

Thermal fluid modeling of small scale open Brayton cycle configurations

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Dissertation submitted in partial fulfilment of the
requirements for the degree *Master of Engineering in
Mechanical Engineering* at the North-West University

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Graduation ceremony: May 2019

Student number: 22238409

Abstract

Title : Thermal fluid modeling of small scale open Brayton cycle configurations
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School : Mechanical and Nuclear Engineering
Degree : Master of Engineering

South Africa's high dependence on coal based power stations for the country's power demand as well as the increase in demand for energy, calls for the development of more efficient energy systems that are capable of utilising renewable energy sources to mitigate the emission of harmful gases produced by the combustion of fossil fuels, which adversely affects the environment and the health of the people. A solution to mitigate the aforementioned problems is utilising small scale open externally fired gas turbines (EFGTs). The EFGT, which is based on the working principle of a Brayton cycle, has grown in interest due to its capability to operate with renewable energy sources such as biomass, and it is drawing much attention now that there is a global trend in shifting towards "green" (environmentally friendly) power generation. The problem with EFGTs is that an efficient power generation system is required if biomass is to be used as a renewable fuel source due to its relatively low heating value compared to fossil fuels. The main objective is to thermodynamically evaluate different open EFGT configurations for small scale power generation in the range of 100 [kW]. The focus of this study is the development of thermal fluid simulation models for different configurations with a computer aided program known as Engineering Equation Solver (EES). In order to make a sensible comparison, the performance of each model was evaluated by generating efficiency graphs which are used to determine the best operating conditions to produce an electrical output of 100 [kW] under all constraints. The results obtained indicated that the regenerative EFGT cycle and the regenerative EFGT cycle with two turbines displayed the best performance with a net electrical efficiency of 0.1965 [-] and 0.2 [-] respectively, at a relatively low heat input to the combustion chamber. The regenerative cycle with reheat gained a third place with an efficiency of 0.1597 [-] while the simple EFGT cycle had the worst performance of all the configurations with a cycle efficiency of 0.1248 [-].

Key Words: Brayton cycle, Thermal fluid, Modeling, Power generation, EFGT.

Opsomming

Titel	: Termo vloeï modellering van klein skaal oop Brayton siklus konfigurasies
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Suid-Afrika se hoë afhanklikheid van steenkool-gebaseerde kragstasies vir die voorsiening in die land se energie-aanvraag en ook die stygende aanvraag na elektrisiteit, noodsaak die ontwikkeling van meer effektiewe energie-stelsels wat oor die vermoë beskik om van hernubare energie bronne gebruik te maak. Sodanige stelsels verminder die vrystelling van skadelike gasse, wat veroorsaak word deur die verbranding van fossiel-brandstowwe, wat 'n nadelige effek het op die omgewing en mense se gesondheid. 'n Moontlike oplossing vir die probleem is die gebruik van 'n klein skaal, oop eksterne verbrandings gas turbine (EVGT). Daar is groeiende belangstelling in die EVGT, wat gebaseer is op die werkbeginsel van 'n Brayton siklus, op grond van sy vermoë om gebruik te maak van hernubare energie-bronne soos biomassa, veral weens 'n globale neiging om te verskuif na 'n sogenaamde "groen" (omgewingsvriendelike) kragopwekking. Die probleem met EVGT's is dat 'n effektiewe kragopwekking-stelsel nodig is om van biomassa gebruik te maak as 'n hernubare brandstof bron aangesien biomassa 'n relatiewe lae hitte-waarde het in vergelyking met fossiel brandstowwe. Die hoof doel van hierdie studie is om verskillende EVGT konfigurasies termodinamies te evalueer vir klein-skaalse kragopwekking in die omvang van 100 [kW]. Die fokus is dus op die ontwikkeling van termo-vloeï simulatie-modelle vir verskillende konfigurasies met behulp van 'n rekenaar-ondersteunde program bekend as Engineering Equation Solver (EES). Die prestasie van elke model word geëvalueer deur effektiwiteits-grafieke te genereer om sodoende 'n sinvolle vergelyking te maak met die doel om die beste werkskondisies vir 'n elektriese uitsetkapasiteit van 100 [kW] te bepaal onder alle beperkings. Die resultate het getoon dat die regeneratiewe EVGT siklus en die regeneratiewe siklus met twee gekoppelde turbines die beste prestasie toon in terme van netto elektriese effektiwiteit van 0.1965 [-] en 0.2 [-] respektiewelik teen 'n relatiewe lae hitte-toevoeging tot die verbrandingskamer. Die regeneratiewe siklus met herverhit het 'n derde plek behaal met 'n

siklus effektiwiteit van 0.1597 [-] terwyl die eenvoudige EVGT siklus die swakste prestasie toon van al die konfigurasies met 'n siklus effektiwiteit van slegs 0.1248 [-].

Sleutel Woorde: Brayton Siklus, Termo-vloei, Modelling, Kragopwekking, EVGT.

Declaration

I, **Jan Willem Lodewyckx (ID: 900516 5012 089)**, declare that this report is a presentation of my own original work. Whenever contributions of others are involved, every effort was made to indicate this clearly, with due reference to the literature. No part of this work has been submitted in the past, or is being submitted, for a degree or examination at any other university or course.

J.W. Lodewyckcx

Witness

Acknowledgements

I would like thank my study leaders, Dr. Philip van Zyl Venter and Prof. Martin van Eldik, for their guidance and support towards the completion of this dissertation. For that, I am forever grateful.

Thank you Nicolè Leeb for your assistance with the verification phase of this dissertation. I'm very grateful for the advice and guidance during the development of the Flownex model that aided in the verification of the results.

Thank you to my parents for the opportunity that they gave me to study; for their undying love, support and prayers.

Thank you to my friends and co-students for their support and motivation until the very end of this study. I will never forget it.

Most importantly, thank Lord Jesus Christ for the perseverance and grace to accomplish this study.

Psalm 107, "Give thanks to the Lord, for he is good; his love endures forever."

Abbreviations

EES	Engineering equation solver
HTHE	High temperature heat exchanger
ISO	International organization of standards
TIT	Turbine inlet temperature
CHP	Combined heat and power
EFGT	Externally fired gas turbine
DFGT	Directly fired gas turbine
HXTD	Heat exchanger temperature difference
NG	Natural gas
UN	United Nations
UNDP	United Nations Development Program
SDG	Sustainable Development Goals

Nomenclature

C_{min}	Minimum heat capacity	J/K·s
C_p	Specific heat capacity at constant pressure	J/kg-K
g	Gravitational acceleration	m/s ²
h	Enthalpy	J/kg
h_{0e}	Total enthalpy at outlet	J/kg
h_{0i}	Total enthalpy at inlet	J/kg
$h_{s,0e}$	Total enthalpy at outlet for an isentropic process	J/kg
L	Incremental Length	m
$\dot{m}$	Mass flow rate	kg/s

$\dot{m}_e$	Mass flow rate at outlet	kg/s
$\dot{m}_i$	Mass flow rate at inlet	kg/s
p_{0e}	Total pressure at outlet	Pa
p_{0i}	Total pressure at inlet	Pa
Pr_c	Compressor pressure ratio	Dimensionless
Pr_T	Turbine pressure ratio	Dimensionless
$\dot{Q}$	Rate of heat transfer	W
$\dot{Q}_{Comb}$	Rate of heat transfer for combustion chamber	W
$\dot{Q}_{max}$	Maximum rate of heat transfer	W
$Regen$	Regeneration process	Dimensionless
t	Time	s
$T_{Comb,Out}$	Combustion chamber outlet temperature	°C
$T_{HXPS,Out}$	Heat exchanger primary side outlet temperature	°C
T_{0pi}	Total temperature at primary side inlet	°C
T_{0si}	Total temperature at secondary side inlet	°C
V	Velocity	m/s
$\dot{W}$	Rate of work transfer	W
$\dot{W}_C$	Compressor rate of work transfer	W
$\dot{W}_{C,s}$	Compressor rate of work transfer for an isentropic process	W
$\dot{W}_{Gen}$	Rate of work transfer by generator	W
$\dot{W}_{Out}$	Rate of work transfer at outlet	W
$\dot{W}_{T,s}$	Turbine rate of work transfer for an isentropic process	W
$\dot{W}_T$	Turbine rate of work transfer	W
z_e	Elevation at outlet	m
z_i	Elevation at inlet	m

Greek symbols

ρ	Density	kg/m ³
ε	Effectiveness	Dimensionless
η	Isentropic efficiency	Dimensionless
$\eta_{Brayton}$	Brayton cycle efficiency	Dimensionless
η_C	Compressor isentropic efficiency	Dimensionless
η_{Gearb}	Gearbox efficiency`	Dimensionless
η_{Gen}	Generator efficiency	Dimensionless
η_T	Turbine isentropic efficiency	Dimensionless
Δp_{0L}	Pressure per drop unit length	Pa
ΔT_{max}	Maximum temperature difference	°C

Subscripts

0	Total
C	Compressor
Comb	Combustion
e	Outlet
gearb	Gearbox
gen	Generator
i	In
max	Maximum
min	Minimum
out	Output

p	Primary
s	Isentropic process
s	Secondary
T	Turbine

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CHAPTER 1

INTRODUCTION

Chapter 1: Introduction

1.1 Background

In September 2015, world leaders adopted 17 Sustainable Development Goals (SDG's) as part of the 2030 Agenda for Sustainable Development. The adoption of these goals showed the commitment of countries to end all forms of poverty, fight inequalities and address climate change worldwide (UN Sustainable Development Goals, 2015). One goal that the United Nations considers important is SDG seven, which is ensuring access to affordable, reliable, sustainable and modern energy for everyone.

According to the United Nations' Sustainable Development Goals, energy plays a vital role in the majority of challenges and opportunities that the world faces today. Access to energy is essential for jobs, security, mitigating climate change, food production and increasing income (UN Sustainable Development Goals, 2015).

Despite incentives by governments and institutions such as the World Bank, United Nations Development Program (UNDP) and the Global Environment Facility (GEF) involved in programs to provide electricity to rural communities in developing countries, millions of people are still without electricity (C.L. Azimoh, 2016). Although, electricity alone, is not a solution to all the development problems that rural communities are facing, it can be argued that without electricity, rural communities cannot benefit from development assistance opportunities (D.F. Barnes, 2011).

The World Bank, (2017) estimated that in 2014, nearly 1.06 billion people still had no access to electricity, while 3.04 billion people relied on solid biomass and kerosene for cooking and heating applications. The global electrification rate stood at 85 [%] of which 96 [%] were in urban areas and 73 [%] in rural areas. On a regional basis, the lack of access to electricity was mainly concentrated in the Sub-Saharan Africa (609 million people had no access to electricity) and South Asia (343 million people had no access to electricity).

The World Energy Outlook, (2017) reported that even though electricity access to people in Sub-Saharan Africa is increasing annually, projections show that, due to the population growth outpacing the electrification rate, by 2030 about 600 million people will still have no electricity, 80 [%] of them living in rural areas.

The State of Electricity Access Report (SEAR, 2017) reported that meeting the demand due to increased access to electricity, has led to two approaches, namely (i) grid-electrification that is connected to urban, peri-urban and rural areas or (ii) off-grid electrification via micro- or mini-grid systems on a community level, or standalone devices and systems at a household level. Both approaches have different capital costs, provides for different population densities and utilize different technologies.

In addition, the major challenges that grid-electrification face are the shortage in generation capacity, inadequate transmission and distribution infrastructure, high related costs to provide rural and remote areas with electricity, poor households that are unable to pay connection fees and the poor financial condition of utilities. Apart from grid-electrification, expanding energy access can also be done through off-grid electrification by means of mini-grid and micro-grid systems. Mini-grids have a generation capacity of less than 10 [MW] that are commonly used to provide for small households and cover an area of up to 50 [km] radius, while micro-grids are much smaller systems that typically operates with a capacity of 100 [kW] or less and generally covers an area of 8 [km] radius.

Both mini- and micro-grid systems can be powered with fossil fuels such as diesel or by utilizing renewable resources such as hydro, solar PV, wind and biomass combustion. Zhao & Li (2016) stated that bioenergy plays a significant role in terms of renewable energy resources and that the development of biomass power generation systems is enjoying attention worldwide. With so many people still making use of solid biomass in rural areas across the globe, biomass can be utilized sustainably for power generation to provide access to electricity which would greatly improve their living standards.

Steyn (2006) reported that globally, there is a growth in interest in small scale power generation systems. The interest is motivated by the need for additional power generation capacity, off-grid power supply to remote areas and standalone power generation that is unaffected by power failures. It was also mentioned that gas turbine machinery is the technology that holds great potential for small scale power generation.

In terms of gas turbine technology, utilising externally fired gas turbines (EFGT) has been considered due to its capability to operate with renewable energy sources such as biomass, and it is drawing much attention now that there is a global trend in shifting towards green (i.e. environmentally friendly) power generation (Al-Attab & Zainal, 2015). In the following section, the advantages and disadvantages, operation and characteristics of an EFGT cycle will be discussed.

1.2 Externally fired gas turbines (EFGT)

A basic gas turbine cycle consists of three main components namely, a compressor, a combustion chamber and a turbine (Saravanamuttoo *et al.*, 1996) as illustrated in Figure 1. Modern gas turbine cycles are based on the closed Brayton cycle. An open cycle can also be analyzed as a closed system by taking the atmosphere as a large heat exchanger operating at a constant atmospheric pressure, without any loss in efficiency (Steyn, 2006).

In a basic gas turbine cycle, the three processes taking place are:

- Isentropic compression (1-2).
- Isobaric heat addition (2-3).
- Isentropic expansion (3-4).

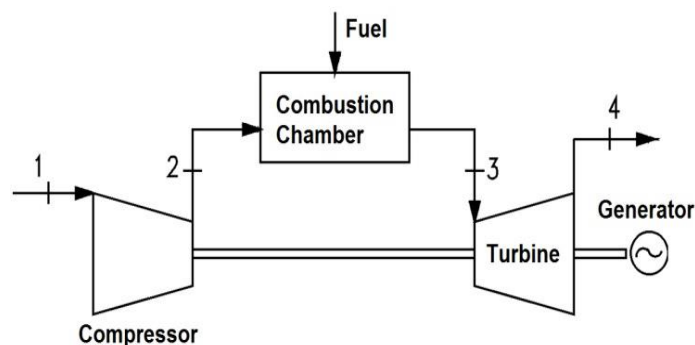


Figure 1 Schematic of an open Brayton cycle (Nascimento *et al.*, 2013).

Ambient air enters the compressor at point 1 in the cycle. In the compressor, shaft work is exerted on the air compressing it isentropically (under ideal conditions) to a higher temperature and pressure between point 1 and point 2. The air receives thermal energy from the combustion chamber, at a constant pressure, between point 2 and point 3. It should be noted that for an EFGT, the combustion process takes place externally and is not in direct contact with the working fluid. The air exits the combustion chamber at a higher temperature and enters the turbine at point 3. From point 3 to point 4, the air is ideally expanded isentropically through the turbine and in the process the temperature and pressure decreases as potential energy is converted into shaft work. The air is then exhausted to the atmosphere. The shaft work can be used for various applications such as driving compressors or electric generators (Borgnakke & Sonntag, 2009).

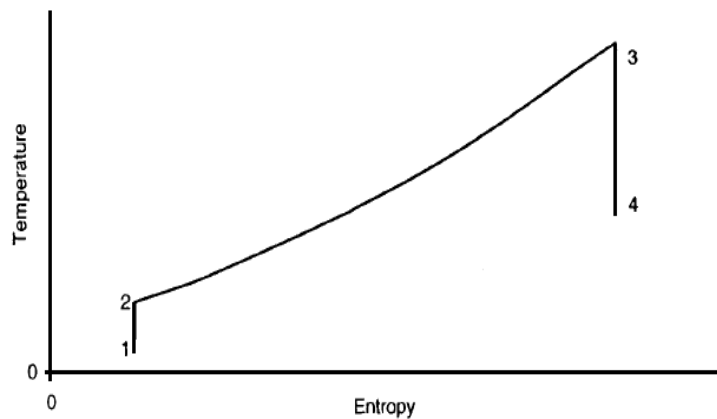


Figure 2 Ideal temperature-entropy diagram for an open Brayton cycle (adapted from: Steyn, 2006).

From Figure 2, the ideal Brayton cycle's temperature versus entropy diagram is illustrated at each point in the cycle. In practice, there is no cycle that operates under ideal conditions. In turbines and compressors, the entropy does in fact change during compression and expansion and there are pressure losses and other losses in the gas turbine components.

An externally fired gas turbine (EFGT) can be divided into two main types namely, an open cycle and a closed cycle which is presented in Figure 3. In terms of efficiency, the open cycle has a higher electrical efficiency, whereas the closed cycle has a higher total efficiency (Gard, 2008). A comparison between the two types of EFGT configurations is presented in Table 1. In Figure 4, a conventional gas turbine, known as a directly fired gas turbine (DFGT) is illustrated.

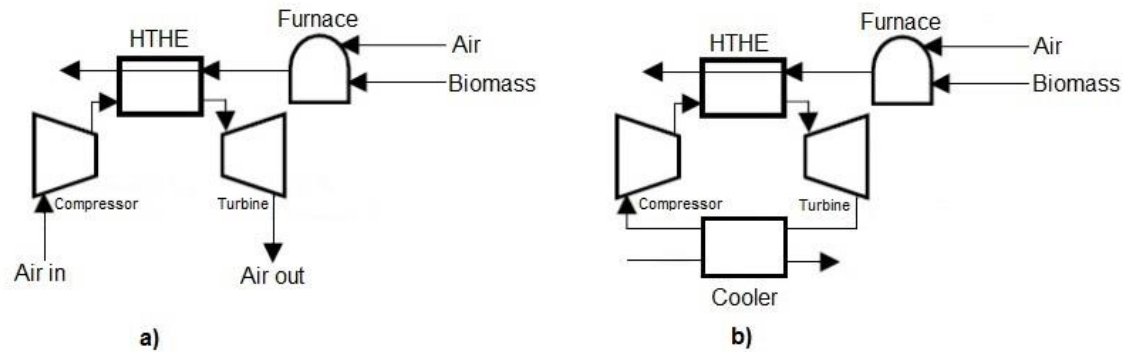


Figure 3 Externally fired gas turbine configuration for an open cycle (a) and a closed cycle (b)
 (adapted from: Al-Attab & Zainal, 2015)

Both EFGT - and DFGT cycles can thermodynamically be described by the Brayton cycle (Al-Attab & Zainal, 2015). The name EFGT implies that the combustion process is done externally within a combustion chamber or furnace. In other words, the process of combustion takes place outside of the working fluid stream (Anheden, 2000). In the EFGT cycle, the flue gases in the combustion chamber is used to heat up the compressed air from the compressor by means of a high temperature heat exchanger (HTHE) (Savola *et al.*, 2015).

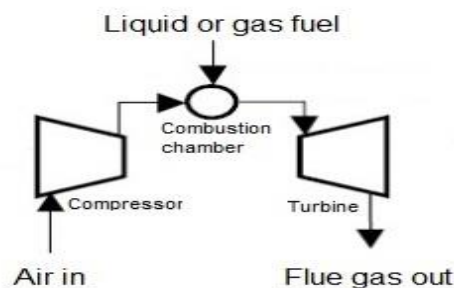


Figure 4 Directly fired gas turbine configuration (adapted from: Al-Attab & Zainal, 2015).

The EFGT has similar advantages to that of conventional gas turbines including a low operating cost, long life expectancy, and a relatively high energy efficiency even for small scale applications (Pantaleo *et al.*, 2013). The difference between the EFGT and DFGT is that the latter can only employ clean fuels. It can also operate with solid fuels such as coal and low quality fuels only after it has gone through a gasification process and also an extensive gas cleaning process. On the other hand, an EFGT can handle a wide range of fuels without requiring equipment for gas cleaning, fuel compression or fuel injection (Al-Attab & Zainal, 2015).

Table 1 Comparison between an open- and a closed cycle EFGT (adapted from: Anheden, 2000; Al-Attab & Zainal, 2015).

Description	Open cycle EFGT	Closed cycle EFGT
Working fluid flow	Fresh ambient air enters the cycle. After flowing through the whole cycle it is exhausted by the turbine into the atmosphere. The process is repeated with new fresh air. The working fluid is thus continuously replaced.	The working fluid is continuously being circulated through the cycle.
Working fluid	Only air is used.	Can use various working fluids, other than air, that have better thermodynamic properties such as helium, carbon dioxide and nitrogen.
Efficiency	Higher electrical efficiency due to a lower compressor inlet temperature (ambient temperature).	Has a higher total efficiency (electrical efficiency and thermal efficiency).
Application	Typically used for power generation.	Ideal for combined heat and power (CHP) generation.
Cost	Less expensive than a closed cycle.	More expensive due to an additional heat exchanger (gas cooler) in the cycle, which is used to pre-cool the working fluid before re-entering the compressor.

An overview of EGFT's is given and they have been compared to conventional gas turbines that use liquid or gas fuels for power generation. Rural areas in Africa, where very poor communities reside, lacks access to even basic services and resources. Connecting these communities to main power grids proves uneconomical due to the hard to reach areas and their inability to pay for electricity services. However, utilizing gas turbine systems that can provide off-grid power supply is a possible alternative. Using liquid or gas fueled turbines would require massive amounts of capital, that are unavailable, to acquire such fuels to keep the power generation systems running. On the other hand, the utilization of EFGT systems seems promising when considering it utilizes low quality fuels. Since billions of people in rural

communities rely on low quality fuels, such as wood, for cooking and heating purposes, the wood can be utilized as a fuel resource for EFGTs that would not only provide electricity, but also result in more efficient cooking and heating methods.

Given the economic situation in rural areas, EFGTs would need to be low in cost and simple, but still efficient. The open cycle EFGT would be the better choice when compared to the closed cycle EFGT as the open cycle has higher electrical efficiencies. What makes the open cycle an even better choice is that it is simple and compact as opposed to a closed cycle that has an additional heat exchanger which requires water or forced air cooling from a fan to cool the working fluid before it cycles through the system, thus making the system more complex, costly and larger in size.

1.3 Problem statement

Currently, there are still rural communities that have no access to electricity. The drawbacks associated with grid-electrification in rural communities calls for an alternative solution that is simplistic yet sustainable and that can provide off-grid electrification by means of standalone power generation systems. Open EFGT systems can provide such a solution. However, limited information is available regarding the desired operating conditions and performance of small-scale, open EFGT systems that are capable of operating with an electrical power output capacity in the proximity of 100 [kW].

1.4 Objective of the study

The objective of this study is to investigate what method and operating conditions, for open EFGT power generation systems, should be incorporated to generate electricity in rural communities. Therefore, the focus will be on the thermodynamic evaluation of different open EFGT configurations, found in the literature, for small scale, off-grid power generation applications in the range of 100 [kW].

1.5 Method of the study

The method that will be followed in order to reach the objective of the study is:

- Conduct a literature survey on open EFGT cycles to obtain information on the work that has been done to date. The literature will contain information regarding typical configurations that are available as well as the assumptions and approaches that have been used to model open EFGT cycles and the evaluation of their performance.
 - Investigate the theoretical knowledge, relevant towards this study, that is based on the components used in EFGT cycles in order to gain an understanding of the theory and equations that they are governed by. Develop simulation models of different EFGT cycle configurations, which are to be solved with the use of the software package, Engineering Equation Solver (EES).
 - Validate a simulation model against an accepted software design package, i.e. Flownex® Simulation Environment.
 - Develop a methodology to determine the preferred operating conditions for different open EFGT configurations.
-

CHAPTER 2

LITERATURE SURVEY

Chapter 2: Literature Survey

2.1 Introduction

This chapter contains a review of the studies found in literature regarding EFGT cycles. Firstly, a review of open Brayton cycles is given in which the operation and basic performance improvement methods of Brayton cycles are discussed. This is followed by a background on research that have been done on different EFGT cycle variants and how they were modelled by different researchers.

2.2 EFGT cycle configurations and their modeling approaches

Ferreira and Nascimento (2001) assessed the performance of four combustion gas turbine configurations that were fuelled by biomass. Two EFGT cycles and two DFGT cycles were considered. The distinction between these cycles were mostly based on the integration of intercooling and regeneration. The EFGT cycles consisted of external combustion chambers with HTHEs, while the DFGT cycles contained gasification systems in which the biomass is first converted into gas and then sent to the combustion chamber. The cycles were modelled with the software package GateCycle and a first law analysis were carried out. Component efficiencies and pressure losses were taken into consideration.

The different configurations were evaluated by determining the optimum pressure ratio for each configuration at which the thermal efficiency is the highest possible. This was followed by an exergy analysis to determine the exergy destruction in the components of each cycle. From the results it was concluded that EFGTs showed good performance at low pressure ratios and were considered as good alternatives for biomass combustion. Another advantage that EFGT cycles have is that they don't require bulk gasification systems and extensive gas cleaning equipment as with the DFGTs. It was also reported that, since exergy destruction influences the cost of the product of a device, EFGTs have a lower generation cost compared to DFGTs. Figure 5 illustrates a schematic layout of the basic EFGT configuration used for the investigation.

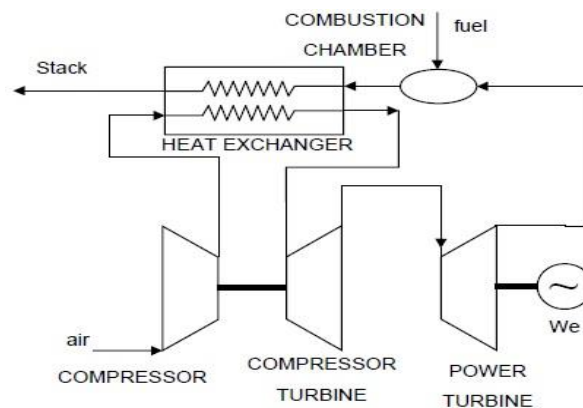


Figure 5 EFGT with a power turbine (Ferreira & Nascimento, 2001).

Saravanamuttoo *et al.* (1996) claimed that using a gas turbine consisting of two turbines in a multi shaft arrangement, as illustrated in Figure 5, is ideal for variable load applications and in electricity generating units. One turbine is used to drive the compressor and the other turbine, also known as the power turbine, is connected to the generator. One disadvantage is that a control system is required for preventive measures when the electrical load is being shed.

