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PROCESS SIMULATION TOOL DEVELOPMENT FOR A SMALL GAS TURBINE

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ABSTRACT

Matti Päivärinne: Process simulation tool development for a small gas turbine
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Power consumption is increasing in the next decades and demands to fulfill this need are expected to turn towards cleaner and more efficient energy production. While renewables are expected to increase, their growth rate cannot compensate for the increase in power consumption alone, and gaseous fuels, such as natural gas, are expected to play a big role along renewables in this transition to produce energy cleaner and more efficiently. Demand for efficient and exact energy also drives decentralization, meaning that energy can be produced where demand is, to fulfill needs precisely and quickly. Many industrial applications also require process heat and with decentralized combined heat and power production, great efficiency increase is possible. Small gas turbines excel in this type of combined heat and power production with versatile fuels, also including natural gas. With this current continuing trend in the energy market, increase in the gas turbine installations can be expected.

Interest towards gas turbines increases the importance of gas turbine performance models. In off-design conditions, performance is significantly affected by the load and ambient conditions. With accurate models, the performance of engines can be predicted for each application and designing costs and time can be reduced. During operation, drive can be optimized to reach higher efficiencies and with engine monitoring, maintenances can be planned to be condition-based, not predictive based.

In this master's thesis, performance prediction model was created for intercooled and recuperated gas turbine process with two spools, both spools including generator, compressor and turbine. The model was requested by the company to replace currently used model, with one which could better correspond to the company's need. The developed model was steady-state, full range performance model which used Newton-Raphson iteration. The developed model was compared to old model and results were in-line. The new model was as requested by the company excluding some attributes which could not be included in the scope of this thesis but will be added to the model later.

Keywords: gas turbine, performance prediction, model, simulation, off-design

The originality of this thesis has been checked using the Turnitin OriginalityCheck service.

TIIVISTELMÄ

Matti Päivärinne: Prosessisimulointityökalun kehittäminen pienelle kaasuturbiinille
Diplomityö
Tampereen yliopisto
Konetekniikan diplomi-insinöörin tutkinto-ohjelma
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Energian käyttö tulee kasvamaan seuraavien vuosikymmenien aikana ja tämä lisääntynyt kasvu halutaan täyttää puhtaammilla ja paremman hyötysuhteen omaavilla energian lähteillä. Uusiutuvien energianlähteiden käyttö tulee nousemaan, mutta ne eivät pysty täyttämään tätä vajetta yksin ja kaasumaisten polttoaineiden, kuten maakaasun, odotetaan kasvavan uusiutuvien rinnalla tuottamaan tarvittavan energian. Tarve tehokkaaseen ja täsmälliseen energiaan ajaa myös hajautetun energian tuotannon kasvua, jotta energia voidaan tuottaa paikallisesti tarvittava määrä nopeasti. Monet teollisuuden prosessit, jotka hyötyvät hajautetusta energian tuotannosta, tarvitsevat lämpöä sähkön kanssa, ja yhteistuotannolla voidaan näissä prosesseissa päästä suuriin energian säästöihin. Pienet kaasuturbiinit toimivat erittäin hyvin tällaisissa prosesseissa käyttäen monipuolisia polttoaineita, suurimpana maakaasu, ja tuottaen samalla sähkö ja lämpöä. Samansuuntaisen trendin jatkuessa energiamarkkinoilla, kaasuturbiinien käyttöönotto tulee kasvamaan tulevaisuudessa.

Kasvanut kiinnostus kaasuturbiineja kohtaan lisää myös niiden toimintaa kuvaavien mallien tärkeyttä. Osakuormilla kaasuturbiinin toiminta on erittäin herkkä kuormanmuutoksille, sekä vallitseville ympäristöolosuhteille. Käyttäen tarkkoja malleja, kaasuturbiinin toimintaa voidaan tarkastella muuttuvien olosuhteiden mukaan ja näin koneiden suunnittelukustannuksia ja aikaa saadaan pienennettyä. Malleilla voidaan optimoida kaasuturbiinin ajo päästen korkeammille hyötysuhteille ja käytönaikaisella monitoroinnilla huoltotoimenpiteitä voidaan tehdä vasta silloin kun niihin on tarve.

Tässä diplomityössä kehitettiin prosessisimulointimalli välijäähdytetylle, rekuperoidulle, kaksi akseliselle kaasuturbiiniprosessille, jossa molemmissa akseleissa on generaattori, kompressori ja turbiini. Mallin tavoite on korvata yhtiöllä nykyisin käytössä ollut malli, vastaamaan paremmin yhtiön tarpeita. Kehitetty malli on steady-state, koko toiminta-alueen kattava malli, joka käyttää Newton-Raphson iterointia löytääkseen toimintapisteen. Kehitettyä mallia verrattiin nykyisin käytössä olleeseen malliin ja tulokset vastasivat toisiaan. Malli vastasi yhtiön odotuksia suurimmalta osalta, muutamaa poikkeusta lukuun ottamatta, joita ei ollut tämän diplomityön puitteissa mahdollista toteuttaa. Puutteet tullaan kuitenkin lisäämään malliin myöhemmin.

Avainsanat: Kaasuturbiini, Toiminnankuvaus, malli, simulaatio, osakuorma

PREFACE

This master's thesis was done for Aurelia Turbines during autumn 2018 and spring 2019. Instructors for this thesis were Toni Hartikainen from Aurelia Turbines and Henrik Tolvanen from Tampere University.

I would like to thank people from Aurelia Turbines for providing such interesting and challenging topic for this thesis, all those tips and support and coffee break conversations which helped me to finish my thesis. I want to thank also my instructors and friends and family for providing me support when I needed it.

Lappeenranta, April 1, 2019

Matti Päivärinne

CONTENTS

LIST OF SYMBOLS AND ABBREVIATIONS

Btu	British thermal unit
CCHP	Combined cooling, heating and power
CCPP	Combined cycle power plant
CFD	Computational fluid dynamics
CHP	Combined heating and power
DC	Direct current
DNA	Dynamic network analysis
EIA	U.S. Energy Information Administration
GPA	Gas path analysis
HP	High-pressure
IEA	International Energy Agency
IRG2	Intercooled, recuperated and 2-spool gas turbine process with 2 generators
LUT	Lappeenranta University of Technology
LP	Low-pressure
NIST	National Institute of Standards and Technology
N-R	Newton-Raphson
OECD	Organization for Economic Co-operation and Development
TIT	Turbine inlet temperature
Ts	Temperature-entropy
T-Mats	Toolbox for the Modeling and Analysis of Thermodynamic Systems
UI	User interface

A	*flow area*	*[m²]*
D	*impeller diameter*	*[m]*
c_p	*specific heat in constant pressure*	*[kgm²/Ks²]*
$effectiveness$	*heat exchanger effectiveness*	*[-]*
h	*enthalpy*	*[kJ/kg]*
s	*entropy*	*[kJ/K]*
m	*mass flow rate*	*[kg/s]*
N	*rotational speed*	*[RPM]*
p	*pressure*	*[Pa]*
q	*specific heat*	*[J/kg]*
Q_{net}	*net calorific value*	*[J/kg]*
R	*gas constant*	*[J/kgK]*
T	*temperature*	*[K]*
t	*maximum temperature ratio*	*[-]*
V	*speed*	*[m/s]*
w	specific work	*[J/kg]*
γ	*ratio between specific heats*	*[-]*
π	*pressure ratio*	*[-]*
ρ	*density*	*[kg/m³]*
η	*efficiency*	*[-]*

subscripts

c	*compressor*
t	*turbine*
pol	*polytropic efficiency*
f	*fuel*

1. INTRODUCTION

Global power consumption and power generation is increasing and will increase heavily in the next decades [1]. There are demands for shifting towards renewable, cleaner and more local energy production due to rising concern of global warming and depleting resources such as coal and oil. The shift to pure renewable energy production cannot be achieved in the next decades since renewable energy suffers high fluctuation of energy production depending on time of the day and season. With renewables, requested baseload cannot be delivered unless alternative power sources and energy storages are used.

Another trend in the energy market is rising demand in energy production by gaseous fuels and with current projections, natural gas is overtaking coal in energy production by 2030 [1]. A major advancement in this is happening in China [2], where the trend has been moving out of coal-fired power plants to cleaner energy production. Moreover, Germany has agreed to end reliance on coal by 2038 [3]. With these changes in energy market, increased interest on decentralizing energy production [4], and EU directive promoting combined heat and power (CHP) applications [5], there is a huge potential for small gas turbines in the future energy production.

Increased installations of gas turbines also lead to increase in designing and maintenance of gas turbines. During designing, it is essential to be able to predict the gas turbine operation in design point as well as off-design point operations. In the literature, many approaches to predict performance are proposed: models with alternative modeling methods [6][7], models concentrating to specific features of the gas turbines *e.g.* variable geometry compressors [8] or turbine cooling [9], and comparing different modeling methods [10]. Gas turbine models are also used for optimizing startups and load transitions to reducing the time and the fuel consumption [11][12][13] as well as performance modeling during operation to optimize maintenance procedure [6][14][15].

The goal in this master's thesis was to develop performance simulation tool for Aurelia Turbines small gas turbine Aurelia® A400. The modeled engine was 400 kW IRG2-process gas turbine. IRG2-process is two-spool, intercooled and recuperated with both turbines driving compressors and generators. Developed performance simulation tool was used to assist commercial team and R&D to predict the performance of the studied gas turbine and aimed to replace currently used Excel-based model with a model corresponding better to the users need. The Developed gas turbine model was created using MATLAB and it uses state variable method and estimated component maps to describe gas turbine operation accurately during design and off-design operation.

This thesis answered following research questions:

- What are small gas turbines, what is the future vision for this type of energy production and in what kind of applications are they used?
- How gas turbine function and its components behave?
- How gas turbines have been modeled and what the users' needs have been?
- What is the best method to develop a performance simulation tool for studied gas turbine to correspond to the users' requirements?
- How the developed model copes when comparing it to the current Excel-based model?

In this thesis, first the market potential of small gas turbines is discussed, followed by listing of applications for small gas turbines. Then the thermodynamics of gas turbines, which were used in the simulation, are explained as well as component behavior and gas turbine operation. After the operation is explained, literature survey of simulation principles is shown. Finally developed model is introduced and its function is compared to currently used Excel model.

2. SMALL GAS TURBINES

A gas turbine is a combustion engine, where continuously burned fuel is converted to thrust or shaft power. Main components in gas turbines are compressor, combustor and turbine and these components can be found in every gas turbine process. Schematic of simple gas turbine process is shown in Figure 1, and as can be seen, first air is lead through an inlet to a compressor where it is compressed and further delivered to a combustion chamber. In the combustion chamber air is mixed with fuel and burned. Burned exhaust gas is lead into the turbine where it is expanded, and work extracted. After the turbine, the hot exhaust gas is lead through outlet for further treatment. The turbine and the compressor are in the same shaft and work extracted from the turbine drives the compressor. After compressor, excess mechanical energy is yet transformed into electrical energy in a generator located at the end of the shaft. In jet engines, instead of electricity, excess energy is transformed into thrust in a nozzle and the turbine extracts only the amount of work the compressor requires. In this master’s thesis, only gas turbines used for power generation are considered.

Gas turbines are categorized by power output they provide and weight. Smallest gas turbines are called microturbines and the power range for these are less than 500 kW weighing less than 10 metric tons. Small gas turbines can be categorized for weighing more than 10 metric tons and outputting less than 2 MW. Turbines outputting more than 2MW are referred as industrial gas turbines.

With electrical application, it is also possible to utilize excess heat coming out of the turbine. This heat provides heating power, which can be used in industrial engines as a steam cycle heat source (combined-cycle power plants, CCPP), or in smaller application to produce steam or hot water (CHP).

In this chapter the future market for small gas turbines, and how recent trends in the energy market permit increase of small gas turbine usage are discussed. Moreover, applications for gas turbines and gas turbine process and components are described.

2.1 Market potential

The change in the energy market is favorable to small gas turbines, since there is a tendency to move towards decentralized energy production, cleaner fuels like natural gas, and combined heat and power generation. With these effects, gas turbines can have a big part in the next decades’ energy production.

