This is a very large heat exchanger and is also not a contact heater like the PBMR-DPP. This would typically be a counter flow heat exchanger overall, but a cross -flow heat -exchanger inside, with the gas on the outside and the water/steam flowing on the inside. The heat exchanger is divided into many different stages e.g. evaporator, super heater, and economiser. The pressure loss through the HRSG is much larger than the loss through the reactor because of pipe friction and large k-factors, due to a variation in pipe diameter. The value of the pressure loss through the HRSG has been estimated at 12,5% of the turbine outlet pressure, based on the friction loss experienced in Rankine cycles and flue gas flow in fossil fired boilers (ESKOM, Lethabo)

The effect of the pressure losses may now be calculated on the ideal cycle. The pressure losses are introduced incrementally and the effect of the different parameters can be seen. The important parameters that have an effect on the combined cycle have been tabulated and the difference in the values is due to the effect of the pressure loss. For the purpose of this illustration the temperature at the compressor inlet is kept constant at 15°C and all other components are ideal (15°C is

used to act as a bench-mark to evaluate the cycles on an even footing). The effect of the pressure drop through the PBMR-DPP will first be examined. The results can be seen in the next table.

Table 3: Effect of pressure loss through the PBMR-DPP

Cycle description	Optimum pressure ratio [-]	Work delivered by the cycle [kJ/Kg]	Brayton efficiency [%]	Heat rejected [kJ/Kg]	Carnot efficiency [%]
Simple ideal cycle	6.094	1682.403	51.46	1587	76.44
1% pressure loss from compressor outlet	6.125	1669.98	51.18	1593	76.44
2% pressure loss from compressor outlet	6.156	1657.44	50.89	1599	76.44

The Brayton efficiency is defined as:

$$\eta_{Brayton} = \frac{W_{nett}}{Q_{in}}$$

And the Carnot efficiency is defined as:

$$\eta_{Carnot} = \frac{T_{max} - T_{min}}{T_{max}}$$

The only parameter that remains constant is the Carnot efficiency. This is due to the fact that the minimum and maximum temperature remains constant. The optimum pressure ratio of the cycle increases from the ideal due to the pressure loss. This happens because the compressor has to do more work to overcome the pressure



loss. The nett work produced by the cycle decreases as the pressure loss increases, this is due to the increase in pressure at the compressor outlet to overcome the pressure loss over the reactor. The CBC efficiency accordingly decreases because more compressor work is needed.

One very important parameter that increases is the heat rejected to the Rankine cycle. The temperatures at point 2 and point 4 remain the same, so the increase is not due to the possibility of introducing a heat -exchanger. The increase in heat comes from the cycle that operates less efficiently. Thus heat that was previously converted to work is now rejected as heat to the Rankine cycle.

The effect of the pressure loss through the HRSG has the same effect as the pressure loss through the reactor. This is because when the loss is much larger, the effect becomes more pronounced, although the tendencies are the same. The results can be seen in the following table.

Table 4: Effect of the pressure loss through the HRSG

Cycle description	Optimum pressure ratio [-]	Work delivered by the cycle [kJ/Kg]	Brayton efficiency [%]	Heat rejected [kJ/Kg]	Carnot efficiency [%]
Simple ideal cycle	6.094	1682.403	51.46	1587	76.44
2.5 % pressure loss from turbine exhaust	6.127	1654.104	50.75	1602	76.44
5 % pressure loss from turbine exhaust	6.252	1618.826	50.00	1619	76.44
7.5 % pressure loss from turbine exhaust	6.336	1585.513	49.23	1635	76.44
10 % pressure loss from turbine exhaust	6.424	1551.101	48.42	1653	76.44
12.5 % pressure loss from turbine exhaust	6.515	1515.522	47.57	1670	76.44
15 % pressure loss from turbine exhaust	6.610	1478.703	46.69	1689	76.44

The effect of the pressure loss can best be seen when plotted for all the parameters that are influenced. This graphical representation can be seen in the following figures:

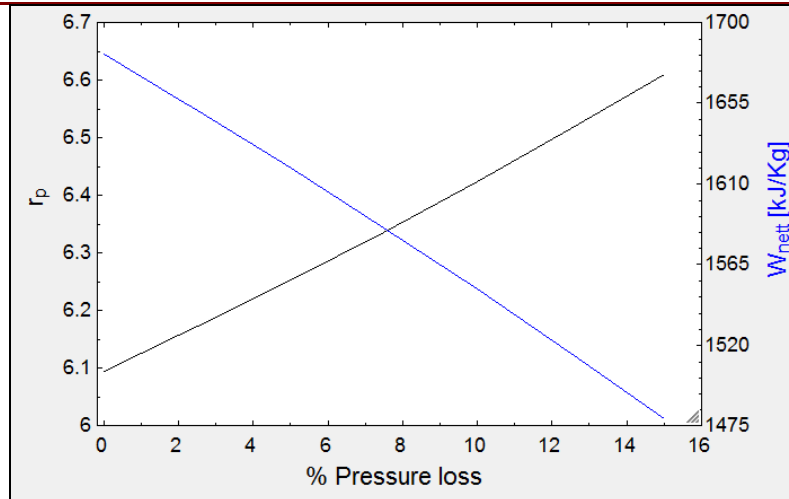


Figure 14: Effect of pressure loss on pressure ratio and work produced

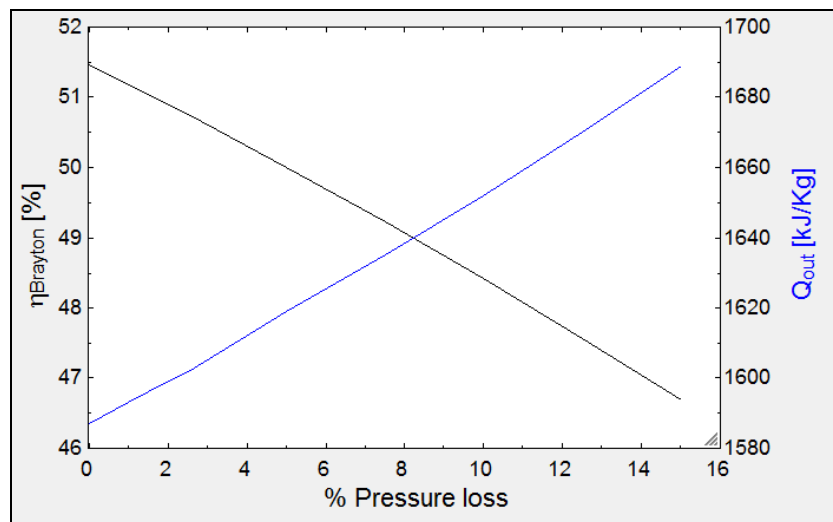


Figure 15: Effect of pressure loss on CBC efficiency and Q_{out}

As mentioned in Chapter 4, the main design criterion for the CBC is maximum work and not efficiency, as is the traditional primary design criterion. In this case, the CBC efficiency is important as well. If it is possible to find a set of circumstances where the efficiency and work are both a maximum at the same pressure ratio, this would be the optimum solution.

If the ideal cycle work has a maximum point, for a pressure ratio that rises above the critical pressure ratio, the work will start to decrease. If efficiency is considered, it can be seen that the efficiency keeps increasing as the pressure ratio increases. Thus for

the optimum pressure ratio, the maximum work must be the optimum for this CBC. This can graphically be seen in the next figure

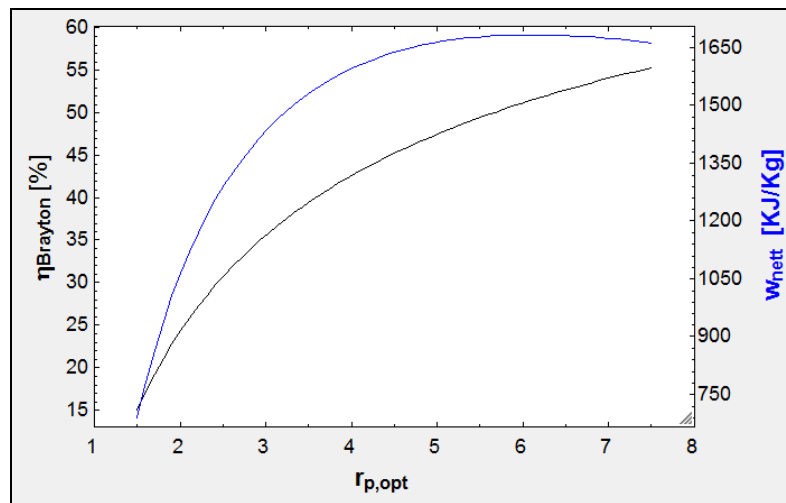


Figure 16: Effect of pressure ratio on the ideal CBC

The pressure loss in the PBMR-MODUL and in the HRSG has an effect on the curves. After this has been determined, a new optimum point of r_p may be defined. In the graph below the effect of the pressure loss can be seen. The tendencies on the graph remain the same but the graphs move to a lower work produced as well as a lower efficiency. The optimum pressure ratio where work is produced can clearly be seen. Thus for this CBC with pressure losses introduced, the optimum pressure ratio will still be the pressure ratio where maximum work is produced.

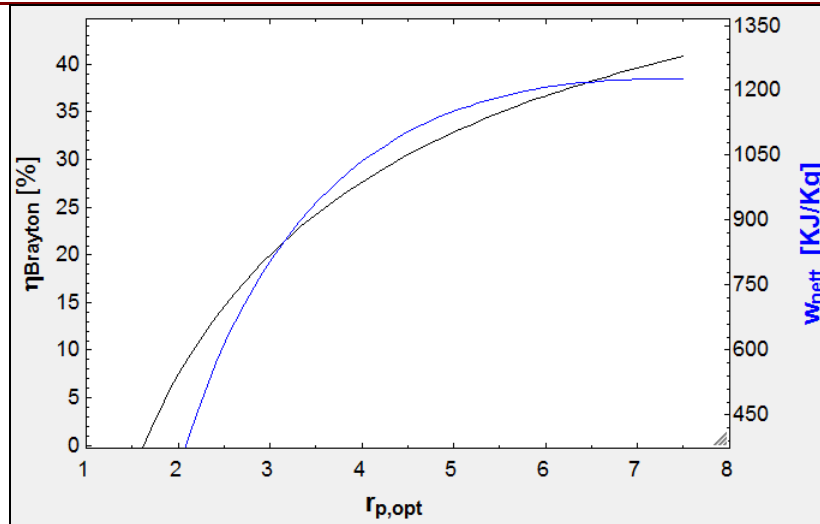


Figure 17: Work and efficiency as a function of pressure ratio.

Isentropic & Polytropic efficiency

Turbo machines such as axial flow turbines and compressors operate at very high efficiencies typically, compared to reciprocating thermodynamic machines. The efficiency of a turbo machine can be expressed as isentropic or polotrophic efficiency. Isentropic efficiency varies as the gas is compressed or expanded by the different stages of a turbo machine.

The varying isentropic efficiencies make it very difficult to use. The solution to this problem is to divide it into an infinite number of small steps each with the same isentropic efficiency. The result of this process delivers the politropic efficiency. The isentropic and politropic efficiencies differ for different processes. For compression the politropic efficiency is higher than the isentropic efficiency, and for expansion the opposite is the case.

The effect of the politropic efficiency on the work produced and the CBC efficiency achieved at different pressure ratios are much more significant than that of the pressure loss. The next table shows the effect of the polytropic efficiency on the different important parameters of the CBC.

Table 5: Effect of politropic efficiency on CBC parameters

Cycle description	Optimum pressure ratio [-]	Work delivered by the cycle [kJ/Kg]	Brayton efficiency [%]	Heat rejected [kJ/Kg]	Carnot efficiency [%]
Simple ideal cycle	6.094	1682.403	51.46	1587	76.44
97.5% Politropic efficiency	5.717	1570.549	47.70	1722	76.44
95% Politropic efficiency	5.349	1458.086	43.91	1863	76.44
92.5% Politropic efficiency	4.990	1345.396	40.12	2008	76.44
90% Politropic efficiency	4.643	1232.908	36.35	2159	76.44
87.5% Politropic efficiency	4.309	1121.107	32.64	2314	76.44

The politropic efficiency has the opposite effect on pressure ratio that the pressure losses have. The optimum pressure ratio drops dramatically as the politropic efficiency decreases. The work produced by the cycle decreases as the politropic efficiency decreases.