Bdour *et al.* (2016) investigated an EFGT with a capacity of 15 [kW] thermal. The cycle was modelled using the software package Aspen Plus. The combustion process was also modelled to obtain more realistic results. Component efficiencies and ambient conditions were taken into account for the simulation. The configuration considered for the investigation was that of a simple EFGT cycle without recuperation and is illustrated in Figure 6. Several parameters have been investigated including the effects of pressure ratio, air mass flow rate, TIT and heat exchanger temperature difference.

The results show that an efficiency of between 5 and 17 [%] can be obtained, which is similar to the results found in other literature. It was explained that the reason for the variation in efficiency was because the investigation was performed for several combustion temperatures, actual load, heat exchanger temperatures and heat transfer efficiencies. It was also concluded that cycle improvement is achievable with more efficient compressors and turbines as well as the option to include waste heat recovery from the turbine exhaust. Additionally, TIT plays an important role in cycle efficiency, but the latter requires the development of high temperature resistant materials for the turbine and heat exchanger.

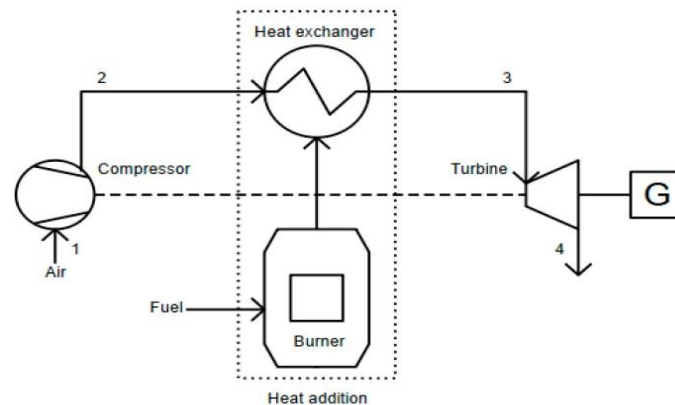


Figure 6 Simple EFGT cycle (Bdour *et al.*, 2016).

Kautz and Hansen (2007) investigated an EFGT cycle for the decentralized use of biomass. A commercial gas turbine, the Turbec T100 with an electrical power output of 100 [kW], was used as design basis. The calculations were carried out with the software package Aspen Plus. In the model of the EFGT, the combustor of the Turbec machine was replaced by a heat exchanger and a furnace operating at atmospheric pressure as illustrated in Figure 7. With the addition of a recuperator (type of heat exchanger), the electrical efficiency could be raised from 16 [%] to [30%]. The main parameters investigated were the pressure ratio, TIT as well as the temperature difference and pressure losses associated with the heat exchanger. ISO standard conditions were assumed and component efficiencies have been taken into account for the simulation. The possibility of using solar energy as an additional heat source has also been reported. From the results, the following main conclusions were made:

- An EFGT cycle is suitable for decentralised CHP plants that utilizes biomass.
- The possible electrical power capacity at which these types of turbines can operate ranges between 30 and 2000 [kW].
- The use of a recuperator to preheat the compressed air with the waste heat exhausted by the turbine improves the cycle efficiency, which is close to the high efficiencies obtained in standard gas turbines with regeneration.
- Optimizing the recuperator and heat exchanger is essential and need to be the main objective for biomass combustion.
- The additional cost of the heat exchanger and atmospheric combustor will have to be compensated for with the use of low cost and low quality biomass waste fuels, as opposed to the use of NG (natural gas) in standard gas turbines.
- An EFGT shows potential in terms of efficiency and investment cost over other standard gas turbines.

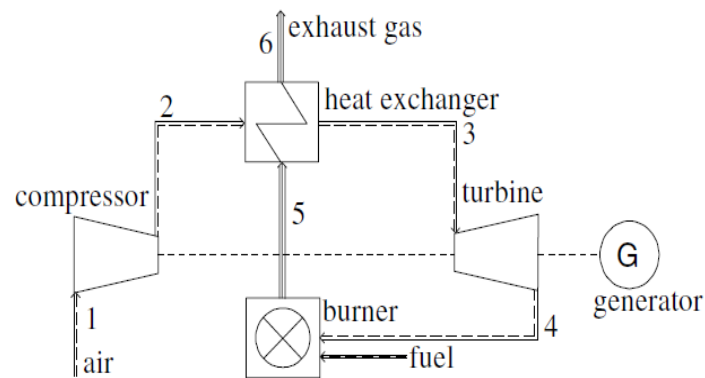


Figure 7 Recuperated EFGT cycle (Kautz & Hansen, 2007).

Goodarzi (2016) did a comparative analysis on a new regenerative Brayton cycle. The investigation focused on three different cycles namely, a basic Brayton cycle, a conventional regenerative Brayton cycle and the newly proposed regenerative Brayton cycle. Schematic layouts of both regenerative cycles are illustrated in Figure 8a and 8b. According to Goodarzi (2016) the main difference between the conventional cycle and the newly proposed regenerative cycle is that hot air is expanded above atmospheric pressure through the first turbine. The hot air then enters the regenerator to preheat the compressed air. From the regenerator the air flows through the second turbine where it is expanded to atmospheric pressure.

It was mentioned that an ideal thermodynamic analysis is sufficient to illustrate the advantages of the newly proposed regenerative cycle. The study was conducted under the assumption that the cycles operate under ideal conditions and therefore isentropic expansion and compression took place, all losses are neglected and the air was assumed ideal with constant thermal properties. The analysis was based on first law thermodynamics for each control volume within the particular cycle. For a comparative analysis, the TIT and ambient temperatures were kept constant while the compressor and first turbine pressure ratios were variable parameters. Several results were obtained for each cycle based on compressor pressure ratios of 5, 10 and 15. The results include dimensionless specific power output, dimensionless specific heat absorption, thermal efficiency, heat absorption per output power, exhausted heat per output power and the reduced temperature of the exhausted airflow (Goodarzi, 2016).

From the results, it was concluded that the newly proposed cycle displayed better thermal and energy performances compared to the conventional regenerative cycle. It was recommended that the first turbine of the new cycle only drives the compressor and the second turbine be used as the power turbine. The new cycle yielded good performance results at low compressor and first turbine pressure ratios.

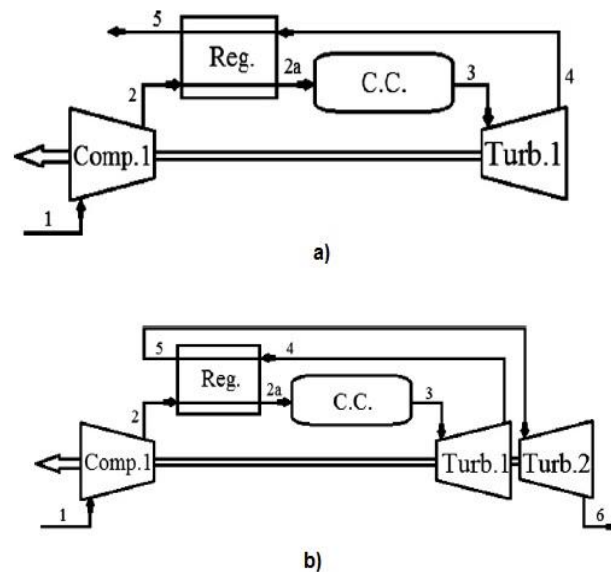


Figure 8 a) Regenerative Brayton cycle and b) New regenerative Brayton cycle (Goodarzi, 2016).

Amirante *et al.* (2015) investigated the performance of two combined cycle configurations by using an EFGT cycle as a basis. The aim was to compare the performance of the cycles when either biomass or methane is used as a fuel source. The data from this study supported the development of an actual EFGT-Rankine combined cycle. The first cycle considered was an open EFGT combined with an open Rankine cycle and is illustrated in Figure 9. Illustrated in Figure 10 is the second case which is an open EFGT cycle combined with a closed Rankine cycle with additional components consisting of a condenser and a degasser. The closed Rankine cycle also has the same operation as a conventional steam turbine. The Rankine as well as the EFGT cycle, in each case, are linked to a generator.

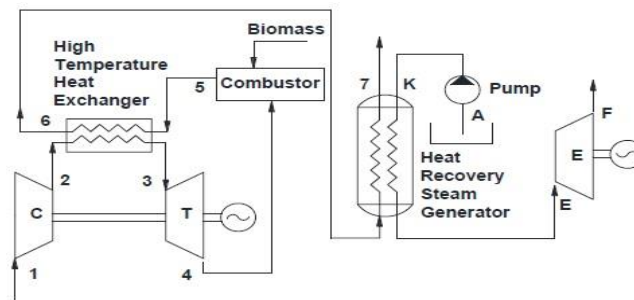


Figure 9 Open EFGT cycle combined with an open Rankine cycle (Amirante *et al.*, 2015).

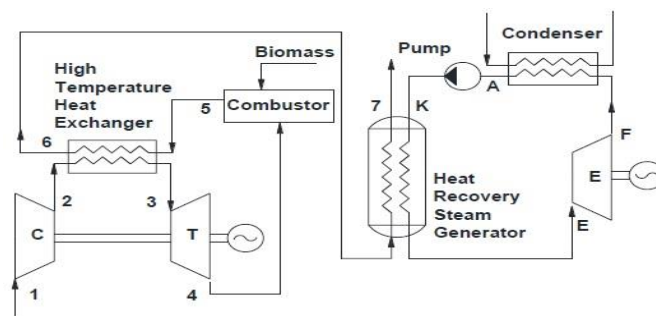


Figure 10 Open EFGT cycle combined with a closed Rankine cycle (Amirante *et al.*, 2015).

For the EFGT cycle, data from a commercial turbocharger (GARRETT GTX5518) was used to model the EFGT along with a ceramic heat exchanger. It was reported that a fluidized bed combustor or a standard furnace would suffice for the EFGT combustion process. Standard components for the Rankine cycle were also used. The analysis of the cycles was conducted with two software packages, namely Excel and GateCycle. The governing equations in Excel were solved with the so called "false position" method and with the assumption that the working fluid is semi-perfect, meaning that the fluid properties are only influenced by a variation of temperature.

From the results obtained, the EFGT cycle produced 77 [kW] of electrical power at a pressure ratio of 3, TIT of 879 [°C] and an air mass flow rate of 0.68 [kg/s]. Additionally, the total plant efficiency, using methane, was determined to be 0.27 [-] for the case with an open Rankine Cycle and 0.296 [-] for the case with a closed Rankine cycle. The effects of using biomass and methane for the first case were compared and the results showed that when using biomass, an overall electrical power output of 89.65 [kW] and a cycle efficiency of 0.25 [-] is achievable and for methane an overall electrical power output of 88.85 [kW] with a cycle efficiency of 0.2747 [-] can be achieved. The models developed in Excel and GateCycle were compared

and the results showed good agreement. Amirante *et al.* (2015) concluded that the availability and reliability of the plant components makes it a feasible solution for both cases in terms of power generation. Furthermore, it was claimed that combined cycles show potential in providing high energy efficiencies for small scale, combined cycle power generation applications.

Vera *et al.* (2011b) compared the performance of an EFGT and a gasifier-turbine system. Both were intended for combined heat and power (CHP)¹ applications. The cycles were modelled using the software package Cycle-Tempo assuming steady state flow conditions for the working fluid. The results were determined for an electrical power output of 30 [kW] and a thermal output of 60 [kW] which was used for the production of sanitary hot water. Component efficiencies and pressure losses were considered in conducting the simulations. For a comparative analysis the TIT, electrical power output and the thermal power output were kept constant for both cycles. The optimum pressure ratio was determined at which the electrical efficiency of each cycle reached a maximum value. The results concluded that the EFGT performed better with a pressure ratio of 4 [-] and an electrical efficiency of 19.1 [%] compared to the gasifier-turbine with a pressure ratio of 3.8 [-] and an electrical efficiency of 12.3 [%]. It was explained that the low gasifier-turbine efficiency was mainly because of the syngas from the gasification process that needed to be compressed before being sent to the combustor. As a result, additional work is required for the compression process. Another factor that affected the gasifier-turbine efficiency is the pressure losses in the complex gas cleaning system.

Vera *et al.* (2011a) decided to conduct another investigation by combining the EFGT and gasifier-turbine systems into a single EFGT cycle integrated with a gasifier. This cycle was also modelled with Cycle-Tempo. The proposed model was set to achieve an electrical power output of 70 [kW] and a thermal power output of 150 [kW]. The combustion process took place at atmospheric pressure and 980 [°C]. Important operating parameters that have been evaluated include the TIT, pressure ratio and heat exchanger temperature difference. From the results it was concluded that the TIT, pressure ratio and heat exchanger temperature difference increase electrical efficiency, and that the cycle was able to achieve an electrical efficiency of 19.6 [%] at an optimum pressure ratio of 4 [-].

¹ CHP is an electricity producing system that recovers heat that would otherwise have been wasted in the form of useful energy by means of a heat exchanging device. The latter is usually to provide steam or hot water for several applications and processes (EPA, 2016).

Sigarchian (2012) conducted the modeling and analysis of several hybrid solar-dish, Brayton gas turbine layouts. Figure 11 and 12 illustrate the typical configurations based on an EFGT which were compared against a conventional simple EFGT cycle without recuperation. The solar section of the system consisted of a parabolic dish concentrator, which reflects solar irradiation to a small point called the focus, and also a solar receiver that absorbs the solar energy that is being reflected by the concentrator. The cycle was thermodynamically analysed based on the first law of thermodynamics with the aim to obtain an electrical power output of 5 [kW]. Several assumptions were made regarding component efficiencies, pressure ratios and losses. The TIT for the cycles in Figure 11 and 12 was assumed to be 1000 [°C]. The results from the thermodynamic analysis showed that the conventional EFGT had an optimised electrical efficiency of 14.2 [%] while electrical efficiencies of 14 [%] and 15 [%] for the cycle layouts in Figure 11 and 12 respectively, where obtained. The results showed that the integration of a solar-dish with an EFGT did not have much influence on the electrical efficiency.

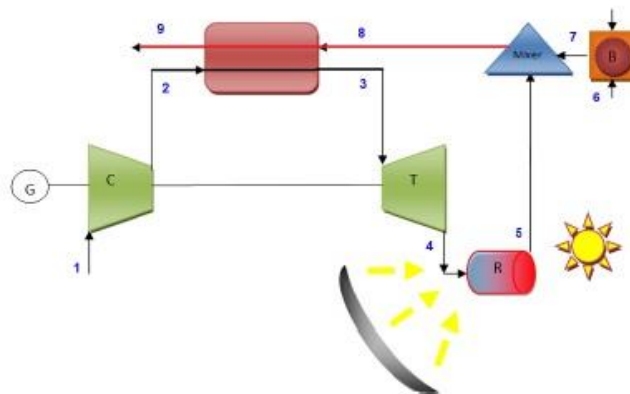


Figure 11 Solar integrated EFGT with the dish positioned after the turbine (Sigarchian, 2012).

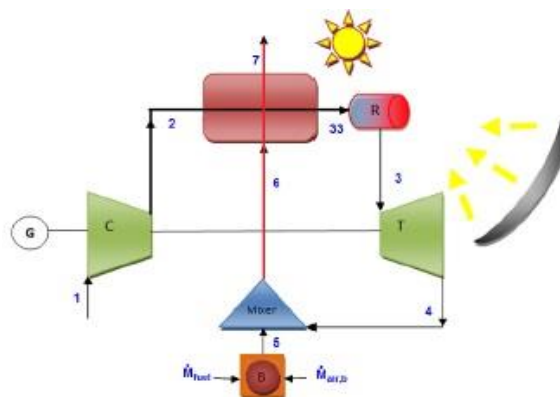


Figure 12 Solar integrated EFGT with the dish positioned before the turbine (Sigarchian, 2012).

2.3 Summary

From the literature survey conducted it is evident that there are numerous methods of utilizing EFGT-based cycles for power production purposes as well as combined heat and power applications. Different approaches also exist that can be used to model and simulate EFGT cycles. The literature review contains different configurations of stand-alone EFGT systems that are capable of generating electricity in rural areas. The assumptions made by the authors, from the literature, regarding the component characteristics and the working fluid will be applied to the EFGT cycles that are used in this study and is discussed in detail in Chapter 4. The table below is a summary of research done in the field of EFGT based power production cycles, some of which was discussed in the literature.

Table 2 Summary of the performance of the different EFGT cycles discussed in the literature.

EFGT Type	Fuel	Power [kW]	TIT [°C]	Pressure Ratio [-]	Efficiency [%]	Reference
Simple	Olive Residues	15	815	4	5 - 17	Bdour <i>et al.</i> (2016)
Simple	Wood fuel	20-30	850	4.5	15	Pritchard (2005)
Gasifier Integrated	Olive Residues	70	850	4	20	Vera <i>et al.</i> (2011b)
Simple	NG	50	750	-	16	Traverso <i>et al.</i> (2006)
CHP, Recuperated	Biomass	77.54	900	-	19.2	Pantaleo <i>et al.</i> (2013)
Recuperated	NG	100	900	4.5	30	Kautz and Hansen (2007)
Integrated biomass rotary dryer	Biomass	100	950	3.5	22-33	Cocco <i>et al.</i> (2006)
Simple	Olive residues	30	830	4	19.4	Vera <i>et al.</i> (2011b)
Gasifier Integrated	Biomass	100	777-1077	2-8	16	Datta <i>et al.</i> (2010)
Recuperated	Biomass	100	950	4.5	20-30	

CHAPTER 3

THEORETICAL BACKGROUND

Chapter 3: Theoretical Background

3.1 Introduction

In this chapter the theoretical background is discussed in order to gain a thorough understanding of the theory that governs Brayton cycles. The theoretical background will consider the relevant theory and governing equations that is necessary for the successful development of thermal fluid simulation models for different EFGT configurations.

3.2 Simulation model

The generic structure for any simulation model, which may be for a single component or for an integrated system that comprise of a number of components, needs to include the following aspects that is fundamental for the development thereof, namely (Rousseau, 2013):

- Conservation laws: mass, momentum and energy.
- Component characteristics: Heat transfer rates, pressure drops, component dimensions.
- Fluid properties: Thermodynamic property tables and gas laws.
- Boundary values: Mass flows, temperatures and pressures.

3.3 Conservation laws

The conservation laws of mass, momentum and energy are part of the fundamental assumptions that is necessary for the development and modeling of thermal fluid systems.

3.3.1 Conservation of mass

For the conservation of mass, the general equation that is applicable to thermal fluid systems is (Borgnakke & Sonntag, 2009):

$$V \frac{\partial \rho}{\partial t} + \dot{m}_e - \dot{m}_i = 0 \quad (3.1)$$

Where V represents the velocity, ρ the density, t the time and $\dot{m}$ the mass flow rate. In this study, subscripts e and i will be used to denote the outlet and inlet respectively.

Given steady state conditions are assumed, the state of the fluid is not influenced over time and $\partial\rho/\partial t = 0$. The general equation for the conservation of mass then reduces to:

$$\dot{m}_e - \dot{m}_i = 0 \quad (3.2)$$

Since there is no change in the condition of the fluid over time when steady state conditions are assumed, the inlet and outlet mass flow rates of the fluid are equal. From the latter it then follows that Eq (3.2) can be rewritten so that the mass flow rate is described by a single value namely:

$$\dot{m}_e = \dot{m}_i = \dot{m} \quad (3.3)$$

3.3.2 Conservation of momentum

For the conservation of momentum, the general equation for incompressible flow that is applicable to thermal fluid systems is (Borgnakke & Sonntag, 2009):

$$\rho L \frac{\partial V}{\partial t} + (p_{oe} - p_{oi}) + \rho g(z_e - z_i) + \Delta p_{oL} = 0 \quad (3.4)$$

Where L represents the incremental length, p the pressure, g the gravitational acceleration, z the elevation height and Δp_{oL} the pressure drop due to frictional and other losses. If steady state conditions are assumed then, $\partial\rho/\partial t = 0$. The general equation for the conservation of momentum then becomes:

$$(p_{oe} - p_{oi}) + \rho g(z_e - z_i) = -\Delta p_{oL} \quad (3.5)$$

Also if there is no change in the elevation height, for the thermal fluid system being considered, and steady state flow still prevails, then Eq (3.2.5) is reduced to:

$$p_{oe} - p_{oi} = -\Delta p_{oL} \quad (3.6)$$

3.3.3 Conservation of energy

For the conservation of energy, the general equation that is applicable to thermal fluid systems is (Borgnakke & Sonntag, 2009):

$$\dot{Q} + \dot{W} = V \frac{\partial(\rho h - p)}{\partial t} + \dot{m}_e h_{oe} - \dot{m}_i h_{oi} + \dot{m}_e g z_e - \dot{m}_i g z_i \quad (3.7)$$

Where $\dot{Q}$ is the total rate of heat transfer to the fluid, $\dot{W}$ the total rate of work done on the fluid² and h the enthalpy. Assuming that steady state conditions apply then $\partial(\rho h - p)/\partial t = 0$ and by substitution of Eq (3.3) into Eq (3.7) the following equation is obtained:

$$\dot{Q} + \dot{W} = \dot{m}(h_{oe} - h_{oi}) + \dot{m}g(z_e - z_i) \quad (3.8)$$

When a fluid flows through compressors, turbines and fans, work is being performed during this process, i.e. due to compression of the fluid or expansion by the fluid. When these components perform work, no external heat is added or extracted during this process, resulting in $\dot{Q} = 0$. It is also assumed for this study that there is no difference in elevation height for any component, thus $z_e - z_i = 0$. From the latter mentioned, Eq (3.8) reduces to:

$$\dot{W} = \dot{m}(h_{oe} - h_{oi}) \quad (3.9)$$

In any heat exchanger, heat transfer from a warm fluid to a cold fluid takes place and therefore no work is being done during this process, resulting in $\dot{W} = 0$. Also, if the difference in elevation height is neglected then Eq (3.8) reduces to:

$$\dot{Q} = \dot{m}(h_{oe} - h_{oi}) \quad (3.10)$$

² Note that for sign convention for this study, the value of the rate of heat added to the fluid and the rate of work done on the fluid is positive, while the rate of work done by the fluid and the rate of heat transfer from the fluid to the surroundings are considered to have a negative value.

3.4 Component Characteristics

The components in a Brayton cycle needs to form an integrated whole so that a complete model of a Brayton cycle can be developed for different configurations. This section progresses from the conservation law of energy by employing simplified component characteristic equations of generic components in Brayton cycles.

3.4.1 Compressors

The purpose of a compressor, illustrated in Figure 13, is to increase the pressure of the fluid that flows through it. For a compressor to compress the fluid, work needs to be performed during the process. For an isentropic process, the total rate of work performed by a generic compressor is defined by (Borgnakke & Sonntag, 2009):

$$\dot{W}_{C,s} = \dot{m} \cdot (h_{s,oe} - h_{oi}) \quad (3.11)$$

With $\dot{W}_{C,s}$ the isentropic compressor work and $h_{s,oe}$ the total enthalpy at the outlet for an isentropic process. The isentropic efficiency for a compressor is defined by:

$$\eta_c = \frac{\dot{W}_c}{\dot{W}_{C,s}} \quad (3.12)$$

With η_c the compressor isentropic efficiency and $\dot{W}_c$ the actual compressor work that is described by Eq. (3.9). If ideal conditions are assumed, then $\eta_c = 1$ and from Eq (3.12) and Eq (3.9) it follows that:

$$\dot{W}_{C,s} = \dot{W}_c = \dot{W} \quad (3.13)$$

For a compressor, the pressure ratio at which a fluid can be compressed is defined by (Borgnakke & Sonntag, 2009):

$$Pr_c = \frac{P_{oe}}{P_{oi}} \quad (3.14)$$

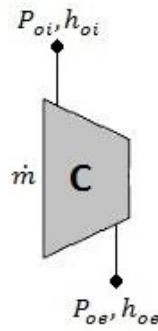


Figure 13 Schematic of a generic compressor (Rousseau, 2013).

3.4.2 Turbines

A turbine is predominantly used to drive other components. When a fluid at an elevated pressure and temperature enters the turbine, the energy contained in the fluid is converted into mechanical energy as it expands through the turbine, resulting in the rotation of a shaft that is connected to other components. During the expansion process the fluid performs work on the turbine blades. There is no heat transfer taking place during this process. For a generic turbine the total rate of isentropic work performed is defined by (Borgnakke & Sonntag, 2009):

$$\dot{W}_{T,s} = \dot{m} \cdot (h_{s,oe} - h_{oi}) \quad (3.15)$$

Where $\dot{W}_{T,s}$ is defined as the isentropic turbine work output. A schematic of a generic turbine is illustrated in Figure 14. The isentropic efficiency for a turbine is defined by:

$$\eta_T = \frac{\dot{W}_{T,s}}{\dot{W}_T} \quad (3.16)$$

With η_T the turbine isentropic efficiency and $\dot{W}_T$ the actual turbine work that is described by Eq. (3.9). If ideal conditions are assumed, then $\eta_T = 1$ and from Eq (3.16) and Eq (3.9) it follows that:

$$\dot{W}_{T,s} = \dot{W}_T = \dot{W} \quad (3.17)$$

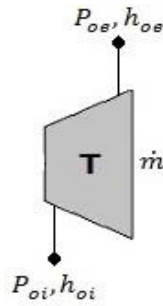


Figure 14 Schematic of a generic turbine (Rousseau, 2013).

For a turbine, the ratio of expansion of the fluid through it is defined by (Borgnakke & Sonntag, 2009):

$$Pr_T = \frac{P_{oe}}{P_{oi}} \quad (3.18)$$

3.4.3 Heat exchangers

Figure 15 illustrates a simplified schematic of a generic heat exchanger that involves heat transfer between two fluid streams. The simplest method of estimating the heat transfer duty for a generic heat exchanger is by writing it as a fraction of the maximum heat transfer that is theoretically possible namely (Incropera *et al.*, 2006):

$$\dot{Q} = \varepsilon \cdot \dot{Q}_{max} \quad (3.19)$$

Where $\dot{Q}$ is the actual heat transfer duty, ε the effectiveness of the heat exchanger and $\dot{Q}_{max}$ the maximum rate of heat transfer that is theoretically possible between two fluid streams. The actual heat transfer duty $\dot{Q}$ can be determined with Eq (3.10).

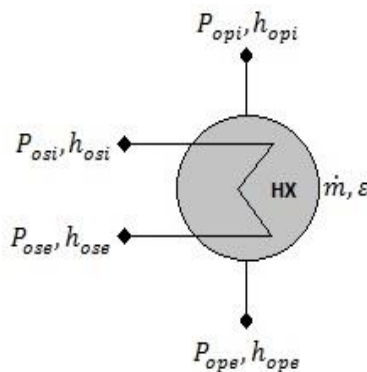


Figure 15 Schematic of a generic heat exchanger (Rousseau, 2013).