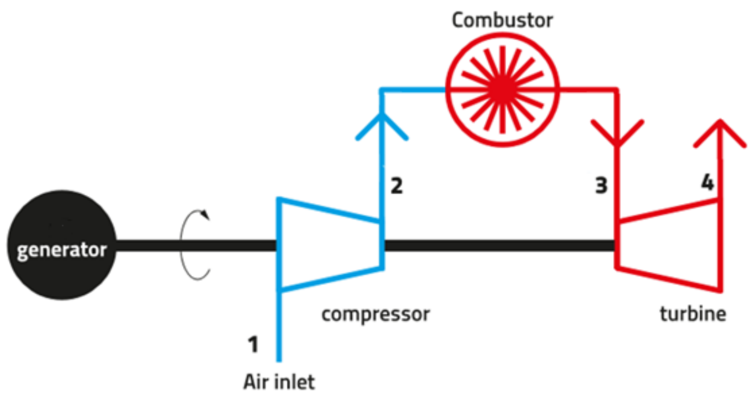


Figure 1. *Schematic of simple gas turbine*

2.1.1 Future energy demand

U.S. Energy Information Administration EIA introduced in their "International energy outlook" [1] their projections and estimations of energy consumption for the next 20 years, Figure 2. In the figure, history and projection for energy consumption is shown for OECD (Organization for Economic Co-operation and Development) and non-OECD countries. Units are quadrillion (10^{15}) Btu (British thermal units). OECD countries energy consumption is estimated to stay approximately at the present level, but non-OECD countries energy consumption is projected to grow 35 %, from present level of 350 quadrillion Btu to 473 quadrillion Btu by 2040.

In the International Energy Outlook, EIA also estimated different fuel consumptions for the same time period, shown in Figure 3. In this study nuclear, renewables, natural gas, coal, and petroleum and other liquids are considered energy source groups, nuclear being the smallest source and petroleum the biggest. Due to rising concerns about global warming and depletion of fossil fuel reserves, in EIA's projection, coal use is expected to diminish, while more clean energy, like natural gas and renewables, are expected to increase the most. EIA's projections for growth in energy consumption can't be compensated only with renewables, meaning that present fossil fuels, especially cleaner and more efficient natural gas, are also playing a big role in next decades' energy market.

International Energy Agency (IEA) is having similar view about future energy market, estimating in their World energy outlook 2018 [16] that natural gas overtakes coal in 2030 and becomes second-largest fuel in the global energy mix. IEA estimated that the use of natural gas among industrial consumers increases 45 %. Demand for natural gas is rising due to interest in developing economies caused by policies to improve air quality through coal to gas boiler conversions, led by China. Asia makes up half of the global growth in natural gas [2]. In 2017, natural gas demand grew 3 %, and China, where demand increased 15 %, leading the growth accounting for nearly one third of the global increase.

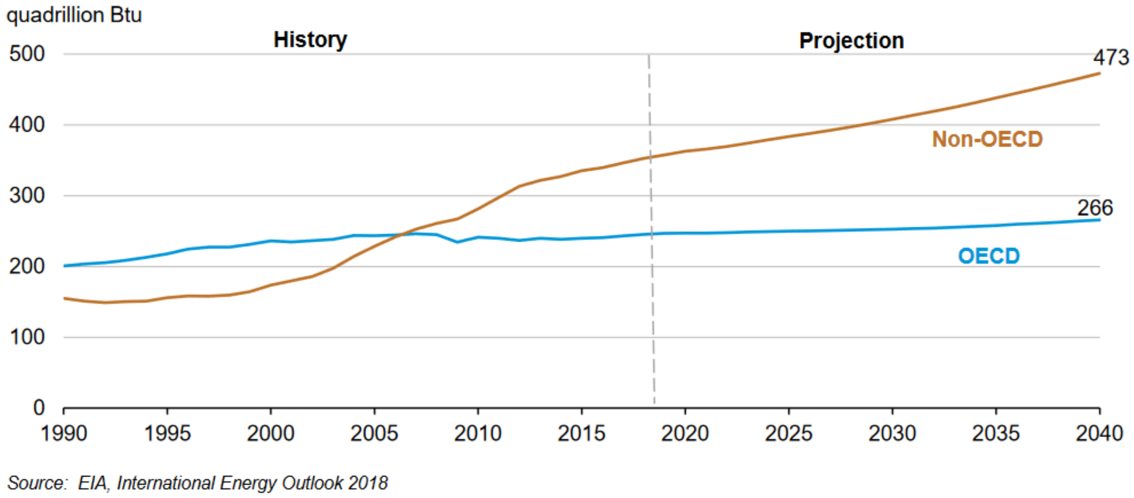


Figure 2. *World energy consumption [1]*

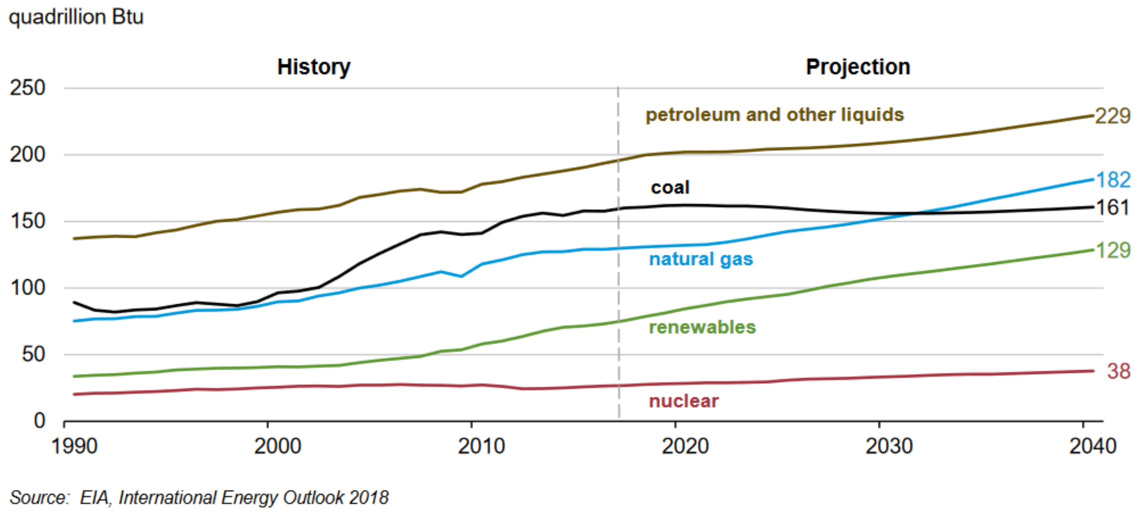


Figure 3. *World energy consumption by energy source [1]*

In 2016, natural gas-fired electricity generation was covering 42 % of total electricity production in the United States [17]. Natural gas-powered generator capacity accounted for 53 % for CCPP, 28 % for gas turbines and 17 % for steam turbines, making gas turbines the largest natural gas utilizer in electricity generation.

In the United states CCPP were the main cause of natural gas utilization growth in the 1990's and 2000s, due to being cleaner and more efficient than coal-fired plants [17] and due to increase in domestic natural gas production, especially with shale gas. The share of shale gas has been increasing, due to improvements in development of the shale gas production in the last decade [18]. However, current trend tends to turn towards decentralized energy production as can be seen from the survey done by Diesel and Gas Turbine Worldwide [19]. Figure 4 shows ordered power generation capacity between reciprocating engines, gas turbines and steam turbines in 2017. The survey included engines starting from 500 kW, and gas and steam turbines from 1 MW. Data was created by requesting sales numbers from manufacturers, however not every manufacturer's numbers were included in this report. The spike in gas turbines in 200 MW range is due the data not containing more specific differentiation of the gas turbines above 180 MW. As can be seen, de-centralized energy generation is already a large market. Using EIAs average of installation cost for gas turbines and engines 1,000 $/kW [20], the market cap was 23 billion dollars in 2017 and it will be increasing in the future as decentralized energy production is expected to keep rising. Currently power generation plants under 3.5 MW are dominated by reciprocating engines due to their higher efficiency compared to gas turbines in this power range. However, in many applications gas turbines are performing better, with new gas turbine technology and increased efficiency making gas turbines are able to supersede engines in small scale applications.

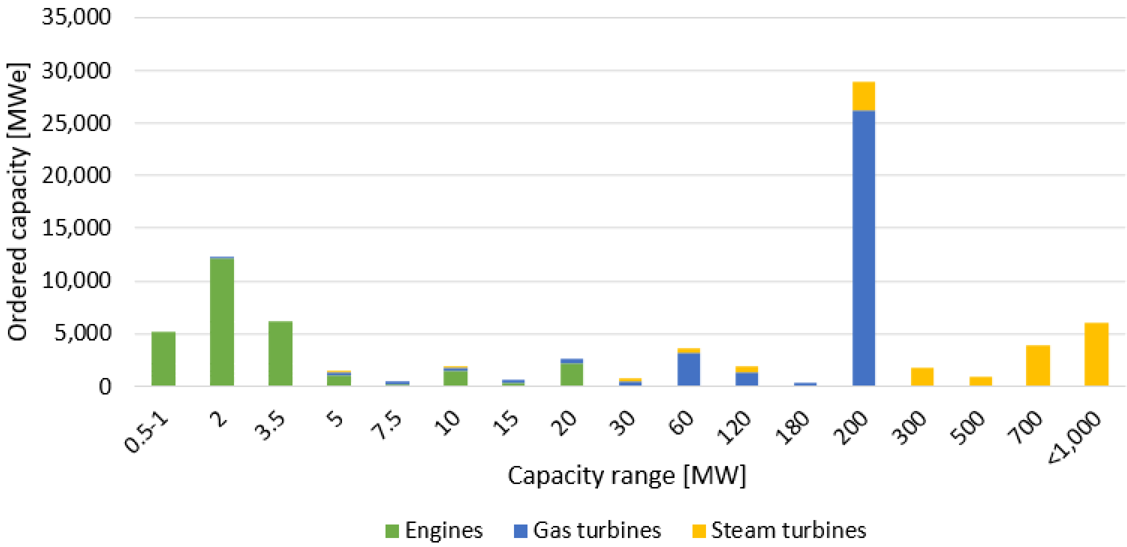


Figure 4. *Annual survey of ordered power generation capacity*

2.1.2 Gas turbine compared with reciprocating engine

As stated in Figure 4, reciprocating engines are dominating today's market in small scale power generation application. This is partly because reciprocating engines can reach better efficiencies than common gas turbines at the same power range in nominal load as well as part loads [21]. Efficiencies of reciprocating engines are between 35 % and 45 % when gas turbines can only reach about 30–33 %. In Figure 5, efficiencies of gas turbines and reciprocating engines in the current market are shown in the power range of 0–2,000 kW engines. Data is obtained from gas turbine manufacturers and ASUE study [22], collected and combined by Aurelia turbines. Black spots show reciprocating

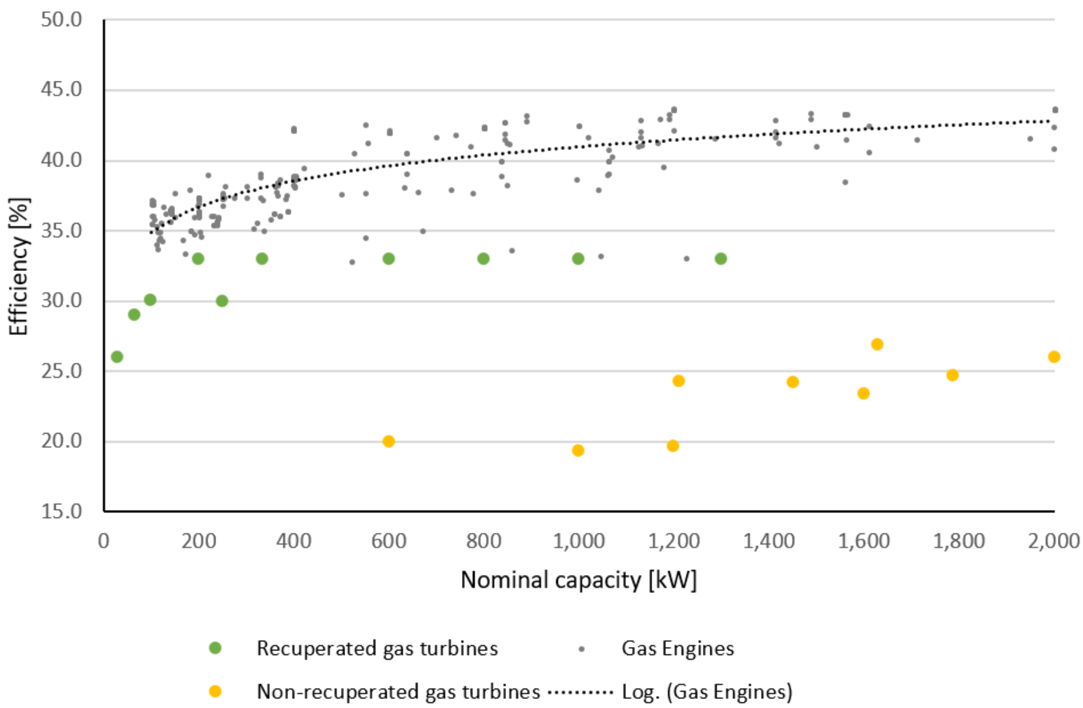


Figure 5. *Gas turbine and reciprocating engine efficiencies*

engines and as can be seen, they dominate gas turbines, recuperated gas turbines, marked with green circles, and non-recuperated gas turbines, marked with yellow circles, in the whole power range. But introducing a new gas turbine process with higher efficiency, like IRG2-process, it is possible for gas turbines to displace reciprocating engines from their dominating position in the market, also providing other benefits of gas turbines described later in this chapter.