On the expansion side of the CBC the turbine is not able to do the same amount of work and this increase the outlet temperature of the gas at the turbine outlet. It also increases the entropy of the turbine outlet. The temperature of the cycle than has to be decreased to reduce the entropy in the cycle to close the CBC. Due to the larger amount of work needed in the compressor and less work produced in the turbine, the CBC efficiency decreases.

One positive effect of the politropic efficiency has on the cycle is that it increases the amount of heat rejected to the Rankine cycle. It also makes use of a heat exchanger possible in the CBC to increase the efficiency. The work produced and efficiency of the cycle as a function of pressure ratio may now be investigated and the effect of the decrease in politropic efficiency may be investigated.

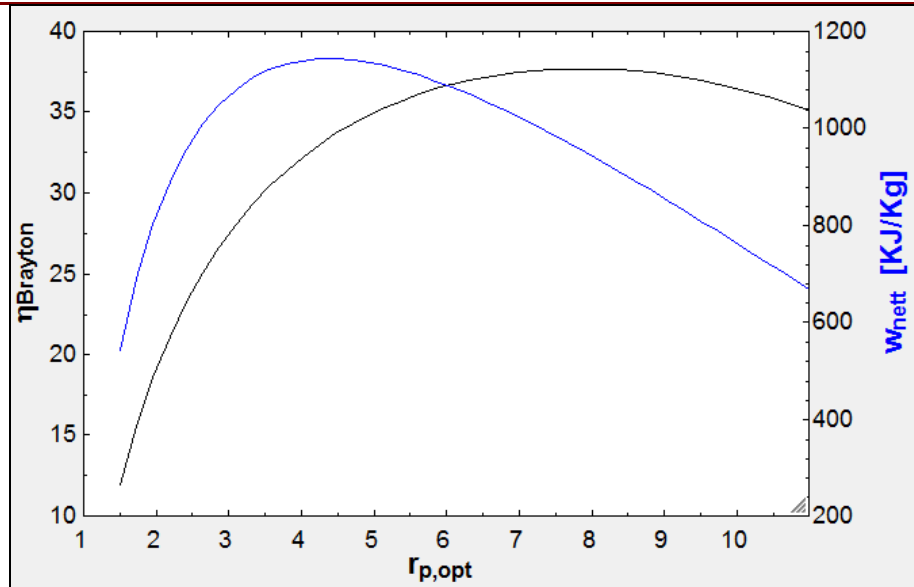


Figure 18: Effect of politropic efficiency

The most significant effect that the effect of politropic efficiency has is that the efficiency curve now also has a turning point with varying r_p . If the optimum of the efficiency curve and the optimum on the work curve can occur at the same pressure ratio the ideal would be reached. The lower the politropic efficiency is, the closer the two optimum points are to each other, an advantage for the CBC. From the ideal cycle to a cycle with a polytypic efficiency 87.5% the loss in CBC efficiency is 18.82% which is an extremely large difference in cycle efficiency.

Heat-exchanger effectiveness:

The **heat exchanger effectiveness** between the turbine outlet and the compressor inlet has a dramatic effect on the cycle efficiency. This **heat exchanger** increases the efficiency of the cycle because less heat is rejected to the Rankine cycle. This reduction in heat rejected to the Rankine cycle is very disadvantageous if the Rankine cycle has a higher efficiency than the Brayton cycle.

As stated before the heat exchanger only improves the cycle efficiency and does not affect the amount of work produced by the cycle. The effect of the heat exchanger can be seen with constant politropic efficiencies and pressure losses. The table below shows the effect of heat-exchanger effectiveness:

Table 6: Effect of ideal heat-exchanger on CBC efficiency

Cycle description	Optimum pressure ratio [-]	Work delivered by the cycle [kJ/Kg]	Brayton efficiency [%]
100% HX-effectiveness	4.974	1027.3514	39.77
97.5% HX-effectiveness	4.974	1027.3514	39.5
95% HX-effectiveness	4.974	1027.3514	39.23
92.5% HX-effectiveness	4.974	1027.3514	38.96
90% HX-effectiveness	4.974	1027.3514	38.70
87.5% HX-effectiveness	4.974	1027.3514	38.44

As was expected, the Brayton efficiency also decreases as the heat-exchanger effectiveness decreases.

Turbine blade cooling:

Turbines are only able to operate at temperatures as high as that in the PBMR-DPP if blade cooling were present. Blade cooling is achieved when helium is tapped from the compressor outlet at a relatively low temperature and high pressure. This helium is then fed through channels inside the turbine blades to cool the centre of the blade. The helium is then released at the end of the blade into the main stream of helium that is being expanded in the downstream turbine stages.

This specific mass-flow that is bled from the compressor outlet does not pass through the PBMR-DPP, which means less helium has to be heated to the maximum cycle temperature. The mass-flow being tapped from the compressor outlet is relatively constant because of a constant maximum cycle temperature.

The mass flow bled for a gas-turbine working at 950°C is approximately 5.7% of the total mass flow (Laurens, 2009). This loss decreases the work as well as the efficiency of the cycle.



Mechanical losses

A mechanical loss in the CBC is due to friction on the main shaft of the turbo machines. These may include a friction loss from the turbine to the compressor and from the turbine to the generator. Mechanical losses also include jacking oil that has to be pumped to ensure sufficient lubrication of the white metal journal bearings. A conservative value of 3% has been chosen for this loss to ensure sufficient power is available for small auxiliary systems present on the CBC.

5.2.4 CBC optimisation

The effect of losses on work and efficiency has been determined. To find the optimum r_p for the CBC, the cycle has to be optimised. The losses mentioned above are summarised in the following table:

Table 7: Table summarising losses

Loss present:	Magnitude of loss:
Pressure drop through the reactor	1.5% of compressor outlet
Pressure drop through the HRSG	12.5% of turbine exhaust
Mechanical loss	3%
Mass-flow loss due to blade cooling	5.7% of compressor outlet
Polytropic efficiency of compressor.	91% (88.59% Isentropic)
Politropic efficiency of turbine	91% (92.69% isentropic)

The only optimisation left is the pressure ratio of the cycle that has to be optimised for maximum work produced in the cycle. This is done by using an iterative procedure over the realistic boundaries of pressure ratio for the CBC. The results obtained during this optimisation will be discussed in the cycle comparison.

5.3 Dual pressure closed Brayton cycle

5.3.1 Solution methodology

This dual pressure CBC will also be solved sequentially with EES. Once again the sequential method of solving will be adequate for this cycle with the combined cycle kept in mind. The T-s diagram of the cycle may be seen in Figure 11.

The first compressor takes suction at point 1 and delivers the working fluid to a specific pressure at point 2. The pressure at point 2 is dependent on the cycle pressure ratio. The cycle pressure ratio is optimised to deliver maximum work. The first compressor does the square-root of the pressure ratio to ensure that the compressors do an equal amount of work.

From point 2 to point 3 the inter-cooling takes place. In this ideal cycle the heat exchanger has to cool the gas from the T_2 to T_1 (the same temperature as the first suction point). The gas is cooled at a constant pressure. Thus the T-s diagram follows a constant pressure line to the minimum cycle temperature.

The second high-pressure compressor (HPC) takes suction at point 3 and compresses the gas to point 4. The same amount of work is done by the second compressor as the first, although the pressure difference is larger in the second compression stage.

From point 4 to 5 the PBMR-DPP heats the gas to the maximum cycle temperature. From point 5 to point 6 the turbine expands the gas to the turbine outlet.

5.3.2 Component losses

The component losses in the dual pressure cycle are analogues to the losses in the simple cycle. In the dual pressure cycle more losses are present due to the addition of a heat-exchanger (inter cooler) and a second compressor.

Inter-cooler losses:

The losses present in the intercooler are twofold. The first loss is a pressure loss through the intercooler. This is the same type of loss as in the PBMR-DPP and the HRGS but it is an additional loss than in the simple cycle. The effect of the pressure losses present in the dual-pressure cycle over a wide variety of pressure ratios may be compared to the simple cycle.

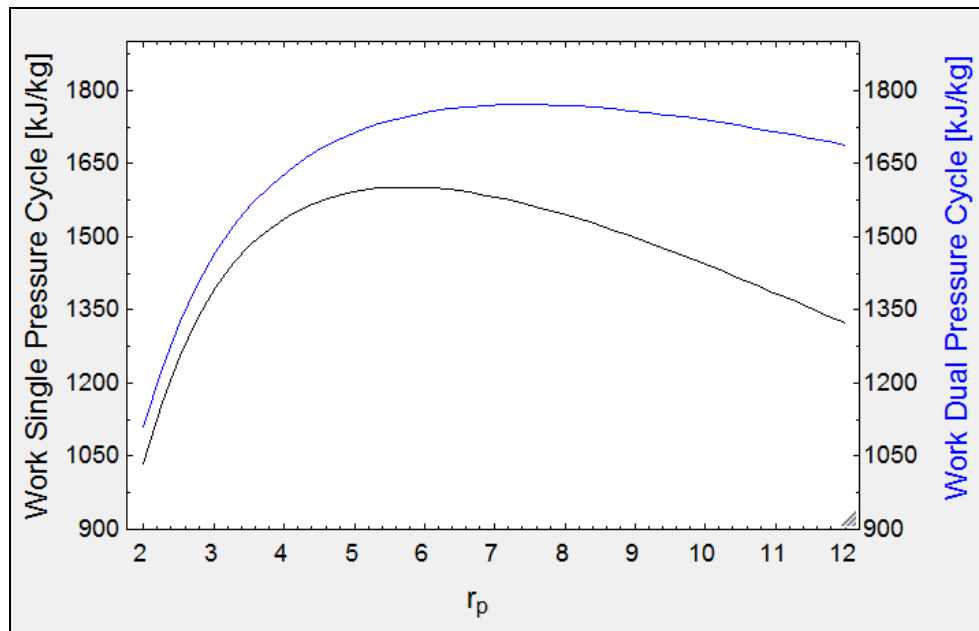


Figure 19: Work curves for both cycles

It can be seen that the dual pressure cycle outperforms the single pressure cycle if all other components were considered ideal. This is due to the higher-pressure ratio of the cycle.

The next inter-cooler loss to be considered is a temperature loss through the heat-exchanger. The minimum cycle temperature at point one is the same temperature to be achieved at point 3, although this is not realised due to the heat-exchanger efficiency present.

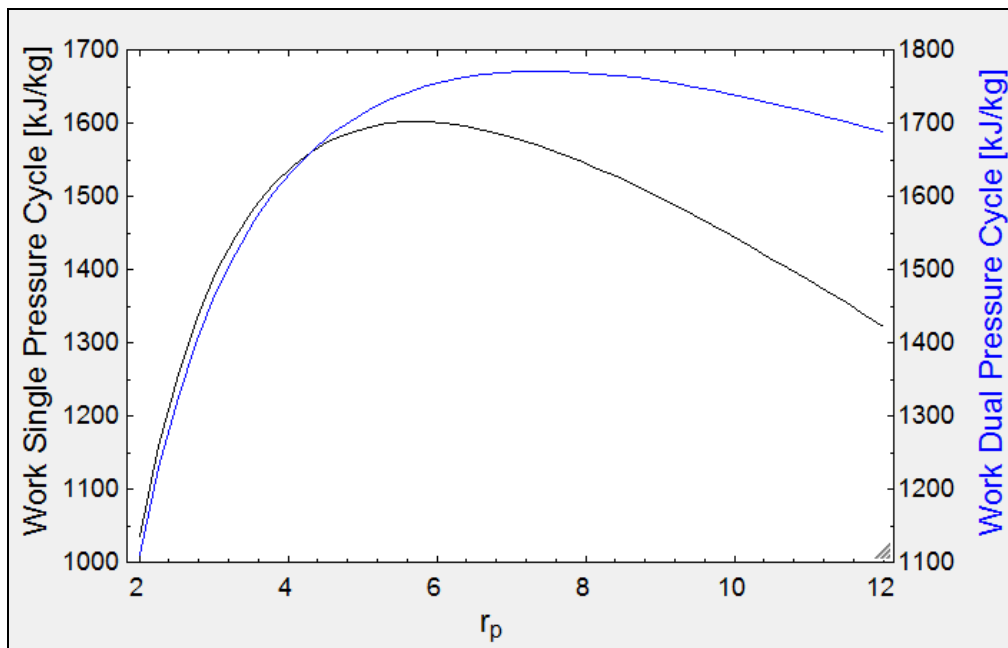


Figure 20: Work curves for all inter-cooler losses introduced

With all the inter-cooler losses introduced, the work produced at the maximum points are much closer together than before.