From Eq (3.19) the maximum rate of heat transfer can be calculated with the following (Incropera *et al.*, 2006):

$$\dot{Q}_{max} = C_{min} \cdot \Delta T_{max} \quad (3.20)$$

With C_{min} the minimum heat capacity of the two fluid streams and ΔT_{Max} the maximum temperature difference between the two fluid streams, thus $\Delta T_{Max} = (T_{opi} - T_{osi})$. From Eq (3.20), the minimum heat capacity is determined by (Incropera *et al.*, 2006):

$$C_{min} = \dot{m} \cdot C_p \quad (3.21)$$

Where C_p is the specific heat value for a constant pressure process. By substituting Eq (3.21) and Eq (3.20) into Eq (3.19) the actual heat transfer duty between the two fluid streams becomes:

$$\dot{Q} = \varepsilon \cdot C_{min} \cdot (T_{opi} - T_{osi}) \quad (3.22)$$

If ideal conditions are assumed, then the heat exchanger effectiveness has a value of 1 and as a result, Eq (3.19) is reduced to:

$$\dot{Q} = \dot{Q}_{max} \quad (3.23)$$

3.5 Cycle efficiency and shaft energy balance

The overall Brayton cycle thermal efficiency is an important parameter as it indicates how efficient the heat input by the combustor is converted into useful work. The Brayton cycle efficiency is calculated with the following (Borgnakke & Sonntag, 2009):

$$\eta_{Brayton} = \frac{\dot{W}_{Out}}{\dot{Q}_{Comb}} \quad (3.24)$$

Where $\dot{W}_{out}$ is the total power output delivered by the cycle and $\dot{Q}_{comb}$ the heat added to the working fluid by the combustion chamber. If it is assumed that there are no gearbox losses to the generator and that the generator is operating at 100 [%] efficiency, then the total power output is equal to the power generated by the generator. Thus:

$$\dot{W}_{out} = \dot{W}_{Gen} \quad (3.25)$$

3.6 Shaft energy balance

Depending on the number of compressors, turbines and a generator that are connected together on a single shaft, the energy balance is given by:

$$\sum \dot{W}_T + \sum \dot{W}_C + \left(\frac{1}{\eta_{Gearb}}\right) \cdot \left(\frac{1}{\eta_{Gen}}\right) \cdot \dot{W}_{Gen} = 0 \quad (3.26)$$

With $\dot{W}_{Gen}$ the net or total power generated by the generator, η_{Gearb} the gearbox efficiency and η_{Gen} the generator efficiency. If ideal conditions are assumed, then Eq (3.26) is reduced to:

$$\sum \dot{W}_T + \sum \dot{W}_C + \dot{W}_{Gen} = 0 \quad (3.27)$$

3.7 Summary

The theoretical background that is necessary for the successful development and operation of thermal fluid simulation models was presented. This included conservation laws, component characteristics as well as cycle thermal efficiency and shaft energy balance equations. The assumptions made and the simplified component characteristic equations can now be incorporated in the modeling of Brayton cycles that consists of the components discussed in this chapter.

CHAPTER 4

BRAYTON CYCLE MODELING

Chapter 4: Brayton Cycle Modeling

4.1 Introduction

In order to compare the performance of the different EFGT cycles, a certain approach needs to be followed so that a sensible comparison can be made. In this chapter the constraints and variable parameters for which the cycles are simulated, are discussed. This is followed by a discussion of a process for the calculation of an efficiency point and subsequently determining the maximum possible efficiency for an electrical output of 100 [kW] under all constraints.

4.2 Constraints

It is important to take into account the constraints that are applicable to the EFGT cycles before they are simulated. This provides for more realistic conditions under which they operate. From the literature discussed in chapter 2, the following constraints have been considered for the simulation of EFGTs:

4.2.1 Heat exchanger and combustion chamber maximum temperature

Even though the TIT of a turbine is limited by the metallurgical constraints of the material that it is made of, it can be controlled by means of a heat source such as a heat exchanger or combustion chamber, depending on the desired configuration and application thereof. It is also a given that a specific heat exchanger or combustion chamber can only reach a certain maximum temperature due to its metallurgical constraints.

4.2.1.1 Heat exchanger

The TIT achievable via a heat exchanger have been reported to be in the range of 700 -1100 [°C]. This value is restricted by the heat exchanger material (Anheden, 2000). As previously mentioned, high TIT is accompanied by heat exchangers with high development costs that would require long payback periods on the whole system. Amirante *et al.* (2015), reported that a TIT of 878.9 [°C] is achievable from the results they obtained for a preliminary design of an EFGT system. Vera *et al.* (2011b) proposed an EFGT system with a heat exchanger that is capable of providing a TIT of 830 [°C]. For this study, the maximum TIT that is achievable by the heat exchanger is limited to 800 [°C].

4.2.1.2 Combustion chamber

The wide variety of biomass that is available for combustion provides all kinds of biomass types with different chemical compositions. Some biomass types have a low ash melting temperature that adversely affects the combustion chamber and its efficiency. Cuellar (2012) reported that when biomass is being combusted, the formation of slag occurs at temperatures between 800 and 1700 [°C]. Since there are a lot of factors that affect the temperature at which slag is formed, the maximum temperature of the working medium at the combustion chamber outlet is limited to 800 [°C].

4.2.2 Pressure ratio

In the literature, the authors carried out parametric studies of several parameters to observe its effect on the cycle performance of an EFGT system with an electrical power capacity of 100 [kW] and less. A common parameter used was the pressure ratio that were mostly varied between 2 and 6 [-] due to the limitations of the turbomachinery investigated. It was reported that for small scale power production, 4 and 4.5 [-] are the optimum pressure ratio for different operating conditions. This can be seen in Table 2 with listed references. For this study, the pressure ratio is fixed at 4.5 [-].

4.3 Variable parameters

Before efficiency graphs can be generated, the variable input and output parameters needs to be known in order to know which parameters can be varied. By varying the input parameters, the best selection of operating conditions, under all constraints, can be determined for each simulation model. The EES code for each simulation model is written with several inputs that are required to determine the outputs of the model. The EES codes are shown in Appendix C. From these inputs the two main parameters that are variable for an EFGT system are mass flow rate and heat input.

4.3.1 Mass flow rate

The mass flow rate can be physically controlled by means of a fan or throttling valve. This way, the air that enters the system can be controlled as the operator sees fit. The mass flow rate is varied to evaluate the effect it has on the cycle efficiency. The preliminary design model of Amirante *et al.* (2015) achieved an air flow rate of 0.68 [kg/s], while Kautz and Hansen

(2007) obtained a value of 0.78 [kg/s]. From literature the mass flow rate has been reported to be less than 1 [kg/s] for an electrical power capacity of 100 [kW] or less. However, Amirante *et al.* (2015) as well as Kautz and Hansen (2007) did not account for combustion chamber losses and pipe losses which would require a larger amount of air mass flow. Therefore, a convenient guess value range of 0-1 [kg/s] has been used as a start-off point to simulate the models for this study.

4.3.2 Heat input

The heat input to the combustion chamber is also a parameter that will be varied to evaluate the effect it has on the cycle efficiency of the simulation model. The rate at which fuel is added to the combustion chamber can be physically managed by means of a conveyer or an auger that feeds the fuel to the combustion chamber. Heat input values of 360 [kW] and 333 [kW] have been reported by Amirante *et al.* (2015) and Kautz and Hansen (2007) respectively for their EFGT models that are integrated with a reheat system. Without reheat in an EFGT the amount of heat input required by the system will be significantly larger. Furthermore, gearbox and generator efficiencies would affect the amount of heat input required to obtain a certain amount electrical power output. For this reason, a convenient range of 50-600 [kW] has been used to simulate the models.

The range of values for the mass flow rate and heat input can be higher than the values mentioned above, however the above mentioned values are used because they are close to the actual values used in literature.

Note that the ambient inlet temperature and pressure have been set as variable inputs and are dependent on the location and weather conditions should an actual EFGT system be manufactured. In this case, all of the simulation models are compared based on the same ambient conditions in order to make a sensible comparison. Therefore, these parameters have been set for ISO standard conditions of 100 [kPa] and 25 [°C] with a humidity of 60 [%] for the working fluid, which is the same assumption made by authors Kautz and Hansen (2007). The humidity of the working fluid is not incorporated for the purpose of this study, but it can be considered for further investigation into EFGT cycles.

4.4 Assumptions

In practice nothing operates under ideal conditions which means no component operates at 100 [%] efficiency. Below are the assumptions for the components used in each simulation model:

- The working fluid flow is assumed to operate under steady state conditions (Vera *et al.*, 2011b).
- The exhaust pressure at the outlet of each system should be higher than that of the atmosphere in order for the air in the cycle to be exhausted into the environment. Therefore, the air pressure at the outlet is assumed to be 10 [kPa] higher than the air pressure of the environment.
- The effect of humidity on the working fluid is neglected.

4.4.1 Turbine and compressor isentropic efficiencies

Turbine and compressor design determine at what efficiencies they are capable of operating. Based on the turbine and compressor values used by Kautz and Hansen (2007) the turbine isentropic efficiency has been set to a value of 83 [%] and the compressor efficiency is fixed at 77 [%].

4.4.2 Heat exchanger effectiveness

The heat exchanger effectiveness determines how well a heat transfer takes place between to working fluids. The heat exchanger effectiveness that were used by Kautz and Hansen (2007) in their article for EFGTs was 87 [%].

4.4.3 Pressure drop in pipes

Pressure drop in pipes are caused by frictional losses as the air flows through bends, valves, fittings and components. The pressure drop in pipes have been reported to be 1 [%] of the pipe's inlet pressure (Steyn, 2006).

4.4.4 Combustion chamber pressure drop

Vera *et al.* (2011b) reported in their list of constraints that the pressure drop in a combustion chamber is 0.5 [%]. This value has also been used as a constraint for this study.

4.4.5 Heat exchanger pressure drop

According to Datta *et al.* (2010), a heat exchanger pressure drop of 2 [%] on the hot side and 3 [%] on the cold side can be assumed for a heat exchanger. The same values will be incorporated in the simulation models.

4.4.6 Gearbox efficiency and generator efficiency

Vera *et al.* (2011b) reported a generator efficiency of 95 [%] which is also assumed for the generator in the EES models. A value of 95 [%] is assumed for the gearbox efficiency.

4.5 Calculation of an efficiency point

The simulation model of the regenerative cycle is used to explain the logic of the calculation of an efficiency point. The cycles have been programmed based on a node and element approach. An element represents a component and each element has two nodes which represent the element's inlet and outlet conditions. Each node is represented by a number. Figure 16 below illustrates the regenerative cycle. The regenerative cycle consists of the following components and their respective nodes:

- Compressor (Nodes 2-3)
- Turbine (Nodes 8-9)
- Heat Exchanger (Nodes 4-5 and 10-11)
- Combustion Chamber (Nodes 6-7)
- Ducting (Nodes 1-2, 3-4, 5-6, 7-8, 9-10, 11-12)
- Generator

The generator does not have any nodes. It is the amount of power that is delivered and is represented by $\dot{W}_{Gen}$ in the programme code.

EES is a general equation-solving program that can numerically solve thousands of coupled non-linear algebraic and differential equations. For each equation there needs to be an equal amount of variables in order for EES to solve the equation. Firstly, the assumptions and known parameters are given in order to know what is fixed and what equations are needed in order to solve the problem. The assumptions and values were obtained from literature and discussed in Sections 4.2 - 4.4. A summary of these values are listed in Sections 4.5.1 and 4.5.2.

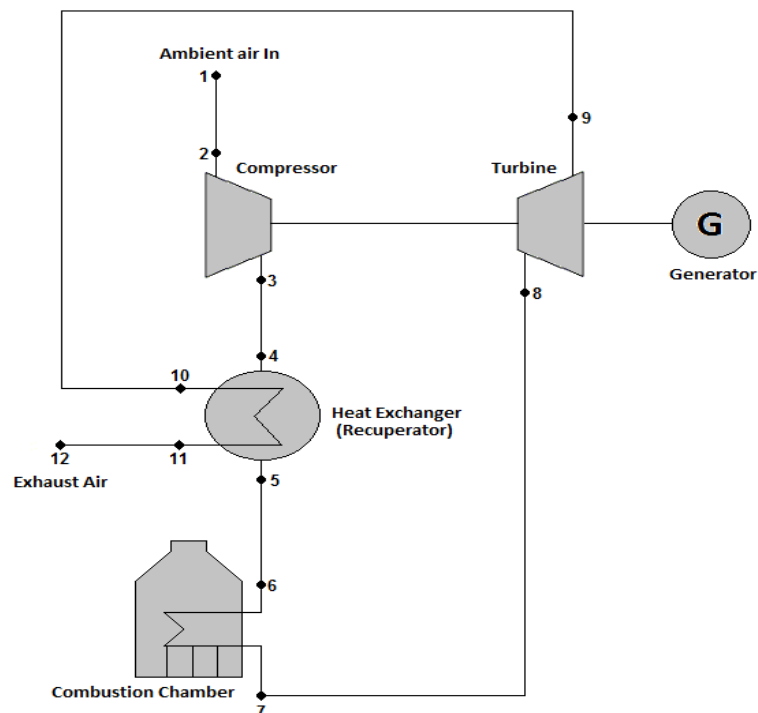


Figure 16 Regenerative cycle

4.5.1 Working fluid properties

The working fluid used is air and it enters the system at ambient conditions. The air is assumed to enter the system at a mass flow rate of 1 [kg/s]. The fluid properties are denoted as:

$$T_{amb} = 25 \text{ [}^\circ\text{C]} \quad (\text{Ambient air temperature})$$

$$P_{amb} = 100 \text{ [kPa]} \quad (\text{Ambient air pressure})$$

$$\dot{m} = 1 \text{ [kg/s]} \quad (\text{Air mass flow rate})$$

4.5.2 Component characteristics

The values below describe the characteristics of the components used which were obtained from literature:

Compressor characteristics

$$Pr_c = 4.5 \text{ [-]} \quad (\text{Compressor pressure ratio})$$

$$\eta_c = 0.77 \text{ [-]} \quad (\text{Compressor isentropic efficiency})$$

Turbine characteristics

$$\eta_c = 0.83 [-] \quad (\text{Turbine isentropic efficiency})$$

Pipe/Ducting characteristics

$$\alpha_p = 0.01 [-] \quad (\text{Fraction of the average pressure drop in pipes/ducting})$$

Heat exchanger characteristics

$$\alpha_{HX,PS} = 0.03 [-] \quad (\text{Fraction of the average pressure drop on primary side})$$

$$\alpha_{HX,SS} = 0.02 [-] \quad (\text{Fraction of the average pressure drop on secondary side})$$

$$\varepsilon_{HX} = 0.87 [-] \quad (\text{Heat exchanger effectiveness})$$

Generator characteristics

$$\eta_{Gen} = 0.95 [-] \quad (\text{Generator efficiency})$$

$$\eta_{gearb} = 0.95 [-] \quad (\text{Gearbox efficiency})$$

Combustion chamber characteristics

$$\dot{Q}_{Comb} = 350 [kW] \quad (\text{Heat added to system by combustion chamber})$$

$$\alpha_{Comb} = 0.005 [-] \quad (\text{Fraction of the average pressure drop in combustion chamber})$$

4.5.3 Node calculation

Now that the known values of the working fluid's properties and component characteristics is given, the next step would be to write down all the relevant equations that can be used with the known values to solve the unknown parameters. The best approach is to use the equations that is applicable to each node. The values obtained for each node will be labelled according to the number of the node. For instance, the temperature at node 1 will be known as $T_{o,1}$. From Figure 16 the equations for each node is given.

The aim is to determine the temperature, pressure, enthalpy and entropy of the air as it flows through each component. With this information, the conservation of energy equation can be used to determine the work and heat for each component in the cycle. As a result, the amount of electricity that the cycle can generate is determined and at which operating conditions the cycle operates.

Node 1 (System inlet)

Node 1 is the inlet to the whole cycle where the air enters at ambient conditions. Therefore,

Cycle inlet air temperature

$$T_{01} = T_{amb}$$

Cycle inlet air pressure

$$P_{01} = P_{amb}$$

Enthalpy of inlet air

$$h_{01} = f(T_{01}, P_{01})$$

Entropy of inlet air

$$s_1 = f(T_{01}, P_{01})$$

With the temperature and pressure known, an EES function can be used to determine the enthalpy and entropy. EES has built-in functions that can be used to determine the properties of different fluids from thermodynamic property tables. The conservation of energy is used to determine the work and heat transfer for the components and therefore the enthalpy and entropy needs to be calculated. The entropy is calculated because the turbine and compressor have isentropic efficiencies. The entropy values are used to determine the actual work done or delivered by compressors and turbines in an EFGT cycle.

Node 2 (Compressor inlet)

Node 2 is where the air flows through the pipe and enters the compressor. Firstly, the pressure at the compressor inlet is determined. This is done by calculating the pressure drop across the pipe section (node 1-2) with Eq. (3.6) which is derived from the conservation of momentum. It is assumed that the pressure drop is a certain fraction of the average pressure between the inlet and outlet of the pipe section. Therefore, the pressure drop through pipe section 1-2 is,

$$\Delta P_{012} = \alpha_p * \left[\frac{(P_{02} + P_{01})}{2} \right]$$

Each pipe section is assumed to have no heat transfer that takes places ($\dot{Q} = 0$). This means that the enthalpy of the fluid that enters the pipe is the same as the enthalpy of fluid at the pipe outlet. From the conservation of energy, if $\dot{Q}$ is zero then Eq. (3.10) reduces to

$$h_{02} = h_{01}$$

Now that the pressure and enthalpy is known the temperature and entropy can be determined with EES built-in functions that determines these values from a thermodynamic property table. Therefore,

Temperature at compressor inlet

$$T_{02} = f(P_{02}, h_{02})$$

Entropy at compressor inlet

$$s_2 = f(T_{02}, P_{02})$$

Node 3 (Compressor outlet)

Node 3 is where the air exits the compressor at a higher pressure and temperature. The pressure of the fluid that enters the compressor and the compression ratio is known. With Eq. (3.14) the value of the compressed air can be calculated. Therefore,

$$P_{03} = P_{02} * Pr_c$$

Given that the pressure have been determined and using the enthalpy of the fluid that enters the compressor, the enthalpy of the compressed fluid for an isentropic process can now be determined with another built-in EES function namely,

$$h_{0s3} = f(s_2, P_{03})$$

The isentropic efficiency of a compressor is defined by Eq. (3.12) which is the actual compressor work divided by the isentropic compressor work. Using both Eq. (3.11) and Eq. (3.9) and solving them simultaneously the enthalpy of the fluid that exits the compressor can be determined. Therefore,

Ideal compressor work

$$\dot{W}_c = \left(\frac{1}{\eta_c} \right) * \dot{m} * (h_{0s3} - h_{02})$$

Actual compressor work

$$\dot{W}_C = \dot{m} * (h_{03} - h_{02})$$

Now that the pressure and enthalpy is known the temperature and entropy can be determined with EES functions. Therefore,

Temperature at compressor outlet

$$T_{03} = f(P_{03}, h_{03})$$

Entropy at compressor outlet

$$s_3 = f(T_{03}, P_{03})$$

Node 4 (HX primary side inlet)

Node 4 is where the air flows through a pipe section before it enters the primary side of the heat exchanger. The properties of the fluid flowing through the pipe section has the same approach as the calculation of the values for node 2. Therefore,

Pressure drop through pipe section 34

$$\Delta P_{034} = \alpha_p * \left[\frac{(P_{04} + P_{03})}{2} \right]$$

Primary side inlet pressure

$$P_{04} = P_{03} - \Delta P_{034}$$

Primary side inlet enthalpy

$$h_{04} = h_{03}$$

Primary side inlet temperature

$$T_{04} = f(P_{04}, h_{04})$$

Primary side inlet entropy

$$s_4 = f(T_{04}, P_{04})$$

Node 5 (HX primary side outlet)

Node 5 is where the air exits the primary side of the heat exchanger at a higher temperature due to the heat that is added to the air. The pressure drop across the heat exchanger primary side is determined by

$$\Delta P_{HX,PS} = \alpha_{HX,PS} * \left[\frac{(P_{05} + P_{04})}{2} \right]$$

With the given fraction of average pressure drop across the heat exchanger primary side. The same approach can be used to determine the pressure as with the pipe sections, because the heat exchanger is modelled as two pipes through which two working fluids, at different temperatures, exchanges heat. The pressure of the air that exits the heat exchanger secondary side is determined with Eq (3.6). Therefore,

$$P_{05} = P_{04} - \Delta P_{HX,PS}$$

The maximum possible heat transfer for the heat exchanger primary side, under ideal conditions, can be determined with Eq (3.20). Thus,

$$\dot{Q}_{HX,PS} = \varepsilon_{HX} * C_{min} * (T_{010} - T_{04})$$

The minimum heat capacity C_{min} is determined by Eq (3.21). A function is used in which EES determines which of the two fluid streams, either on the primary side or the secondary side, has the smallest heat capacity value and it is then used in the equation above.

$$C_{min} = \min(\dot{m} * Cp_4, \dot{m} * Cp_{10})$$

However, since ideal conditions are not assumed and in reality there exist no heat exchanger that gives a 100% heat transfer, the actual heat transfer is determined with Eq (3.22). Therefore,

$$\dot{Q}_{HX,PS} = \dot{m} * (h_{05} - h_{04})$$

The heat transfer equations are solved simultaneously. As a result, the enthalpy of the fluid that exit the heat exchanger primary side is determined. Once again EES built-in functions are used to determine the temperature and entropy. Thus,

$$T_{05} = f(P_{05}, h_{05})$$

$$s_5 = f(T_{05}, P_{05})$$

Node 6 (Combustion chamber inlet)

Node 6 is where the air flows through a pipe section before it enters the combustion chamber. The properties of the fluid flowing through the pipe section has the same approach as the calculation of the values for node 2. Therefore,

Pressure drop through pipe section 56

$$\Delta P_{056} = \alpha_p * \left[\frac{(P_{06} + P_{05})}{2} \right]$$

Pressure at combustion chamber inlet

$$P_{06} = P_{05} - \Delta P_{056}$$

Enthalpy at combustion chamber inlet

$$h_{06} = h_{05}$$

Temperature at combustion chamber inlet

$$T_{06} = f(P_{06}, h_{06})$$

Entropy at combustion chamber inlet

$$s_6 = f(T_{06}, P_{06})$$

In order to better distinguish that T_{06} is the HX primary side outlet temperature, the property name T_{06} is returned as $T_{HXPS,Out}$. Therefore,

$$T_{HXPS,Out} = T_{06}$$

The reason this value's name is changed is because it is an important indicator of what the maximum temperature is at which the heat exchanger can operate. In Section 4.2 an assumption was made that the maximum temperature at which a heat exchanger can operate is 800 [°C] due to metallurgical constraints. When this value is over 800 [°C] then the operating conditions of the cycle is not valid. This means that other operating conditions need to be considered.

Node 7 (Combustion chamber outlet)

Node 7 is where the air exits the combustion chamber at a higher temperature. The pressure drop across the combustion chamber is determined by

$$\Delta P_{067} = \alpha_{Comb} * \left[\frac{(P_{07} + P_{06})}{2} \right]$$

With the given fraction of average pressure drop across the heat exchanger primary side. The pressure of the fluid that exits the heat exchanger primary side is determined with Eq (3.6) which is similar to determining the pressure of the fluid flowing through a pipe section. Therefore,

$$P_{07} = P_{06} - \Delta P_{067}$$

From Eq. 3.10 the heat added to the fluid in the combustion chamber is

$$\dot{Q}_{Comb} = \dot{m} * (h_{07} - h_{06})$$

From the equation above the enthalpy of the air at the combustion chamber outlet can be determined since the value of the heat added is known. With the pressure and enthalpy known EES built-in functions are used to determine the temperature and entropy of the air at the combustion chamber outlet. Therefore,

$$T_{07} = f(P_{07}, h_{07})$$

$$s_7 = f(T_{07}, P_{07})$$

In order to better distinguish that T_{07} is the combustion outlet temperature of the working fluid, the property name T_{07} is returned as $T_{Comb,Out}$. Therefore,

$$T_{Comb,Out} = T_{07}$$

The temperature of the air, as it flows out of the combustion chamber is also an important indicator because the turbine also has a metallurgical limit. If the temperature of the air is higher than the specified limit of the turbine, then it is unrealistic for the cycle to operate under these conditions. This would require a reconsideration of the operating conditions selected for the cycle.

Node 8 (Turbine inlet)

Node 8 is where the air flows through a pipe section before it enters the turbine. The properties of the fluid flowing through the pipe section has the same approach as the calculation of the values for node 2. Therefore,

Pressure drop through pipe section 78

$$\Delta P_{078} = \alpha_p * \left[\frac{(P_{08} + P_{07})}{2} \right]$$

Pressure at turbine inlet

$$P_{08} = P_{07} - \Delta P_{078}$$

Enthalpy at turbine inlet

$$h_{08} = h_{07}$$

Temperature at turbine inlet

$$T_{08} = f(P_{08}, h_{08})$$

Entropy at turbine inlet entropy at turbine inlet

$$s_8 = f(T_{08}, P_{08})$$

Node 9 (Turbine outlet)

Node 9 is where the air exits the turbine at a lower pressure. From Eq (3.18) the pressure of the air that expands through the turbine is determined by,

$$P_{09} = P_{08} * \left(\frac{1}{Pr_t} \right)$$

Using the enthalpy of the air as it enters the turbine and with the pressure now known, the enthalpy of the expanded fluid, for an isentropic process, can now be determined with a built-in EES function namely,

$$h_{0s9} = f(s_8, P_{09})$$

The isentropic efficiency of a turbine is defined by Eq (3.16) which is the ideal turbine work divided by the actual turbine work. Using both Eq (3.15) and Eq (3.17) and solving them simultaneously the enthalpy of the fluid that exits the turbine can be determined. Therefore,

Ideal turbine work

$$\dot{W}_T = \eta_t * \dot{m} * (h_{0s9} - h_{08})$$

Actual turbine work

$$\dot{W}_T = \dot{m} * (h_{09} - h_{08})$$

Now that the pressure and enthalpy is known the temperature and entropy can be determined with EES functions.