Reciprocating engines are internal combustion engines, where difference to gas turbines is that in reciprocating engines expansion and compression of air and fuel is carried out in a cylinder fitted with a piston. These types of engines use mainly diesel as a fuel in power generation. They work great in local power generation with moderately developed heat recovery and with close to standards fuels [23].

Problems with reciprocating engines arise when the application requires variable fuels or when steam production is required. Reciprocating engines can only be used with very small variation in fuel, with all distinctly different types of fuel needing a new engine design. In CHP applications reciprocating engines begin to struggle because the components downstream of the engine create changing back pressure to the engine outlet and reciprocating engines are sensitive to keeping back pressure constant. The engines consist of multiple cylinders and the exhaust gas consist of flow streams, flow is not continuous and relatively small, which makes them struggle in combined heat and power applications. Having multiple cylinders with pistons also increases the amount of moving parts in the engine, leading to a more complicated engine design, which requires more maintenance.

Power generation gas turbines produce electrical energy and hot flue gas which can be used as heat power and German Energy Agency states that process heat is the most energy intensive application in industrial energy use, accounting for up to 64 % of total industrial energy consumption [24]. Small gas turbines are used in processes which require local energy production or when local energy is desirable. They can be also used solely to produce electrical energy, but they excel at applications which require electrical power along with heating or cooling power. With combined CHP applications, it is typical to reach efficiencies of over 80 % [25]. Heat demand is common in industry processes which require also small-scale energy production. With gas turbines, heat can be supplied in forms of hot water or steam and applications vary from food processing, brewery and dairy industry to pharmaceutical and chemistry [26].

Gas turbines can also operate in applications which require combined cooling, heat and power (CCHP). In these applications, it is possible to convert exhaust gas heat to cool water with an absorption chiller. Cooling is commonly required in air-conditioning and industrial freezers and these applications use great amount of energy if the cooling is achieved using electricity. In most of these cases there is the need for electricity, for example, supermarkets need electricity in lightning and other electrical devices and a great amount of cooling, in freezers and cold areas. In another example can be found in public buildings, *e.g.* Israeli airport is deploying 9.2 MW gas turbine for providing electricity and air-conditioning [27]. In these kinds of applications, electricity can be produced with gas turbines and the cooling can be achieved with gas turbine exhaust gas cooling can be fulfilled without extra energy input.

Another good application for gas turbines can be found when there is a distinct need for direct current (DC). Locally produced DC does not have disadvantages of transmission and DC can be generated inside the gas turbines or by transformer located near the turbine. Common DC applications are for example data centers, where the required power is well defined, meaning there are no fluctuation in the power requirements, and there is a need for cooling. An example of can be found in an Australian data center, which deployed a 1 MW gas turbine for their power and cooling requirements [28].

Small gas turbines work great in the same applications which reciprocating engines struggle. The reason why reciprocating engines are used in those applications, is their better electrical efficiency, but when modern small gas turbine process is considered, the benefit from better electrical efficiency is no longer a valid reason. When the efficiency of gas turbines rises above most of the reciprocating engines, gas turbines become very competitive also in pure electrical power generation. Gas turbines are much simpler than reciprocating engines, having only one or two rotating parts, thus requiring less maintenance.

With increasing demand for energy, tendency to turn towards cleaner energy sources and desire for decentralized energy production both indicate, that small gas turbines can fill a major share of the future energy requirements.

2.2 Process

Main parameters affecting gas turbine efficiency are the pressure ratio and the turbine inlet temperature (TIT), this is discussed more in chapter 3. However, the pressure ratio and TIT can only be increased to a certain limit, dictated by material tolerances and the cost of manufacturing. Better efficiencies can be reached by modifying the process and introducing other components, which will be discussed in this chapter more.

2.2.1 Components

Components which are used in gas turbines, as mentioned earlier, are a compressor, a combustor and a turbine. Because each component in the gas turbine is separate and has a different task, each component can be designed and tested individually. This also allows modification and the possibility to add new components in the gas turbine process and most of these modifications are aimed to increase efficiency or specific work output. Compression can be done in stages with multiple compressors to reach higher pressure ratios while keeping the efficiency high. Expansion can also be split into several stages to allow for a better efficiency in higher pressure ratios. In small gas turbines, efficiency can be increased in moderate pressure ratios with a recuperator, which is a heat exchanger located between the compressor and the combustor, using excess heat from the turbine to preheat air for the combustor. This leads to a decreased need for fuel and a higher efficiency but only in moderate pressure ratios. With multiple turbines, specific work output can be increased with reheating the air between turbines in another combustor, and with multiple compressors, intercooler can be used to increase specific work by decreasing the inlet temperature of the second compressor. These also increase the efficiency, when used with a recuperator in moderate pressure ratios.

Gas turbine combustion requires a continuous flow, which sets demands for the compressor. From different compressor types, axial, centrifugal and positive displacement compressors, only axial and centrifugal compressors can provide sufficient air flow for combustion. Figure 6 shows a visualization of different types of compressors and their pressure ratio in comparison to mass flow rate. As can be seen, positive displacement compressors have the highest pressure ratio among compressors, but while flow is not continuous and relatively small, it is not used with gas turbines. In the axial compressor, air flows axially through the compressor where a blade row, also called stage, increases the pressure of the air. With axial compressors, pressure ratio is relatively small, around 1.05–1.45 per stage [29], but mass flow rate is large and thus these types of compressors are used with industrial size gas turbines. Pressure ratio can be increased by adding multiple stages to a compressor, and while pressure ratio is small and no direction changes to the air flow occur, their efficiency is higher than centrifugal compressors.

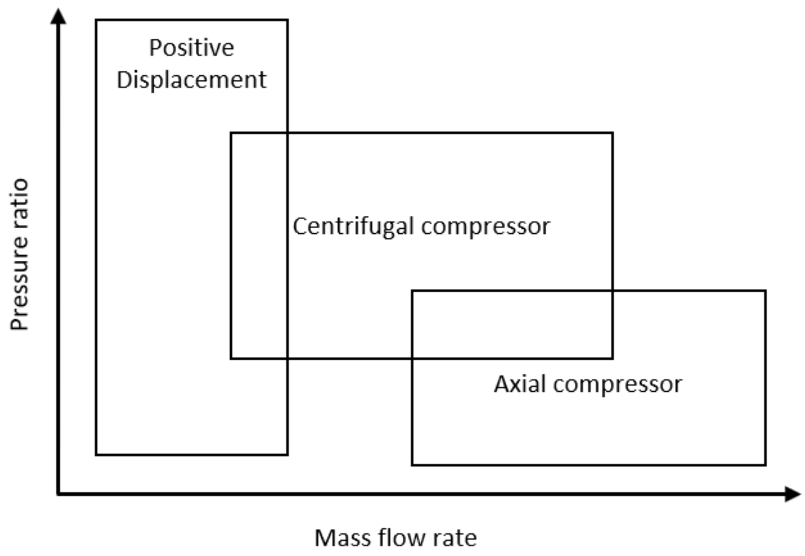


Figure 6. *Performance characteristics of different types of compressors, modified from [29]*

Centrifugal compressor, also called radial compressor, is another type of compressor used with gas turbines and this is more common with smaller gas turbines, where mass flow is not sufficient to use axial compressors efficiently. In a radial compressor, air enters the compressor in an axial direction, where rotating impeller increases the speed and pressure and changes the direction of the air to radial. After the impeller, air is lead to a diffuser where its' speed is decreased, and its' pressure further increased. Radial compressors can increase pressure ratio more than axial compressors and in small gas turbines, which use radial compressors, only one or two stages are used. In this type of processes, the pressure ratio is usually 3–5.

As with compressors, turbines have axial and radial types. Unlike in compressors, where work is done on gas, in turbine flowing gas does work by expanding and rotating turbine blades. In axial turbines, gas flow goes axially through the turbine, whereas in radial turbines gas enters radially and exits axially. Axial turbines allow greater mass flow and thus they are used in larger scale gas turbine engines. In general, engines using axial gas turbines have greater efficiency, because axial turbines allow higher TIT by using a blade cooling technique and blade coating with thermal resistance [30]. However, advancements in additive manufacturing can overcome manufacturing difficulties with cooling in radial turbines allowing higher TIT also in radial turbines in the future [31] [32]. Small and micro gas turbines are using radial turbines because axial turbines have lower efficiency at smaller mass flow rates. Advantages of radial turbines are a short and compact rotor, whereas the lower pressure ratio of axial turbines requires multiple stages, and these add length to the engine. Radial turbines also come at a lower cost, due to fewer parts and easier manufacturing.

All gas turbine combustors have the same function: increasing the temperature of pressurized gas and it is achieved with a continuous process where fuel is mixed with air coming from the compressor and ignited. After the gas turbine is started, the flame is self-sustaining, and it won't require an external igniter. After fuel is burnt, gas is then lead out of combustor towards the turbine. Design of the combustor range from annular, where combustion happens through the whole annulus of the gas turbine, to silo type combustors, where the combustor is an external silo. Silo type combustor is shown in Figure 9. The most recent development is done towards combustors with lower NOx emissions [33].

The efficiency of a gas turbine with a moderate pressure ratio of 4–10, can be increased with introducing a recuperator to the process. The effect on efficiency is discussed in chapter 3.2.1. The recuperator is a heat exchanger used to heat compressed air before the combustor with flue gas from turbines. Air heated with flue gas reduces fuel required to increase the temperature of the gas to the designed turbine inlet temperature, thus leading to an increased efficiency. The recuperator is shown in Figure 8 and as can be seen from the figure, the recuperator increases the size of the whole engine, while the physical size of the recuperator is large compared to other components. Passing hot expanded gas with low density, great mass flow and small pressure losses requires large dimensions from the heat exchanger.

When processes have high pressure ratio, compression can be done in multiple compressors to increase compressor efficiency. In these kinds of processes, specific work can be increased by installing an intercooler between the compressor stages. The intercooler is a heat exchanger where cold fluid cools warm air coming from the previous compressor. As described in chapter 3, the compressor does not only increase pressure, but also the temperature of the air, and with warmer inlet air the specific work of compressor decreases. The objective of an intercooler is to chill the air coming from the previous compressor to reduce specific work of the second compressor. Using only ab intercooler in the process reduces the efficiency, but when used with recuperator as shown in chapter 3, the efficiency is increased.

Another way to increase specific work output of the gas turbine process is use to reheating of the gas between multiple turbines. This is done by adding a second combustor between turbine stages. The effect to the process is similar to that of intercooling, decreasing efficiency used alone but combined with recuperator leading to an even further increased efficiency with moderate pressure ratios. This process of reheating the gas between multiple turbines is called a reheat.