5.4 Cycle Comparison

5.4.1 Work considerations

These cycles may now be compared on an actual basis with all the losses introduced. The polytropic efficiencies in both cycles are the same for compressors and turbines. The dual-pressure cycle has an additional loss due to the larger number of compressors present as well as the intercooler. The figure below shows the working curves for the two different cycles.

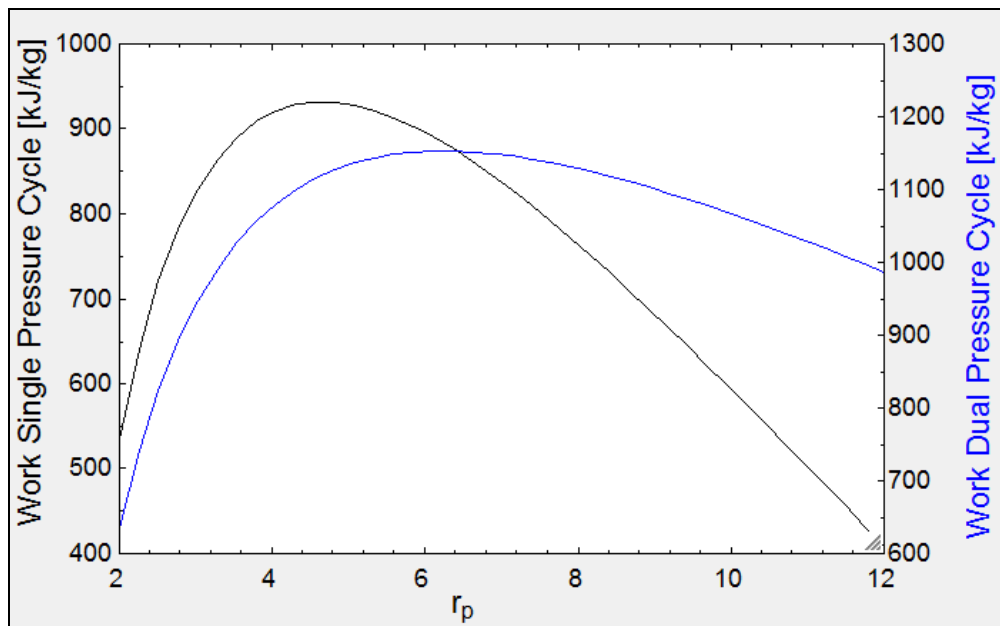


Figure 21 Working curves for both cycles

With all the losses introduced, the single-pressure cycle produces more work at the optimum pressure ratio. This is largely due to the extra pressure loss through the inter-cooler, the temperature loss through the inter cooler and the extra work required due to the larger component losses. The single-pressure cycle produces more work over a large pressure ratio range, thus making it more flexible than the dual pressure cycle for of design conditions.

5.4.2 Efficiency consideration

The efficiency curves for these cycles may now be considered. Cycle efficiency is not the main design parameter for this cycle due to the use of a Rankine cycle as a bottoming cycle. The dual-pressure cycle produces work at a higher efficiency over the first section of pressure ratios, where the maximum work is produced. Where the single pressure cycle produces more, work is at a higher efficiency.

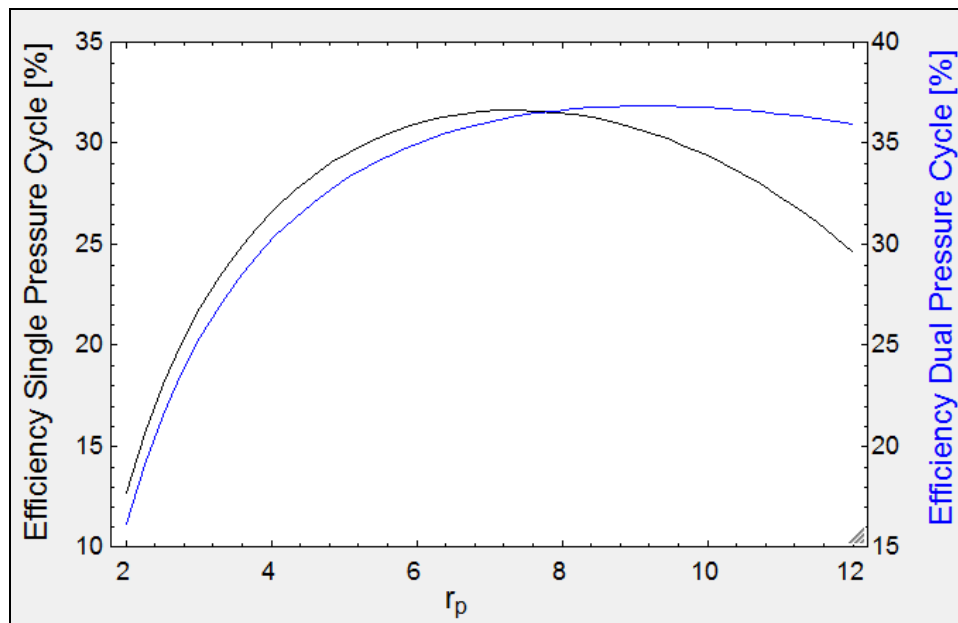


Figure 22 Efficiency curves for both cycles

5.4.3 Heat considerations

The heat rejected from the cycle to the Rankine cycle is a very important parameter, because the heat is used to produce work in the Rankine cycle. As stated in section 5.3.1 the heat rejected to the Rankine cycle by the dual pressure cycle would be less due to the heat rejected from the inter cooler and is rejected to the atmosphere. The following figure shows the heat rejected from both cycles.

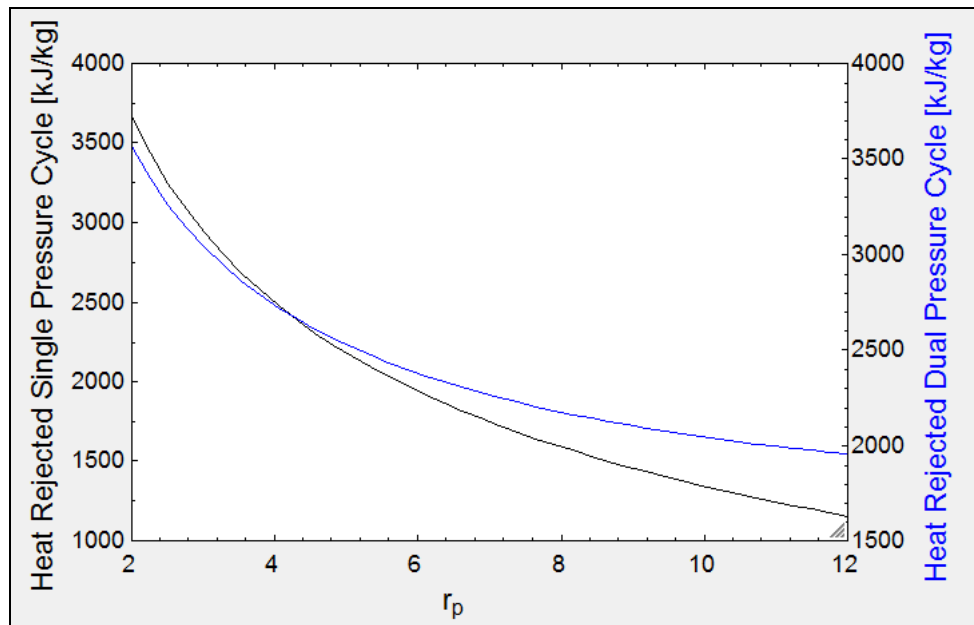


Figure 23 Heat rejected curves from both cycles

Considering the figure above it would be more advantageous to run the cycle at a lower pressure ratio of design conditions due to the higher amount of heat rejected from the cycle. The following table shows the most important parameters at optimum design conditions.

Table 8: Cycle parameter at optimum operating conditions

	Minimum cycle temperature [°C]	Optimum pressure ratio [r_p]	Maximum work produced [kJ/kg]	Brayton efficiency obtained [%]	Heat rejected [kJ/kg]
Single Pressure Cycle	25	4.719	979.399	29.77	2263.866
Dual Pressure Cycle	93	5.101	966.833	32.00	2266.160

It can clearly be seen that the single pressure cycle produces more work (main design parameter) than the dual-pressure cycle does. Power-generating plants regularly operate at off-design parameters, due to the load following conditions. The following table shows the two cycles operating at 85%.

Table 9: Cycle parameters at 85% work produced

	Minimum cycle temperature [°C]	Optimum pressure ratio [r_p]	Maximum work produced [kJ/kg]	Brayton efficiency obtained [%]	Heat rejected [kJ/kg]
Single pressure cycle	25	2.821	833.161	21.30	3049.152
Dual pressure cycle	93	2.833	821.694	24.07	2822.926

The off design conditions show that the simple cycle produces more work at a lower pressure ratio and rejects more heat to the Rankine cycle. The cycle operates at a lower efficiency than expected.

5.5 Conclusion

After a thorough investigation of the possible cycles two cycles were identified for possible use on the combined cycle. Although the dual pressure cycle operates at a higher efficiency the best cycle choice is the simple cycle. The simple cycle produces more work and rejects more heat to the Rankine cycle, at both design and off-design conditions. It would also be less expensive due to less turbo machinery, while the control system would be less complex and reliable for transient and start-up conditions.



6 CHAPTER 6: RANKINE CYCLE DEVELOPMENT AND DESIGN RESULTS

6.1 Introduction

The Rankine cycle is the most common cycle used in South Africa for power generation. The focus of this chapter will be on the different variations of the Rankine cycle, as well as the development of a Rankine cycle for the combined cycle.

The optimisation of this cycle is critical for the best possible combined cycle efficiency since, according to the First Law, the bottoming cycle should reject the least amount of heat. The design of a Rankine cycle is dependent on many different parameters. In this chapter the parameters resulting from the Brayton cycle design that will dictate the design of the Rankine cycle have to be identified. The cycle layout may then be considered with regard to feed-heating as well as the use of a steam feed pump turbine.

The losses present in the cycle have to be quantified to determine their impact as well as the effect on work produced and the cycle efficiency. If everything has been taken into account, the design results may be evaluated.

6.2 Cycle Discussion

6.2.1 Introduction

There are many variations of the Rankine cycle that may possibly be used in the CC. The basis of the Rankine cycle is a theoretical cycle called the Carnot cycle. The Carnot cycle has the maximum theoretical efficiency that may be achieved by any Rankine cycle under the same conditions. The Carnot cycle's efficiency effectively becomes the new optimum working point for the Rankine cycle. Although the Carnot cycle is the most efficient cycle possible it does not produce much work.

The Rankine cycle is developed from the Carnot cycle to the point where work can effectively be produced. The Rankine cycle, which is used with coal as fuel supply to generate power in South Africa may be seen in the following T-s Diagram.