Temperature at turbine outlet

$$T_{09} = f(P_{09}, h_{09})$$

Entropy at turbine outlet

$$s_9 = f(T_{09}, P_{09})$$

Node 10 (HX secondary side inlet)

Node 10 is where the air flows through a pipe section before it enters the heat exchanger secondary side. The properties of the fluid flowing through the pipe section has the same approach as the calculation of the values for node 2. Therefore,

Pressure drop through pipe section 910

$$\Delta P_{0910} = \alpha_p * \left[\frac{(P_{010} + P_{09})}{2} \right]$$

Secondary side inlet pressure

$$P_{010} = P_{09} - \Delta P_{0910}$$

Secondary side inlet enthalpy

$$h_{010} = h_{09}$$

Secondary side inlet temperature

$$T_{010} = f(P_{010}, h_{010})$$

Secondary side inlet entropy

$$s_{10} = f(T_{010}, P_{010})$$

Node 11 (HX secondary side outlet)

Node 11 is where the air exits the secondary side of the heat exchanger at a lower temperature due to the heat transfer between the fluids in the secondary and primary side of the heat exchanger.

The pressure drop across the heat exchanger secondary side is determined by

$$\Delta P_{HX,SS} = \alpha_{HX,SS} * \left[\frac{(P_{011} + P_{010})}{2} \right]$$

With the given fraction of average pressure drop across the heat exchanger secondary side. The same approach can be used to determine the pressure as with the pipe sections, because the heat exchanger is modelled as two pipes through which two working fluids, at different temperatures, exchanges heat. The pressure of the air that exits the heat exchanger secondary side is determined with Eq (3.6). Therefore,

$$P_{011} = P_{010} - \Delta P_{HX,SS}$$

The maximum possible heat transfer for the heat exchanger secondary side, under ideal conditions, can be determined with Eq (3.20). Thus,

$$\dot{Q}_{HX,SS} = \varepsilon_{HX} * C_{min} * (T_{05} - T_{10})$$

The minimum heat capacity C_{min} is determined by Eq (3.21). A function is used in which EES determines which of the two fluid streams, either on the primary side or the secondary side, has the smallest heat capacity value which is then returned to the equation above.

$$C_{min} = \min(\dot{m} * Cp_5, \dot{m} * Cp_{10})$$

However, since ideal conditions are not assumed and in reality there exist no heat exchanger that gives a 100% heat transfer, the actual heat transfer is determined with Eq (3.22). Therefore,

$$\dot{Q}_{HX,SS} = \dot{m} * (h_{011} - h_{010})$$

The heat transfer equations are solved simultaneously. As a result, the enthalpy of the fluid that exits the heat exchanger secondary side is determined. EES built-in functions are used to determine the temperature and entropy. Thus,

$$T_{011} = f(P_{011}, h_{011})$$

$$s_{11} = f(T_{011}, P_{011})$$

Node 12 (Cycle Exhaust)

Node 12 is where the air is exhausted into atmosphere. The same equations apply here as with other pipe sections. The cycle is repeated with fresh air entering the cycle.

Pressure Drop through Pipe 1112

$$\Delta P_{01112} = \alpha_p * \left[\frac{(P_{012} + P_{011})}{2} \right]$$

Pipe Section 1112 Pressure

$$P_{012} = P_{011} - \Delta P_{01112}$$

Pipe Section 1112 Pressure

$$P_{012} = P_{amb} + 10$$

The exhaust pressure, P_{012} is assumed to be 10 kPa higher than atmosphere so that the air in the EFGT system is exhausted instead of being drawn into the system.

Pipe Section 1112 Enthalpy

$$h_{012} = h_{011}$$

Pipe Section 1112 Temperature

$$T_{012} = f(P_{012}, h_{012})$$

Pipe Section 1112 Entropy

$$s_{12} = f(T_{012}, P_{012})$$

4.5.4 Other calculations

Energy balance

An energy balance equation is used to ensure that the calculations are correct. An energy balance equation is based on the principle that energy cannot be created nor can it be destroyed. Therefore, the amount of energy put into the system should equal to the amount of energy taken out of the system.

$$E = (\dot{m} * h_{01}) + \dot{W}_C + \dot{W}_T + \dot{Q}_{Comb} - (\dot{m} * h_{012})$$

Specific heat

The specific heat values, at a constant pressure and volume process, for the air at each node is also calculated as it is required to solve the amount of heat added or given by the heat exchanger. Not all specific heat values are used and only serves as additional information to the cycle.

Duplicate i = 1,12

$$Cp_i = f(T_i, P_i)$$

$$Cv_i = f(T_i, P_i)$$

End

Shaft energy balance

Since the compressor, turbine and generator are driven by a single shaft. The shaft energy balance is an additional equation used to solve the whole cycle. From Eq (3.26),

$$\dot{W}_T = -\dot{W}_C - \left[\frac{\dot{W}_{Gen}}{(\eta_{Gen} * \eta_{Gearb})} \right]$$

Total power output

The total power output is the amount of power that is available after all other power requiring components have been subtracted from the total amount of power generated by the system. Therefore,

$$\dot{W}_{Gen} = \dot{W}_{Out}$$

Overall cycle efficiency

The overall cycle efficiency is an important indicator of how well a cycle performs. It is defined as the ratio of the amount of heat added to the amount of work delivered. It follows from Eq (3.24) that,

$$\eta_{Brayton} = \frac{\dot{W}_{Out}}{\dot{Q}_{Comb}}$$

Procedure

A procedure is used to implement an analytical relationship between two or more variables. In this case, the procedure is used to determine whether the air flowing through the primary side of the heat exchanger is heated or cooled. If the air flowing through the primary side is being heated the temperature of the air should be higher than the air. An if function is used to return a set a values depending whether the statement is true or false. The equations used in the procedure also appear at some of the nodes. The equations need not be duplicate and they are only discussed at the relevant nodes to explain the logic of determining the properties of the air flowing through the given component. The procedure is discussed in more detail in Appendix B.

Procedure test($\varepsilon_{HX}, C_{min}, T_{04}, T_{010}, h_{04}, \dot{m} : Regen, \dot{Q}_{HX,SS}, \dot{Q}_{HX,PS}, h_{05}$)

(If heat exchange is taking place)

if ($T_{010} - T_{04} \geq 15$) *Then*

Regen = 1 **(A value of 1 is returned if the statement is true)**

$$\dot{Q}_{HX,SS} = \varepsilon_{HX} * C_{min} * (T_{04} - T_{010})$$

$$\dot{Q}_{HX,PS} = -\dot{Q}_{HX,SS}$$

$$h_{05} = \left(\frac{\dot{Q}_{HX,PS}}{\dot{m}} \right) + h_{04}$$

Else

(If heat exchange is not taking place)

Regen = 0 (A value of 0 is returned if the statement is false)

$$\dot{Q}_{HX,PS} = 0$$

$$\dot{Q}_{HX,SS} = 0$$

$$h_{05} = h_{04}$$

Endif

End

Call procedure

A call procedure is a function in EES which returns the values of the equations used in the procedure. Therefore,

$$Call\ test(\varepsilon_{HX}, C_{min}, T_{04}, T_{010}, h_{04}, \dot{m} : Regen, \dot{Q}_{HX,SS}, \dot{Q}_{HX,PS}, h_{05})$$

The modeling of the regenerative cycle is now complete. The programme is initiated by clicking run and the equations are iteratively solved until they converge. The results obtained are not final as only one efficiency point is determined. The next step in the modeling process is to determine what the maximum possible efficiency would be for and an electrical power output of 100 [kW] under all constraints. The latter is discussed in detail in Chapter 5.

4.6 Conclusion

In this chapter the logic of determining an efficiency point, with the aid of a computer software program named EES, have been discussed. The modeling approach is the same for all relevant EFGT cycles that have been modelled for this study. In the next chapter the methodology of determining the maximum possible efficiency, for the given constraints, is discussed.

CHAPTER 5

BRAYTON CYCLE EFFICIENCY CALCULATION

Chapter 5: Brayton Cycle Efficiency Calculation

5.1 Introduction

In Chapter 4 the logic of determining an efficiency point have been discussed for the regenerative cycle, which was modelled in EES. The EES model can generate any number of efficiencies for a given set of conditions. In order to obtain meaningful results, the maximum possible efficiency, for an electrical power output of 100 [kW] under all constraints, needs to be determined. A certain methodology needs to be followed in order to determine the maximum possible efficiency.

5.2 Regenerative cycle results

The regenerative cycle has been modelled in EES and the logic towards the calculation of a single efficiency point have been discussed in Chapter 4. The results obtained are shown in the table below. The results, in terms of the cycle performance have also been determined and is illustrated in the Table 4.

Table 3 Results obtained from EES for each node.

Node	T_0[i] [C]	P_0[i] [kPa]	h_0s[i] [kJ/kg]	h_0[i] [kJ/kg]	s_[i] [kJ/kg-K]	Cp_[i] [kJ/kg-K]	Cv_[i] [kJ/kg-K]
1	25	100		298,4	6,862	1,006	0,7179
2	25	99		298,4	6,865	1,006	0,7179
3	230,7	445,5	459,1	507,2	6,965	1,032	0,7435
4	230,7	441,1		507,2	6,968	1,032	0,7434
5	521,6	428,1		816,7	7,461	1,098	0,8104
6	521,6	423,8		816,7	7,464	1,098	0,8104
7	831,2	421,7		1167	7,836	1,16	0,8725
8	831,2	417,5		1167	7,839	1,16	0,8725
9	575,4	114,5	816,4	876	7,912	1,11	0,8226
10	575,4	113,4		876	7,915	1,11	0,8226
11	287,8	111,1		566,4	7,476	1,042	0,7551
12	287,8	110		566,4	7,479	1,042	0,7551

Table 4 Regenerative cycle operating conditions.

Description	Parameter	Value	Unit
Air Mass Flow Rate	$m_{\dot{}}$	1	[kg/s]
Combustion Chamber Heat	$Q_{\dot{}}_{\text{Comb}}$	350	[kW]
Cycle Efficiency	η_{Brayton}	0,2023	[-]
Power Generated	$W_{\dot{}}_{\text{Gen}}$	70,79	[kW]
Combustion Chamber Outlet Temperature	$T_{\text{Comb_Out}}$	831,2	[°C]
Heat Exchanger Primary Side Outlet Temperature	$T_{\text{HXPS_Out}}$	521,6	[°C]
Compressor Work	$W_{\dot{}}_{\text{C}}$	208,8	[kW]
Turbine Work	$W_{\dot{}}_{\text{T}}$	-290,7	[kW]
Turbine Pressure Ratio	Pr_{t}	3,646	[-]
Heat Exchange Indication	Regen	1	[-]

The results, from Table 4, show that only 70.79 [kW] of power have been generated by the cycle with an efficiency of 0.2023 (20%). At the current conditions, the combustion chamber outlet temperature is 831.2 [°C] which is higher than the assumed constraint of 800 [°C]. This means that the operating conditions at which the cycle is currently operating are not satisfying the specifications for this study. There is a need to generate 100 [kW] of electricity within the constraints mentioned in Section 4.2. In order to achieve the latter some parameters need to be changed to get the desired results.

5.3 Maximum possible efficiency calculation

The air mass flow rate of the cycle can be varied by using a blower or a fan. Also, the heat added to the cycle can be varied by means of varying the amount of fuel (biomass or wood) added to the combustion chamber. These variable parameters were discussed in detail in Section 4.3. The ambient conditions can also be varied depending on the geographical location of the actual system. However, if two cycles were to be compared with each other at the same rural area then the ambient conditions will have no effect on the results of the cycles as they remain the same for both cycles. It is possible, if more efficient equipment is obtained, that better results will be observed, but more than one cycle are being evaluated and therefore the components and their characteristics are fixed so that a sensible comparison between the cycles can be made.

It is clear that the mass flow rate and the heat added to the system are the only parameters that can be varied to obtain the desired operating conditions. It is also important that there may be a number of combination of values for the mass flow rate and the heat added to the system that is able to achieve the desired operating conditions and within the constraints. Therefore, the best approach would be to determine the operating conditions for the regenerative cycle that can generate 100 [kW] within the constraints and at a maximum efficiency.

A certain methodology needs to be followed in order to determine what amount of heat needs to be added to the system and what the air mass flow rate needs to be for the cycle to achieve a maximum possible efficiency for an electrical output of 100 [kW] under all constraints. Figure 17 shows a step-by-step process that has been followed in order to determine the maximum efficiency.

Before discussing the methodology it should be clear that the goal, as previously mentioned, is to generate efficiency graphs in order to choose the best operating conditions to produce an electrical output of 100 [kW], under all constraints, for the EFGT models considered with the constraints being that the maximum pressure ratio is kept at 4.5 [-] and that the combustion chamber and heat exchanger primary outlet temperature do not exceed a value of 800 [°C] as these components are limited by their metallurgical properties.

Step 1

The first step in determining the maximum efficiency point is to guess a mass flow rate value. A value between 0 and 1 [kg/s] can be selected. The reason for this range is because, from the literature review, the mass flow rate for an EFGT cycle with an electrical power output of 100 [kW] have been less than 1 [kg/s] and therefore it would serve as a good starting point to determine the mass flow rate of the air that flows through the regenerative cycle. A value of 0.2 [kg/s] has been selected.

The next part of step 1 is to vary the amount of heat added to the cycle while the mass flow rate is fixed for each value of heat input. A range of 50 [kW] to 600 [kW] is used for the heat input. The reason for using this heat input range is discussed in Section 4.3.2. Step 1 is now complete, the program is activated to start the calculation process, and the results obtained are shown in Table 5.

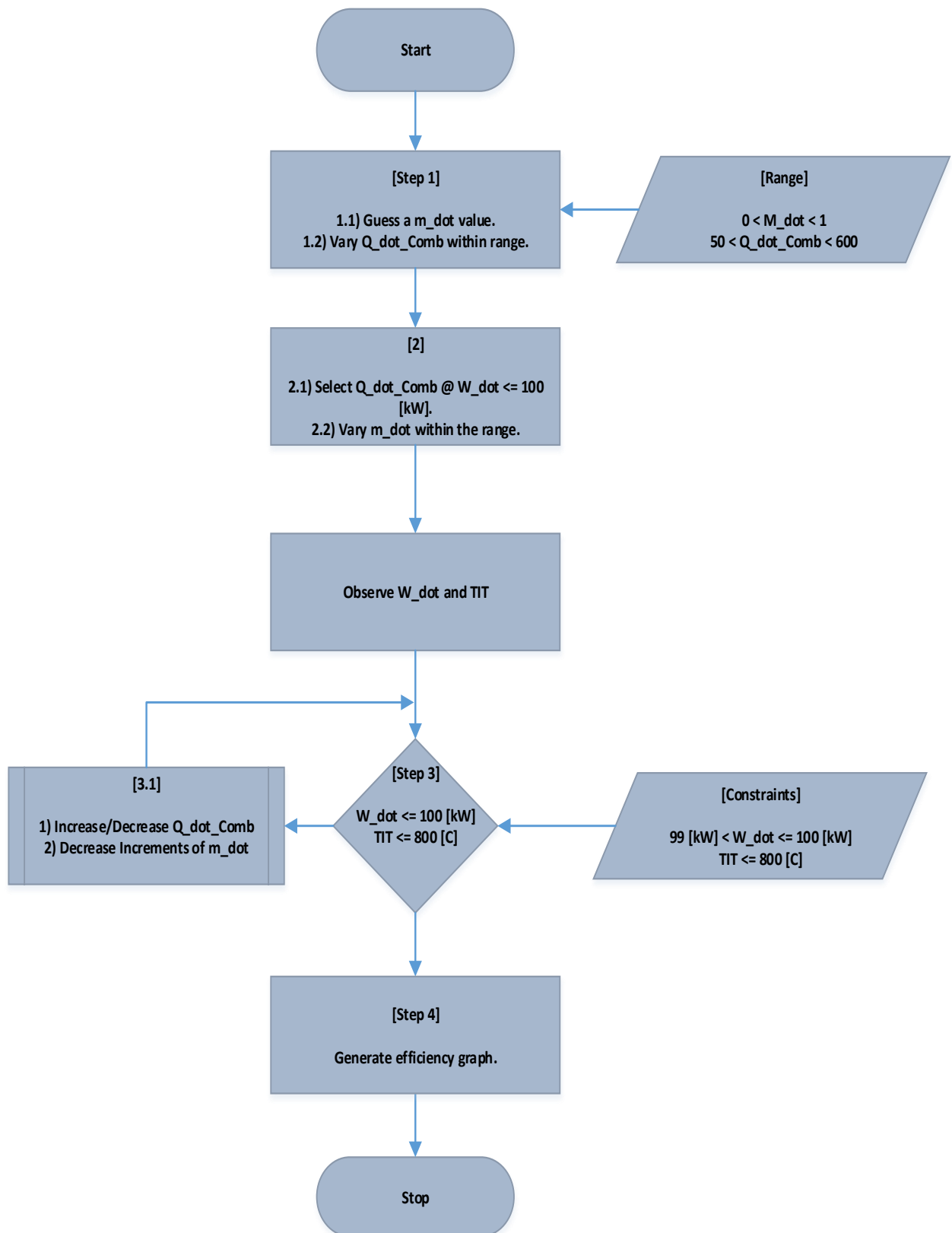


Figure 17 Process flow diagram for determining the maximum possible efficiency for an EFGT cycle with EES.

Note that when the mass flow rate is guessed there is a chance that EES will give a convergence error, due to the guess values that it assigns to the unknown variables in an iterative method. In this case the user should change the guess values or re-run the simulation model with other values for the mass flow rate or heat input until it successfully calculates the unknown values in the table. After this, the guess values need to be updated. The EES user manual can be consulted for further details on the latter.

Table 5 EES results from step 1

Nr.	m_dot	Q_dot_Comb	Eta_Brayton	W_dot_Gen	T_Comb_Out	T_HXPS_Out	Regen
	[kg/s]	[kW]	[-]	[kW]	[°C]	[°C]	[-]
1	0,2	50	0,06471	3,236	593,6	363,8	1
2	0,2	78,95	0,2491	19,67	931,3	586,9	1
3	0,2	107,9	0,3207	34,6	1237	782	1
4	0,2	136,8	0,3556	48,66	1525	960,4	1
5	0,2	165,8	0,3749	62,15	1802	1128	1
6	0,2	194,7	0,3864	75,25	2070	1289	1
7	0,2	223,7	0,3937	88,07	2333	1444	1
8	0,2	252,6	0,3984	100,7	2590	1595	1
9	0,2	281,6	0,4015	113,1	2844	1742	1
10	0,2	310,5	0,4035	125,3	3095	1887	1
11	0,2	339,5	0,4047	137,4	3342	2029	1
12	0,2	368,4	0,4055	149,4	3587	2169	1
13	0,2	397,4	0,4058	161,3	3830	2307	1
14	0,2	426,3	0,4059	173,1	4071	2443	1
15	0,2	455,3	0,4058	184,8	4310	2578	1
16	0,2	484,2	0,4056	196,4	4548	2712	1
17	0,2	513,2	0,4052	208	4783	2844	1
18	0,2	542,1	0,4048	219,4	5018	2975	1
19	0,2	571,1	0,4043	230,9	5251	3105	1
20	0,2	600	0,4037	242,2	5483	3234	1

Step 2

The next step is to evaluate the amount of power generated from Table 5. Select the row that has a power output closest or equal to 100 [kW]. Close observation shows that row 7 matches the criteria with a generated power output of 88.07 [kW] and a heat input value of 223.7 [kW]. It is also observed that the combustion chamber and heat exchanger primary side outlet

temperature exceeds the limit of 800 [°C] by far. In other words, a new mass flow rate value needs to be selected for the cycle to operate within the constraints.

The next part of step 2 is to use the heat input value from row 7 and this time, vary the mass flow rate. The mass flow rate is varied between 0 [kg/s] and 1 [kg/s] while the heat input is selected as 223.7 [kW]. The program is activated to start the calculation process, and the results obtained are shown in Table 6.

Table 6 EES results from step 2

Nr.	m_dot	Q_dot_Comb	Eta_Brayton	W_dot_Gen	T_Comb_Out	T_HXPS_Out	Regen
	[kg/s]	[kW]	[-]	[kW]	[°C]	[°C]	[-]
1	0,1	223,7	0,4059	90,8	4245	2541	1
2	0,1474	223,7	0,4031	90,17	3035	1852	1
3	0,1947	223,7	0,3949	88,33	2387	1476	1
4	0,2421	223,7	0,3831	85,69	1979	1234	1
5	0,2895	223,7	0,3686	82,45	1696	1064	1
6	0,3368	223,7	0,3519	78,72	1486	936,4	1
7	0,3842	223,7	0,3334	74,59	1324	836,1	1
8	0,4316	223,7	0,3134	70,1	1194	754,7	1
9	0,4789	223,7	0,2919	65,31	1087	687	1
10	0,5263	223,7	0,2693	60,24	997,3	629,5	1
11	0,5737	223,7	0,2456	54,94	920,7	580	1
12	0,6211	223,7	0,2209	49,42	854,3	536,7	1
13	0,6684	223,7	0,1954	43,72	796,1	498,6	1
14	0,7158	223,7	0,1692	37,85	744,7	464,6	1
15	0,7632	223,7	0,1423	31,83	698,8	434,1	1
16	0,8105	223,7	0,1148	25,68	657,5	406,6	1
17	0,8579	223,7	0,08683	19,42	620,3	381,7	1
18	0,9053	223,7	0,0584	13,06	586,4	358,9	1
19	0,9526	223,7	0,02956	6,614	555,5	338,1	1
20	1	223,7	0,000384	0,08581	527,1	319	1

Step 3

Observing the values obtained from Table 6. Row 13 shows a combustion outlet temperature of 796.1 [°C], which is within the constraints. However, the amount of power generated at the conditions displayed in row 13 is only 43.72 [kW]. The problem here is that the combination of the heat input and mass flow rate is not correct. In order to get the desired results, the next

step is through a trial and error process in which the mass flow rate and heat input are either increased or decreased until a generated power output of 100 [kW] is achieved and within the temperature limit of 800 [°C] for the combustion chamber outlet as well as the heat exchanger primary side outlet. The latter need to be determined at the maximum efficiency possible.

Step 1 and 2 was necessary to get close to the desired operating conditions for the regenerative cycle. For step 3, an iterative process is required to get the operating conditions under the specified constraints. The best approach in this iterative process is by doing the following:

- Increase the heat input value with small increments of say 5 [kW] until a power output of 100 [kW] is obtained.
- While the heat input is being increased the combustion outlet temperature should be observed for each increment. If the temperature starts to deviate from 800 [°C] by more than 10 [°C] then the mass flow range should change accordingly until the variable parameters reaches a value of $99 \leq \dot{W}_{out} \leq 100 [kW]$, $790 [^{\circ}C] \leq T_{Comb,Out} \leq 800 [^{\circ}C]$ and $\eta_{Brayton} = \text{maximum possible}$.

After a few iterative steps the desired operating conditions have been determined and is shown in Table 7. The results have also been plotted on a graph to show how the efficiency changes as the mass flow rate changes. This is illustrated in Figure 18.

It can be seen in Table 7 that the values shown in row 10 are the operating conditions at which the cycle needs to operate as they are within the constraints and meets the specifications. The temperature of the combustion chamber outlet (799.8 [°C]) and the heat exchanger primary side outlet (501 [°C]) are within the temperature limit of 800 [°C]. The power generated, for a mass flow rate of 1.514 [kg/s] and a heat input of 509 [kW], is 100.4 [kW]. At these operating conditions, the maximum possible efficiency is at 0.1972 [-]. The maximum possible efficiency point is marked in red on the efficiency graph in Figure 18. From the graph it is clear that there are higher values of cycle efficiency however, the operating conditions of the cycle at these efficiencies are not within the constraints.

Table 7 EES results from step 3

Nr.	m_dot [kg/s]	Q_dot_Comb [kW]	Eta_Brayton [-]	W_dot_Gen [kW]	T_Comb_Out [°C]	T_HXPS_Out [°C]	Regen [-]
1	1,49	509	0,2028	103,2	812,1	509,1	1
2	1,493	509	0,2022	102,9	810,7	508,2	1
3	1,495	509	0,2016	102,6	809,3	507,2	1
4	1,498	509	0,2009	102,3	807,9	506,3	1
5	1,501	509	0,2003	102	806,6	505,4	1
6	1,503	509	0,1997	101,6	805,2	504,6	1
7	1,506	509	0,199	101,3	803,8	503,7	1
8	1,508	509	0,1984	101	802,5	502,8	1
9	1,511	509	0,1978	100,7	801,1	501,9	1
10	1,514	509	0,1972	100,4	799,8	501	1
11	1,516	509	0,1965	100	798,4	500,1	1
12	1,519	509	0,1959	99,71	797,1	499,2	1
13	1,522	509	0,1953	99,39	795,8	498,4	1
14	1,524	509	0,1946	99,07	794,4	497,5	1
15	1,527	509	0,194	98,74	793,1	496,6	1
16	1,529	509	0,1934	98,42	791,8	495,7	1
17	1,532	509	0,1927	98,1	790,5	494,9	1
18	1,535	509	0,1921	97,78	789,2	494	1
19	1,537	509	0,1915	97,45	787,9	493,1	1
20	1,54	509	0,1908	97,13	786,5	492,3	1

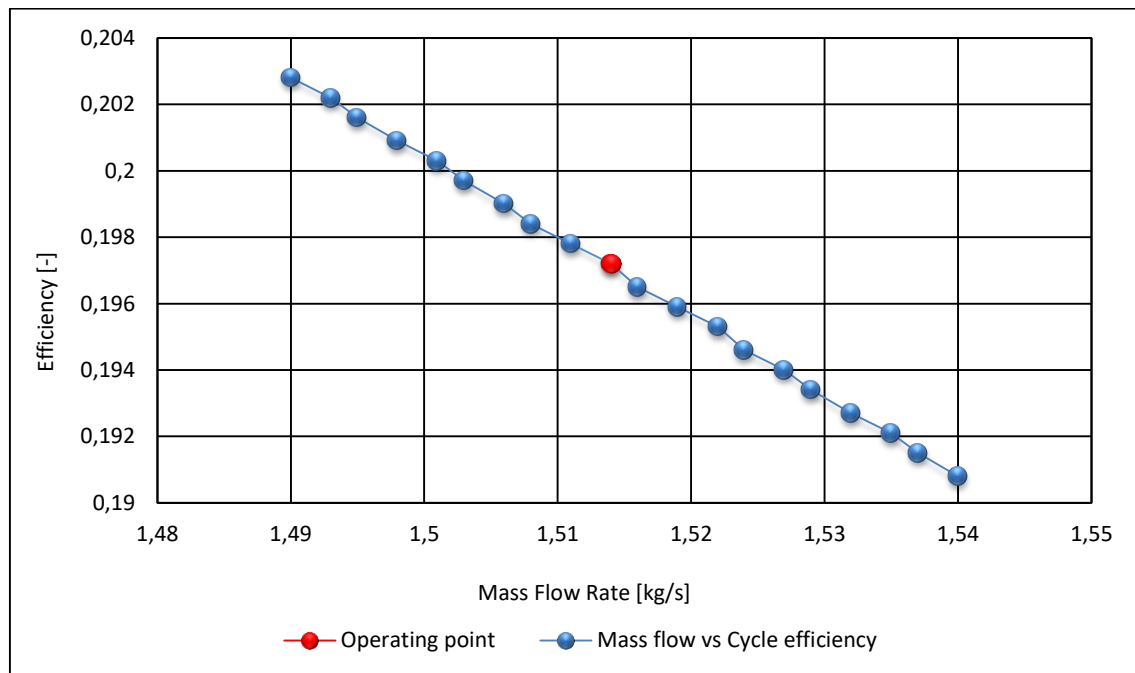


Figure 18 Mass flow rate versus efficiency graph for the regenerative cycle.