2.2.2 Common parameters

A common microturbine is recuperated, and it has one spool with a centrifugal compressor, a radial turbine and a generator [34][35][36]. Schematic for such process is shown in Figure 7. This type of process is usual in gas turbines under 400 kW and its' efficiency varies from 30 % to 33 %. In microturbines, all parameters are relatively close to each others across different manufacturers, pressure ratio being 4.5, mass flow rate being 1.3–2.1 kg/s, and TIT being 950 ºC, which means that those parameters are optimized, and any improvements would be costly or at least risky.

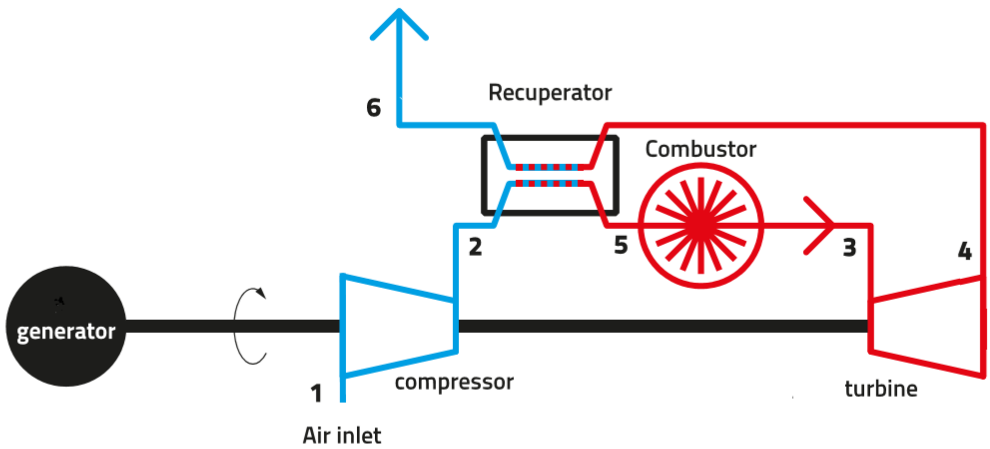


Figure 7. *Recuperated single-spool gas turbine*

Common microturbines use technology that has been in use for the last 50 years. Without any breakthrough in material science, the efficiency of this cycle will not be notably improved. With modifications to the small gas turbine cycle, it is possible to reach better efficiencies with the same parameters. One modified small gas turbine cycle is

Aurelia® A400 gas turbine, whose schematic is shown in Figure 8 and visualization with components specified in Figure 9. In Figure 9, TURU stands for turbine unit, including associated turbine, compressor and generator mounted to a shaft. Performance model created in this master's thesis was created for this process. This gas turbine process is twin-spool, with both spools including a compressor, turbine, and generator. The compressor and the turbine are both radial types. Generators are permanent magnet generators; they ca operate at variable speeds and they are directly linked to the spool without a gearbox.

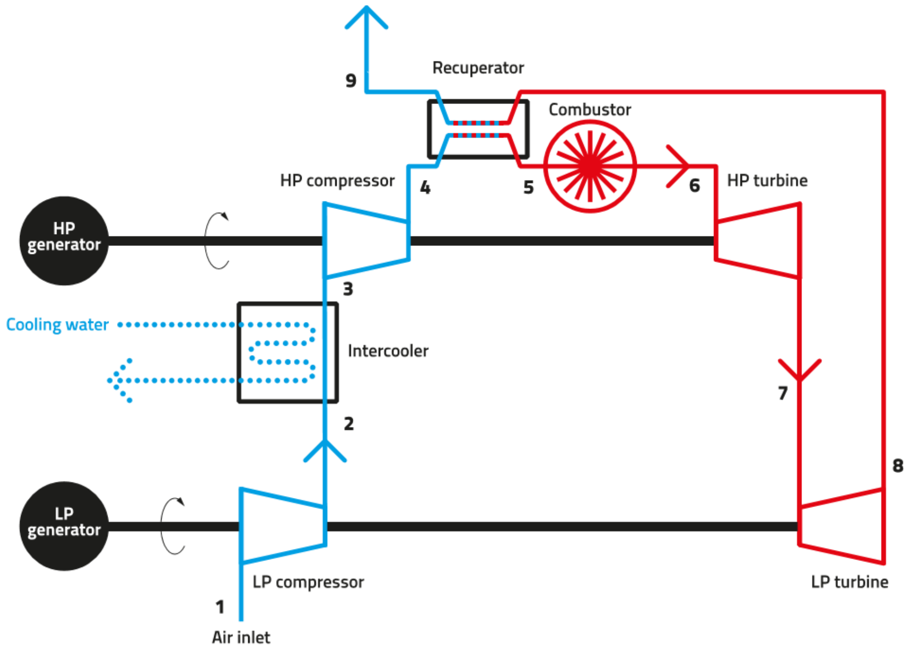


Figure 8. *Aurelia turbines IRG2-process schematic*

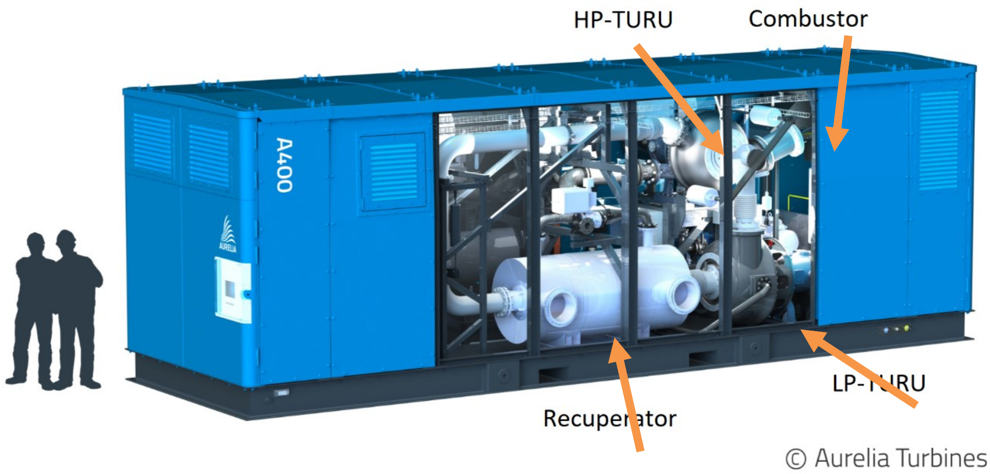


Figure 9. *Aurelia® A400 gas turbine*

The process also includes a recuperator and requires an external intercooler. The combustor is modular and can be changed if necessary, to be able to burn variety of fuels. To further increase the efficiency both spools have active magnetic bearings to reduce friction and maintenance needs. The active magnetic bearings also remove the need for oil pumps. With this kind of process, it is possible to reach efficiencies of about 40 %, with the same magnitude of TIT and pressure ratios as other small gas turbines.

3. GAS TURBINE CYCLES

Simulation of a gas turbine is done with mathematically describing thermodynamic relations of the engine. In this chapter these thermodynamic principles, for processes and component introduced in the previous chapter, are described. The behavior of each component is discussed and the change to efficiency and to specific work is described with idealized processes. Idealized processes were chosen to have an understanding why each component are used with gas turbines. In chapter 6, these thermodynamic principles are used to create a simulation of the process shown in Figure 8.

3.1 Simple cycle

Simple cycle gas turbine, as the name states, is the most basic gas turbine process. This type of process is only including essential components for the operation of a gas turbine, compressor, combustor and turbine, and schematic for the simple cycle is shown in Figure 1. In this chapter, idealized simple cycle, Brayton cycle, and its operation are described. Idealizing gas turbine cycles include the following simplification [33]:

- compression and expansion are reversible and adiabatic, i.e. isentropic,
- the change of kinetic energy of the working fluid between the inlet and outlet of each component is negligible,
- there are no pressure losses in the inlet ducting, combustion chambers, heat exchangers, intercooler, exhaust ducting and ducts connecting the components,
- the working fluid has the same composition throughout the cycle and is a perfect gas with constant specific heats,
- the mass flow of gas is constant throughout the cycle,
- heat transfer in a heat-exchanger is perfect, so that heat increase in the cold side is maximum possible and equals the temperature drop in the hot side.

Temperature-entropy-diagram (Ts-diagram) for the Brayton cycle is shown in Figure 10. Ts-diagram shows temperature, entropy and pressure relation with air. This method is an effective way to describe a gas turbine process. In the diagram, temperature is at y-axis, entropy is at x-axis and parabolic constant pressure lines are at the background of diagram. Numbers refer to the same points than in Figure 1, number 1 being the inlet, 2 after compression, 3 after combustion and 4 at the outlet. In Ts-diagram curves p0 and p1 are constant pressure lines, p0 [Pa] is atmospheric pressure and p1 is the pressure delivered by a compressor. Gas turbine process starts with air entering compressor and in ideal case, compression happens in constant entropy, from point 1 to 2. After compression is heat addition in a combustor. During this time in Ts-diagram are moving

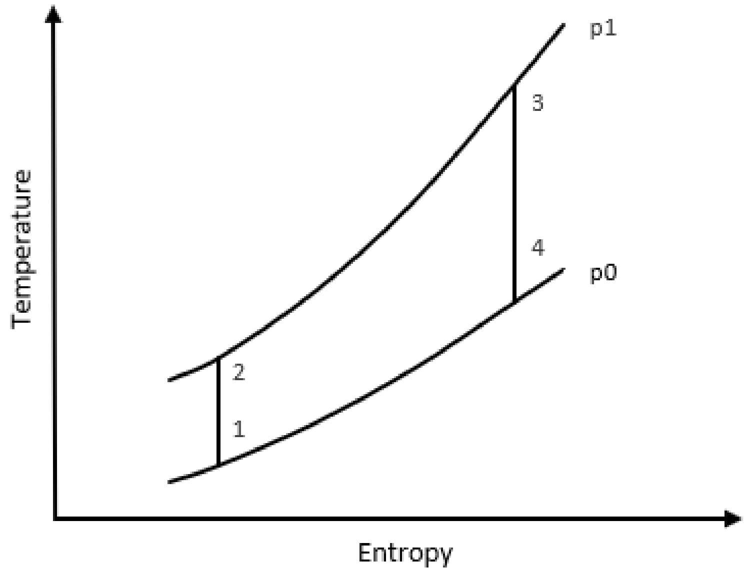


Figure 10. *Ts-diagram for Brayton cycle*

along constant pressure lines, from point 2 to 3. After combustion air is guided to a turbine to expand in constant entropy, point 3 to 4. Rest of the heat is released to the atmosphere, point 4 to 1.

Like stated earlier, work extraction in a gas turbine is done with expansion in the turbine. Some of this work is used to drive the compressor and to provide pressure in the cycle. Cycle need also external energy source and it is done by heat addition in the form of chemical energy of fuel added in the combustor. These work and heat transfers can be expressed in the change of enthalpy

$$w_c = -(h_2 - h_1) \tag{1}$$

$$w_t = h_3 - h_4 \tag{2}$$

$$\text{q}= h_3 - h_2 \tag{3}$$

Work required by compressor per unit mass flow, i.e. specific work, is w_c [kJ/kg], specific work produced by a turbine is w_t [kJ/kg] and heat addition per unit mass flow in a combustor is q [kJ/kg]. h [kJ/kg] is enthalpy and subscripts are referring to Figure 10. For a perfect gas, the relation between temperatures T [K] and enthalpy is $h = c_pT$, c_p [kgm^2/Ks^2] being specific heat capacity in constant pressure and variation of c_p can be assumed to be negligible with small changes in temperature. Now with equations (1) –(3), ideal cycle efficiency can be expressed as net work output per heat supplied

$$\eta = \frac{W}{Q} = \frac{c_p(T_3 - T_4) - c_p(T_2 - T_1)}{c_p(T_3 - T_2)}. \tag{4}$$

In the equation above, η is efficiency and T is temperature in point where subscripts are referring. In equation (4) variation of c_p need to be taken in to account, because specific heat is a function of temperature and temperature difference between compressor and turbines are high enough.