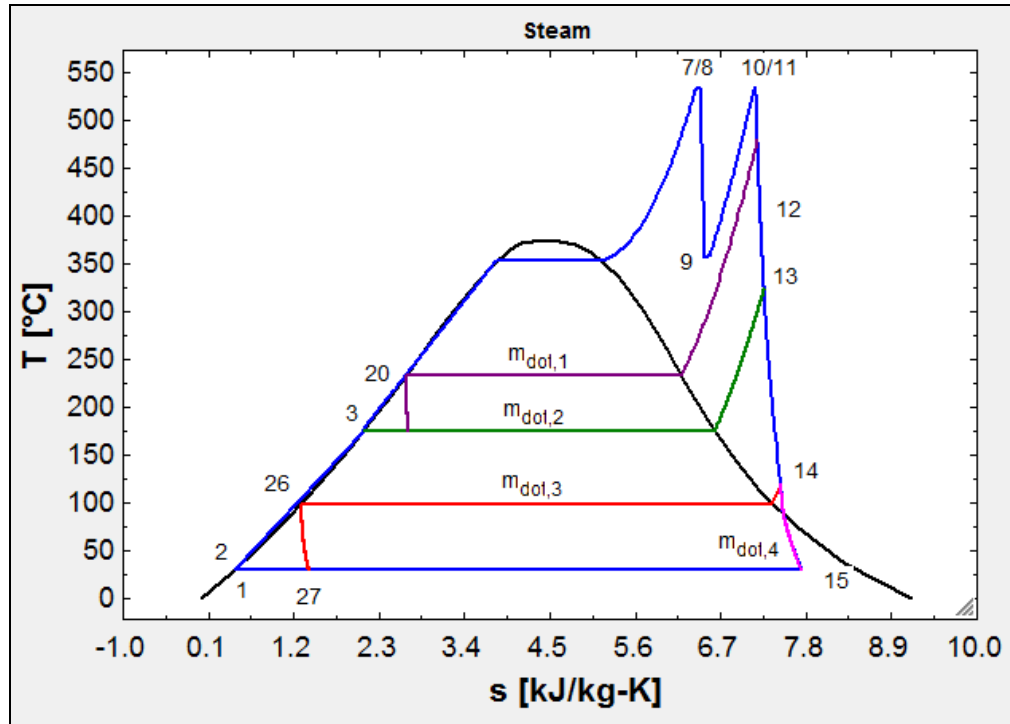


Figure 24: Rankine cycle T-s diagram

Point 1 on the Rankine cycle is saturated water at the minimum cycle pressure and temperature. In a typical Rankine cycle application (wet cooling tower) the pressure at point 1 could be as low as 4.5 kPa – 12 kPa (absolute) with a temperature close to the ambient temperature.

The second point on the Rankine cycle is after the condensate pump has raised the pressure to 600 kPa – 1200 kPa. The working fluid is compressed to a sub cooled liquid. The rise in temperature may be determined with the isentropic relation; the small increase in temperature is not even noticeable on the T-s diagram.

From point 2 to point 3 the pressure remains constant (with the exception of the pressure loss) and the temperature is raised by the low-pressure feed heaters. The feed heaters use steam extracted from the intermediate pressure and low-pressure



turbine. The less heat used to heat the working fluid, from the external heat source, the more efficiently the cycle will operate.

To raise the pressure from 800 kPa – 1200 kPa to the maximum cycle pressure of 21 MPa requires a feed pump. The reason for the second pump is due to the type of feed heater (dearrator) that is used at that temperature. The contact feed heater that is used requires de-aeration of the working fluid. Once again the temperature rise in the fluid is very small in the compression stage.

The heat rejected from the Brayton cycle heats the working fluid from point 12 to point 4 as a liquid. The heat added from point 4 to 5 changes working fluid from water to steam. The steam is super-heated from point 5 to point 6 to the maximum cycle temperature, all by rejected heat from the Brayton.

Point 6 is known as the turbine inlet, the conditions at this point are 535°C and approximately 16.1 MPa (ESKOM, Letkabo). From point 6 to 7 the high-pressure turbine (HPT) expands the steam to around 4.2 MPa. Re-heating then takes place to heat the gas to the maximum cycle temperature but at a lower pressure.

Point 8 is considered the intermediate pressure turbine (IPT) inlet; the steam is expanded from the IPT through the low pressure turbine (LPT) to the condenser inlet. Points 9, 10 and 11 are the feed-heater inputs. This brief discussion of a typical Rankine cycle is only illustrative and to compare results with the Rankine cycle used for the CC.

6.2.2 Main design parameter

The design parameters for the Rankine cycle are mostly gained from the Brayton cycle calculations. The main design input that is obtained from the Brayton cycle is the heat that is rejected as well as the maximum temperature at which the heat is rejected. The maximum temperature that will be available for the Rankine cycle is a few degrees lower than the maximum Brayton outlet temperature due to the heat-exchanger efficiency.



Traditionally, the heat-exchanger efficiency is used to calculate the outlet temperature of the working fluid after it has passed through the heat-exchanger. In this case, all the heat has to be extracted from the gas down to a specific temperature. The design of the heat exchanger has to compensate for the loss to ensure that the correct temperature is obtained. Thus, the gas and steam will have to exchange heat to a lower temperature to overcome the heat-exchanger efficiency.

A mass energy balance has to be carried out to ensure that the mass-flow in the Rankine cycle is calibrated to absorb all possible energy from the gas. Because of the high specific heat value of water the water will absorb more energy per mass unit than the gas in the Brayton cycle provides per mass unit. The result is a lower mass-flow in the Rankine cycle than in the Brayton cycle.

The main design parameter for the Rankine cycle is optimum cycle efficiency, the reason being that the heat not converted to work in the Rankine cycle is rejected to the atmosphere. The more heat rejected to the atmosphere, the larger the negative effect on the surrounding environment would be.

6.2.3 Feed heating

Feed heating reduces the heat input to the cycle by using bled-steam from the turbines to pre-heat the condensate. When considering the Rankine cycle on the T-s diagram in the previous section the first stage of feed-heating will be the feed heating taking place at the highest temperature.

This stage of feed-heating take steam at a high temperature and pressure (just below the IPT inlet) and uses this heat to raise the temperature from point 3 to point 12. The HRSG heats the gas from point 12 to the maximum cycle temperature. After the heat was used to heat the saturated liquid in a closed feed-heater, the bled steam is rejected via a constant enthalpy process to the next feed heater.

The next feed heater is a contact feed heater. Instead of using a closed heat exchanger, the bled steam is directly released into the de-aeration tank. This tank is



used to allow the air trapped in the working fluid to escape, also serving as a suction point for the main feed pump.

The third and last feed heater has an extraction point just before the working fluid is expanded to the two-phase condition. The outlet of this feed heater is to the condenser. Due to this addition of the mass flow in the condenser, the energy balance in the condenser has to be calculated accordingly.

6.2.4 Steam feed pump turbine (SFPT)

The steam feed pump may be powered in two different ways, the first using the electrical energy produced by the cycle to power electrical motors that pump the fluid. The second way to power these pumps is to make use of a separate SFPT. The turbine runs in parallel with the LPT but on a separate shaft. The shaft power is then used to power the feed pump. Using the SFPT is much more efficient due to among others the further losses in the generator and, transformer.

6.3 Cycle input parameters

The cycle input parameters dictates the design of the Rankine cycle with regard to the size of the plant as well as possible reheating that may take place. Total heat supply of the PBMR-DPP is 200 MWth and that is available for the CC. The heat is transferred from the reactor via this contact heat exchanger to the Brayton cycle. The work is then produced in the turbine and the waste heat is rejected to the Rankine cycle via the HRSG.

The 200 MWth is not the only supply of energy into the cycle. A large amount of work is needed to compress the gas from the minimum Brayton cycle pressure to the maximum cycle pressure. As no energy may be generated in the cycle an energy balance must be completed to ensure that the First Law of Thermodynamics is satisfied. Thus, all the energy that has been added to the cycle (work and heat) has to be equal to the energy extracted from the cycle.



$$\begin{aligned}E_{in} &= Q_{in} + W_{in} \\E_{out} &= Q_{out} + W_{out} \\E_{in} &= E_{out}\end{aligned}$$

In this case the value's obtained from the Brayton cycle may be substituted into the above equations to ensure that no energy is generated.

$$\begin{aligned}E_{in} &= 200 + 130.278 = 330.278MW \\E_{out} &= 162.521 + 167.757 = 330.278MW\end{aligned}$$

Thus having balanced the mass energy according to the First Law of Thermodynamics it can clearly be seen that the heat added to the Rankine cycle is 162.521 MWth. The Brayton cycle operates at extremely high temperatures, thus the temperature where the heat is available for the Rankine cycle is also high enough. The Brayton cycle turbine outlet temperature is approximately 520°C, thus the Rankine cycle may operate at temperatures as high as 500°C.

One other very important cycle input parameter is the quality of the steam on the LPT outlet. This variable cannot be fixed at a specific value, the effect of this parameter on cycle efficiency and work has to be determined. There is however a limit to the lowest values of quality for such an LPT. If the quality is too low water droplets will erode the turbine blades.

The cooling tower that cools the exhaust steam of the LPT to a saturated liquid may be a dry or wet cooling tower. The choice of cooling tower determines the condenser back pressure and temperature. The choice of cooling tower is determined by the location of the CC. If water is freely available (e.g. river, sea), the best option would be a natural circulating wet cooling tower. If this is, however, not available, a forced draft dry cooling tower can be used



6.4 Solution methodology

The cycle has to be solved with the optimisation of the cycle as consideration. The main design variation is the pressure that the system operates at. There are many different parameters that impact on the system pressure that has to be considered. The traditional sequential method of solving the Rankine cycle is not going to be adequate in this case.

The solution methodology for this cycle is to solve the cycle from the LPT outlet and then move to the LPT inlet and from there move in the reverse order to the last point, being the condenser outlet.

The reason for this unusual solving method is to ensure complete control over the quality of the steam on the LPT outlet. The maximum cycle pressure may then be calculated from the quality using the reverse solution method. By using this method the calculation of the effect of isentropic efficiencies is more difficult and results in more calculations to be carried out to obtain the same results. However the gain in control over LPT outlet conditions justifies the solution methodology.

6.5 Cycle losses

The ideal cycle has to be considered first so that the impact of each loss may be identified and quantified. This identification of the cycle losses will improve the optimisation and ensure that the cycle is fully understood.

6.5.1 Ideal cycle

The ideal cycle has been modelled and optimised. The optimisation was carried out with the cycle efficiency as the main design parameter. The parameter that was used to optimise the cycle is the quality of the steam at the LPT outlet. By changing the LPT quality it directly affects the cycle pressure. The pressure is calculated from the LPT outlet to the IPT inlet and continually in reverse order to the condenser outlet.

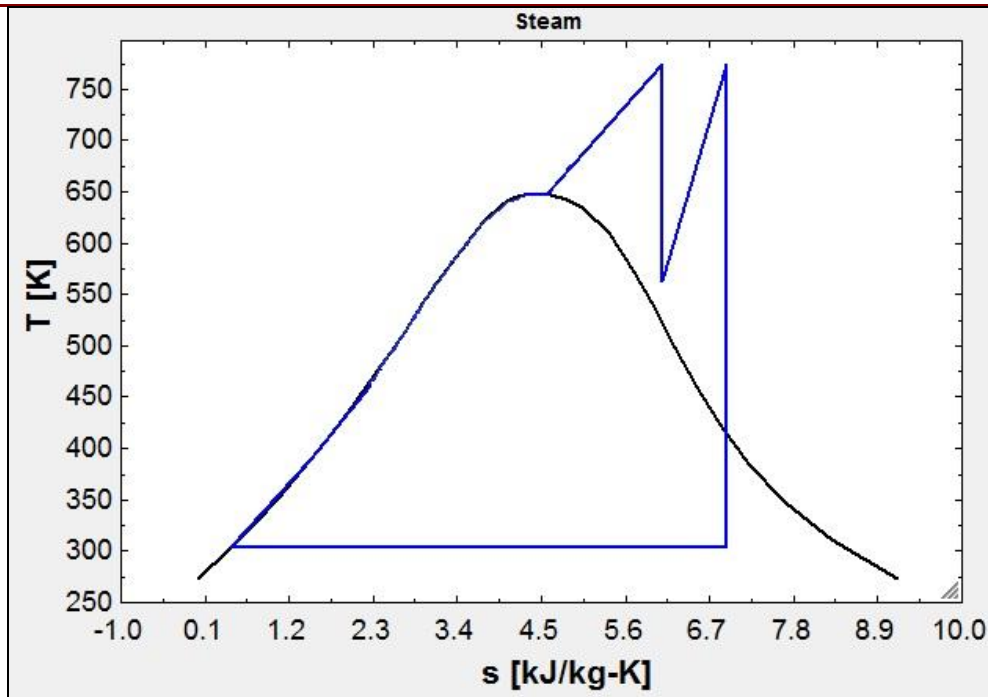


Figure 25 Ideal Rankine cycle T-s diagram

For this ideal cycle the optimization carried out resulted in the maximum pressure possible in the boiler, and effectively a very wet LPT outlet. Theoretically the more wet the steam on the turbine outlet is the higher the cycle efficiency. This is limited by the critical pressure of water to ensure that the boiler is not a super critical boiler. The effect of higher quality steam at the LPT outlet decreases the amount of work done as well as the Rankine cycle efficiency.