The high mass flow rate is due to the high heat input required to keep the combustion outlet temperature within the constraints. For a fixed heat input a higher mass flow rate will result in a lower combustion outlet temperature, because there is less time to transfer heat energy to the working fluid as it flows through the combustion chamber. The high heat input requirement is due to component losses and the efficiencies at which they operate that requires more heat energy in order to maintain a power output of 100 [kW].

Using the mass flow rate and heat input for the newly obtained efficiency of the regenerative cycle, the values of each node at this efficiency point can be determined and is listed in Table 8 below.

Table 8 Regenerative cycle results for each node at the maximum possible efficiency.

Node	T ₀ [i]	P ₀ [i]	h _{0s} [i]	h ₀ [i]	s _[i]	C _p [i]	C _v [i]
	[C]	[kPa]	[kJ/kg]	[kJ/kg]	[kJ/kg-K]	[kJ/kg-K]	[kJ/kg-K]
1	25	100		298,4	6,862	1,006	0,7179
2	25	99		298,4	6,865	1,006	0,7179
3	230,7	445,5	459,1	507,2	6,965	1,032	0,7435
4	230,7	441,1		507,2	6,968	1,032	0,7434
5	500,2	428,1		793,3	7,431	1,093	0,8054
6	500,2	423,8		793,3	7,434	1,093	0,8054
7	798,6	421,7		1129	7,802	1,154	0,867
8	798,6	417,5		1129	7,805	1,154	0,867
9	549,3	114,5	789,3	847,1	7,877	1,104	0,8167
10	549,3	113,4		847,1	7,88	1,104	0,8167
11	282,5	111,1		561	7,466	1,041	0,7539
12	282,5	110		561	7,469	1,041	0,7539

Similarly, the operating conditions of the regenerative cycle at this efficiency point is also determined and listed in Table 9. With these newly obtained operating conditions, an evaluation will be made by comparing these results against the results obtained from other EFGT cycle configurations.

Table 9 Regenerative cycle operating conditions at maximum possible efficiency.

Description	Parameter	Value	Unit
Air Mass Flow Rate	m_dot	1,514	[kg/s]
Combustion Chamber Heat	Q_dot_Comb	509	[kW]
Cycle Efficiency	Eta_Brayton	0,1972	[-]
Power Generated	W_dot_Gen	100,4	[kW]
Combustion Chamber Outlet Temperature	T_Comb_Out	799,8	[°C]
Heat Exchanger Primary Side Outlet Temperature	T_HXPS_Out	501	[°C]
Heat Exchange Indication	Regen	1	[-]
Compressor Work	W_dot_C	316,1	[kW]
Turbine Work	W_dot_T	-427,2	[kW]
Turbine Pressure Ratio	Pr_t	3,646	[-]

5.4 Conclusion

In this chapter the methodology of determining the maximum possible efficiency, under all constraints and for an electrical power output of 100 [kW], have been discussed. The methodology can be applied to any EFGT cycle. The efficiencies and operating conditions of the EFGT cycles selected for the purpose of this study have been determined with this methodology and the results obtained are shown in Chapter 6. Furthermore, in Chapter 6, the EFGT cycles are compared to determine which of the cycles can be used for power generation in rural areas. The simple EFGT cycle that have been modelled with EES is also validated against Flownex[®], a valid thermal-fluid simulation software package.

CHAPTER 6

CYCLE COMPARISON AND MODEL VERIFICATION

Chapter 6: Cycle Comparison and Model Verification

6.1 Introduction

The previous chapter focused on the logic behind the calculation of an efficiency point for an EFGT cycle as well as determining the maximum efficiency, under all constraints, for an electrical power output of 100 [kW]. In this chapter, the operating conditions for different EFGT cycle configurations have been determined by using the same approach that were discussed in Chapter 4. The EFGT cycle configurations are compared against each other in order to determine which of these configurations is the most suitable to provide electricity in rural areas. Included in this chapter is the verification of the results obtained for the simple EFGT cycle against Flownex®.

6.2 EFGT cycle configurations

Four EFGT cycle configurations have been selected to evaluate their performance against each other in order to determine the best configuration to give an electrical power output of 100 [kW] under all constraints for the purpose of providing electricity in rural areas. Each cycle has been modelled in EES and the approach is the same as the regenerative cycle that were discussed in Chapter 4. Each cycle has a different configuration and the amount of nodes and components are determined by the configuration. The EES code for each cycle is shown in Appendix B. The four EFGT cycle configurations are:

- Simple cycle.
- Regenerative cycle.
- Regenerative cycle with two turbines in series.
- Regenerative cycle with reheating.

6.2.1 EFGT cycle configurations and their results

The same assumptions and constraints are applied to each EFGT cycle configuration as with the regenerative cycle that were discussed in Chapter 4. Each cycle should also have an electrical power output of 100 [kW]. For each cycle the operating conditions are determined as well the maximum possible efficiency under all constraints. Each cycle is also briefly

discussed. The regenerative cycle's results have already been determined and will only be shown in Table 19 where the results of each cycle is listed.

6.2.1.1 Simple cycle

Figure 19 illustrates a schematic layout of a simple EFGT cycle configuration. This configuration is made up of a compressor, an external combustion chamber, a turbine and a generator. A similar model was presented by Kautz and Hansen, (2007) as well as Bdour et al., (2016). It was reported that the relatively poor efficiency of this cycle is mainly due to the working fluid that exits the system with a relatively high temperature. As a results the cycle releases a large amount of unutilized heat into the atmosphere. This configuration is used for the purpose of illustrating the performance of the most basic EFGT that is available against other configurations that operates more efficiently. The results obtained for the simple EFGT cycle is listed in Table 10,11 and 12. The mass flow rate versus efficiency graph is plotted and illustrated in Figure 20.

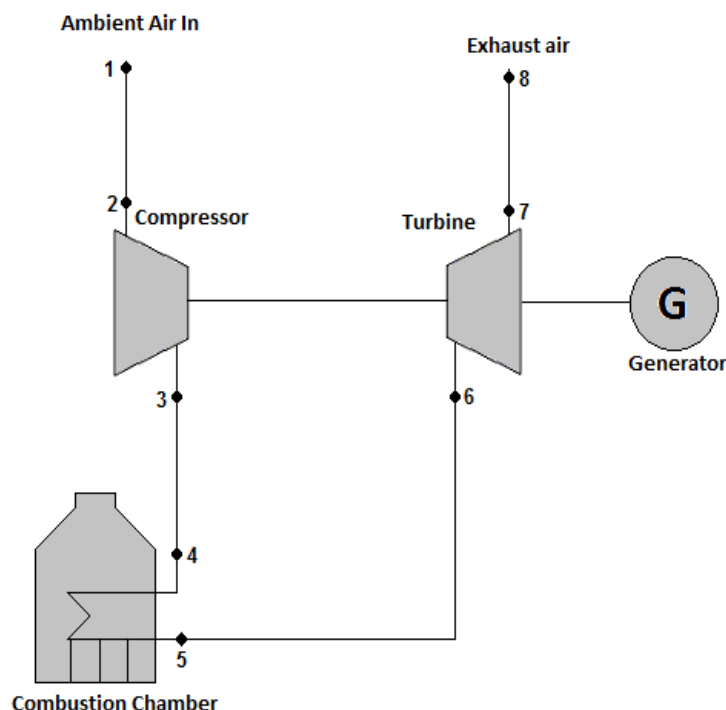


Figure 19 Simple EFGT cycle configuration

Table 10 Performance of a simple cycle for different mass flow rate values.

Nr.	$\dot{m}$ [kg/s]	$\dot{Q}_{\text{Comb}}$ [kW]	η_{Brayton} [-]	$\dot{W}_{\text{Gen}}$ [kW]	$T_{\text{Comb_Out}}$ [°C]
1	1,2	800	0,1311	104,8	837,3
2	1,211	800	0,1303	104,2	832,3
3	1,221	800	0,1295	103,6	827,4
4	1,232	800	0,1287	103	822,6
5	1,242	800	0,1279	102,4	817,8
6	1,253	800	0,1272	101,7	813,1
7	1,263	800	0,1264	101,1	808,5
8	1,274	800	0,1256	100,5	804
9	1,284	800	0,1248	99,84	799,6
10	1,295	800	0,124	99,21	795,2
11	1,305	800	0,1232	98,58	790,8
12	1,316	800	0,1224	97,95	786,6
13	1,326	800	0,1216	97,32	782,4
14	1,337	800	0,1209	96,68	778,3
15	1,347	800	0,1201	96,05	774,2
16	1,358	800	0,1193	95,41	770,2
17	1,368	800	0,1185	94,78	766,3
18	1,379	800	0,1177	94,14	762,4
19	1,389	800	0,1169	93,5	758,5
20	1,4	800	0,1161	92,87	754,8

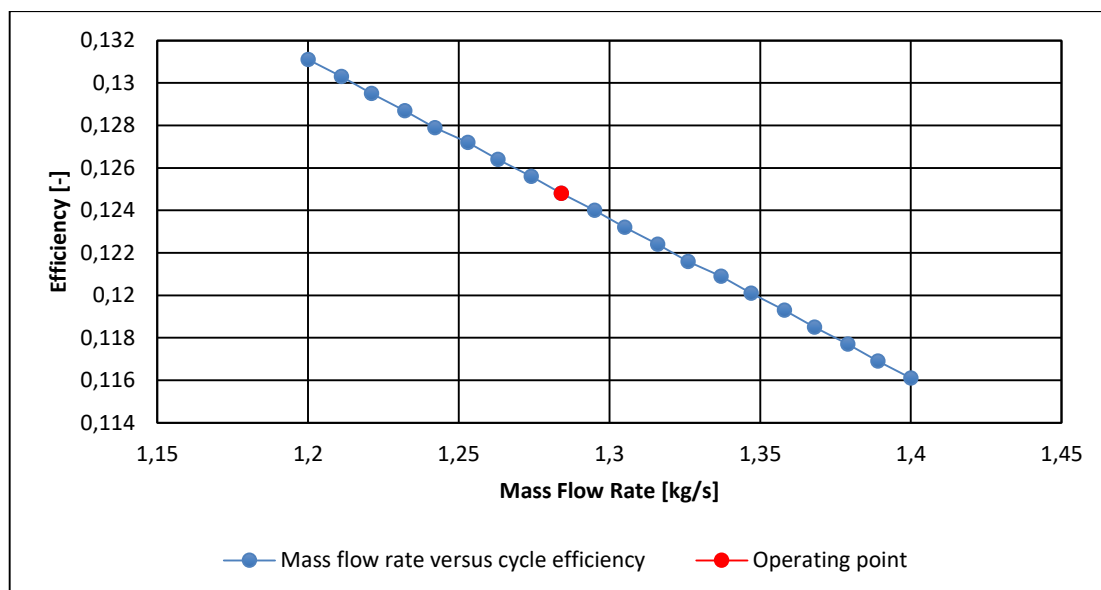


Figure 20 Mass flow rate versus efficiency graph for the simple cycle.

Table 11 Simple cycle result for each node at the maximum possible efficiency.

Node	T ₀ [i] [C]	P ₀ [i] [kPa]	h _{0s} [i] [kJ/kg]	h ₀ [i] [kJ/kg]	s _[i] [kJ/kg-K]	Cp _[i] [kJ/kg-K]	Cv _[i] [kJ/kg-K]
1	25	100		298,4	6,862	1,006	0,7179
2	25	99		298,4	6,865	1,006	0,7179
3	230,7	445,5	459,1	507,2	6,965	1,032	0,7435
4	230,7	441,1		507,2	6,968	1,032	0,7434
5	799,6	438,9		1130	7,791	1,154	0,8672
6	799,6	434,5		1130	7,794	1,154	0,8672
7	538,6	111,1	774,9	835,3	7,871	1,101	0,8142
8	538,6	110		835,3	7,874	1,101	0,8142

Table 12 Simple cycle operating conditions at maximum possible efficiency.

Description	Parameter	Value	Unit
Air Mass Flow Rate	m _{dot}	1,284	[kg/s]
Combustion Chamber Heat	Q _{dot} _Comb	800	[kW]
Cycle Efficiency	Eta_Brayton	0,1248	[-]
Power Generated	W _{dot} _Gen	99,85	[kW]
Combustion Chamber Outlet Temperature	T_Comb_Out	799,6	[°C]
Heat Exchanger Primary Side Outlet Temperature	T_HXPS_Out	N/A	[°C]
Heat Exchange Indication	Regen	N/A	[-]
Compressor Work	W _{dot} _C	268,1	[kW]
Turbine Work	W _{dot} _T	-378,7	[kW]
Turbine Pressure Ratio	Pr _t	3,911	[-]

6.2.1.2 Regenerative cycle with two turbines in series

This configuration is similar to the regenerative cycle discussed in Chapter 4, but has an additional turbine in the cycle. It therefore employs the same calculations and modeling approach except for the shaft energy balance as this configuration consists of two individual turbines with shafts. One shaft is connected to the first turbine and the compressor while the other shaft is connected to the second turbine and the generator. A schematic layout of this regenerative cycle with two series connected turbines is illustrated in Figure 21.



Table 13 Performance of a regenerative cycle, containing two turbines in series, for different mass flow rate values.

Nr.	m_dot	Q_dot_Comb	Eta_Brayton	W_dot_Gen	T_Comb_Out	T_HXPS_Out	Regen
	[kg/s]	[kW]	[-]	[kW]	[°C]	[°C]	[-]
1	1,25	500	0,2553	127,7	939,3	590,6	1
2	1,279	500	0,2487	124,4	919,4	577,8	1
3	1,308	500	0,242	121	900,4	565,4	1
4	1,337	500	0,2352	117,6	882	553,5	1
5	1,366	500	0,2284	114,2	864,4	542,1	1
6	1,395	500	0,2215	110,7	847,4	531	1
7	1,424	500	0,2145	107,2	831	520,3	1
8	1,453	500	0,2074	103,7	815,2	510	1
9	1,482	500	0,2003	100,2	799,9	500	1
10	1,511	500	0,1932	96,58	785,2	490,4	1
11	1,539	500	0,1859	92,96	770,9	481	1
12	1,568	500	0,1786	89,31	757,1	472	1
13	1,597	500	0,1713	85,64	743,8	463,2	1
14	1,626	500	0,1639	81,94	730,9	454,7	1
15	1,655	500	0,1564	78,21	718,3	446,5	1
16	1,684	500	0,1489	74,45	706,2	438,5	1
17	1,713	500	0,1413	70,67	694,4	430,7	1
18	1,742	500	0,1337	66,87	683	423,2	1
19	1,771	500	0,1261	63,04	671,9	415,9	1
20	1,8	500	0,1184	59,19	661,1	408,8	1

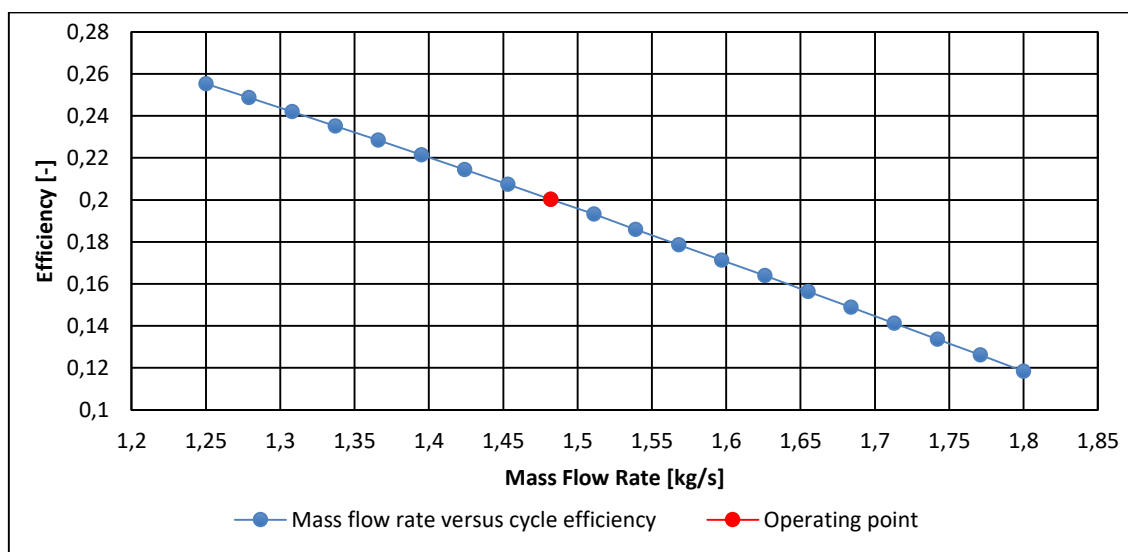


Figure 22 Mass flow rate versus efficiency for a regenerative cycle containing two turbines in series.

Table 14 Regenerative cycle, containing two turbines in series, result for each node at the maximum possible efficiency.

Node	T ₀ [i] [C]	P ₀ [i] [kPa]	h _{0s} [i] [kJ/kg]	h ₀ [i] [kJ/kg]	s _[i] [kJ/kg-K]	Cp _[i] [kJ/kg-K]	Cv _[i] [kJ/kg-K]
1	25	100		298,4	6,862	1,006	0,7179
2	25	99		298,4	6,865	1,006	0,7179
3	230,7	445,5	459,1	507,2	6,965	1,032	0,7435
4	230,7	441,1		507,2	6,968	1,032	0,7434
5	499,9	428,1		792,9	7,43	1,093	0,8054
6	499,9	423,8		792,9	7,433	1,093	0,8053
7	799,7	421,7		1130	7,803	1,154	0,8672
8	799,7	417,5		1130	7,806	1,154	0,8672
9	616,2	167,5	878,7	921,5	7,855	1,119	0,8315
10	616,3	165,8		921,5	7,858	1,119	0,8315
11	548,9	114,5	831,3	846,6	7,877	1,104	0,8166
12	548,9	113,4		846,6	7,879	1,104	0,8166
13	282,5	111,1		560,9	7,466	1,041	0,7539
14	282,5	110		560,9	7,469	1,041	0,7539

Table 15 Regenerative cycle, containing two turbines in series, operating conditions at maximum possible efficiency.

Description	Parameter	Value	Unit
Air Mass Flow Rate	m _{dot}	1,482	[kg/s]
Combustion Chamber Heat	Q _{dot} _Comb	500	[kW]
Cycle Efficiency	Eta_Brayton	0,2003	[-]
Power Generated	W _{dot} _Gen	100,2	[kW]
Combustion Chamber Outlet Temperature	T_Comb_Out	799,9	[°C]
Heat Exchanger Primary Side Outlet Temperature	T_HXPS_Out	500	[°C]
Heat Exchange Indication	Regen	1	[-]
Compressor Work	W _{dot} _C	309,4	[kW]
Turbine 1 Work	W _{dot} _T1	-309,4	[kW]
Turbine 2 Work	W _{dot} _T2	-110,9	[kW]
Turbine 1 Pressure Ratio	Pr_t1	2,493	[-]
Turbine 2 Pressure Ratio	Pr_t2	1,448	[-]

A newly proposed regenerative cycle was reported by Goodarzi, (2016). It was claimed that this configuration has a better thermodynamic performance than the conventional regenerative cycle. The newly proposed regenerative cycle utilise one heat exchanger that operates as a recuperator and serves as a reheat for the two turbines. A schematic layout of this configuration is illustrated in Figure 23. The modeling approach towards this configuration is the same as the regenerative cycle with two series connected turbines. Apart from the different layout of this newly proposed regenerative cycle, there is no additional equations or theory that needs to be employed to develop a simulation model in EES. The results obtained for the regenerative EFGT cycle with two series connected turbines is listed in Table 16,17 and 18. The mass flow rate versus efficiency graph is plotted and illustrated in Figure 24.

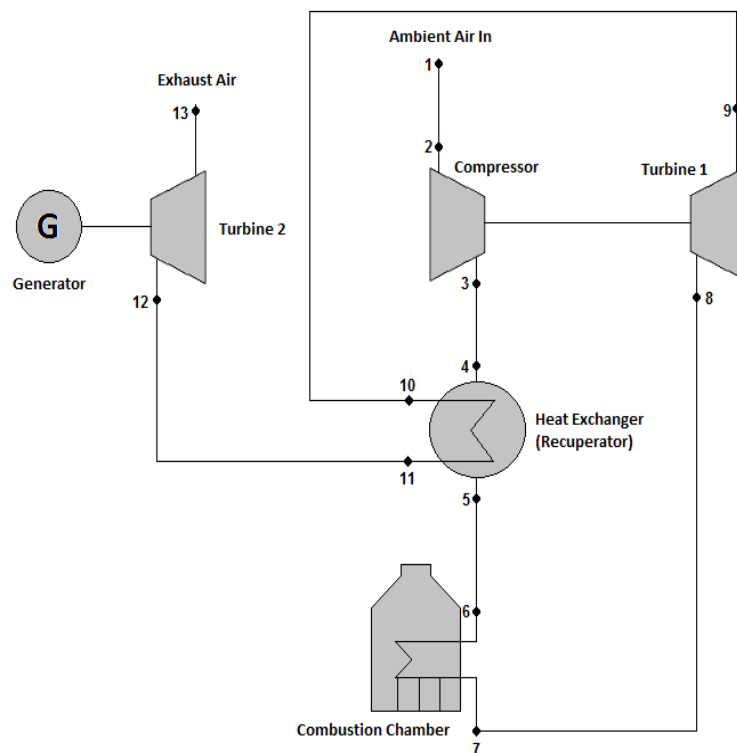


Figure 23 Regenerative EFGT cycle configuration with reheat.

Table 16 Performance of a regenerative cycle, with reheating, for different mass flow rate values.

Nr.	$\dot{m}$ [kg/s]	$\dot{Q}_{\text{Comb}}$ [kW]	η_{Brayton} [-]	$\dot{W}_{\text{Gen}}$ [kW]	$T_{\text{Comb_Out}}$ [°C]	$T_{\text{HXPS_Out}}$ [°C]	Regen [-]
1	2,1	625	0,1833	114,6	887,1	627,4	1
2	2,116	625	0,1812	113,3	878	620	1
3	2,132	625	0,179	111,9	869	612,6	1
4	2,147	625	0,1768	110,5	860,1	605,2	1
5	2,163	625	0,1745	109,1	851,2	597,8	1
6	2,179	625	0,1722	107,6	842,4	590,5	1
7	2,195	625	0,1698	106,1	833,7	583,2	1
8	2,211	625	0,1673	104,6	825	576	1
9	2,226	625	0,1648	103	816,3	568,8	1
10	2,242	625	0,1622	101,4	807,8	561,6	1
11	2,258	625	0,1595	99,69	799,2	554,5	1
12	2,274	625	0,1568	97,97	790,8	547,4	1
13	2,289	625	0,1539	96,2	782,3	540,3	1
14	2,305	625	0,151	94,38	774	533,2	1
15	2,321	625	0,148	92,51	765,6	526,2	1
16	2,337	625	0,1449	90,59	757,3	519,2	1
17	2,353	625	0,1418	88,61	749,1	512,2	1
18	2,368	625	0,1385	86,58	740,8	505,2	1
19	2,384	625	0,1352	84,48	732,7	498,3	1
20	2,4	625	0,1317	82,33	724,5	491,3	1

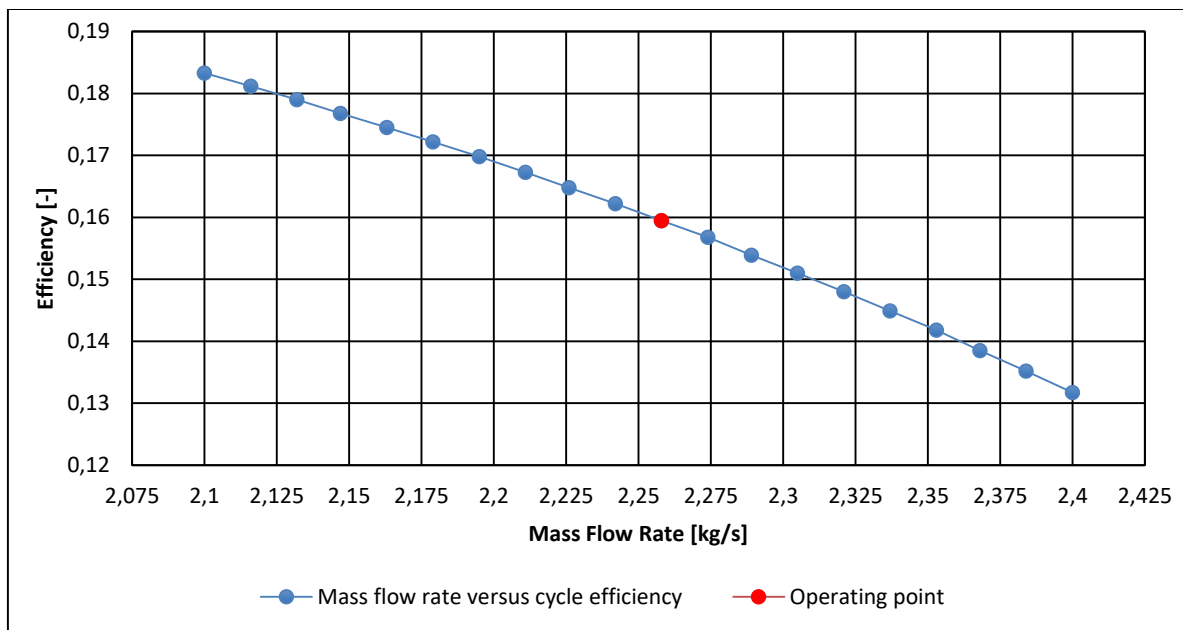


Figure 24 Mass flow rate versus efficiency for a regenerative cycle with reheating.