Isentropic pressure-temperature relation is

$$\frac{T_2}{T_1} = \left(\frac{p_1}{p_0}\right)^{(\gamma-1)/\gamma} = \frac{T_3}{T_4} \tag{5}$$

where p_1/p_0 is pressure ratio, subscripts referring to Figure 10 constant pressure lines and γ is ratio between specific heats and for air $\gamma = 1.4$. With p-T relation efficiency can be expressed

$$\eta = 1 - \left(\frac{1}{\pi}\right)^{(\gamma-1)/\gamma} \tag{6}$$

where π is pressure ratio p_1/p_0. As equation (6) states, efficiency is only depending on pressure ratio and ratio between specific heats. When simplification with gas nature is made, and γ for air in the whole process is used, only pressure ratio is affecting to cycle efficiency. Variation in efficiency compared to pressure ratio in the process is showed in Figure 11.

As can be seen, efficiency increases with pressure ratio, but in high pressure ratios efficiency increase is not significant enough to compensate material costs to further increase pressure ratio.

For a simple cycle, specific work output is turbine work subtracted by specific work output required by the compressor and it can be expressed in a non-dimensional form in a function of pressure and temperature ratio and ratio between specific heats

$$\frac{W}{c_p T_1} = t\left(1 - \frac{1}{\pi^{(\gamma-1)/\gamma}}\right) - \left(\pi^{(\gamma-1)/\gamma} - 1\right). \tag{7}$$

Inlet temperature T_1 is not a major parameter for gas turbine performance and it is then convenient to show specific work output in a non-dimensional form. Specific work output is plotted in Figure 15 as a functions of pressure ratios and temperature ratios for air. In the figure, specific work output increase for intercooled cycle, solid line, is shown compared to simple cycle, dashed line.

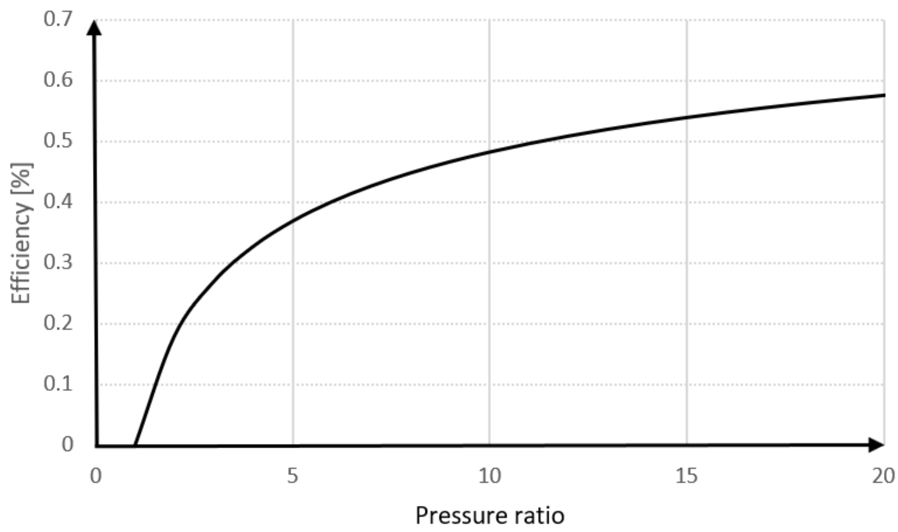


Figure 11. *Efficiency of a simple cycle*

3.2 Modified cycle

With bringing modification to process, it is possible to reach higher efficiencies and specific work outputs. With introducing recuperator in a one-spool process, a spool is a shaft which the turbine rotates, it is possible to increase efficiency in a relatively small pressure ratio. Gas turbine process having multiple spools, an intercooler can be used between compression stages to cool compressed air from the first stage or reheat between turbines to increase the temperature of gas before the second turbine. Both of these cycles increase specific work output of the process and they also can be used together. In this chapter these modifications are explained in detail with the changes in efficiencies and specific work output.

3.2.1 Recuperated cycle

Schematic for recuperated single-spool gas turbine process was shown in Figure 7. In recuperated gas turbine process compressed air is heated with flue gas heat in a heat exchanger. In this case, heat exchanger is assumed to be ideal cross-flow heat exchanger. In cross-flow heat exchanger hot and cold fluids, in this case cold air from the compressor and hot gas from turbines, flow in counter directions. In ideal case all heat from hot gas is transferred to cold air, meaning that cold air is at the exit at the same temperature than gas entering heat exchanger.

Ts-diagram for ideal gas turbine cycle with recuperator is shown in Figure 12, where numbers are referring to Figure 7. Compression happens between point 1–2, heat

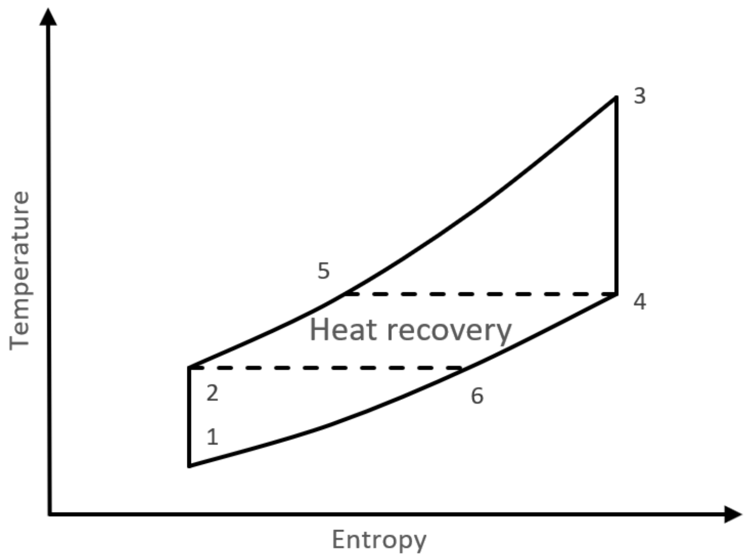


Figure 12. *Ts-diagram for a recuperated cycle*

addition in recuperation between points 2 and 5, combustion chamber heat addition between points 5–3, expansion in turbine between 3–4 and heat from exhaust gas is transferred to compressed air in recuperator between lines 4–6. Heat recovery from recuperator is showed in are 2–5–4–6.

Now external heat addition in a combustor is done between points 5 and 3, thus efficiency becomes

$$\eta = \frac{W}{Q} = \frac{c_p(T_3-T_4)-c_p(T_2-T_1)}{c_p(T_3-T_5)}. \tag{8}$$

In ideal heat exchanger, hot gas inlet temperature is the same as cold air outlet temperature, $T_4 = T_5$, using nomenclature $t = T_3/T_1$ for maximum temperature ratio and equation (5) isentropic p-T relation, efficiency can be expressed in terms of pressure ratio, temperature ratio and ratio between specific heats

$$\eta = 1 - \frac{\pi^{(\gamma-1)/\gamma}}{t}. \tag{9}$$

In maximum temperature ratio T_1 is almost always ambient temperature and about constant, efficiency increases as maximum temperature increases and decreases while pressure ratio increases. Efficiency curves for different values of t are shown in Figure 13 with air isentropic expansion factor.

In Figure 13 dashed line is simple cycle efficiency and solid lines are efficiencies with temperature ratios from 2 to 5. Benefit from recuperator is greatest at lower pressure ratios and after pressure ratio increases further certain point, point where in Figure 12 pressure ratio increases so that T_2 exceeds T_4, recuperator start to lower efficiency.

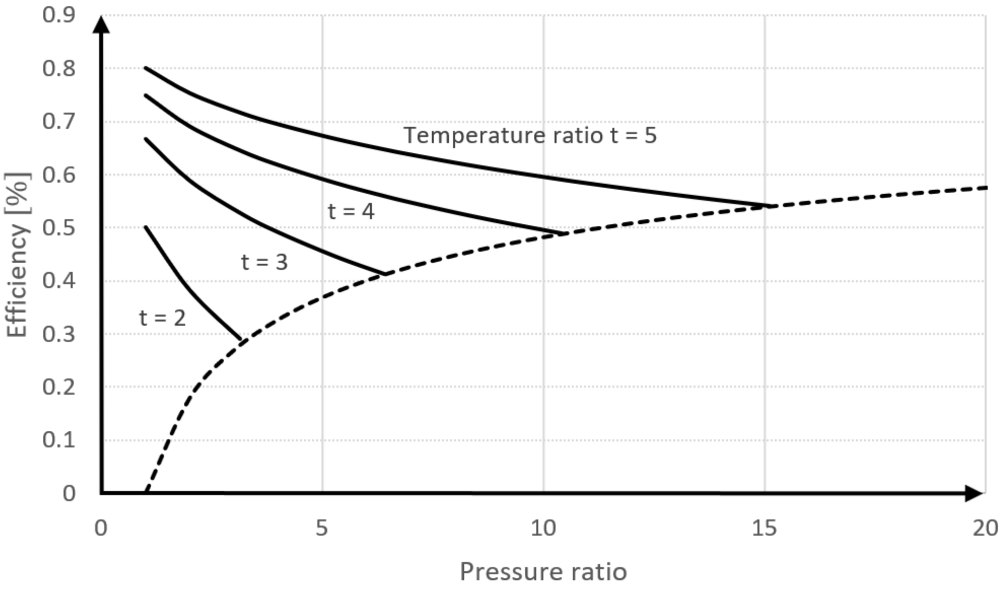


Figure 13. *Efficiency of a recuperated cycle*

When pressure ratio is at 1, efficiency is showing 80 % in higher temperature ratios, but this is only to the ideal case. Pressure ratio near 1 would bring very close to 0 specific work, Figure 15, and it would require a huge power plant, decreasing efficiency further. Real efficiency peak is around pressure ratio of 5 [33].

3.2.2 Intercooled cycle

Specific work output can be increased with intercooling between low-pressure (LP) and high-pressure (HP) compressors. T-s diagram for intercooled part of the process is shown in Figure 14, numbers refer to Figure 8, intercooled and recuperated cycle. Specific work output is increased due to decrease in required compression work. Compression work is decreased, because of vertical distance between a pair of constant pressure lines increases as the entropy increases, thus $(T_2 - T_1) + (T_4 - T_3) < T_{2'} - T_1$. [33]

In intercooled gas turbine process, compression is done in 2 stages, first air at atmospheric pressure is compressed in LP-compressor to pressure p1, Figure 14. After compression temperature of the air rise from T_1 to T_2, and the air is taken into intercooler, where the temperature is lowered in constant pressure, ideally to $T_3 = T_1$. After intercooler, air is compressed with HP-compressor to the final pressure, p2.

Assuming air is cooled to T_1, Saravanamuuttoo *et al.* [33] claim that specific work output is a maximum when the pressure ratios of the LP and HP compressors, and hence temperature drop and work transfer, are equal. With these in mind, it is possible to derive

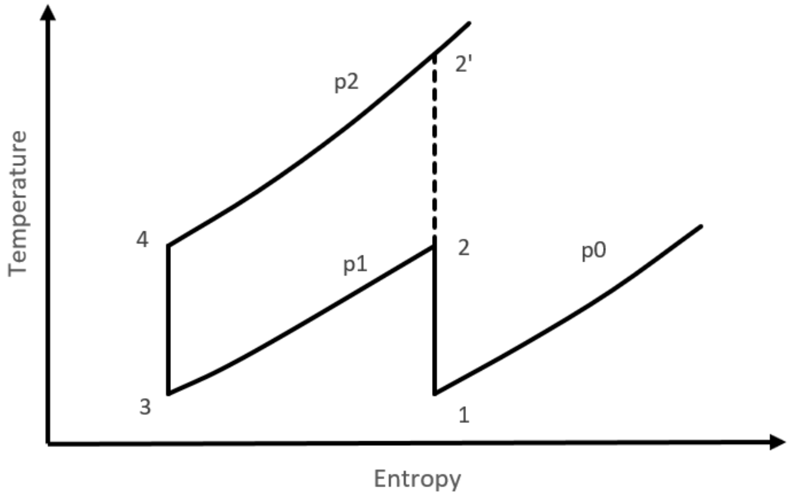


Figure 14. *Ts-diagram for an intercooled compression*

simple cycle specific work equation (7) and efficiency (6). Using nomenclature $c = \pi^{(\gamma-1)/\gamma}$ they become

$$\frac{W}{c_p T_1} = t\left(1 - \frac{1}{c}\right) - 2\sqrt{c} + 2 \tag{10}$$

and (11)

$$\eta = \frac{t(1-1/c) - 2\sqrt{c} + 2}{t - \sqrt{c}}. \tag{12}$$

In Figure 15 specific work output increase in intercooled process is shown compared to simple cycle. Simple cycle specific work output is shown in dashed lines and intercooled cycle is drawn with solid lines. Specific work output only depends on maximum temperature ratio and pressure ratio, when gas properties are assumed to stay constant. Again, curves for different maximum temperature ratios are plotted against pressure ratio to show dimensionless specific work output. As can be seen specific work output increases substantially in higher pressure and temperature ratios.