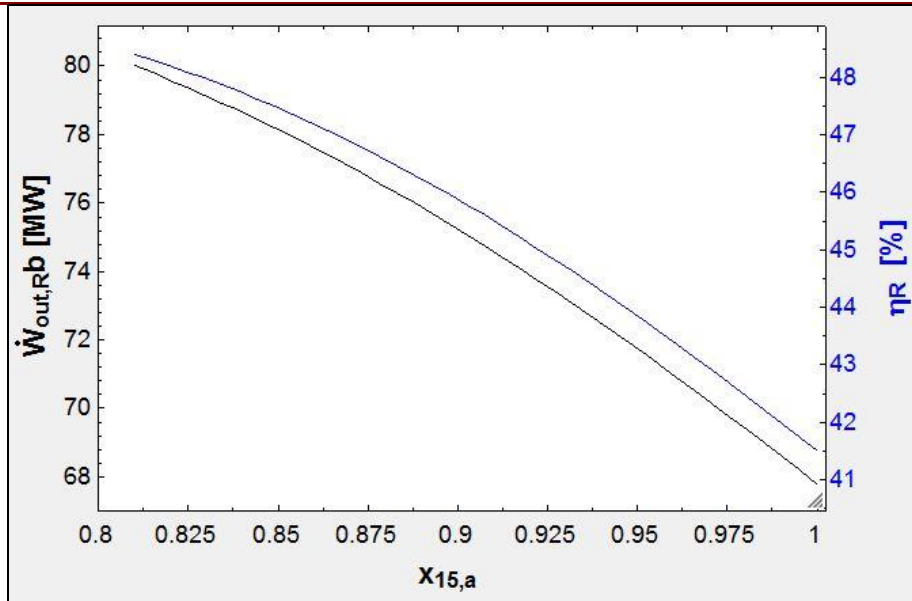


Figure 26 Effect of steam quality at the LPT outlet on work and efficiency

The following table summarises the results of the ideal cycle

Table 10: Ideal Rankine cycle Results

	LPT steam quality	Work produced [KW]	Cycle efficiency [%]
Ideal Rankine Cycle	0.81	78 666	48.40

The cycle efficiency is very high. This is due to the fact that normal losses present in a Rankine cycle have been ignored. The quality of the steam on the LPT outlet is very low and would cause damage on the last stages of the turbine.

6.5.2 Isentropic efficiencies

Isentropic efficiencies are present on all turbo machines, axial flow compressors and turbines. The efficiencies cause the compressors to require more work from the cycle to do the same amount of compression. The turbines have to make use of a higher flow rate to do the same amount of work as the ideal. The effect on the T-s diagram can clearly be seen below.

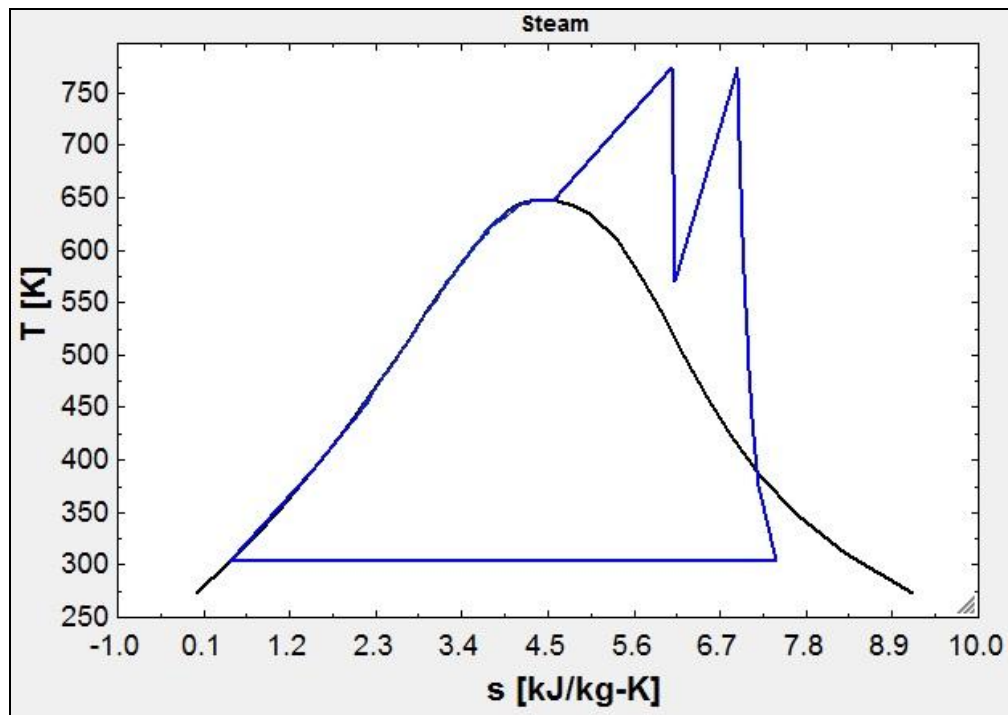


Figure 27 Rankine cycle T-s diagram with isentropic efficiencies introduced

Although the effect on the pump may not be seen the effect on the turbines is very dramatic. The slopes to the right on the turbine curves are due to the isentropic efficiencies. The quality on the LPT outlet is as small as possible to increase the maximum amount of work done and the best efficiency achieved.

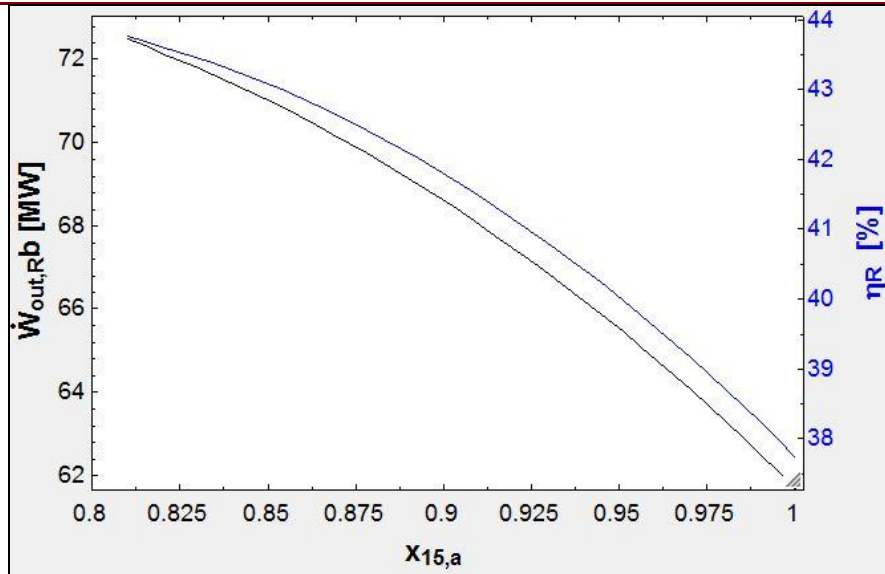


Figure 28 Effect of quality on work and cycle efficiency

From the work and efficiency curves above it may be seen that much less work is done by the Rankine cycle at a lower efficiency if the isentropic efficiencies are introduced. The tendencies of the curves are identical to the ideal cycle but at lower values. Thus the lower the quality at the LPT outlet and effectively a higher pressure can be used in the cycle, the more efficiently the cycle will operate.

6.5.3 Pressure losses

The pressure losses in the Rankine cycle have to be incorporated to model the cycle as accurately as possible. The common use of Rankine cycles in South Africa makes the acquisition of the data for pressure losses easily available (the magnitude's of the cycle parameters are show in Table 12). The pressure losses in the cycle are expressed as a percentage of the inlet pressure. This is useful becuase if the pressure in the cycle decreases the pressure losses remain in proportion to the cycle pressure.

The main pressure losses in the cycle are present in the heating of the saturated liquid, the super heater as well as the re-heater. The losses are mainly due to the friction coefficient in the pipes of the heat exchangers. The design of the HRSG heat exchanger will be similar to the design of a traditional coal-fired power station heat exchanger.

Shown from calculations, the effect of the pressure losses is an increase of steam quality at the LPT outlet, effectively lowering the maximum cycle pressure. The smallest quality that is possible for the ideal cycle is 0.81, but if pressure losses were introduced, the minimum quality moves to 0.823. The optimisation delivers a cycle pressure that is reduced from 21.995 MPa to a more acceptable 19.774 MPa (this was shown in calculations attached in section 9.1).

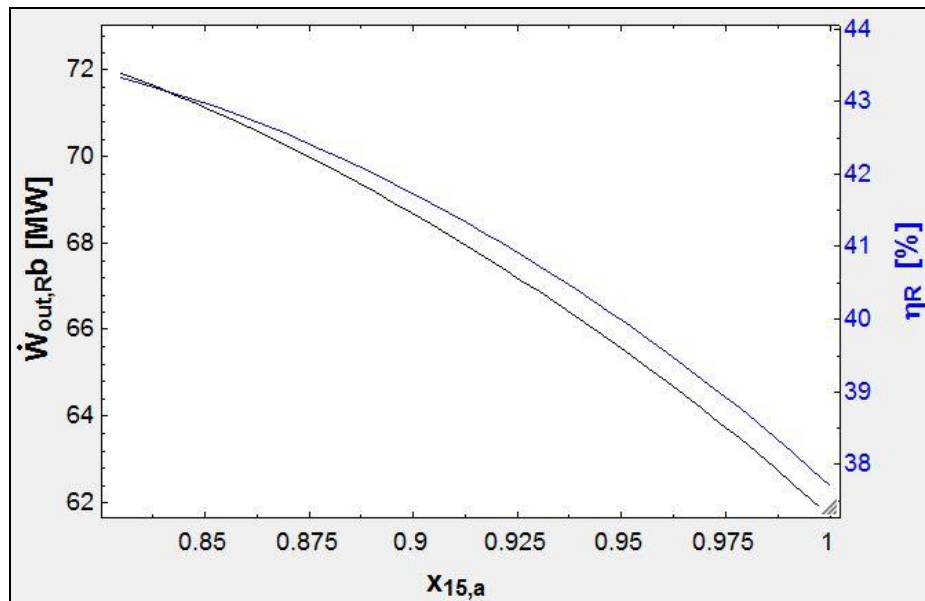


Figure 29 Cycle performance curve for all current losses introduced

The figure above shows the Rankine cycle with the losses introduced that have been covered. It can still be seen that with the losses introduced, the lower the quality on the LPT outlet, the more work is produced.

6.5.4 Turbine valve losses

The steam that was heated by the HRSG has to flow at high pressure and temperature to the turbines. Before the steam enters the turbine it passes through control valves and emergency stop valves (ESV), to choke the flow from the boiler if needed. Even if the valve were fully open, there is a pressure drop over the valves (shown in Table 12). This pressure drop reduces the pressure that the turbine

should work at. This pressure has the same effect as the pressure losses through the heat exchangers.

6.5.5 Steam Attenuation

Steam attenuation is a very important parameter in the Rankine cycle. Saturated water is tapped from the main feed pump at high pressure and low temperature. This water bypasses the heating process and is relayed to the inlet of the super-heater and the re-heater. The water is used to protect the super heater inlet and to maintain good control over the maximum steam temperature (a very important parameter).

Although the water is added to the steam before the super heater and the re-heater the temperature measurement is carried out after the heat transfer has taken place. This is done to ensure that incipient metal temperature excursions do not occur in the heat exchangers and steam temperatures are accurately maintained.