Table 17 Regenerative cycle, with reheating, result for each node at the maximum possible efficiency.

Node	T ₀ [i] [C]	P ₀ [i] [kPa]	h _{0s} [i] [kJ/kg]	h ₀ [i] [kJ/kg]	s _[i] [kJ/kg-K]	Cp _[i] [kJ/kg-K]	Cv _[i] [kJ/kg-K]
1	25	100		298,4	6,862	1,006	0,7179
2	25	99		298,4	6,865	1,006	0,7179
3	230,7	445,5	459,1	507,2	6,965	1,032	0,7435
4	230,7	441,1		507,2	6,968	1,032	0,7434
5	569	428,1		868,9	7,524	1,109	0,8212
6	569	423,8		868,9	7,527	1,109	0,8212
7	816,5	421,7		1150	7,821	1,157	0,8701
8	816,5	417,5		1150	7,824	1,157	0,8701
9	633,6	170,2	898,2	940,9	7,872	1,122	0,8352
10	633,6	168,5		940,9	7,875	1,122	0,8352
11	300	165,1		579,2	7,384	1,045	0,7578
12	300	163,5		579,2	7,387	1,045	0,7578
13	250,6	110	517,4	527,9	7,407	1,035	0,7471

Table 18 Regenerative cycle, with reheating, operating conditions at maximum possible efficiency.

Description	Parameter	Value	Unit
Air Mass Flow Rate	m _{dot}	2,258	[kg/s]
Combustion Chamber Heat	Q _{dot} _Comb	625	[kW]
Cycle Efficiency	Eta_Brayton	0,1595	[-]
Power Generated	W _{dot} _Gen	99,69	[kW]
Combustion Chamber Outlet Temperature	T_Comb_Out	799,2	[°C]
Heat Exchanger Primary Side Outlet Temperature	T_HXPS_Out	554,5	[°C]
Heat Exchange Indication	Regen	1	[-]
Compressor Work	W _{dot} _C	471,4	[kW]
Turbine 1 Work	W _{dot} _T1	-471,4	[kW]
Turbine 2 Work	W _{dot} _T2	-110,4	[kW]
Turbine 1 Pressure Ratio	Pr_t1	2,494	[-]
Turbine 2 Pressure Ratio	Pr_t2	1,462	[-]

6.3 Evaluation and comparison of different EFGT cycles

Now that the results of different cycle configurations have been determined. The results for each configuration is listed in Table 19 in order to evaluate and compare them against each other.

Table 19 Performance results for different EFGT configurations.

Description	Parameter	Unit	Simple	Regenerative	Regenerative Two Turbines in Series	Reheat
Air Mass Flow Rate	m_dot	[kg/s]	1,284	1,514	1,482	2,258
Combustion Chamber Heat	Q_dot_Comb	[kW]	800	509	500	625
Cycle Efficiency	Eta_Brayton	[-]	0,1248	0,1972	0,2003	0,1595
Power Generated	W_dot_Gen	[kW]	99,85	100,4	100,2	99,69
Combustion Chamber Outlet Temperature	T_Comb_Out	[°C]	799,6	799,8	799,9	799,2
Heat Exchanger Primary Side Outlet Temperature	T_HXPS_Out	[°C]	N/A	501	500	554,5
Heat Exchange Indication	Regen	[-]	N/A	1	1	1
Compressor Work	W_dot_C	[kW]	268,1	316,1	309,4	471,4
Turbine 1 Work	W_dot_T1	[kW]	-378,7	-427,2	-309,4	-471,4
Turbine 2 Work	W_dot_T2	[kW]	N/A	N/A	-110,9	-110,4
Turbine 1 Pressure Ratio	Pr_t1	[-]	3,911	3,646	2,493	2,494
Turbine 2 Pressure Ratio	Pr_t2	[-]	N/A	N/A	1,448	1,462

Table 19 shows the performance of the EFGT cycles that are subject to an electrical power output capacity of 100 [kW] under all constraints. A cycle thermal efficiency of 0.3 [-] that was achieved by Kautz and Hansen (2007) is considered to be a very good cycle efficiency. The simple EFGT cycle, having a cycle thermal efficiency of 0.1248 [-], performed the worst of all the configurations considered. The reason for this poor performance is that a large amount of unused thermal heat energy is exhausted into the atmosphere that could have otherwise been put to good use by means of a heat exchanger to increase the temperature of the working fluid and produce more power at a lower rate of heat input.

The regenerative and reheat cycles performed significantly better due to the heat exchanger that they are equipped with, that recovers the thermal energy exhausted by the turbine to increase the compressed air that flows from the compressor to the combustion chamber. The

EFGT cycle with reheat was the third best performer with a cycle thermal efficiency of 0.1595 [-]. The relative low efficiency of this cycle is due to the first turbine that is limited to a TIT of 800 [°C]. The thermal energy exhausted by the first turbine is circulated through a heat exchanger which heats up the compressed air that exits the compressor. Since the air temperature can only be increased to a certain extent because of the constraints, the remaining heat energy is exhausted by the heat exchanger and flows to the second turbine which produces the nett electrical power output. The temperature of the air that enters the second turbine is much lower than 800 [°C] resulting in a less efficient production of power compared to the regenerative cycle as well as the regenerative cycle with two series connected turbines.

The regenerative cycle and the regenerative cycle with two series connected turbines were the best performers of all the configurations considered. The regenerative cycle with two turbines in series displayed the best performance with a cycle thermal efficiency of 0.2 [-], while the regenerative cycle yielded an efficiency of 0.1972 [-].

Both the regenerative cycle and regenerative cycle with two turbines in series are eligible for providing electricity in rural communities. However, for further investigation into these configurations, the costing and payback period of each cycle application can be analysed to determine which of these two cycles would be the most feasible with the fastest payback period. It would be obvious that the regenerative cycle with two turbines in series have additional costs to buy an extra turbine as well as other extra related costs such as installing extra ducting and the labour that goes with it.

A simple calculation can be made as to how much additional electricity does the regenerative cycle with two turbines in series generate compared to the regenerative cycle with one turbine. For example, it is required that the daily operation of these cycles need to be 18 hours at a capacity of 100 [kW/h]. On an annual basis the amount of power generated by these cycles are then 657 [MWh], considering a year has 365 days. The cycle efficiency is the amount of power generated over the amount of heat input. Therefore, the amount heat input required by each cycle to deliver this amount of electricity is 3285 [MWh] for the regenerative cycle with two turbines in series while the regenerative cycle with one turbine requires 3332 [MWh] of heat input. The difference is 47 [MWh].

If 47 [MWh] of heat is added to the regenerative cycle with two turbines in series. It would be able to generate 9.4 [MWh] additional electricity compared to the regenerative cycle with one turbine. A feasibility study needs to be conducted to determine which of the two cycles is the most feasible option. However, the operating conditions for both cycles have been determined and any one of these two cycles is capable of providing electricity to rural communities.

6.4 Model verification

A simulation model is not complete until it has been verified against a valid software package. The EES simulation model of a simple EFGT configuration was used to verify it against Flownex[®] Simulation Environment, a valid analysis and design software package. The aim of verification is to ensure that the EES simulation models calculate the correct operating conditions with a good accuracy. Since all of the simulation models were developed from the same theory and principles, only one of the simulation models needs to be verified. Furthermore, the requirements for a detail and more complex Flownex model is not necessary as it only needs to verify that the results obtained with EES are correct.

Flownex is a simulation tool that is used for the simulation of thermal-fluid systems such as gas turbines and nuclear reactors. The National Nuclear Regulator (NNR) in South Africa reviewed the Flownex[®] software Verification & Validation (V&V) status and found it to be acceptable to use to support the design and safety case for the PBMR (Pebble Bed Modular Reactor) (Flownex, 2017) and is therefore deemed as an acceptable software package to use as a verification tool for the simulation models that were developed with EES. Figure 25 illustrates the simple EFGT configuration developed in Flownex. This configuration corresponds with that of a simple EFGT cycle developed in EES which is illustrated in Appendix A.

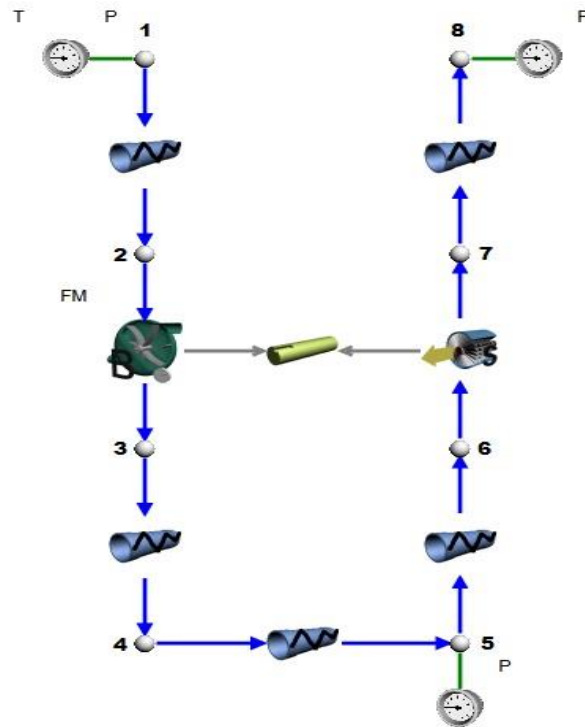


Figure 25 Flownex model of a simple EFGT cycle configuration.

The table below describes the symbols for each component used to develop the simple EFGT model in Flownex. Each symbol is represented by a set of programmable code that is used to perform certain functions. Flownex uses a node and element system for which an element represents a component and a node represents the inlet and outlet conditions of that element.

Table 20 Component symbols used in Flownex.

	Symbol	Component description
1.		Boundary condition
2.		Node
3.		Flow resistance

	Symbol	Component description
4.		Basic centrifugal pump
5.		Simple turbine
6.		Shaft

The symbols listed in Table 20 are:

Boundary condition

The boundary condition element defines the boundary condition of a component. The user has the choice of assigning a boundary pressure, temperature or mass flow value for a component. Either all of them can be used or individually depending on the application. With the boundary condition element, the ambient conditions can be specified at the inlet and outlet of the simulation model similar to what has been done in EES.

Node

Nodes serve as end points of an element to which other elements can be connected in any random way. The inlet and outlet conditions of a component can be viewed at the nodes.

Flow resistance

The flow resistance element is used to model the pressure drop in various components such as pipes, heat exchangers, ducts etc. This element is convenient to model a component with little geometrical information or to speed up the solving time in complex modeling systems. The pipes in the EES simulation models have not been modelled based on their geometry which makes the flow resistance element ideal to model the pipes with in Flownex.

Basic centrifugal pump

The basic centrifugal pump element is an element that is used to model a pump or fan of which the pump or fan characteristic curves are not yet known. This element is mostly used for preliminary designs. Since no characteristic curves were used in the EES models, this element is convenient to model a compressor because the pump element is developed to increase the

pressure of the working fluid that flows through it. This is the same function that a compressor provides. The efficiency at which the working fluid is pressurized as well as the mass flow can be specified for this element.

Simple turbine

Modeling the turbine with EES required no turbine chart which makes the use of the simple turbine element appropriate to use in Flownex as it does not require the characteristic curve of a turbine as well. It determines the pressure and flow behavior through various methods including the Ellipse law. The turbine power output is determined from the pressure ratio and the turbine isentropic efficiency which is similar to the power output that was determined by the turbine developed in EES.

Shaft

The shaft element serves as a connecting element to rotating components such as turbines, compressors and pumps. The shaft element is used to promote the interaction, such as the transfer of mechanical power, between rotating components. The shaft has been used to determine the net power that the turbine in the simulation model delivers after subtracting the power requirements of the basic centrifugal pump.

6.4.1 Variable inputs and assumptions

6.4.1.1 Variable inputs

The variable parameters that can be varied for the Flownex model are the ambient inlet temperature and pressure, mass flow rate, pump isentropic efficiency, system heat input, pressure ratio, turbine isentropic efficiency and also the exhaust pressure of the cycle. These parameters are also variable parameters in the EES model which means that when a parameter is varied in EES, the same changes in the values of the parameters can be brought on to the Flownex model and vice versa.

6.4.1.2 Assumptions

The assumptions made for the EES models also applies to the Flownex model. However, there are some assumptions that were not considered for the Flownex model namely:

- The gearbox and generator efficiency were considered to be ideal.
- The pressure drop in components caused by frictional and other losses are also neglected.

Not considering the above-mentioned assumptions for the Flownex model will not affect the end results. The simple EFGT model in EES also needs to not take these effects into account when comparing the results to the results obtained from the simple EFGT model in Flownex.

6.4.2 Working principle and simulation

The working principle of the Flownex simple EFGT simulation model in Figure 27 is as follow:

Firstly, a boundary condition element is positioned at the inlet of the simple EFGT simulation model. With the boundary condition the ambient inlet temperature and pressure specified as 25 [°C] and 100 [kPa] respectively. A flow resistance element is used as a pipe connection to the basic centrifugal pump. For each flow resistance element, the working fluid that flows through it is specified as air and the flow admittance through the element is set to a value of 100 [%] meaning that there are no frictional or other losses taking place.

For the basic centrifugal pump element, the mass flow rate and the pump isentropic efficiency is specified as 1.068 [kg/s] and 77 [%] respectively. The working fluid is also set as air for the basic centrifugal pump as it uses it to determine the entropy and enthalpy from fluid property charts. Another flow resistance element is used to serve as connection to the combustion chamber. Since there is no combustion chamber element available in Flownex, a flow resistance element is used to model a combustion chamber. For this flow resistance element, the heat input option is used and set to a value 663 [kW]. As a result, the flow resistance element transfers heat energy to the working fluid, thus increasing the air temperature.

Connected to the outlet of the flow resistance element, that is modelled as a combustion chamber, is another boundary condition element. With this element the pressure is set to a value of 450 [kPa]. This element is used to control the pressure ratio for the simulation model. Another flow resistance element is used to connect the "combustion chamber" to the simple turbine element. For the simple turbine element, the turbine isentropic efficiency, mass flow rate, pressure ratio, inlet temperature and inlet pressure to the turbine is specified. The turbine inlet temperature and pressure is obtained from the results calculated in EES while the rest are known values. The simple turbine determines the power output, outlet temperature and outlet pressure according to the Ellipse law and thus requires the pressure and temperature at the turbine inlet.

A flow resistance element is connected to the simple turbine outlet with a boundary condition element. With the boundary condition element, the system exhaust pressure can be controlled and is set to a value of 110 [kPa] according to the assumptions. When the simulation is run the values are determined. These values are the same as in Figure 27.

6.4.3 Results

It is a given that in order to compare the results obtained with the models developed in EES and Flownex, the assumptions and inputs should be the same. Neglecting the effect of pressure drop, gearbox and generator efficiencies and following the approach discussed in Chapter 4, the newly obtained results for the simple EFGT model in EES would then be as illustrated in Table 21. For the same inputs and assumptions, the results obtained from the Flownex model is also listed in the table below. The results obtained from the two software packages are also illustrated in diagram format in Figure 26 and Figure 27.

Table 21 EES versus Flownex compared results.

Component	Properties [Unit]	EES Value	Flownex Value	Error [%]
Inlet	T_amb [C]	25.000	25.000	0.000
	P_amb [kPa]	100.000	100.000	0.000
	m_dot [kg/s]	1.068	1.068	0.000
Compressor	T_comp [C]	230.700	230.687	0.006
	P_comp [kPa]	450.000	450.000	0.000
	Pr_comp [-]	4.500	4.500	0.000
	W_comp [kW]	223.000	222.982	0.008
Combustor	T_comb [C]	797.700	797.510	0.024
	P_comb [kPa]	450.000	450.000	0.000
	Q_comb [kW]	663.000	663.000	0.000
Turbine	T_turb [C]	529.700	529.053	0.122
	P_turb [kPa]	110.000	110.000	0.000
	W_turb [kW]	323.000	321.523	0.457
Other	Nett Work [kW]	100.000	98.541	1.459
	Cycle Eff [-]	0.151	0.149	1.459

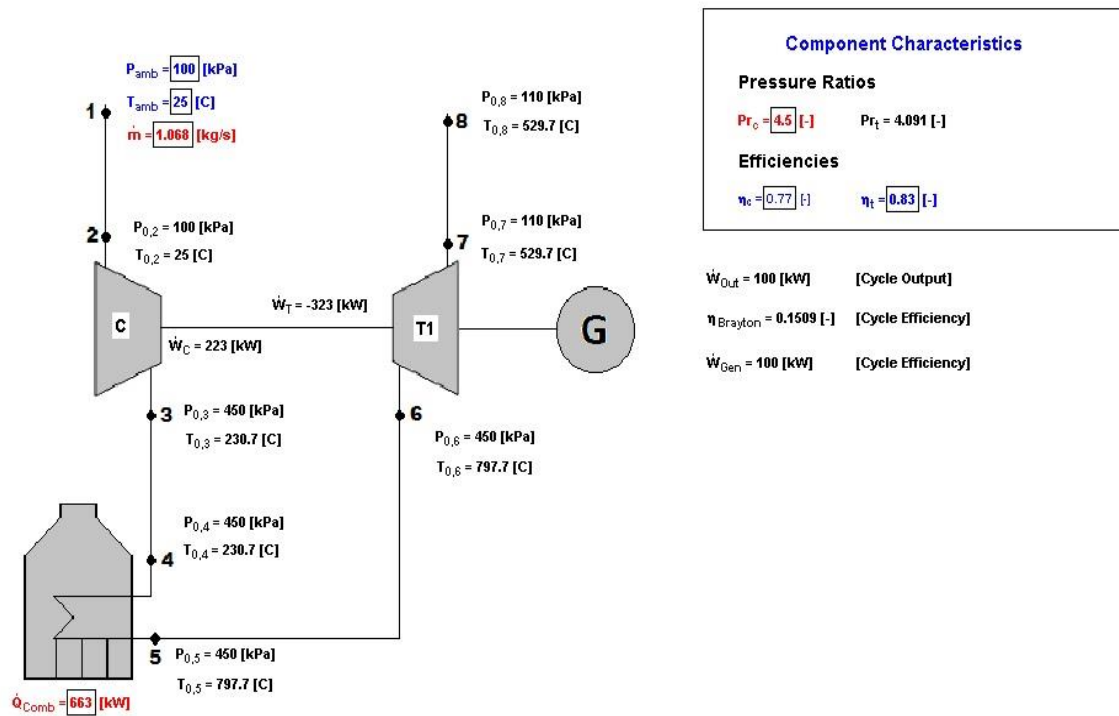


Figure 26 EES results for a simple EFGT cycle.

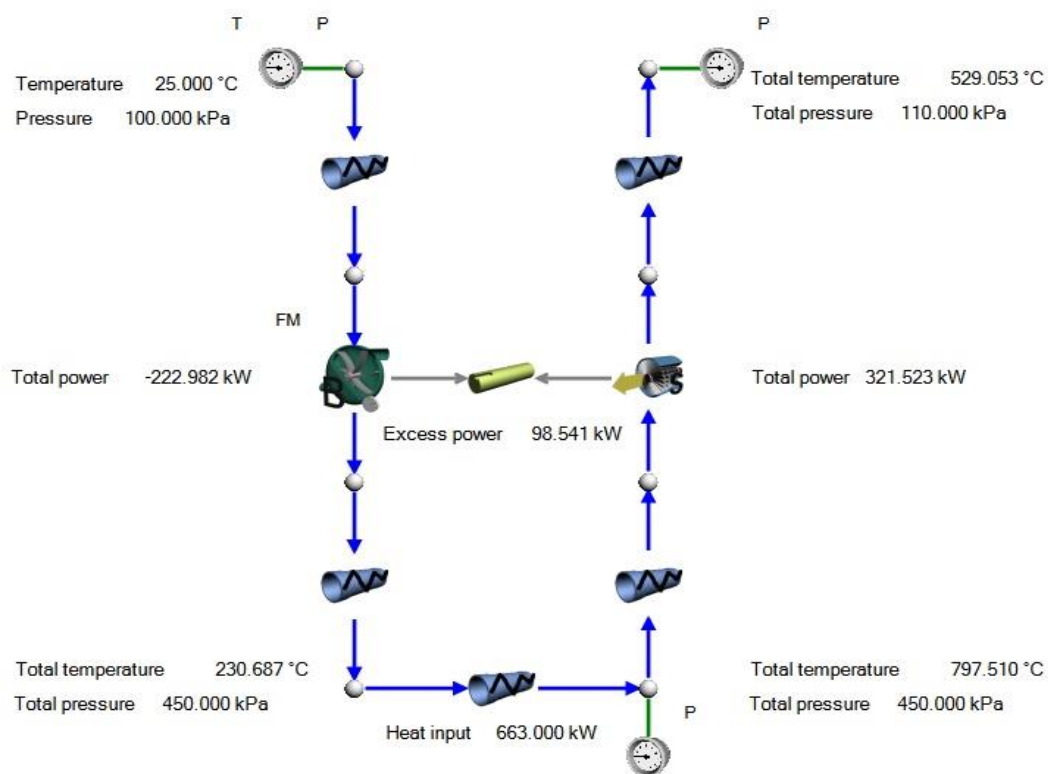


Figure 27 Flownex results for a simple EFGT model.

From the results obtained in Table 21, it is evident that there is a good agreement between the values determined by the two software packages. The results show a maximum error percentage of 1.46 [%] for the cycle efficiency and nett work done. This means that the cycle efficiency and nett work values from EES are accurate within 98.54 [%] compared to that of Flownex. The error is very small and can be explained by the calculation method of each software package in terms of the number of decimals used and the rounding of the values throughout the calculation process. The conclusion that can be made is that the simulation models developed in EES is verified against a valid software package and is considered an acceptable tool to use for the performance calculation of the EFGT cycles.

6.5 Conclusion

A sensible comparison of the EFGT cycles showed that the regenerative EFGT as well as the regenerative EFGT with two turbines in series can be utilized for electrical power output of 100 [kW]. The simple EFGT simulation model developed in EES was verified against a valid thermal fluid software package, Flownex® Simulation Environment. The results obtained with the EES model showed good agreement with the results obtained by the Flownex model.

CHAPTER 7

SUMMARY AND CONCLUSIONS

Chapter 7: Summary and Conclusions

7.1 Introduction

This chapter contains a summary of each chapter as well as the most important results that were obtained in the previous chapters. This is followed by a conclusion of the results after which recommendations are given for future research.

7.2 Summary

The first chapter focused on the purpose and objective of this study. In short the aim was to determine which EFGT cycle configurations will be the best to produce an electrical power output of 100 [kW] under all constraints. The models would be sensibly compared through the generation of efficiency graphs. In the second chapter a literature survey was carried out. It was found that there are a variety of possible configurations to use. Using a heat exchanger significantly increases the performance of an EFGT cycle. The main limitations of these EFGT configurations were the material constraints of the components such as turbines and heat exchangers that were limited to operating temperatures of 1100 [°C]. For a compressor, the optimum pressure ratio for small scale applications in the region of 100 [kW] were 4.5. The combustion section of an EFGT can either be a furnace or a gasifier that first converts the fuel into a gas before combustion. The main drawback of a gasifier, however, is that an intense cleaning system is required to prevent damage to the turbine and heat exchanger components.

Chapter 3 focused on the relevant theory and background of thermal fluid systems which served as building blocks that was necessary to develop simulation models for different EFGT configurations. The models were developed with EES, a computer software package. Chapter 4 discussed the integration of the theory and equations with EES to develop a simulation model. The regenerative cycle was used as an example of how a simulation model were developed and an explanation for using each equation.

In chapter 5, a step wise approach was used to explain the process of generating efficiency graphs. This formed the main focus of the study. The results for each simulation model were determined according to the procedure discussed in chapter 4 and 5. Different EFGT cycles

have been compared and the results were discussed in Chapter 6. The results showed that the simple EFGT cycle performed the worst of all the simulation models with a cycle efficiency of 0.1248 [-]. The regenerative EFGT cycle with reheat performed the second worst with a cycle efficiency of 0.1597 [-]. The regenerative cycle as well as the regenerative cycle with two turbines connected in series, yielding a cycle efficiency of 0.1965 [-] and 0.2 [-] respectively, performed the best of all the configurations considered for the study. The EES model of the simple EFGT cycle was verified against a valid software package known as Flownex® Simulation Environment. The latter were discussed in Chapter 6. Only the simple EFGT cycle was verified because the modeling approach was the same for every configuration. The simple EFGT cycle was modelled in Flownex as well. The results of both software packages have been compared against each other and showed good agreement with a maximum error percentage of 1.46 [%] for the nett work done as well as the cycle efficiency.

7.3 Conclusion

The problem is that there are still rural communities that have no access to electricity. Since grid-electrification is impossible, a solution to provide electricity to rural communities is by using open EFGT systems. However, limited information is available regarding the desired operating conditions and performance of small-scale, open EFGT systems that are capable of operating with an electrical power output capacity in the proximity of 100 [kW].

The study objective was to investigate what method and operating conditions for EFGT cycles should be incorporated to generate electricity in rural communities. This is done by the thermodynamic evaluation of different open EFGT configurations, found in the literature, for small scale, off-grid power generation applications in the range of 100 [kW]. The results showed that the regenerative cycle and the regenerative cycle with two turbines in series have good efficiencies relative to the other configurations in this study. Both cycles can be incorporated to generate electricity in rural communities. The most suitable cycle between the regenerative cycle and the regenerative cycle with two turbines in series needs to be determined by doing a feasibility study. This however is not within the scope and can be pursued for future research.

The simple EFGT cycle, that were modelled in EES, were also validated using a valid thermal-fluid software package known as Flownex. The EES model showed good agreement compared to the Flownex model.