Increase in specific work output is increased in expense on a decrease in efficiency. The decrease in efficiency is shown in Figure 16. It is expected while intercooler adds one less efficient cycle, cycle 2─2'─4─3 in Figure 14. The dashed line indicates simple cycle efficiency and solid lines are efficiency lines with different maximum temperature ratios. As maximum temperature increases, with intercooled cycle efficiency reduction becomes less severe.

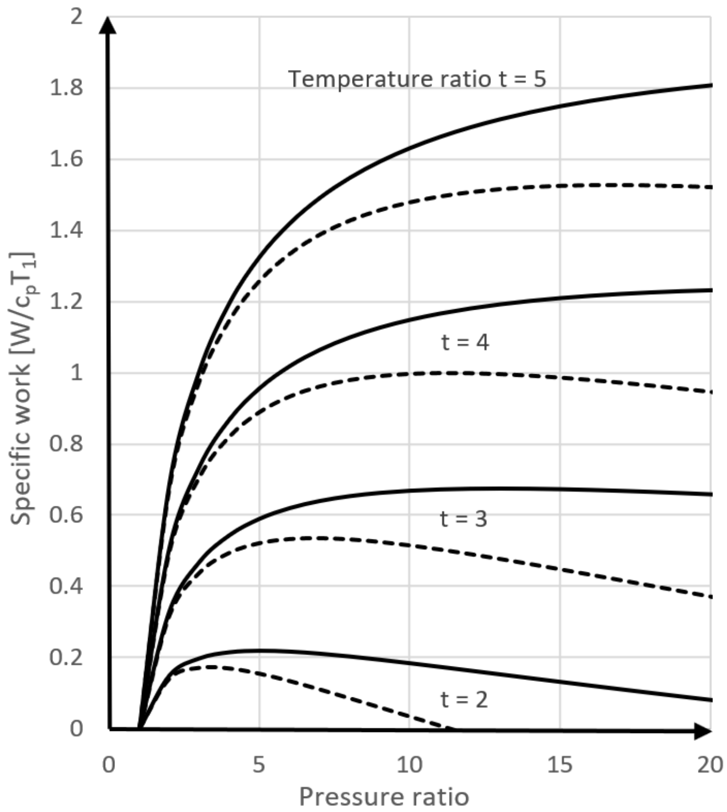


Figure 15. *Specific work output of a simple and intercooled cycle*

3.2.3 Recuperated intercooled cycle

Combining recuperated and intercooled cycle, efficiency loss in intercooling can be overcome. Efficiency becomes in this cycle, as expressed with c and t,

$$\eta = 1 - \frac{2\sqrt{c}-2}{t-t/c}. \tag{13}$$

Figure 17 shows the efficiency of recuperated and intercooled cycle. Dashed lines represent intercooled cycle efficiency from Figure 16 and solid lines recuperated and intercooled cycle. Multiple maximum temperature ratio (t) curves are plotted against pressure ratio and efficiency. As in recuperated cycle, benefit in efficiency is greatest at lower pressure ratios like explained in the chapter concerning recuperator. Cycle with recuperator and intercooler also have maximum pressure ratio, similar to recuperated cycle, where further increasing pressure ratio leads to lower efficiencies than just intercooled cycle.

Looking at Figure 17 and Figure 13, where recuperated, and intercooled and recuperated cycle efficiencies are shown, the increase in efficiency in the latter cycle is noticeable. Gain in efficiency increases as maximum temperature ratio increases while pressure

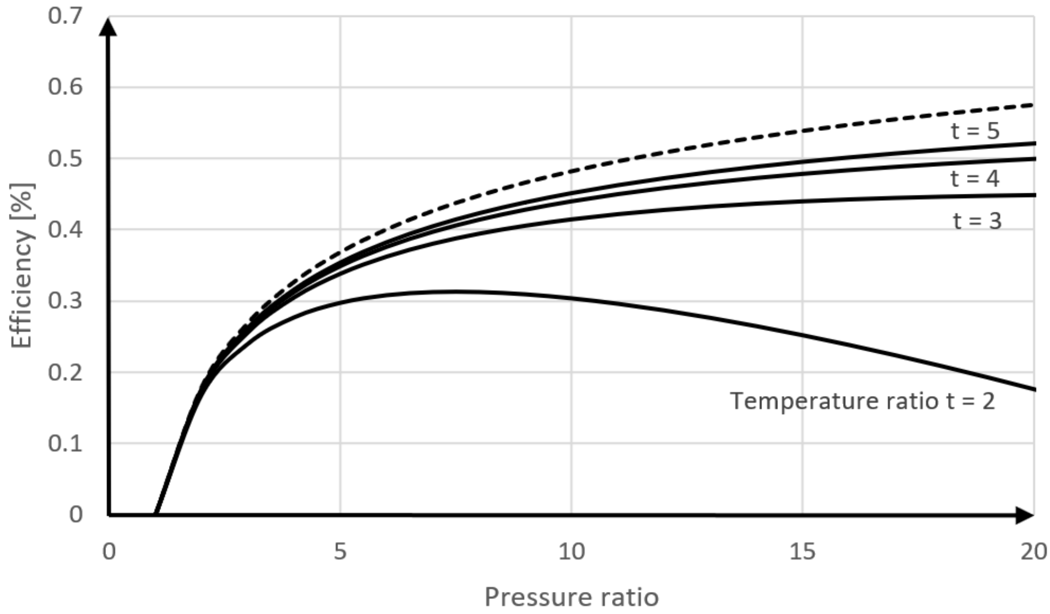


Figure 16. *Efficiency of an intercooled cycle*

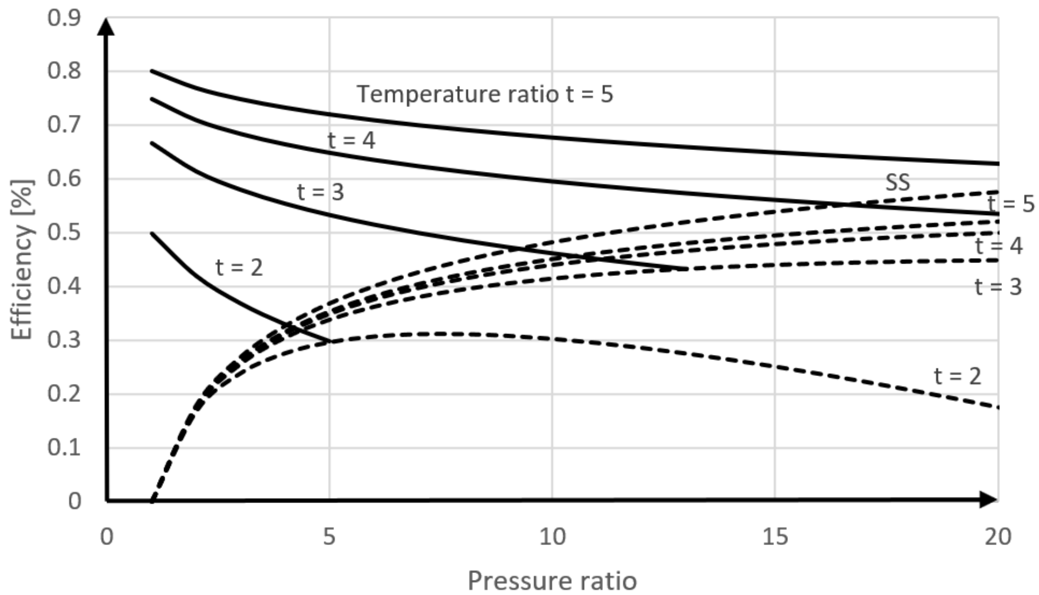


Figure 17. *Efficiency of an Intercooled and recuperated cycle*

ratio is at moderate range, about 10–20. Efficiency is increased because compressors work is reduced, and the increased heat addition due to decreased temperature from the outlet of compressors is filled by excess heat from turbines, meaning that while added heat remains constant, compressor work is reduced.

3.2.4 Reheated cycle

In reheated gas turbine cycle, expansion is split with multiple turbines, and between turbines, gas is heated again. Splitting expansion and heating gas between, brings increased turbine work, similar to decreased compressor work in intercooler due increase in vertical distance between pair of constant pressure lines as entropy increase in Ts-diagram. This is worth noting, but in this master's thesis no efficiency and specific work outputs are shown, while studied gas turbine process does not include reheat. When temperature increase is assumed to be equal to TIT of the first turbine, almost identical curves to intercooled cycle specific work and efficiency, Figure 15 and Figure 16, can be obtained. In the reheated cycle, increase in specific work output and decrease in efficiency is greater than with intercooler, because high temperature region is more sensitive to modifications compared to lower temperature range. This is due already mentioned increase in vertical space between constant pressure lines in Ts-diagram.

3.3 Thermodynamics of components

In chapters 3.1 and 3.2 ideal cycles were introduced and how each modification in the process change efficiency and specific work output. Also, simplifications to the idealized process were described. In ideal cycles it was noticed, that higher pressure ratio and maximum temperature ratio increased efficiency and specific heat output and in recuperated cycles lower pressure ration lead to higher efficiency.

Gas turbine process simulation models are most commonly using thermodynamic relations of each component. It is important to know how pressure and temperature changes in each component and how mass flow rate and air composition affect to these, to describe gas turbine operation accurately. In this chapter thermodynamic principles of each component are discussed in detail, and explained equations used to create performance model.

3.3.1 Compressor

In earlier chapters, compression process was assumed to be reversible and adiabatic. However, in real processes increasing and decreasing air speed cannot be transformed to pressure without some of the energy transforming to heat. These losses increase the entropy in the process, and it is shown in Ts-diagram in Figure 18.

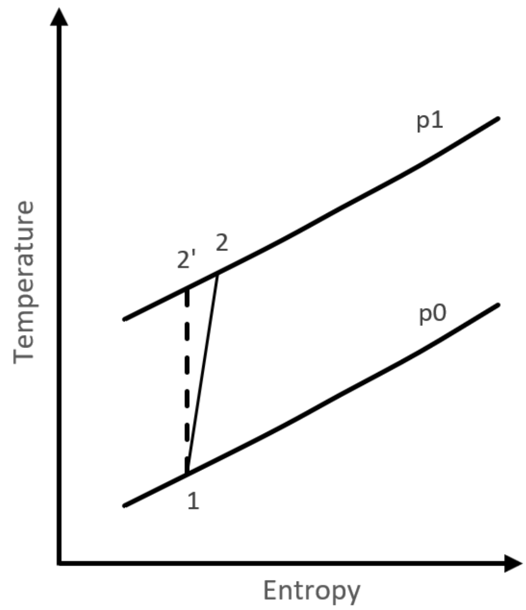


Figure 18. *Isentropic and real compression*

In Figure 18, 2' express isentropic pressure rise, and 2 real pressure rise. As it can be seen real process needs more work to gain the same pressure rise. Compressor isentropic efficiency for this can be expressed as ratio between isentropic work and real work

$$\eta_c = \frac{W\prime}{W} = \frac{T\prime_2 - T_1}{T_2 - T_1}. \tag{14}$$

Apostrophe units, T' and W', implies to isentropic value and subscripts are referring to Figure 18, 1 being ambient pressure before compressor and 2 working pressure after compressor.