6.6 Cycle design results

The Rankine cycle proposed for the combined cycle can be seen in the figure below:

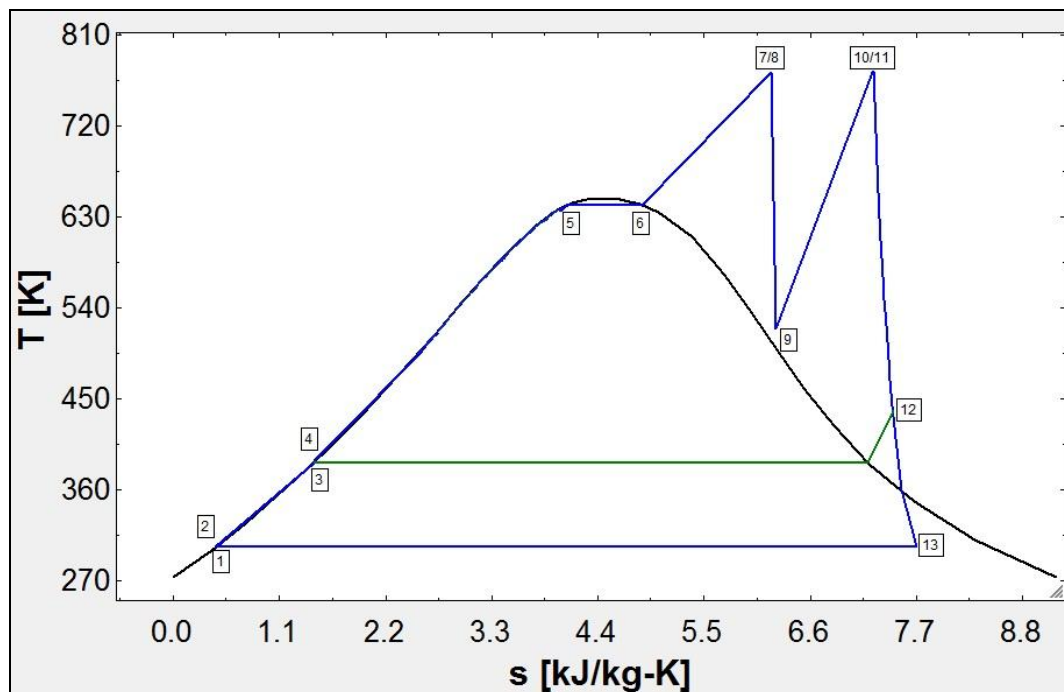


Figure 30: Proposed Rankine cycle

If the temperature of point one of the cycle were considered (condenser outlet) in optimisation the boundaries are very limited. The temperature depends on the type of cooling tower that will be used with the Rankine cycle. The following table summarises the results of work and efficiency achieved by the cycle with a variation in the condenser outlet temperature.

Table 11: Effect of condenser outlet temperature

Condenser outlet temperature [°C]	Optimum quality at LPT outlet	Work produced by the cycle. [MW]	Cycle efficiency achieved [%]
20	0.8717	74.2	44.71
25	0.8747	73.29	44.11
30	0.8826	72.19	43.43
35	0.8904	71.08	42.75
40	0.8981	69.98	42.07

The table above clearly show that as the condenser outlet temperature decreases both work and efficiency increases. The choice of cooling tower is then motivated by the cooling tower that produces the lowest possible condenser outlet temperature. This would be a wet cooling tower. If a large water source is available such as a river or the ocean a realistic condenser outlet temperature is 30°C.

From point 1 the saturated water is pumped with a condensate pump to a specific pressure at point 2. The pressure at point 2 has to be carefully chosen because it determines the pressure at which the steam is tapped from the IPT for the contact feed heater. The reason is that the contact feed heater has to inject steam into the de-aeration tank where the pressure of the steam entering has to be larger than the pressure in the tank.

The feed heater has to heat the saturated liquid from the condensate pump delivery point to the main feed pump suction point. Thus with the use of the First Law of Thermodynamics an energy balance can be used to determine the mass flow needed in the feed-heater (dearator). The pressure at which the saturated liquid is delivered

after the main feed pump is determined by the maximum cycle pressure, which in turn is determined by the optimisation of the quality of the steam at the LPT outlet.

The next very important point is the temperature at point 9. The HPT has to do as much work as possible to ensure maximum cycle efficiency, but if the temperature were to fall too low, water-logging in the re heater will occur. To avoid water-logging, the temperature must be kept high enough to ensure that the working fluid does not change phase.

The losses introduced into the cycle have a dramatic effect on the performance of the cycle. These parameters have to be chosen very carefully to ensure that the cycle operates as closely as possible to the realistic values. The table below shows all these parameters (ESKOM, Duvha)

Table 12: Magnitudes of cycle parameters

Input parameter	Value
$T_{\min}$	30 [°C]
$T_{\max}$	499.85 [°C]
ΔT_{Pinch}	30 [°C]
ΔP_{TIV}	100 [kPa]
$\eta_{\text{IsentropicHP}}$	92.5 [%]
$\eta_{\text{Isentropic IP/LP}}$	89 [%]
ΔP_{RHRT}	10.5 [%]
ΔP_{SHRT}	11.5 [%]
ΔP_{SL}	22.1 [%]
η_{BFP}	85 [%]

From the Brayton cycle design, the temperature available for the Rankine cycle is approximately 530°C. There is, however, a temperature drop of 30°C through the heat exchanger, thus the maximum temperature available is 500°C.

The isentropic efficiency of the turbine is divided into two main groups; the first is the HPT efficiency which is much higher than the IPT and the LPT. The reason for the difference in efficiencies is due to the pressure and temperature that the turbines operate at. The higher the temperature and pressure the more effective the turbine will operate, (The HP handles no wet steam, as the LP does).

From point 4 to point 6 in the cycle there is a 22.1% drop in pressure due to pipe friction as well as potential head. The super heating phase from point 6 to 7 has a pressure loss of 11.5%, with a further 100 kPa pressure loss from point 7 to point 8 over the turbine inlet valve. The final pressure loss occurs in the re heater. A total of 10.5% of the pressure is lost in the re heater.

With all the input parameters fixed, only the steam quality on the LPT exhaust remains to be optimised. The design results may now be obtained. The following table shows the important design results obtained from the Rankine cycle.

Table 13: Rankine design results

Cycle parameter	Value
x_{LPT}	0.8826
W_{In}	1.605 [MW]
W_{Nett}	70.584 [MW]
T_{HPT}	290.65 [°C]
$\eta_{Rankine}$	43.43 [%]
Q_{Out}	91.937 [MW]
$\dot{m}$	49.54 [Kg/s]

The first value mentioned in the table is a result of optimisation. The steam quality on the LPT outlet is calculated to be 0.8826. This value is very similar to the practical results obtained in the actual Rankine cycle in a coal-fired power station (ESKOM, Lethabo). The work required to power the feed-pumps is 1.605 MW, which is done by the steam-feed pump turbine. This relatively low value is due to the cycle mass flow of approximately 50 kg/s.



The nett work produced in the cycle is 70.5 MW at a cycle efficiency of 43.4%. This high efficiency is due to the optimisation of the cycle pressure, as well as the small amount of power required in auxiliary power. The auxiliary power usually presents in a coal-fired boiler such as coal mills, primary and secondary air fans are not present due to the heat source of this Rankine cycle, coming from the HRSG.

The parameter T_{HPT} is the temperature of the steam on the outlet of the HPT. This must be higher than the saturation temperature at the same pressure to avoid water-logging. The heat rejected from the cycle is still very large if the heat input of 162.5 MW were considered. The heat still available in the steam is available at a very low temperature compared to the ambient; this makes it very difficult to use the energy available.

6.7 Conclusions

The design parameters of the cycle has been established and motivated. The effect of all the losses has been considered and the magnitude has been established. Having designed and optimised the cycle it is now ready to be combined with the Brayton cycle to form the combined cycle.



7 CHAPTER 7: COMBINED CYCLE CONSIDERATIONS

7.1 Introduction

Combining the Brayton and Rankine cycle to form a combined cycle is a very difficult and complex procedure. The Brayton cycle acts as the input parameter for the Rankine cycle, that has to be adapted for this specific application. The heat transfer in the HRSG which links the two cycles has to be determined correctly to ensure that both cycles operate at the optimum working point.

This chapter will cover the cycle matching as well as the design results obtained from the combined cycle calculations. To obtain a combined cycle efficiency of more than 50 % the designed cycle has to be optimised to ensure optimum efficiency.

7.2 Combining the cycles

7.2.1 HRSG input parameters

The cycles operate independently from one another with the only exception where the cycles transfer heat in the HRSG. The HRSG design is therefore critical to match the cycles perfectly. The HRSG design can only be carried out once the amount of heat that needs to be transferred is known.

Generally, in thermodynamics when a cycle is considered, the mass flow in the cycle is assumed to be 1 kg/s. This assumption of unity provides a way to directly compare different cycles as well as to ensure an energy balance. In the case of the combined cycle, this assumption would not be adequate to compare the amount of energy transferred between the Brayton and Rankine cycles. The reason for this is that the working fluids of the two cycles are very different.

The easiest way to overcome this problem is to transfer the heat units from kJ/kg into kW or MW. This can easily be done by multiplying the heat available in the gas with the mass-flow of the gas. In this way the heat available in the helium may be transferred to the steam and the mass flow of the steam may subsequently be calculated.

The heat exchanger effectiveness of the HRSG may be expressed in two ways. Firstly the effectiveness may be relative to the steam and secondly, relative to helium. The heat exchanger effectiveness is then a result of the pinch point temperature. The pinch point of the combined cycle is a temperature difference present between the highest temperature on the wet saturation line (Rankine cycle) and the gas turbine exhaust temperature at the same entropy value as in the Rankine cycle ($s_{\text{Rankine}} = s_{\text{Brayton}}$).

7.2.2 Rankine cycle mass flow considerations

It is known that the amount of heat available in a gas is the specific amount of heat in the gas multiplied by the mass-flow, as can be seen in the next equation.

$$\begin{aligned}\dot{Q}_{in} &= q_{in} \times \dot{m} \\ \dot{m}_{\text{Rankine}} &= \frac{\dot{Q}_{\text{Brayton_in}}}{q_{\text{Rankine_in}}}\end{aligned}$$

This equation may be adapted (as above) for the combined cycle to convert the amount of heat available in the helium to the amount of mass-flow present in the Rankine cycle if the specific amount of heat is known.

7.2.3 Mass – Energy Balance

To ensure that no energy is produced or destroyed the combined cycle has to be balanced using the First Law of Thermodynamics. Thus, all the energy inputs and outputs to the cycle must be equal



$$\dot{Q}_{in} + \dot{W}_{in} = \dot{Q}_{out} + \dot{W}_{out}$$

If a 50 % combined cycle efficiency is obtained then exactly 100 MW heat must be rejected to the atmosphere and 100 MW work has to be produced.

7.3 Optimisation

A correct solution methodology has been followed to ensure that the optimisation of the Brayton and Rankine cycles may take place as fluently as possible. It was possible to solve the cycles so that only one parameter was left as a possible optimisation parameter in both cycles.

The optimisation of the combined cycle thus has two degrees of freedom before the combination of the two cycles is considered. If the cycles were to be combined, one more optimisation parameter is added. This parameter is the minimum temperature at which the cycles exchange heat. This temperature is very important because it determines the temperature and subsequently the pressure where feed heating takes place. This in turn then determines the condensate pump delivery pressure of the Rankine cycle. The pinch point requirement also has to be satisfied

To find the optimum working point of the combined cycle these three parameters have to be optimised. The difficulty of this optimisation is that all the cycle parameters are interdependent. Thus if the minimum HRSG temperature were chosen the steam quality on the LPT outlet of the Rankine cycle has to be optimised. The pressure ratio of the Brayton cycle is also affected by the minimum HRSG temperature.

There are several practical implications that have to be considered. If a too low minimum HRSG temperature is chosen, the feed heating on the Rankine cycle will not be as effective. If the bleed steam tapping point is inside the dry saturation line, the feed heater will not perform optimally. An iterative process has to be used to ensure that the entire physical requirement is met as well as the design requirement set to the combined cycle.