7.4 Future recommendations

The following are considered as recommendations for further research:

- Feasibility of using an EFGT system by determining the payback period of manufacturing a 100 [kW] EFGT system and the potential savings that is achievable to produce electricity with biomass instead of buying electricity from a local energy supplier.
- The effect of humidity and ambient conditions on an EFGT system as it differs depending on location and climate.
- Detail component modeling to aid with the selection of off-the-shelf components when a prototype is being considered.
- Manufacturing a prototype of a regenerative EFGT system for the purpose of experimental tests to verify computer simulated EFGT systems and components.

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Appendix A: Procedure for recuperation

EES is a general equation-solving program that can numerically solve thousands of coupled non-linear algebraic and differential equations. It however only solves equations based on the code and instructions it is given by the user. In a recuperator heat is exchanged between two fluid streams in which the hot fluid stream transfers heat to the cold fluid stream for the purpose of increasing its temperature. The problem is EES needs to be programmed to determine when heat transfer is taking place or not when a simulation is carried out for certain input parameters. This is because the values of the input parameters selected for a simulation analysis may either be sufficient such that heat transfer is possible in the recuperator or it may not. This is why a procedure needs to be used for a case as this. A procedure contains input and output variables that acts upon an instruction when a certain condition is met. The procedure is discussed based on a schematic diagram of a recuperator illustrated in Figure 28.

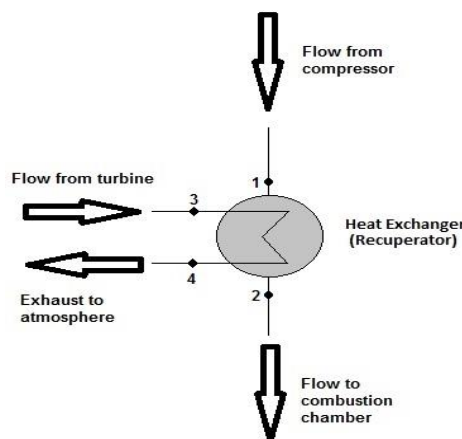


Figure 28 Schematic of a recuperator.

In Figure 28 it is shown that the flow from the compressor is the cold fluid stream that enters the heat exchanger and the flow from the turbine is the hot fluid stream. If the temperature at node 3 is the same as the temperature at node 1 then there can be no heat transfer taking place as the air is in a state of thermal equilibrium, thus $T_{0,3} - T_{0,1} = 0$ [°C]. If there is a temperature difference between the two inlet sides, then heat transfer is bound to take place. It is important to consider a value above 0 [°C] that would give a meaningful rate of heat to the cold fluid stream. This value was selected to be 15 [°C].

In the procedure two pathways are followed when the conditions of heat transfer or no heat transfer is taking place. These conditions are determined with an if-then-else function namely:

```

1 If  $(T_{0,3} - T_{0,1} \geq 15)$  then
2
3 Regen = 1
4  $\dot{Q}_{HX,SS} = C_{min} \cdot (T_{0,1} - T_{0,3})$ 
5  $\dot{Q}_{HX,PS} = -\dot{Q}_{HX,SS}$ 
6  $\dot{Q}_{HX,PS} = \dot{m} \cdot (h_{0,2} - h_{0,1})$ 
7
8 Else
9
10 Regen = 0
12  $\dot{Q}_{HX,SS} = 0$ 
13  $\dot{Q}_{HX,PS} = 0$ 
14  $h_{0,2} = h_{0,1}$ 
15
16 Endif
17 End
  
```

The line that is numbered 1 gives the condition that needs to be met. If this condition is held true, then heat transfer takes place. Line 3 is used as an indicator to show that the condition is met. Lines 4-6 is the equations used when heat transfer takes place which is the same calculations discussed in the Chapter 4. Line 8 is the second pathway when the condition in line 1 is not met. Line 10 indicates that no heat transfer takes place with a value of zero. Lines 12 and 13 gives the values of zero to the heat transfer duty on both sides. Line 14 sets the enthalpy at the inlet and outlet of the primary side equal to each other. Line 16 ends the if-then-else condition and line 17 ends the procedure.

Appendix B: EES model codes

B.1 Simple EFGT model

```

#####

Simple EFGT model

#####

"___ Input Properties ___"

Fluid Properties

T_amb = 25 [C]           "Ambient Air Temperature"
P_amb = 100 [kPa]        "Ambient Air Pressure"
m_dot = 1 [kg/s]         "Air Mass Flow Rate (Guess Value)"

"___ Component Characteristics ___"

Compressor

Pr_c = 4.5 [-]           "Compressor Pressure Ratio"
Eta_c = 0.77 [-]         "Compressor Efficiency"

Turbine

//Pr_t = 4.5 [-]         "Turbine Pressure Ratio"
Eta_t = 0.83 [-]         "Turbine Efficiency"

Pipe Sections

Alpha_p = 0.01 [-]       "Fraction of Average Pressure Drop in Pipes"

Generator

//W_dot_Gen = 100 [kW]   "Electrical Generation Capability of System Excluding Losses and
Pressure Drops"
Eta_Gen = 0.95 [-]       "Generator Efficiency"
Eta_Gearb = 0.95 [-]     "Gearbox Efficiency"

Combustion Chamber

Q_dot_Comb = 150 [kW]    "Heat added by Combustion Chamber (Guess Value)"
Alpha_Comb = 0.005       "Fraction of Average Pressure Drop in Combustion Chamber"

#####

"___ Calculations ___"

"System Inlet (Point 1)"

T_0[1] = T_amb           "Inlet Air Temperature"
P_0[1] = P_amb           "Inlet Air Pressure"
h_0[1] = enthalpy(Air_ha,T = T_0[1],P = P_0[1]) "Enthalpy of Inlet Air"
s_0[1] = entropy(Air_ha,T = T_0[1],P = P_0[1]) "Entropy of Inlet Air"

"Pipe Section 12: Compressor Inlet (Point 2)"

DELTA_P0_12 = Alpha_p * ((P_0[1] + P_0[2]) / 2) "Pressure Drop through Pipe Section 12"
P_0[2] = P_0[1] - DELTA_P0_12 "Pressure at Compressor Inlet"

```

```

h_0[2] = h_0[1]
T_0[2] = temperature(Air_ha,P = P_0[2],h = h_0[2])
s_0[2] = entropy(Air_ha,T = T_0[2],P = P_0[2])

"
"

"Compressor (Point 3)"

P_0[3] = P_0[2] * Pr_c
h_0s[3] = enthalpy(Air_ha,s = s_0[2],P = P_0[3])

W_dot_C = (1 / Eta_c) * m_dot * (h_0s[3] - h_0[2])
W_dot_C = m_dot * (h_0[3] - h_0[2])
T_0[3] = temperature(Air_ha,h = h_0[3],P = P_0[3])
s_0[3] = entropy(Air_ha,T = T_0[3],P = P_0[3])

"
"

"Pipe Section 34: Combustion Chamber Inlet (Point 4)"

DELTA_P0_34 = Alpha_p * ((P_0[4] + P_0[3]) / 2)
P_0[4] = P_0[3] - DELTA_P0_34
h_0[4] = h_0[3]
T_0[4] = temperature(Air_ha,P = P_0[4],h = h_0[4])
s_0[4] = entropy(Air_ha,T = T_0[4],P = P_0[4])

"
"

"Combustion Chamber (Point 5)"

DELTA_P0_45 = Alpha_Comb * ((P_0[5] + P_0[4]) / 2)
P_0[5] = P_0[4] - DELTA_P0_45

//DELTA_P_Comb = 0.005
//P_0[5] = P_0[4] - DELTA_P_Comb

Q_dot_Comb = m_dot * (h_0[5] - h_0[4])
T_0[5] = temperature(Air_ha,h = h_0[5],P = P_0[5])
s_0[5] = entropy(Air_ha,T = T_0[5],P = P_0[5])

T_Comb_Out = T_0[5]

"
"

"Pipe Section 56: Turbine Inlet (Point 6)"

DELTA_P0_56 = Alpha_p * ((P_0[6] + P_0[5]) / 2)
P_0[6] = P_0[5] - DELTA_P0_56
h_0[6] = h_0[5]
T_0[6] = temperature(Air_ha,P = P_0[6],h = h_0[6])
s_0[6] = entropy(Air_ha,T = T_0[6],P = P_0[6])

"
"

"Turbine (Point 7)"

P_0[7] = P_0[6] * (1 / Pr_t)

h_0s[7] = enthalpy(Air_ha,s = s_0[6],P = P_0[7])

W_dot_T = Eta_t * m_dot * (h_0s[7] - h_0[6])
W_dot_T = m_dot * (h_0[7] - h_0[6])
T_0[7] = temperature(Air_ha,h = h_0[7],P = P_0[7])
s_0[7] = entropy(Air_ha,T = T_0[7],P = P_0[7])

"
"

"Work Required to Drive the Compressor and the Generator"

W_dot_T = - W_dot_C - W_dot_Gen / (Eta_Gearb * Eta_Gen)

```

```

"
"

"Pipe Section 78: Exhaust (Point 8)"

DELTA_P0_78 = Alpha_p * ((P_0[8] + P_0[7]) / 2)
P_0[8] = P_0[7] - DELTA_P0_78
P_0[8] = P_Amb + 10
h_0[8] = h_0[7]
T_0[8] = temperature(Air_ha,P = P_0[8],h = h_0[8])
s_0[8] = entropy(Air_ha,T = T_0[8],P = P_0[8])

"Pressure Drop through Pipe Section 78"
"Pressure at Exhaust"
"Pressure at Exhaust"
"Enthalpy at Exhaust"
"Temperature at Exhaust"
"Entropy at Exhaust"

"
"

"Energy Balance"

E = (m_dot * h_0[1]) + W_dot_C + W_dot_T + Q_dot_Comb - (m_dot * h_0[8])

"
"

"Specific Heat Values"

Duplicate i = 1,8

Cp_[i] = cp(Air_ha,T = T_0[i], P = P_0[i])
Cv_[i] = cv(Air_ha,T = T_0[i], P = P_0[i])

End

"
"

"Total Power Output"

W_dot_Out = W_dot_Gen

"Overall Cycle Efficiency"

Eta_Brayton = (W_dot_Out) / (Q_dot_Comb)

"
"

```

B.2 Regenerative EFGT model

```
"#####"
```

Regenerative EFGT model

```
"#####"
```

Procedure

Procedure test(Epsilon_HX, C_min, T_0[4], T_0[10], h_0[4], m_dot : Regen, Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])

"The Procedure is used to determine whether the Air flowing through the primary side of the Recuperator is heated or cooled"

If (T_0[10] - T_0[4] >= 15) Then

Regen = 1

Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[4] - T_0[10])

Q_dot_HX_PS = - Q_dot_HX_SS

h_0[5] = (Q_dot_HX_PS / m_dot) + h_0[4]

Else

Regen = 0

Q_dot_HX_SS = 0

Q_dot_HX_PS = 0

h_0[5] = h_0[4]

Endif

End

" ____ Input Properties ____ "

Fluid Properties

T_amb = 25 [C]

P_amb = 100 [kPa]

m_dot = 1 [kg/s]

"Ambient Air Temperature"

"Ambient Air Pressure"

"Air Mass Flow Rate (Guess Value)"

" ____ Component Characteristics ____ "

Compressor

Pr_c = 4.5 [-]

Eta_c = 0.77 [-]

"Compressor Pressure Ratio"

"Compressor Efficiency"

Turbine

//Pr_t = 4.5 [-]

Eta_t = 0.83 [-]

"Turbine 1 Pressure Ratio"

"Turbine 1 Efficiency"

Pipe Sections

Alpha_p = 0.1 [-]

"Fraction of Average Pressure Drop in Pipes"

Heat Exchangers

Alpha_HX_PS = 0.3 [-]

Alpha_HX_SS = 0.2 [-]

Epsilon_HX = 0.87 [-]

"Fraction of Average Pressure Drop in Heat Exchanger"

"Fraction of Average Pressure Drop in Heat Exchanger"

"Heat Exchanger Effectiveness"

Generator

//W_dot_Gen = 100 [kW]
Drops"

Eta_Gen = 0.95 [-]

Eta_Gearb = 0.95 [-]

"Electrical Generation Capability of System Excluding Losses and Pressure Drops"

"Generator Efficiency"

"Gearbox Efficiency"

Combustion Chamber

$Q_{\text{dot_Comb}} = 150$ [kW]
 $\alpha_{\text{Comb}} = 0.005$

"Heat added by Combustion Chamber (Guess Value)"
"Fraction of Average Pressure Drop in Combustion Chamber"

#####

Calculations

System Inlet (Point 1)

$T_{0[1]} = T_{\text{amb}}$
 $P_{0[1]} = P_{\text{amb}}$
 $h_{0[1]} = \text{enthalpy}(\text{Air_ha}, T = T_{0[1]}, P = P_{0[1]})$
 $s_{[1]} = \text{entropy}(\text{Air_ha}, T = T_{0[1]}, P = P_{0[1]})$

"Inlet Air Temperature"
"Inlet Air Pressure"
"Enthalpy of Inlet Air"
"Entropy of Inlet Air"

"

Pipe Section 12: Compressor Inlet (Point 2)

$\Delta P_{0_12} = \alpha_p * ((P_{0[2]} + P_{0[1]}) / 2)$
 $P_{0[2]} = P_{0[1]} - \Delta P_{0_12}$
 $h_{0[2]} = h_{0[1]}$
 $T_{0[2]} = \text{temperature}(\text{Air_ha}, P = P_{0[2]}, h = h_{0[2]})$
 $s_{[2]} = \text{entropy}(\text{Air_ha}, T = T_{0[2]}, P = P_{0[2]})$

"Pressure Drop through Pipe Section 12"
"Pressure at Compressor Inlet"
"Enthalpy at Compressor Inlet"
"Temperature at Compressor Inlet"
"Entropy at Compressor Inlet"

"

Compressor (Point 3)

$P_{0[3]} = P_{0[2]} * Pr_c$
 $h_{0s[3]} = \text{enthalpy}(\text{Air_ha}, s = s_{[2]}, P = P_{0[3]})$
 $W_{\text{dot_C}} = (1 / \eta_c) * m_{\text{dot}} * (h_{0s[3]} - h_{0[2]})$
 $W_{\text{dot_C}} = m_{\text{dot}} * (h_{0[3]} - h_{0[2]})$
 $T_{0[3]} = \text{temperature}(\text{Air_ha}, h = h_{0[3]}, P = P_{0[3]})$
 $s_{[3]} = \text{entropy}(\text{Air_ha}, T = T_{0[3]}, P = P_{0[3]})$

"Pressure at Compressor Outlet"
"Compressor Enthalpy for an Isentropic Process"
"Ideal Compressor Work"
"Actual Compressor Work"
"Temperature at Compressor Outlet"
"Entropy at Compressor Outlet"

"

Pipe Section 34: HX Primary Side Inlet (Point 4)

$\Delta P_{0_34} = \alpha_p * ((P_{0[4]} + P_{0[3]}) / 2)$
 $P_{0[4]} = P_{0[3]} - \Delta P_{0_34}$
 $h_{0[4]} = h_{0[3]}$
 $T_{0[4]} = \text{temperature}(\text{Air_ha}, P = P_{0[4]}, h = h_{0[4]})$
 $s_{[4]} = \text{entropy}(\text{Air_ha}, T = T_{0[4]}, P = P_{0[4]})$

"Pressure Drop through Pipe Section 34"
"Primary Side Inlet Pressure"
"Primary Side Inlet Enthalpy"
"Primary Side Inlet Temperature"
"Primary Side Inlet Entropy"

"

Heat Exchanger: Primary Side (Point 5)

$\Delta P_{\text{HX_PS}} = \alpha_{\text{HX_PS}} * ((P_{0[5]} + P_{0[4]}) / 2)$
 $\Delta P_{\text{HX_PS}} = 20$ [kPa]
 $P_{0[5]} = P_{0[4]} - \Delta P_{\text{HX_PS}}$
 $\dot{Q}_{\text{dot_HX_PS}} = \epsilon_{\text{HX}} * C_{\text{min}} * (T_{0[10]} - T_{0[4]})$
 $\dot{Q}_{\text{dot_HX_PS}} = m_{\text{dot}} * (h_{0[5]} - h_{0[4]})$
 $C_{\text{min}} = \min(Cp_{[4]} * m_{\text{dot}}, Cp_{[10]} * m_{\text{dot}})$
 $T_{0[5]} = \text{temperature}(\text{Air_ha}, P = P_{0[5]}, h = h_{0[5]})$
 $s_{[5]} = \text{entropy}(\text{Air_ha}, T = T_{0[5]}, P = P_{0[5]})$

"Pressure Drop through the HX Primary Side"
"Pressure Drop through the HX Primary Side: Given as 20 [kPa]"
"Primary Side Outlet Pressure"
"Heat Transfer Duty by HX Primary Side"
"Heat Transfer Duty by HX Primary Side"
"Minimum Heat Capacity"
"Primary Side Outlet Temperature"
"Primary Side Outlet Entropy"

"

Pipe Section 56: Combustion Chamber Inlet (Point 6)

$\Delta P_{0_56} = \alpha_p * ((P_{0[6]} + P_{0[5]}) / 2)$
 $P_{0[6]} = P_{0[5]} - \Delta P_{0_56}$
 $h_{0[6]} = h_{0[5]}$

"Pressure Drop through Pipe Section 56"
"Pressure at Combustion Chamber Inlet"
"Enthalpy at Combustion Chamber Inlet"

$T_{0[6]} = \text{temperature}(\text{Air_ha}, P = P_{0[6]}, h = h_{0[6]})$	"Temperature at Combustion Chamber Inlet"
$s_{[6]} = \text{entropy}(\text{Air_ha}, T = T_{0[6]}, P = P_{0[6]})$	"Entropy at Combustion Chamber Inlet"
$T_{\text{HXPS_Out}} = T_{0[6]}$	"Set HX Primary Side Outlet Temperature To $T_{\text{HXPS_Out}}$ "
"	
"Combustion Chamber (Point 7)"	
$\text{DELTA_P0_67} = \text{Alpha_Comb} * ((P_{0[7]} + P_{0[6]}) / 2)$	"Pressure Drop through Pipe Section 78"
$P_{0[7]} = P_{0[6]} - \text{DELTA_P0_67}$	"Pressure at Combustion Chamber Outlet"
//DELTA_P_Comb = 0	"Pressure Drop through Combustion Chamber"
//P_0[7] = P_0[6] - DELTA_P_Comb	"Pressure at Combustion Chamber Outlet"
$Q_{\text{dot_Comb}} = m_{\text{dot}} * (h_{0[7]} - h_{0[6]})$	"Heat added by Combustion Chamber"
$T_{0[7]} = \text{temperature}(\text{Air_ha}, h = h_{0[7]}, P = P_{0[7]})$	"Temperature at Combustion Chamber Outlet"
$s_{[7]} = \text{entropy}(\text{Air_ha}, T = T_{0[7]}, P = P_{0[7]})$	"Entropy at Combustion Chamber Outlet"
$T_{\text{Comb_Out}} = T_{0[7]}$	"Set Combustion Outlet Temperature To $T_{\text{Comb_Out}}$ "
"	
"Pipe Section 78: Turbine Inlet (Point 8)"	
$\text{DELTA_P0_78} = \text{Alpha_p} * ((P_{0[7]} + P_{0[8]}) / 2)$	"Pressure Drop through Pipe Section 78"
$P_{0[8]} = P_{0[7]} - \text{DELTA_P0_78}$	"Pressure at Turbine Inlet"
$h_{0[8]} = h_{0[7]}$	"Enthalpy at Turbine Inlet"
$T_{0[8]} = \text{temperature}(\text{Air_ha}, P = P_{0[8]}, h = h_{0[8]})$	"Temperature at Turbine Inlet"
$s_{[8]} = \text{entropy}(\text{Air_ha}, T = T_{0[8]}, P = P_{0[8]})$	"Entropy at Turbine Inlet"
"	
"Turbine (Point 9)"	
$P_{0[9]} = P_{0[8]} * (1 / \text{Pr_t})$	"Pressure at Turbine Outlet"
$h_{0s[9]} = \text{enthalpy}(\text{Air_ha}, s = s_{[8]}, P = P_{0[9]})$	"Turbine Enthalpy for an Isentropic Process"
$W_{\text{dot_T}} = \text{Eta_t} * m_{\text{dot}} * (h_{0s[9]} - h_{0[8]})$	"Ideal Work Done by Turbine"
$W_{\text{dot_T}} = m_{\text{dot}} * (h_{0[9]} - h_{0[8]})$	"Actual Work Done by Turbine"
$T_{0[9]} = \text{temperature}(\text{Air_ha}, h = h_{0[9]}, P = P_{0[9]})$	"Temperature at Turbine Outlet"
$s_{[9]} = \text{entropy}(\text{Air_ha}, T = T_{0[9]}, P = P_{0[9]})$	"Entropy at Turbine Outlet"
"	
"Work Required by Turbine to Drive the Compressor and the Generator"	
$W_{\text{dot_T}} = -W_{\text{dot_C}} - W_{\text{dot_Gen}} / (\text{Eta_Gen} * \text{Eta_Gearb})$	
"	
"Pipe Section 910: HX Secondary Side Inlet (Point 10)"	
$\text{DELTA_P0_910} = \text{Alpha_p} * ((P_{0[10]} + P_{0[9]}) / 2)$	"Pressure Drop through Pipe Section 910"
$P_{0[10]} = P_{0[9]} - \text{DELTA_P0_910}$	"Secondary Side Inlet Pressure"
$h_{0[10]} = h_{0[9]}$	"Secondary Side Inlet Enthalpy"
$T_{0[10]} = \text{temperature}(\text{Air_ha}, P = P_{0[10]}, h = h_{0[10]})$	"Secondary Side Inlet Temperature"
$s_{[10]} = \text{entropy}(\text{Air_ha}, T = T_{0[10]}, P = P_{0[10]})$	"Secondary Side Inlet Entropy"
"	
"Heat Exchanger: Secondary Side (Point 11)"	
$\text{DELTA_P_HX_SS} = \text{Alpha_HX_SS} * ((P_{0[11]} + P_{0[10]}) / 2)$	"Pressure Drop through the HX Secondary Side"
//DELTA_P_HX_SS = 20 [kPa]	"Pressure Drop through the HX Secondary Side: Given as 20 [kPa]"
$P_{0[11]} = P_{0[10]} - \text{DELTA_P_HX_SS}$	"Secondary Side Outlet Pressure"
//Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[5] - T_0[10])	"Heat Transfer Duty by HX Secondary Side"
$Q_{\text{dot_HX_SS}} = m_{\text{dot}} * (h_{0[11]} - h_{0[10]})$	"Heat Transfer Duty by HX Secondary Side"

```
//C_min = MIN(Cp_[5] * m_dot, Cp_[10] * m_dot)
T_0[11] = temperature(Air_ha, h = h_0[11], P = P_0[11])
s_[11] = entropy(Air_ha, T = T_0[11], P = P_0[11])

"Minimum Heat Capacity"
"Secondary Side Outlet Temperature"
"Secondary Side Outlet Entropy"

"
"

"Pipe Section 1112: Exhaust (Point 12)"
a
DELTA_P0_1112 = Alpha_p * ((P_0[12] + P_0[11]) / 2)
P_0[12] = P_0[11] - DELTA_P0_1112
P_0[12] = P_amb + 10
h_0[12] = h_0[11]
T_0[12] = temperature(Air_ha, P = P_0[12], h = h_0[12])
s_[12] = entropy(Air_ha, T = T_0[12], P = P_0[12])

"Pressure Drop through Pipe 1112"
"Pipe Section 1112 Pressure"
"Pipe Section 1112 Pressure"
"Pipe Section 1112 Enthalpy"
"Pipe Section 1112 Temperature"
"Pipe Section 1112 Entropy"

"
"

"Energy Balance"

E = (m_dot * h_0[11]) + W_dot_C + W_dot_T + Q_dot_Comb - (m_dot * h_0[12])

"
"

"Specific Heat Values"

Duplicate i = 1,12

Cp_[i] = cp(Air_ha, T = T_0[i], P = P_0[i])
Cv_[i] = cv(Air_ha, T = T_0[i], P = P_0[i])

End

"
"

"Total Power Output"

W_dot_Out = W_dot_Gen

"Overall Cycle Efficiency"

Eta_Brayton = (W_dot_Out) / (Q_dot_Comb)

"Call Procedure"

Call test(Epsilon_HX, C_min, T_0[4], T_0[10], h_0[4], m_dot : Regen, Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])

"
"
```

B.3 Regenerative EFGT model with two turbines in series

```
#####
```

Brayton Cycle: Regenerative Cycle With Two Turbines

```
#####
```

```
"Procedure"
```

```
Procedure test(Epsilon_HX, C_min, T_0[4], T_0[12], h_0[4], m_dot : Regen , Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])
```

```
"The Procedure is used to determine whether the Air flowing through the primary side of the Recuperater is heated or cooled"
```

```
If (T_0[12] - T_0[4] >= 15) Then
```

```
Regen = 1
```

```
Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[4] - T_0[12])
```

```
Q_dot_HX_PS = - Q_dot_HX_SS
```

```
h_0[5] = (Q_dot_HX_PS / m_dot) + h_0[4]
```

```
Else
```

```
Regen = 0
```

```
Q_dot_HX_SS = 0
```

```
Q_dot_HX_PS = 0
```

```
h_0[5] = h_0[4]
```

```
Endif
```

```
End
```

```
" ____ Input Properties ____"
```

Fluid Properties

```
T_amb = 25 [C]
```

```
P_amb = 100 [kPa]
```

```
m_dot = 1 [kg/s]
```

```
"Ambient Air Temperature"
```

```
"Ambient Air Pressure"
```

```
"Air Mass Flow Rate (Guess Value)"
```

```
" ____ Component Characteristics ____"
```

Compressor

```
Pr_c = 4.5 [-]
```

```
Eta_c = 0.77 [-]
```

```
"Compressor Pressure Ratio"
```

```
"Compressor Efficiency"
```

Turbine

```
//Pr_t1 = 4.5 [-]
```

```
//Pr_t2 = 4.5 [-]
```

```
"Turbine 1 Pressure Ratio"
```

```
"Turbine 2 Pressure Ratio"
```

```
Eta_t1 = 0.83 [-]
```

```
Eta_t2 = 0.83 [-]
```

```
"Turbine 1 Efficiency"
```

```
"Turbine 2 Efficiency"
```

Pipe Sections

```
Alpha_p = 0.01 [-]
```

```
"Fraction of Average Pressure Drop in Pipes"
```