Combining isentropic p-T relation, equation (5), and isentropic efficiency, equation (14), temperature equivalencies of the work transfer for a given pressure ratio is

$$T_2 - T_1 = \frac{T_1}{\eta_c}\left[\pi^{(\gamma-1)/\gamma} - 1\right]. \tag{15}$$

When air is compressed in multiple stages and each stage is composed of similar compressor or blade row, it is reasonable to assume that each successive compressor has equal isentropic efficiency. Similar to the increase in vertical distance between constant pressure lines while entropy increase, when pressure rise is kept constant, in higher pressures and vertical distance between these pressure lines increases. This causes the efficiency of the whole compression to be lower than the efficiency of individual stages. For this reason, it is convenient to use polytropic efficiency to describe the efficiency of multi-stage compression and expansion. Polytropic efficiency is defined as the isentropic efficiency of an infinitesimal stage in the process such that efficiency is constant throughout the whole process. For the whole compression, polytropic efficiency is

$$\eta_{pol} = \frac{dT\prime}{dT} = constant \tag{16}$$

where dT' is isentropic temperature rise for whole compression and dT actual temperature rise after whole compression.

With equation (16) and isentropic p-T relation, equation (5), polytropic p-T relation can be derived to

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma\eta_{pol}}, \tag{17}$$

where γ is specific heat ratio and subscripts 1 and 2 refers to inlet and outlet of the compressor respectively.

Using polytropic p-T relation, equation (15) becomes

$$T_2 - T_1 = T_1\left[\pi^{(\gamma-1)/\gamma\eta_{pol}} - 1\right]. \tag{18}$$

With equation (18), multistage compression can be calculated by using one efficiency and there is no need to know the isentropic efficiency of each individual compressor. This makes calculation easier with gas turbine processes having multiple compression stages when compression can be considered as one incidence.

Another application for polytropic efficiency is when gas turbine performance calculation is done with a range of pressure ratio, i.e. determining the optimal pressure ratio for the given process. In these kind of applications polytropic efficiency can be used as a constant, when isentropic efficiency varies while pressure ratio is changing. This is also a result of diverging pressure lines, with increasing temperatures. [33]

3.3.2 Turbine

Similar to compressors, ideal turbines do not exist, and they cannot change all enthalpy from expansion to power. This leads to an increase in entropy during expansion and decrease in work output compared to the isentropic process is shown in Figure 19. Numbers are referring to simple cycle process, 3 is gas after combustion entering in turbine and in 4 gas is released from turbine.

Isentropic efficiency can be calculated to turbines, unlike with compressors, ratio between actual work and isentropic work

$$\eta_t = \frac{W}{W\prime} = \frac{T_3 - T_4}{T_3 - T\prime_4}. \tag{19}$$

With isentropic p-T relation temperature equivalence for a turbine is

$$T_3 - T_4 = \eta_t T_3 \left[1 - \left(\frac{1}{\pi}\right)^{(\gamma-1)/\gamma}\right]. \tag{20}$$

For a turbines polytropic efficiency can be used similar to compressors and temperature equivalence using polytropic efficiency becomes

$$T_3 - T_4 = T_3 \left[1 - \left(\frac{1}{\pi}\right)^{(\gamma-1)/\gamma\eta_{pol}}\right]. \tag{21}$$

3.3.3 Heat exchanger

Both heat exchangers in gas turbine process, recuperator and intercooler, are using the same principles, transferring heat from warmer fluid to cooler fluid. In gas turbines, 3 types of heat exchangers are commonly used: counter-flow and cross-flow heat exchanger, and regenerator. In counter-flow and cross-flow heat exchangers, fluids are transferring heat through the separating wall, and the name of each type tell the direction of both fluids. In regenerator, fluids are brought cyclically to a matrix which alternately absorbs and rejects heat.

In all of these heat exchanger types, cold hot gases reject the heat at the rate of $m_t c_p(T_4 - T_2)$, where m_t is a mass flow through a turbine and subscripts are referring to Figure 7. Cold air receives this heat in a rate of $m_c c_p(T_5 - T_2)$, where m_c is mass flow through a compressor. The ratio of whose rejection and receiving can be expressed as heat exchanger effectiveness. With assuming mass flow rate and specific heat capacity being constant through the gas turbine, efficiency can be expressed as

$$effectiveness = \frac{T_5 - T_2}{T_4 - T_2}. \tag{22}$$

Effectiveness describes the efficiency of a heat exchanger in terms of temperature. Mass flow rate and specific heat capacity changes can be also assumed to be negligible in intercooler calculation, where cooling fluid can be different than air. This time changes in mass flow rate and specific heat capacity can be included in $effectiveness$. [33]

Using heat exchanger in the process also introduces pressure losses, while gases flow through narrow passages. Depending on the heat exchanger and simulation program, losses can be taken into account as a percentage of incoming pressure or constant pressure losses.

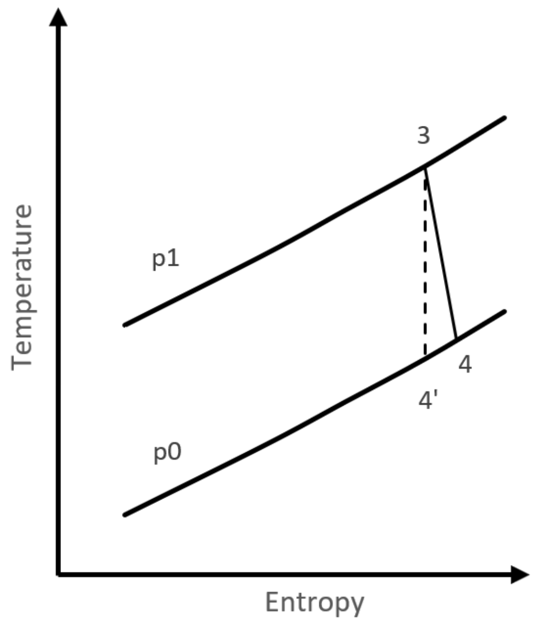


Figure 19. *Isentropic and real expansion*

3.3.4 Combustor

Combustor chamber mixes fuel with air and ignites it increasing enthalpy on gas. Temperature rise in a combustion chamber is calculated with net calorific value Q_{net} [kJ/kg] and mass flow rate of fuel m_f [kg/s]

$$m_f Q_{net} = mc_p(T_3 - T_2), \quad (23)$$

where subscripts are referring to simple cycle, Figure 1. Net calorific value can be replaced with a lower heat value of the fuel. In this time, also retain accuracy, burning efficiency of fuel must be considered. [33]

Similar to hear exchangers, combustor causes some pressure losses to process, and these losses can be considered as a percentage of total pressure or constant pressure losses.

4. OPERATION AND COMPONENT CHARACTERISTICS

Before models can be discussed, first some operational and component behavior must be explained. Main components which affect significantly to the performance of gas turbine are compressors and turbines. The behavior of these components can be described with component characteristics, which describe component behavior in varying conditions, creating a map of component operation. Linking components together is called component matching. Component matching sets restrictions on gas turbine component operation points, where components mounted on the same shaft set same rotational speeds to every component as well as restrictions in mass flow rate and pressure ratio. This limit operational range of each component significantly and creates an operational running line for the gas turbine. Lastly, operation of a gas turbine is discussed and explained how it affects to the simulation of gas turbines and inputs and output of models as well as operation in partial loads.

4.1 Component characteristics

Performance of gas turbine is mostly affected by its driven compressor and turbine. Performance of compressors and turbines can be express with 2 set of graphs: plotting mass flow rate against temperature ratio and pressure ratio, values represented with constant speed curves. These graphs are called component characteristics or maps. A common way to represent temperature ratio in compressor characteristics is to derive efficiency from temperature and pressure ratio from in chapter 3 showed equation (15). These efficiency lines can be plotted to own graph along the pressure ratio graph or superimpose to pressure ratio graph as contour lines. Compressor characteristics, also referred to as compressor map, is showed in Figure 20. This map is generated from the developed MATLAB performance model and it presents the investigated Aurelia® A400 gas turbine LP compressor. All values are hidden in the figure for confidential reasons. In Figure 20, mass flow rate is at x-axis, pressure ratio is at y-axis and 6 constant speed lines N are shown as curves. The lowest operation speed of the compressor is showed with dark blue line bottom of the figure and each higher line is representing higher rotational speed. Efficiency lines are superimposed to figure by expressing them as dotted contour lines and in addition surge line is shown in red dashed line. All values in the figure are hidden for confidential reasons. With this type of map, operation of

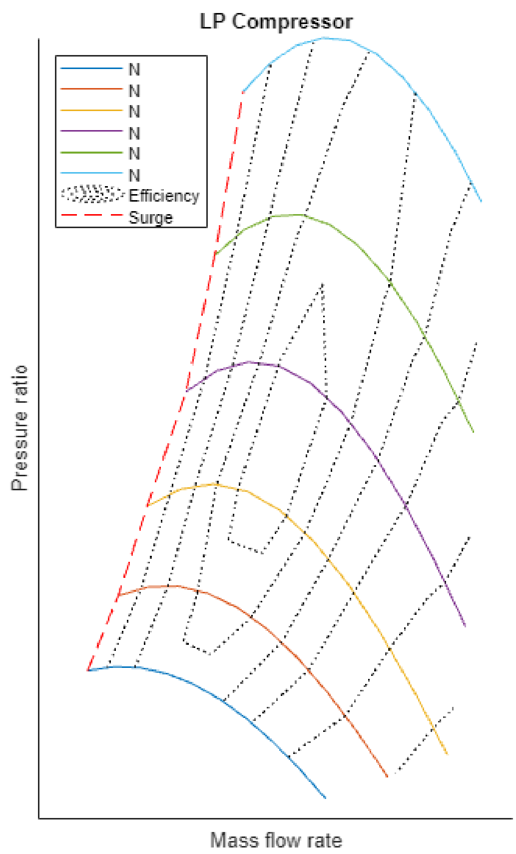


Figure 20. *LP compressor characteristics*

compressor can be fully defined with two parameters. Usually rotational speed and mass flow determines the pressure rise and efficiency of the compressor.

In component characteristics, plain values for mass flow rate, pressure ratio and rotational speeds are rarely used, while inlet conditions, namely temperature and pressure, change the form of the compressor characteristics. Due to that phenomenon, mass flow values and rotational speeds are represented as reference values, where used inlet pressure and temperature are compared to reference values which were used to create the map. This way one component map can be used with all variations of inlet conditions, more of this is discussed in chapter 4.1.2.

Characteristics of turbines can also be described with two set of curves. In Figure 21 set of common characteristics are shown, left side pressure ratio correlations with mass flow rate and on right side pressure ratio correlations with efficiency. Rotational speed is expressed in non-dimensional form. As can be seen, turbines are not as sensitive to rotational speed than compressors and usual way to describe turbine mass flow rate in modeling to use only one rotational speed curve to express turbine maps in their whole operational range [6]. In Figure 21 this simplification is showed on a blue dashed line.

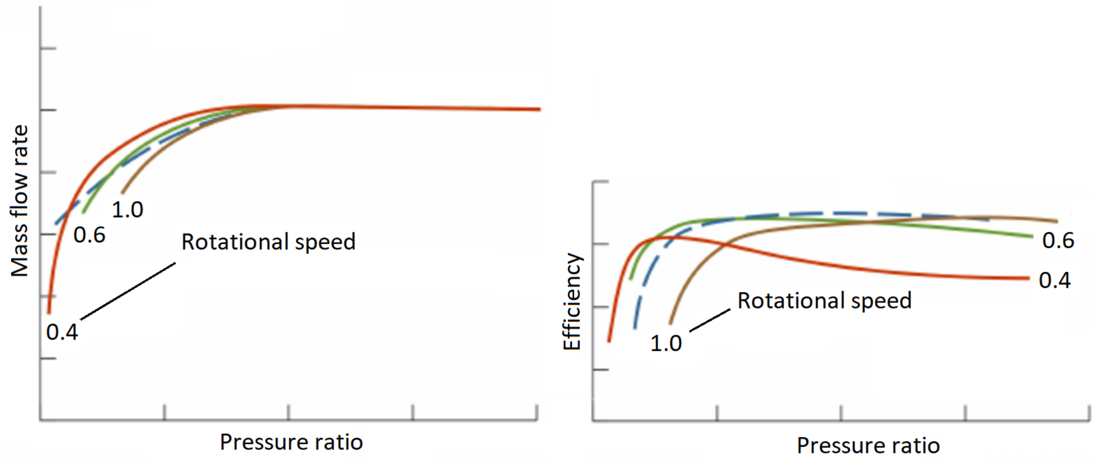


Figure 21. *The turbine characteristics, modified from [37]*

While determining the efficiency, the effect of rotational speed is more prominent and it needs to be used with pressure ratio to accurately predict efficiency. The Shape of turbine map expressing mass flow rate can be explained by thinking the turbine as a valve in a gas turbine. When a pressure difference is existing across the turbine, mass flow is commenced from higher pressure to lower and turbine characteristics determine how much air can pass the turbine. When pressure ratio increases, as increases mass flow rate what is going through the turbine. At certain pressure ratio mass flow rate caps due speed of gas to approaching the speed of sound. This is called choke of the turbine and no increase in mass flow rate can be achieved in the turbine, even if pressure ratio is increased. With compressor, choke is more complex and it is discussed with a surge in the next chapter.