7.4 Cycle discussion

The most efficient way to illustrate the combined cycle is by using a T-s diagram as was the case with the Brayton and Rankine cycles. The first problem encountered when trying to plot the T-s diagram of two different working fluids (He and H₂O) on the same axis is the reference point of entropy. The difference in entropy that is calculated for the Brayton and Rankine cycle should be in the same order and relatively close to each other to best illustrate the cycle.

When working with helium and steam the entropy calculated does not match closely so a valid T-s diagram cannot be obtained. As steam tables are freely available and it is a commonly used substance it was chosen that the reference point of helium has to be moved in order to obtain a valid T-s diagram. This effectively entropy-scaled T-s diagram can be seen in the figure below

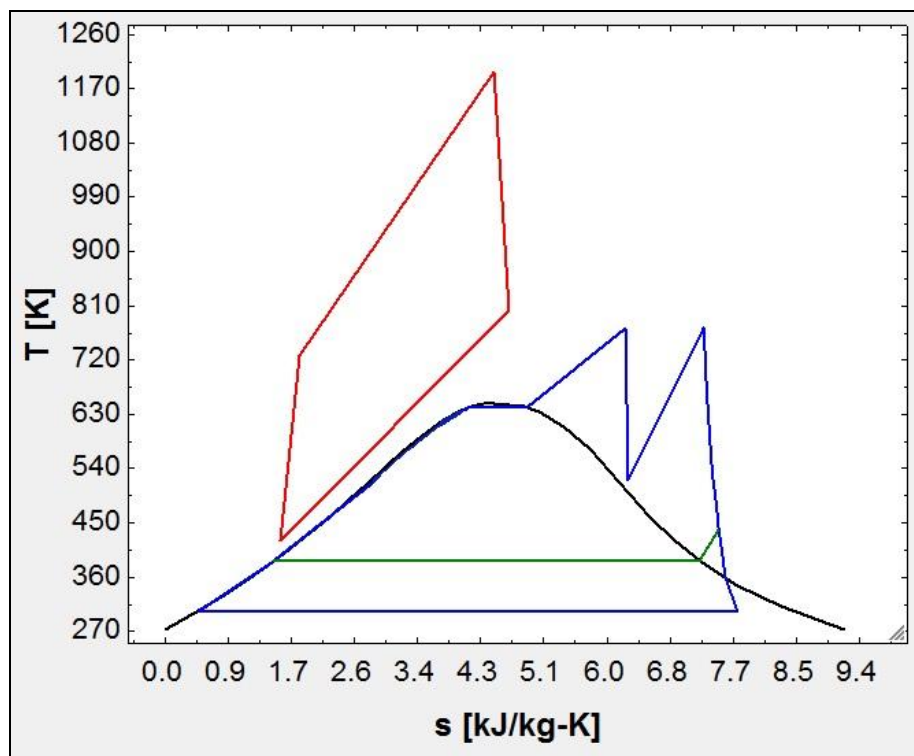


Figure 31: Combined cycle T-s Diagram



The Brayton cycle (in red) operates from the maximum reactor temperature (1223.15K) to just above the feed heater outlet temperature of the Rankine. This is due to the heat exchanger effectiveness, causing the Rankine cycle to exchange heat to a lower temperature, to extract all the heat from the Brayton cycle.

7.5 Results

As mentioned in section 7.3, the optimisation of the cycle is complex due to the three parameters that have to be optimised simultaneously. A solution to the cycle optimisation has to be found to incorporate all three parameters to be incorporated simultaneously. The three parameters to be optimised are:

- Closed Brayton cycle pressure ratio
- Steam quality on LPT steam turbine
- Minimum temperature in the HRSG

The pressure ratio of the CBC is dependent on the minimum CBC temperature that is determined by the minimum temperature in the HRSG. The steam quality on the LPT outlet determines the maximum pressure present in the Rankine cycle. If the quality drops below a specific value the Rankine cycle will be super-critical and not practically achievable.

The optimisation will thus take place for different runs of minimum temperature in the HRSG. If this temperature were kept constant and the steam quality on the LPT steam turbine is varied over a specific range, the optimum pressure ratio may be determined for every instance. The next table shows the results for a constant HRSG temperature of 106.85°C and a variation of the steam quality from 0.8897 to superheated steam.

Table 14: Combined cycle results at the first temperature interval

T_{MinHRSG}	x_{LPT}	r_p	$\eta_{\text{CC}} [\%]$	$W_{\text{Nett}} [\text{MW}]$
106.85 °C	0.8879	3.6120	53.6028	107.2060
106.85 °C	0.8980	3.4900	53.5900	107.1900
106.85 °C	0.9163	3.4850	53.3544	106.7090
106.85 °C	0.9344	3.4850	53.0541	106.1080
106.85 °C	0.9525	3.4850	52.7013	105.4030
106.85 °C	0.9704	3.4860	52.2949	104.5900
106.85 °C	0.9884	3.4860	51.8374	103.6740
106.85 °C	1.0000	3.4850	51.3296	102.6590

The reason for the particular figure of the steam quality on the LPT outlet is 0.8879 is the lowest quality that may be used to ensure that the maximum cycle pressure does not exceed the critical pressure of water. From the table above it can be seen that the lower the quality of the steam on the LPT exhaust the more efficient the cycle operates. The change in pressure ratio is relatively small, giving consistent performance of the cycle.

The combined cycle is producing 107 MW of electrical energy from the 200 MWth that was received from the PBMR-DPP, thus operating at an efficiency higher than 50%. The table above only looks at specific conditions. To complete the optimisation it must be determined if the CC operates more efficiently at higher temperatures. The tendency of the cycle to operate more efficiently if the steam quality on the LPT outlet is lower has to be confirmed at all possible temperatures.

The next table shows the cycle optimisation at a higher temperature than before

Table 15: Combined cycle results at the second temperature interval

T_{MINHRSG}	x_{LPT}	r_p	$\eta_{\text{CC}} [\%]$	$W_{\text{Nett}} [\text{MW}]$
126.85 °C	0.8876	3.672	52.9154	105.831
126.85 °C	0.9000	-	-	-
126.85 °C	0.9272	3.2680	52.9399	105.880
126.85 °C	0.9453	3.2680	52.6197	105.2390
126.85 °C	0.9633	3.2680	52.2357	104.4710
126.85 °C	0.9813	3.2690	51.7890	103.5780
126.85 °C	0.9992	3.2690	51.2837	102.5670

A very peculiar tendency is noticed in this optimisation at 126.85°C. The first iteration at the lowest possible LPT steam quality delivers the highest efficiency as previously. This efficiency is achieved at a higher Brayton pressure ratio. The optimisation at the next LPT steam quality interval does not deliver viable results for the cycle. The optimum pressure ratio that the Brayton cycle should operate at, forces the Rankine cycle to become super critical.

The result of the above optimisation can be seen on a T-s diagram below

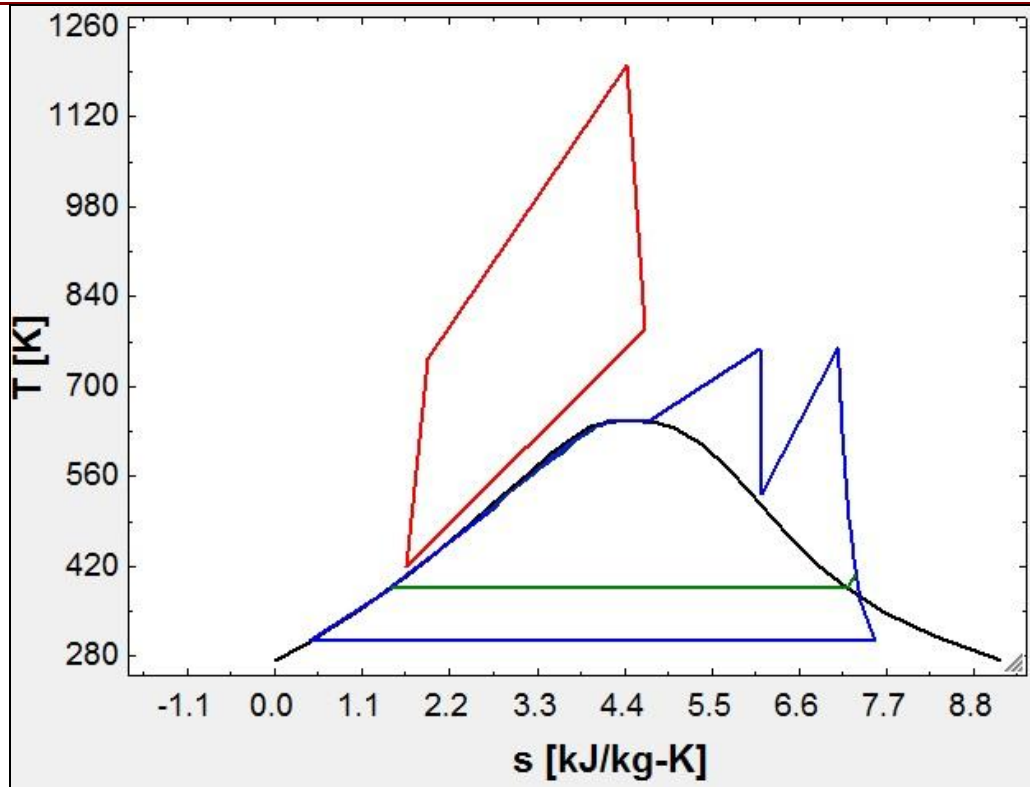


Figure 32: T-s Diagram of optimised cycle parameters

The steam used for feed heating is tapped from the LPT just before the steam is expanded into the two-phase region. The maximum pressure is very high in the evaporator to ensure the minimum steam quality on the LPT outlet.

The next optimisation iteration is carried out at 166.85°C. The results can be seen in the table below

Table 16: Combined cycle results at the third temperature interval

T_{MINHRSG}	x_{LPT}	r_p	$\eta_{\text{CC}} [\%]$	$W_{\text{Nett}} [\text{MW}]$
166.85 °C	0.8879	3.6580	50.9689	101.938
166.85 °C	0.9078	3.4460	51.2874	102.5750
166.85 °C	0.9282	3.1760	51.6451	103.2900
166.85 °C	0.9484	2.9540	51.7737	103.5470
166.85 °C	0.9669	2.9020	51.4698	102.9400
166.85 °C	0.9849	2.9020	50.9859	101.9720

This higher temperature used for the minimum HRSG temperature delivers a completely different efficiency distribution in the cycle. Previously, the lower the steam quality on the LPT outlet, the higher the efficiency would be. This is not applicable for the case of the raised temperature. The efficiency distribution can be seen in the figure below