Heat Exchangers

```
Alpha_HX_PS = 0.03 [-]
```

```
Alpha_HX_SS = 0.02 [-]
```

```
Epsilon_HX = 0.8 [-]
```

```
"Fraction of Average Pressure Drop in Heat Exchanger"
```

```
"Fraction of Average Pressure Drop in Heat Exchanger"
```

```
"Heat Exchanger Effectiveness"
```

Generator

/W_dot_Gen = 100 [kW] "Electrical Generation Capability of System Excluding Losses and Pressure Drops"
Eta_Gen = 0.95 [-] "Generator Efficiency"
Eta_Gearb = 0.95 [-] "Gearbox Efficiency"

Combustion Chamber

Q_dot_Comb = 500 [kW] "Heat added by Combustion Chamber (Guess Value)"
Alpha_Comb = 0.005 "Fraction of Average Pressure Drop in Combustion Chamber"

"#####"

"_____ Calculations _____"

"System Inlet (Point 1)"

T_0[1] = T_amb "Inlet Air Temperature"
P_0[1] = P_amb "Inlet Air Pressure"
h_0[1] = enthalpy(Air_ha, T = T_0[1], P = P_0[1]) "Enthalpy of Inlet Air"
s_0[1] = entropy(Air_ha, T = T_0[1], P = P_0[1]) "Entropy of Inlet Air"

"_____"

"Pipe Section 12: Compressor Inlet (Point 2)"

DELTA_P0_12 = Alpha_p * ((P_0[2] + P_0[1]) / 2) "Pressure Drop through Pipe Section 12"
P_0[2] = P_0[1] - DELTA_P0_12 "Pressure at Compressor Inlet"
h_0[2] = h_0[1] "Enthalpy at Compressor Inlet"
T_0[2] = temperature(Air_ha, P = P_0[2], h = h_0[2]) "Temperature at Compressor Inlet"
s_0[2] = entropy(Air_ha, T = T_0[2], P = P_0[2]) "Entropy at Compressor Inlet"

"_____"

"Compressor (Point 3)"

P_0[3] = P_0[2] * Pr_c "Pressure at Compressor Outlet"
h_0s[3] = enthalpy(Air_ha, s = s_0[2], P = P_0[3]) "Compressor Enthalpy for an Isentropic Process"
W_dot_C = (1 / Eta_c) * m_dot * (h_0s[3] - h_0[2]) "Ideal Compressor Work"
W_dot_C = m_dot * (h_0[3] - h_0[2]) "Actual Compressor Work"
T_0[3] = temperature(Air_ha, h = h_0[3], P = P_0[3]) "Temperature at Compressor Outlet"
s_0[3] = entropy(Air_ha, T = T_0[3], P = P_0[3]) "Entropy at Compressor Outlet"

"_____"

"Pipe Section 34: HX Primary Side Inlet (Point 4)"

DELTA_P0_34 = Alpha_p * ((P_0[4] + P_0[3]) / 2) "Pressure Drop through Pipe Section 34"
P_0[4] = P_0[3] - DELTA_P0_34 "Primary Side Inlet Pressure"
h_0[4] = h_0[3] "Primary Side Inlet Enthalpy"
T_0[4] = temperature(Air_ha, P = P_0[4], h = h_0[4]) "Primary Side Inlet Temperature"
s_0[4] = entropy(Air_ha, T = T_0[4], P = P_0[4]) "Primary Side Inlet Entropy"

"_____"

"Heat Exchanger: Primary Side (Point 5)"

DELTA_P_HX_PS = Alpha_HX_PS * ((P_0[5] + P_0[4]) / 2) "Pressure Drop through the HX Primary Side"
//DELTA_P_HX_PS = 20 [kPa] "Pressure Drop through the HX Primary Side: Given as 20 [kPa]"
P_0[5] = P_0[4] - DELTA_P_HX_PS "Primary Side Outlet Pressure"
//Q_dot_HX_PS = Epsilon_HX * C_min * (T_0[12] - T_0[4]) "Heat Transfer Duty by HX Primary Side"
//Q_dot_HX_PS = m_dot * (h_0[5] - h_0[4]) "Heat Transfer Duty by HX Primary Side"
C_min = MIN(Cp_0[4] * m_dot, Cp_0[12] * m_dot) "Minimum Heat Capacity"
T_0[5] = temperature(Air_ha, P = P_0[5], h = h_0[5]) "Primary Side Outlet Temperature"
s_0[5] = entropy(Air_ha, T = T_0[5], P = P_0[5]) "Primary Side Outlet Entropy"

"	
"Pipe Section 56: Combustion Chamber Inlet (Point 6)"	
DELTA_P0_56 = Alpha_p * ((P_0[6] + P_0[5]) / 2)	"Pressure Drop through Pipe Section 56"
P_0[6] = P_0[5] - DELTA_P0_56	"Pressure at Combustion Chamber Inlet"
h_0[6] = h_0[5]	"Enthalpy at Combustion Chamber Inlet"
T_0[6] = temperature(Air_ha, P = P_0[6], h = h_0[6])	"Temperature at Combustion Chamber Inlet"
s_0[6] = entropy(Air_ha, T = T_0[6], P = P_0[6])	"Entropy at Combustion Chamber Inlet"
T_HXPS_Out = T_0[6]	"Set HX Primary Side Outlet Temperature To T_HXPS_Out"
"	
"Combustion Chamber (Point 7)"	
DELTA_P0_67 = Alpha_Comb * ((P_0[7] + P_0[6]) / 2)	"Pressure Drop through Pipe Section 67"
P_0[7] = P_0[6] - DELTA_P0_67	"Pressure at Combustion Chamber Outlet"
//DELTA_P_Comb = 0	"Pressure Drop through Combustion Chamber"
//P_0[7] = P_0[6] - DELTA_P_Comb	"Pressure at Combustion Chamber Outlet"
Q_dot_Comb = m_dot * (h_0[7] - h_0[6])	"Heat added by Combustion Chamber"
T_0[7] = temperature(Air_ha, h = h_0[7], P = P_0[7])	"Temperature at Combustion Chamber Outlet"
s_0[7] = entropy(Air_ha, T = T_0[7], P = P_0[7])	"Entropy at Combustion Chamber Outlet"
T_Comb_Out = T_0[7]	"Set Combustion Outlet Temperature To T_Comb_Out"
"	
"Pipe Section 78: Turbine 1 Inlet (Point 8)"	
DELTA_P0_78 = Alpha_p * ((P_0[8] + P_0[7]) / 2)	"Pressure Drop through Pipe Section 78"
P_0[8] = P_0[7] - DELTA_P0_78	"Pressure at Turbine 1 Inlet"
h_0[8] = h_0[7]	"Enthalpy at Turbine 1 Inlet"
T_0[8] = temperature(Air_ha, P = P_0[8], h = h_0[8])	"Temperature at Turbine 1 Inlet"
s_0[8] = entropy(Air_ha, T = T_0[8], P = P_0[8])	"Entropy at Turbine 1 Inlet"
"	
"Turbine 1 (Point 9)"	
P_0[9] = P_0[8] * (1 / Pr_t1)	"Pressure at Turbine 1 Outlet"
h_0s[9] = enthalpy(Air_ha, s = s_0[8], P = P_0[9])	"Turbine 1 Enthalpy for an Isentropic Process"
W_dot_T1 = Eta_t1 * m_dot * (h_0s[9] - h_0[8])	"Ideal Work Done by Turbine 1"
W_dot_T1 = m_dot * (h_0[9] - h_0[8])	"Actual Work Done by Turbine 1"
T_0[9] = temperature(Air_ha, h = h_0[9], P = P_0[9])	"Temperature at Turbine 1 Outlet"
s_0[9] = entropy(Air_ha, T = T_0[9], P = P_0[9])	"Entropy at Turbine 1 Outlet"
"	
"Pipe Section 910: Turbine 2 Inlet (Point 10)"	
DELTA_P0_910 = Alpha_p * ((P_0[10] + P_0[9]) / 2)	"Pressure Drop through Pipe Section 910"
P_0[10] = P_0[9] - DELTA_P0_910	"Pressure at Turbine 2 Inlet"
h_0[10] = h_0[9]	"Enthalpy at Turbine 2 Inlet"
T_0[10] = temperature(Air_ha, P = P_0[10], h = h_0[10])	"Temperature at Turbine 2 Inlet"
s_0[10] = entropy(Air_ha, T = T_0[10], P = P_0[10])	"Entropy at Turbine 2 Inlet"
"	
"Turbine 2 (Point 11)"	
P_0[11] = P_0[10] * (1 / Pr_t2)	"Pressure at Turbine 2 Outlet"
h_0s[11] = enthalpy(Air_ha, s = s_0[10], P = P_0[11])	"Turbine 2 Enthalpy for an Isentropic Process"
W_dot_T2 = Eta_t2 * m_dot * (h_0s[11] - h_0[10])	"Ideal Work Done by Turbine 2"
W_dot_T2 = m_dot * (h_0[11] - h_0[10])	"Actual Work Done by Turbine 2"
T_0[11] = temperature(Air_ha, h = h_0[11], P = P_0[11])	"Temperature at Turbine 2 Outlet"

```

s_[11] = entropy(Air_ha,T = T_0[11],P = P_0[11])           "Entropy at Turbine 2 Outlet"

"Generated Work"

W_dot_Gen = - W_dot_T2 * Eta_Gearb * Eta_Gen

"Work Required by Turbine 1 to Drive the Compressor"

W_dot_T1 = - W_dot_C

"Pipe Section 1112: HX Secondary Side Inlet (Point 12)"

DELTA_P0_1112 = Alpha_p * ((P_0[12] + P_0[11]) / 2)         "Pressure Drop through Pipe Section 1112"
P_0[12] = P_0[11] - DELTA_P0_1112                         "Secondary Side Inlet Pressure"
h_0[12] = h_0[11]                                          "Secondary Side Inlet Enthalpy"
T_0[12] = temperature(Air_ha,P = P_0[12],h = h_0[12])     "Secondary Side Inlet Temperature"
s_[12] = entropy(Air_ha,T = T_0[12],P = P_0[12])          "Secondary Side Inlet Entropy"

"Heat Exchanger: Secondary Side (Point 13)"

DELTA_P_HX_SS = Alpha_HX_SS * ((P_0[13] + P_0[12]) / 2)   "Pressure Drop through the HX Secondary Side"
//DELTA_P_HX_SS = 20 [kPa]                                "Pressure Drop through the HX Secondary Side: Given as 20 [kPa]"
P_0[13] = P_0[12] - DELTA_P_HX_SS                         "Secondary Side Outlet Pressure"

//Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[4] - T_0[12])  "Heat Transfer Duty by HX Secondary Side"
Q_dot_HX_SS = m_dot * (h_0[13] - h_0[12])                "Heat Transfer Duty by HX Secondary Side"

//C_min = MIN(Cp_[4] * m_dot, Cp_[12] * m_dot)           "Minimum Heat Capacity"
T_0[13] = temperature(Air_ha, h = h_0[13], P = P_0[13])   "Secondary Side Outlet Temperature"
s_[13] = entropy(Air_ha, T = T_0[13], P = P_0[13])        "Secondary Side Outlet Entropy"

"Pipe Section 1314: Exhaust (Point 14)"

DELTA_P0_1314 = Alpha_p * ((P_0[14] + P_0[13]) / 2)       "Pressure Drop through Pipe Section 1314"
P_0[14] = P_0[13] - DELTA_P0_1314                        "Pipe Section 1314 Pressure"
P_0[14] = P_amb + 10                                     "Pipe Section 1314 Pressure"
h_0[14] = h_0[13]                                         "Pipe Section 1314 Enthalpy"
T_0[14] = temperature(Air_ha,P = P_0[14], h = h_0[14])    "Pipe Section 1314 Temperature"
s_[14] = entropy(Air_ha, T = T_0[14], P = P_0[14])        "Pipe Section 1314 Entropy"

"Energy Balance"

E = (m_dot * h_0[1]) + W_dot_C + W_dot_T1 + W_dot_T2 + Q_dot_Comb - (m_dot * h_0[14])

"Specific Heat Values"

Duplicate i = 1,14

Cp_[i] = cp(Air_ha,T = T_0[i],P = P_0[i])
Cv_[i] = cv(Air_ha,T = T_0[i],P = P_0[i])

End

"Total Power Output"

W_dot_Out = W_dot_Gen

```

"Overall Cycle Efficiency"

$$\text{Eta_Brayton} = (W_dot_Out) / (Q_dot_Comb)$$

"Call Procedure"

Call test(Epsilon_HX, C_min, T_0[4], T_0[12], h_0[4], m_dot : Regen , Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])

"

"

B.4 Regenerative EFGT model with reheating

```
#####
```

```
    Brayton Cycle: Reheat
```

```
#####
```

```
"Procedure"
```

```
Procedure test(Epsilon_HX, C_min, T_0[4], T_0[10], h_0[4], m_dot : Regen , Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])
```

```
"The Procedure is used to determine whether the Air flowing through the primary side of the Recuperater is heated or cooled"
```

```
If (T_0[10] - T_0[4] >= 15) Then
```

```
    Regen = 1
```

```
    Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[4] - T_0[10])
```

```
    Q_dot_HX_PS = - Q_dot_HX_SS
```

```
    h_0[5] = (Q_dot_HX_PS / m_dot) + h_0[4]
```

```
Else
```

```
    Regen = 0
```

```
    Q_dot_HX_SS = 0
```

```
    Q_dot_HX_PS = 0
```

```
    h_0[5] = h_0[4]
```

```
Endif
```

```
End
```

```
" ____ Input Properties ____"
```

```
"Fluid Properties"
```

```
T_amb = 25 [C]
```

```
P_amb = 100 [kPa]
```

```
m_dot = 1 [kg/s]
```

```
"Ambient Air Temperature"
```

```
"Ambient Air Pressure"
```

```
"Air Mass Flow Rate (Guess Value)"
```

```
" ____ Component Characteristics ____"
```

```
"Compressor"
```

```
Pr_c = 4.5 [-]
```

```
Eta_c = 0.77 [-]
```

```
"Compressor Pressure Ratio"
```

```
"Compressor Efficiency"
```

```
"Turbine"
```

```
//Pr_t1 = 4.5 [-]
```

```
//Pr_t2 = 4.5 [-]
```

```
"Turbine 1 Pressure Ratio"
```

```
"Turbine 2 Pressure Ratio"
```

```
Eta_t1 = 0.83 [-]
```

```
Eta_t2 = 0.83 [-]
```

```
"Turbine 1 Efficiency"
```

```
"Turbine 2 Efficiency"
```

```
"Pipe Sections"
```

```
Alpha_p = 0.01 [-]
```

```
"Fraction of Average Pressure Drop in Pipes"
```

```
"Heat Exchangers"
```

```
Alpha_HX_PS = 0.03 [-]
```

```
Alpha_HX_SS = 0.02 [-]
```

```
Epsilon_HX = 0.87 [-]
```

```
"Fraction of Average Pressure Drop in Heat Exchanger"
```

```
"Fraction of Average Pressure Drop in Heat Exchanger"
```

```
"Heat Exchanger Effectiveness"
```

```
"Generator"
```

//W_dot_Gen = 100 [kW] "Electrical Generation Capability of System Excluding Losses and Pressure Drops"
Eta_Gen = 0.95 [-] "Generator Efficiency"
Eta_Gearb = 0.95 [-] "Gearbox Efficiency"

Combustion Chamber

Q_dot_Comb = 500 [kW] "Heat added by Combustion Chamber (Guess Value)"
Alpha_Comb = 0.005 "Fraction of Average Pressure Drop in Combustion Chamber"

#####

"_____ Calculations _____"

"System Inlet (Point 1)"

T_0[1] = T_amb "Inlet Air Temperature"
P_0[1] = P_amb "Inlet Air Pressure"
h_0[1] = enthalpy(Air_ha, T = T_0[1], P = P_0[1]) "Enthalpy of Inlet Air"
s_0[1] = entropy(Air_ha, T = T_0[1], P = P_0[1]) "Entropy of Inlet Air"

"_____"

"Pipe Section 12: Compressor Inlet (Point 2)"

DELTA_P0_12 = Alpha_p * ((P_0[2] + P_0[1]) / 2) "Pressure Drop through Pipe Section 12"
P_0[2] = P_0[1] - DELTA_P0_12 "Pressure at Compressor Inlet"
h_0[2] = h_0[1] "Enthalpy at Compressor Inlet"
T_0[2] = temperature(Air_ha, P = P_0[2], h = h_0[2]) "Temperature at Compressor Inlet"
s_0[2] = entropy(Air_ha, T = T_0[2], P = P_0[2]) "Entropy at Compressor Inlet"

"_____"

"Compressor (Point 3)"

P_0[3] = P_0[2] * Pr_c "Pressure at Compressor Outlet"
h_0s[3] = enthalpy(Air_ha, s = s_0[2], P = P_0[3]) "Compressor Enthalpy for an Isentropic Process"
W_dot_C = (1 / Eta_c) * m_dot * (h_0s[3] - h_0[2]) "Ideal Compressor Work"
W_dot_C = m_dot * (h_0[3] - h_0[2]) "Actual Compressor Work"
T_0[3] = temperature(Air_ha, h = h_0[3], P = P_0[3]) "Temperature at Compressor Outlet"
s_0[3] = entropy(Air_ha, T = T_0[3], P = P_0[3]) "Entropy at Compressor Outlet"

"_____"

"Pipe Section 34: HX Primary Side Inlet (Point 4)"

DELTA_P0_34 = Alpha_p * ((P_0[4] + P_0[3]) / 2) "Pressure Drop through Pipe Section 34"
P_0[4] = P_0[3] - DELTA_P0_34 "Primary Side Inlet Pressure"
h_0[4] = h_0[3] "Primary Side Inlet Enthalpy"
T_0[4] = temperature(Air_ha, P = P_0[4], h = h_0[4]) "Primary Side Inlet Temperature"
s_0[4] = entropy(Air_ha, T = T_0[4], P = P_0[4]) "Primary Side Inlet Entropy"

"_____"

"Heat Exchanger: Primary Side (Point 5)"

DELTA_P_HX_PS = Alpha_HX_PS * ((P_0[5] + P_0[4]) / 2) "Pressure Drop through the HX Primary Side"
//DELTA_P_HX_PS = 20 [kPa] "Pressure Drop through the HX Primary Side: Given as 20 [kPa]"
P_0[5] = P_0[4] - DELTA_P_HX_PS "Primary Side Outlet Pressure"
//Q_dot_HX_PS = Epsilon_HX * C_min * (T_0[11] - T_0[4]) "Heat Transfer Duty by HX Primary Side"
//Q_dot_HX_PS = m_dot * (h_0[5] - h_0[4]) "Heat Transfer Duty by HX Primary Side"
C_min = min(Cp_0[4] * m_dot, Cp_0[11] * m_dot) "Minimum Heat Capacity"
T_0[5] = temperature(Air_ha, P = P_0[5], h = h_0[5]) "Primary Side Outlet Temperature"
s_0[5] = entropy(Air_ha, T = T_0[5], P = P_0[5]) "Primary Side Outlet Entropy"

"_____"

"Pipe Section 56: Combustion Chamber Inlet (Point 6)"

DELTA_P0_56 = Alpha_p * ((P_0[6] + P_0[5]) / 2)
P_0[6] = P_0[5] - DELTA_P0_56
h_0[6] = h_0[5]
T_0[6] = temperature(Air_ha, P = P_0[6], h = h_0[6])
s_0[6] = entropy(Air_ha, T = T_0[6], P = P_0[6])

"Pressure Drop through Pipe Section 56"
"Pressure at Combustion Chamber Inlet"
"Enthalpy at Combustion Chamber Inlet"
"Temperature at Combustion Chamber Inlet"
"Entropy at Combustion Chamber Inlet"

T_HXPS_Out = T_0[6]

"Set HX Primary Side Outlet Temperature To T_HXPS_Out"

"

"Combustion Chamber (Point 7)"

DELTA_P0_67 = Alpha_Comb * ((P_0[7] + P_0[6]) / 2)
P_0[7] = P_0[6] - DELTA_P0_67

"Pressure Drop through Pipe Section 67"
"Pressure at Combustion Chamber Outlet"

//DELTA_P_Comb = 0
//P_0[7] = P_0[6] - DELTA_P_Comb

"Pressure Drop through Combustion Chamber"
"Pressure at Combustion Chamber Outlet"

Q_dot_Comb = m_dot * (h_0[7] - h_0[6])
T_0[7] = temperature(Air_ha, h = h_0[7], P = P_0[7])
s_0[7] = entropy(Air_ha, T = T_0[7], P = P_0[7])

"Heat added by Combustion Chamber"
"Temperature at Combustion Chamber Outlet"
"Entropy at Combustion Chamber Outlet"

T_Comb_Out = T_0[7]

"Set Combustion Outlet Temperature To T_Comb_Out"

"

"Pipe Section 78: Turbine 1 Inlet (Point 8)"

DELTA_P0_78 = Alpha_p * ((P_0[8] + P_0[7]) / 2)
P_0[8] = P_0[7] - DELTA_P0_78
h_0[8] = h_0[7]
T_0[8] = temperature(Air_ha, P = P_0[8], h = h_0[8])
s_0[8] = entropy(Air_ha, T = T_0[8], P = P_0[8])

"Pressure Drop through Pipe Section 78"
"Pressure at Turbine 1 Inlet"
"Enthalpy at Turbine 1 Inlet"
"Temperature at Turbine 1 Inlet"
"Entropy at Turbine 1 Inlet"

"

"Turbine 1 (Point 9)"

P_0[9] = P_0[8] * (1 / Pr_t1)
h_0s[9] = enthalpy(Air_ha, s = s_0[8], P = P_0[9])

"Pressure at Turbine 1 Outlet"
"Turbine 1 Enthalpy for an Isentropic Process"

W_dot_T1 = Eta_t1 * m_dot * (h_0s[9] - h_0[8])
W_dot_T1 = m_dot * (h_0[9] - h_0[8])
T_0[9] = temperature(Air_ha, h = h_0[9], P = P_0[9])
s_0[9] = entropy(Air_ha, T = T_0[9], P = P_0[9])

"Ideal Work Done by Turbine 1"
"Actual Work Done by Turbine 1"
"Temperature at Turbine 1 Outlet"
"Entropy at Turbine 1 Outlet"

"

"Pipe Section 910: HX Secondary Side Inlet (Point 10)"

DELTA_P0_910 = Alpha_p * ((P_0[10] + P_0[9]) / 2)
P_0[10] = P_0[9] - DELTA_P0_910
h_0[10] = h_0[9]
T_0[10] = temperature(Air_ha, P = P_0[10], h = h_0[10])
s_0[10] = entropy(Air_ha, T = T_0[10], P = P_0[10])

"Pressure Drop through Pipe Section 910"
"Secondary Side Inlet Pressure"
"Secondary Side Inlet Enthalpy"
"Secondary Side Inlet Temperature"
"Secondary Side Inlet Entropy"

"

"Heat Exchanger: Secondary Side (Point 11)"

DELTA_P_HX_SS = Alpha_HX_SS * ((P_0[11] + P_0[10]) / 2)
//DELTA_P_HX_SS = 10 [kPa]
[kPa]
P_0[11] = P_0[10] - DELTA_P_HX_SS

"Pressure Drop through the HX Secondary Side"
"Pressure Drop through the HX Secondary Side: Given as 20"
"Secondary Side Outlet Pressure"

//Q_dot_HX_SS = Epsilon_HX * C_min * (T_0[4] - T_0[10])
Q_dot_HX_SS = m_dot * (h_0[11] - h_0[10])

"Heat Transfer Duty by HX Secondary Side"
"Heat Transfer Duty by HX Secondary Side"

```
//C_min = min(Cp_[4] * m_dot, Cp_[10] * m_dot)
T_0[11] = temperature(Air_ha, h = h_0[11], P = P_0[11])
s_[11] = entropy(Air_ha, T = T_0[11], P = P_0[11])

"Minimum Heat Capacity"
"Secondary Side Outlet Temperature"
"Secondary Side Outlet Entropy"

"Pipe Section 1112: Turbine 2 Inlet (Point 12)"

DELTA_P0_1112 = Alpha_p * ((P_0[12] + P_0[11]) / 2)
P_0[12] = P_0[11] - DELTA_P0_1112
h_0[12] = h_0[11]
T_0[12] = temperature(Air_ha, P = P_0[12], h = h_0[12])
s_[12] = entropy(Air_ha, T = T_0[12], P = P_0[12])

"Pressure Drop through Pipe Section 1112"
"Pressure at Turbine 2 Inlet"
"Enthalpy at Turbine 2 Inlet"
"Temperature at Turbine 2 Inlet"
"Entropy at Turbine 2 Inlet"

"Turbine 2 (Point 13)"

P_0[13] = P_0[12] * (1 / Pr_t2)
P_0[13] = P_amb + 10
h_0s[13] = enthalpy(Air_ha, s = s_[12], P = P_0[13])

"Pressure at Turbine 2 Outlet"
"Pressure at Turbine 2 Outlet"
"Turbine 2 Enthalpy for an Isentropic Process"

W_dot_T2 = Eta_t2 * m_dot * (h_0s[13] - h_0[12])
W_dot_T2 = m_dot * (h_0[13] - h_0[12])
T_0[13] = temperature(Air_ha, h = h_0[13], P = P_0[13])
s_[13] = entropy(Air_ha, T = T_0[13], P = P_0[13])

"Ideal Work Done by Turbine 2"
"Actual Work Done by Turbine 2"
"Temperature at Turbine 2 Outlet"
"Entropy at Turbine 2 Outlet"

"Generated Work"

W_dot_Gen = - W_dot_T2 * Eta_Gearb * Eta_Gen

"Work Required by Turbine 1 to Drive the Compressor"

W_dot_T1 = - W_dot_C

"Energy Balance"

E = (m_dot * h_0[1]) + W_dot_C + W_dot_T1 + W_dot_T2 + Q_dot_Comb - (m_dot * h_0[13])

"Specific Heat Values"

Duplicate i = 1,13

Cp_[i] = cp(Air_ha, T = T_0[i], P = P_0[i])
Cv_[i] = cv(Air_ha, T = T_0[i], P = P_0[i])

End

"Total Power Output"

W_dot_Out = W_dot_Gen

"Overall Cycle Efficiency"

Eta_Brayton = (W_dot_Out) / (Q_dot_Comb)

"Call Procedure"

Call test(Epsilon_HX, C_min, T_0[4], T_0[10], h_0[4], m_dot : Regen, Q_dot_HX_SS, Q_dot_HX_PS, h_0[5])
```