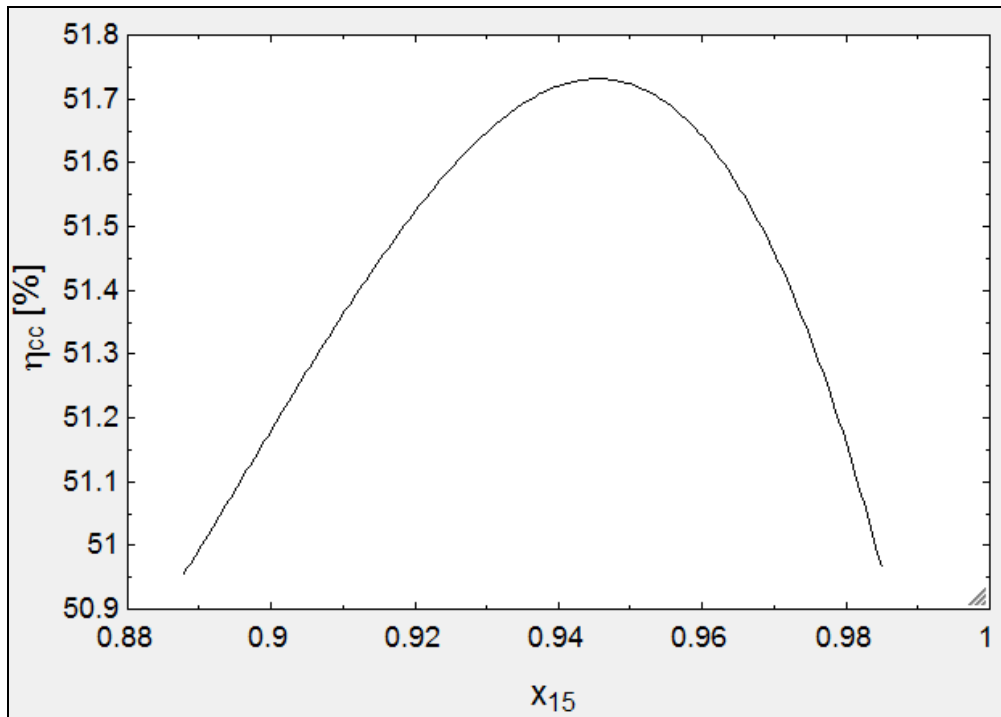


Figure 33: Efficiency distribution at a raised minimum HRSG temperature

The T-s diagram of the raised HRSG temperature can be seen below

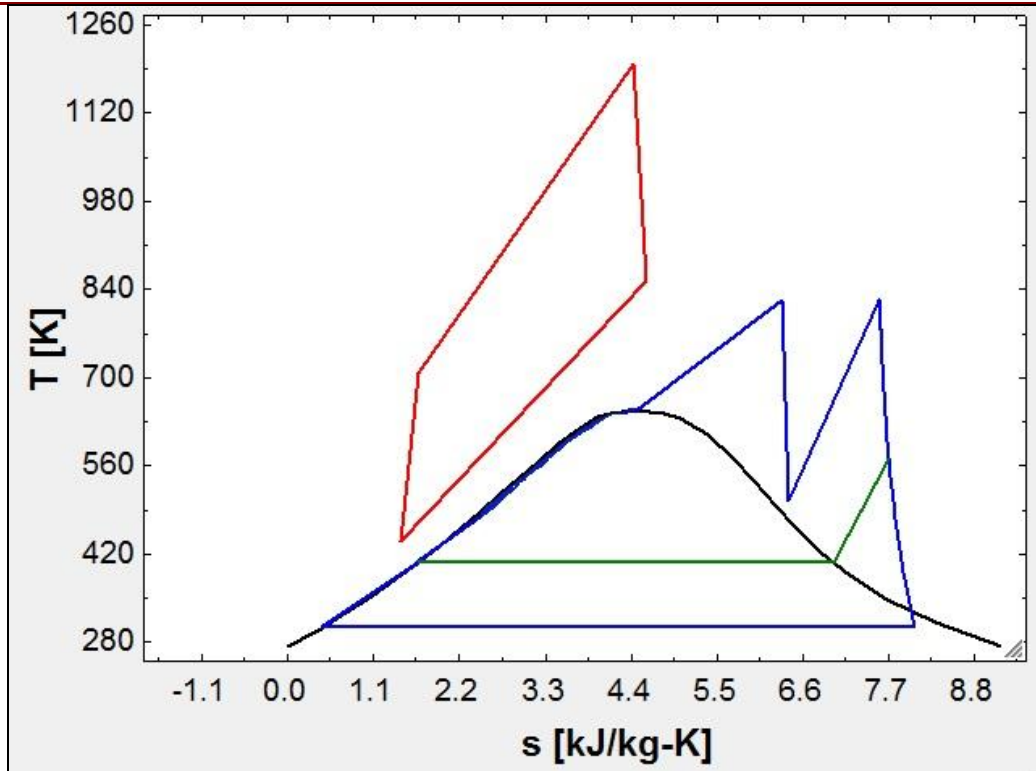


Figure 34: T-s diagram of raised HRSG temperature

The effect of this raised temperature raises the temperature to where the steam is tapped from the LPT. The higher the mass-flow tapped from the LPT, the less steam is available to produce work in the turbine. The higher the temperature that the steam is tapped the lower the mass-flow needed to raise the temperature of the condensate to the required temperature. Thus an optimum temperature has to be determined where the steam should be tapped from the LPT. If the three tables above were considered, the tendency is that the lower the HRSG temperature, the higher the cycle efficiency.

In the first of the iterations used, two phase liquid heat the condensate; in this case a large mass flow is needed to have enough energy to heat the condensate. The last cycle uses steam at a very high temperature to heat the condensate. A small mass flow is needed but it is extracted at a high temperature, thus less work is produced in the LPT. The best case to be used is thus a cycle that uses a relatively high temperature and low mass flow. The middle cycle would thus provide the best solution.



7.6 Conclusions

The optimisation of the cycle has delivered a wide variety of operating conditions for the CC. There are many different parameters that influence the performance of the CC. The tabulated results included in this chapter show the optimum working of the CC at the following conditions:

Table 17: Combined cycle design results

T_1	X_{LPT}	r_p	$\eta_{CC} [\%]$	$W_{Nett} [MW]$
126.85 °C	0.8876	3.672	52.9154	105.831

For this HRSG outlet temperature, the minimum steam quality that can be used is 0.8876, at which the cycle operates the most efficiently. The result obtained in this design satisfies the design objectives of this cycle.



8 CHAPTER8: CONCLUSIONS

8.1 Introduction

This chapter aims to conclude the dissertation as well as give a review of the results obtained from the design process. The different areas that will be looked at are the three cycles that were designed and optimised in the same order.

8.2 Design results

8.2.1 Closed Brayton cycle

Four different CBC's were inspected for possible use with the CC. The performance comparison proved that the simplest cycle will deliver the most work. Although this cycle operates at a lower efficiency it was still used due to the main design parameter being the work produced in the cycle.

The maximum temperature of this cycle was set by the reactor's maximum outlet temperature. This was, however, reduced by making use of blade cooling in the cycle. The following table shows the results obtained from the CBC design.

Table 18: CBC design results

T_{Max}	T_{Min}	W_{Nett}	η_{Carnot}	$\eta_{Brayton}$
924 °C	126.85 °C	38.355 MW	66.58%	18.21%

8.2.2 Rankine cycle

Many different Rankine cycles operate in South Africa to produce the majority of the electrical power available. The Rankine cycle designed for this CC makes use of a re-heater as well as one contact feed heater. The feed pumps are powered by a steam feed-pump turbine to improve the cycle efficiency. Only a single contact feed-heater is used to improve the cycle efficiency and to raise the temperature to where heat is exchanged with the gas via the HRSG.

The main design parameter for the Rankine cycle is efficiency because this cycle is the last process where work may be extracted from the steam. The cycle input parameters were gained from the CBC design as well as a choice of a wet cooling tower. The cycle was solved in such a way that the maximum cycle pressure is calculated from the LPT exhaust steam quality. This improves the optimisation of the Rankine cycle as well as the combined cycle. The design results may be seen in the table below:

Table 19: Rankine design results

T_{Max}	T_{Min}	W_{Nett}	η_{Rankine}	x_{LPT}
484°C	30°C	67.469 MW	42.6 %	0.8876

8.2.3 Combined cycle

Combining the CBC with the Rankine cycle posed problems in optimising the combined cycle. The three parameters that were identified that should be optimised are dependent upon each other. Thus an optimisation methodology had to be found to optimise the cycle. An iterative procedure was developed where the one parameter was kept constant the second varied over a specific range and the third optimised for a specific situation. The three parameters that had to be optimised are:

- Minimum temperature in the HRSG;
- steam quality on the LPT outlet and
- Brayton cycle pressure ratio

The optimisation procedure was carried out on the pressure ratio of the Brayton cycle with the minimum HRSG temperature kept constant and the steam quality varied between a minimum value and super-heated steam. The results obtained from this optimisation can be seen in the following table

Table 20: Combined cycle design results

T_1	x_{LPT}	r_p	$\eta_{CC} [\%]$	$W_{Nett} [MW]$
126.85 °C	0.8876	3.672	52.9154	105.831

8.3 Off-design conditions

As the combined cycle is a power-producing unit, it will have to operate at load-following conditions. The load-following conditions force the cycle to operate at off-design conditions. The cycle has to operate at maximum efficiency at all load situations.

If the CBC were considered, the cycle performance curve shows that there is no dramatic drop-off in cycle efficiency or work produced when the pressure ratio of the Brayton cycle changes. With load changes the mass flow of the cycle may be changed to incorporate the load change without changing the pressure ratio.

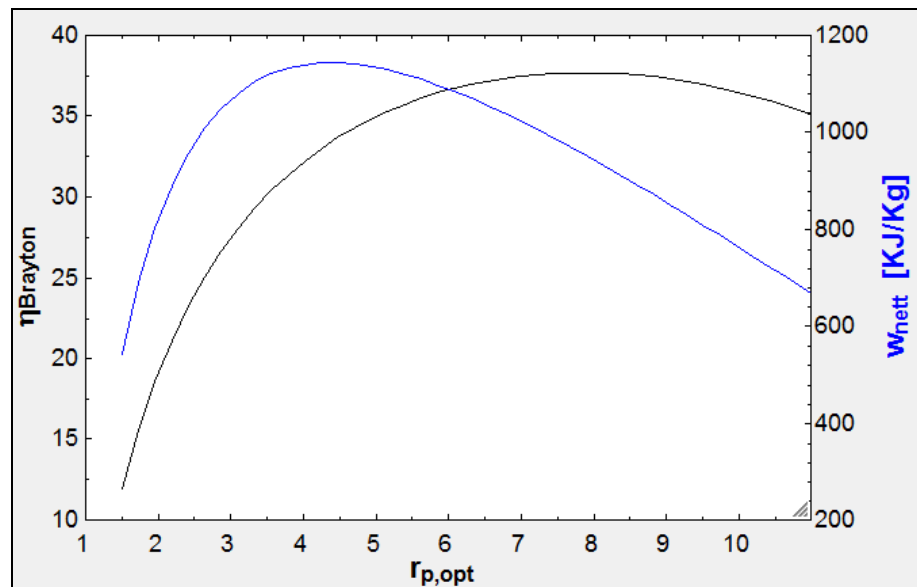


Figure 35: Cycle performance curves for the CBC

When the Rankine cycle is considered the same tendency is noticed. Neither work nor efficiency drops off dramatically as the steam quality is varied. The combination of the cycles continues this tendency as can be seen from the following table

Table 21: Combined cycle performance chart for off-design conditions

T_1	x_{LPT}	r_p	$\eta_{cc} [\%]$	$W_{Nett} [MW]$
126.85 °C	0.8876	3.672	52.9154	105.831
126.85 °C	0.9000	-	-	-
126.85 °C	0.9272	3.2680	52.9399	105.880
126.85 °C	0.9453	3.2680	52.6197	105.2390
126.85 °C	0.9633	3.2680	52.2357	104.4710
126.85 °C	0.9813	3.2690	51.7890	103.5780
126.85 °C	0.9992	3.2690	51.2837	102.5670

8.4 Future study

As this is only a cycle design, the detailed component design was not addressed. The design of the Brayton cycle components, such as the axial flow compressor and turbine (with blade cooling incorporated), has to be developed.

The heat that will be exchanged between the two cycles in the HRSG is a very critical part of the CC. The design of the HRSG is a very important and complex due to the re-heaters that are required for the Rankine cycle. Helium molecules are very small; this makes it very difficult to keep the helium away from the steam in the heat exchanger.

The design of the Rankine cycle component, such as the turbo machines, condensator and the cooling towers has to be done once the location of the operating plant has been established.

8.5 Conclusions

The development of the new generation IV nuclear reactors that is currently being designed requires an efficient PCU to extract the energy efficiently from the reactor. The design of an efficient PCU increases the development of this reactor and may possibly improve the reactor design.



A Brayton-Rankine combined cycle proved to be an efficient PCU to be incorporated with the PBMR MODUL-200. The combined cycle has proved to operate much more efficiently than a Rankine coal fired power plant. The Brayton-Rankine combined cycle also operates more efficiently as a combined cycle than they would as a Brayton and Rankine cycle operated separately (applying the First Law).

Different studies available from literature have been used to improve the design of the combined cycle. The optimisation of the cycle has delivered high efficiency (above 50% efficient) as well as stable load following conditions within acceptable bounds. This combined cycle may be implemented to use with the PBMR-DPP.



9 APPENDIX

9.1 *Combined Cycle EES Calculations*





































9.2 Combined Cycle Excel Calculations







9.3 Single Pressure Cycle EES Calculations

















9.4 Dual Pressure Cycle EES Calculations

















9.5 Rankine Cycle EES Calculations

































9.6 References

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