4.1.1 Surge and choke

The shape of constant speed curves in a compressor map can be explained by the character of pressure. In steadily flowing gas total pressure consist of static pressure, which is the pressure of the moving gas and it is equal in all directions, and dynamic pressure, which denotes the velocity of the moving gas:

$$p_0 = p + \frac{\rho V^2}{2}, \tag{24}$$

where p_0 is total pressure, p is static pressure and term $\rho V^2/2$ denotes to dynamic pressure. Another way to describe the effect of kinetic energy on pressure is stagnation pressure [30]. The difference between stagnation pressure and total pressure is that total pressure is reversible and adiabatic, where stagnation pressure is not reversible or adiabatic. Difference between total and static pressure is about 11 % [33]. Physically total pressure is noticed when a gas stream is brought to rest adiabatically and without work transfer, the pressure rise is equal to dynamic pressure.

Now with knowledge about how pressure is consisting, pressure rise in a compressor can be explained in detail. In centrifugal compressor air is sucked in at impeller eye. By quick rotation of impeller, air is given a centrifugal acceleration before leaving from impeller at trailing edge. At impeller, stagnation pressure of air is increased in the form of static and dynamic pressure. After impeller, air is lead into diverging passages of the diffuser, where speed is decelerated and most of the dynamic pressure is transformed to static pressure. According to Saravanamuttoo *et al*. [33], normal practice is to design the compressor so that half the pressure rise occurs in the impeller and half in the diffuser. After diffuser, the speed of air should be about the same than air entering the compressor. Friction in the diffuser causes some losses in stagnation pressure created in the impeller.

Saravanamuttoo *et al*. [33] described compressor characteristics with theoretical compressor, rotated in a constant speed and having a valve downstream of the compressor. While the compressor is running at constant speed and the valve is slowly being opened, Figure 22 shows how constant speed curve would form in a compressor map. In real compressors however, both ends of curves would be unattainable due surge and stall. When the valve is closed and mass flow being zero, and compressor running constant speed, pressure in the gas would be equal to the amount of pressure head produced by the rotation of impeller on the air trapped between the vanes, point A. When the valve is slowly opened, and mass flow commences, static pressure rise starts to take place also in the diffuser. In the vertex B, efficiency of the compressor reaches maximum, leading maximum pressure ratio and further Increase in mass flow leads to greater losses and lower pressure ratio. Maximum pressure rise occurs, because in a point B, optimal pressure rise happens at impeller and diffuser, increasing mass low rate all kinetic energy cannot be transformed to pressure at diffuser. When mass flow rate of the air greatly exceeds the design mass flow rate, mass flow rate in point B, vane angles of diffuser widely differentiate angles of coming air. This leads to breakaway of the air and rapid fall of efficiency loss and constant speed curve start to decline rapidly. In this case, pressure ratio falls eventually to inlet pressure, point C, where the valve is fully opened, and all the power is absorbed in overcoming internal frictional resistances.

As can be seen from the actual compressor map, Figure 20, most of the curve from A to B cannot be reached due to phenomena called surge. Surging is associated with a

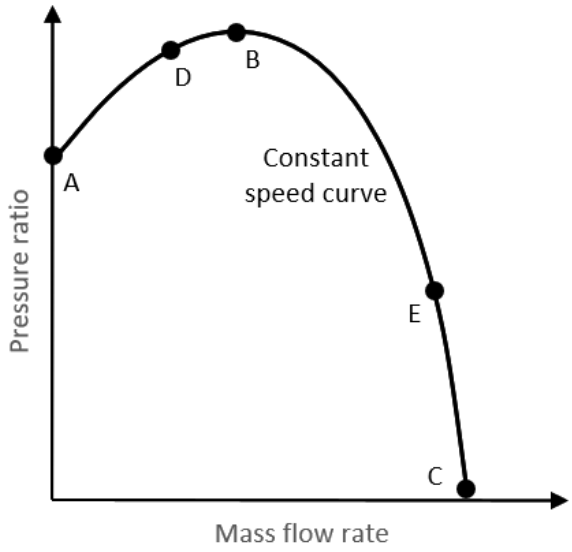


Figure 22. *Theoretical compressor characteristics, modified from [33]*

sudden drop in delivery pressure and rapid air flow pulsation throughout the wholeengine. It is estimated that it happens with the change of positioning in compressor characteristics. When the compressor is operating in a positive slope area, near point D, a decrease in mass flow means a decrease in delivery pressure. Pressure may not drop fast enough in downstream components of the compressor, so pocket with lower pressure is created between the compressor and the following components. This causes the flow to reverse its direction towards pressure gradient and creating an aerodynamic pulse, along with reverse bending on nearly all gas turbine components [38]. After this occurs, pressure ratio drops rapidly, and the compressor is able to create pressure again and repeat the cycle. The surge cycle would continue at a frequency of five to ten times per second, eventually leading to engine damage [39].

Surge line in the compressor map is not immediately on the left side of vertex point B, because the pressure in a downstream component may fall quicker than compressor delivery pressure, but eventually when the mass flow rate is reduced surging happens. Surge line is showed in Figure 20 and it is usually straight curve with all rotational speeds. Location of the surge line depends on mass flow capacity of components downstream the compressor and manufacturer can only give estimations when the surge is going to happen. Operation near this curve should be avoided in every situation, because of the negative impact surging can create to the gas turbine.

When operating on the right side of point B in Figure 22, mass flow increases and pressure decreases. Keeping in mind the continuity equation

$$\mathrm{m} = \rho \mathrm{AV}, \tag{25}$$

where m denotes mass flow rate, ρ density [kg/m^3], A flow area [m^2] and V speed of fluid [m/s], while mass flow rate increases and pressure decreases, meaning density decreases, in fixed size compressor velocity of fluid needs to increase. At constant rotational speed, this means that incidence angle, where air leaves impeller blade changes and while this angle varies from design point angle, efficiency drop increases. At the point where the velocity of fluid approaches the speed of sound, mass flow cannot be increased and pressure ratio and efficiency drop rapidly. This is called choke and it is limiting phenomena of compressor performance another end of constant speed curve.

4.1.2 Non-dimensional analysis

Compressor's and turbine's outlet parameters are highly dependent on inlet temperature, pressure and properties of the working fluid. This means that component characteristics is valid only to one inlet conditions and while one or all of inlet parameters change, the component map also need to change to keep accuracy. Any attempt to describe the full working performance of components with a full range of inlet conditions would require an excess number of component maps. This would bring problems with calculating maps as well as the presentation of results. Component maps can be universalized, to one map including every inlet condition using dimensional analysis, by replacing variables with non-dimensional parameters.

Non-dimensional analysis starts with listing variables, which influence and depend upon the behavior of the compression. Then the function is made from these variables and equated it to zero. Function and depending variables are

$$Function(D, N, m, p_1, p_2, RT_1, RT_2) = 0, \tag{26}$$

where D is impeller diameter [m], N is rotational speed [RPM], m mass flow rate, p_1 and p_2 are inlet and outlet pressures and RT_1 and RT_2 is temperature associated with gas constant [J/kgK] in inlet and outlet. These base variables consist only with 3 units: mass [kg], length [m] and time [t]. Using dimensional analysis, as Buckingham PI theorem, parameters can be reduced to smaller dimensionless parameter groups of $7 - 3 = 4$, 7 variables and 3 units, non-dimensional groups. Variables of equation (26) can be formed non-dimensional parameter groups in a multiple way but most suitable groups to describe the performance of compressor are pressure ratio, temperature ratio, non-dimensional mass flow rate and non-dimensional rotational speed

$$\frac{p_2}{p_1}, \frac{T_2}{T_1}, \frac{m\sqrt{RT_1}}{D^2 p_1}, \frac{ND}{\sqrt{RT_1}}. \tag{27}$$

When compressor size is fixed, and we are only concerned performance using a specific working fluid, R and D may be omitted from the group so that the function of non-dimensional parameter groups is

$$Function\left(\frac{p_2}{p_1}, \frac{T_2}{T_1}, \frac{m\sqrt{T_1}}{p_1}, \frac{N}{\sqrt{T_1}}\right) = 0. \tag{28}$$

Parameters $\left(m\sqrt{T_1}\right)/p_1$ and $N/\sqrt{T_1}$ are not truly dimensionless, but they are usually termed as “non-dimensional” mass flow rate and rotational speed with fixed size compressor and fixed working fluid.

Replacing “dimensional” mass flow rate and rotational speed with “non-dimensional” values in compressor map, compressor map becomes universal to be valid for any inlet condition. Performance of the compressor can be read from non-dimensional map with changing actual compressor mass flow rate and rotational speed to non-dimensional form with inlet pressure and temperature.

Another way to create compressor characteristics is to use reference values of mass flow rate and rotational speed. Reference values are inlet conditions which were used to create the map. Reference mass flow rate and rotational speed are

$$\frac{m\sqrt{T_1/T_{ref}}}{p_1/p_{ref}}, \frac{N}{\sqrt{T_1/T_{ref}}} \tag{29}$$

where T_{ref} and p_{ref} are reference values and T_1 and p_1 are current application initial values. The advantage of this method can be seen when comparing characteristics using non-dimensional values and reference values. Characteristics using reference values display numbers on the axis in recognizable quantities, similar to actual values. [33]

4.2 Component matching

Now when looking at component maps, both compressor and turbine has a wide operating range with mass flow rate, pressure ratio and rotational speed, Figure 20 and Figure 21. When the gas turbine is built and components are linked together, the operating range of each component is restricted substantially. With linked component mass flow rate must be equal in all components in the gas turbine. Another matching parameter is rotational speed of the components mounted to the same shaft. The rest of the matching parameters are pressure ratio, where pressure increase created from compressor needs to be compensated by turbines, other component losses and back pressure, also power matching in gas generation turbine must equal power used with the compressor.

When keeping in mind how compressor act when a valve is brought downstream of the compressor, the function of a gas turbine can be explained. When gas turbine is build and components are linked together, the turbine can be considered as a valve. This means that the turbine determines how much mass flow can be flow through the process by given pressure ratio determined by turbine characteristics. When considering constant rotational speed, this in this case mass flow is determined, thus compressor pressure rise is known. This pressure rise must be utilized in the form of pressure losses of combustor, pipes and outlet as well as expansion in the turbine. Pressure losses can be treated as constants throughout the gas turbine operating range, so the main parameter for utilizing pressure ratio is the turbine. If pressure ratio is not equaling mass flow rate in the turbine map, mass flow rate will adjust itself to correspond current pressure ratio. Changed mass flow rate leads again to new pressure ratio from the compressor and this loop is continued until components are agreeing with mass flow rate and pressure ratio.

While designing gas turbine, it is important that components will work together in high efficiency and in robust working area at component maps at design point as well as the off-design point. In poorly designed gas turbine, turbine can either allow too much air flow to the given compressor, resulting compressor choke and leading to large loss in efficiency or in the case where turbine restrict air flow causing compressor surge and leading possibly to component breakdown.

Using more than one compressor or turbine in the process resulting similar matching procedure with increased restrictions. With multiple compressors, mass flow rate is still determined same throughout the compressors and each compressor increases pressure determined by their characteristics. With turbines, mass flow might be determined only with other turbine or with other components, but both turbines utilize mass flow specified by their characteristics with used mass flow rate.

4.3 Operating gas turbine

It is essential to understand how the gas turbine is operated to cope with models and understand how they are created, and which parameters are important to the user of the model. As explained in the previous chapter, gas turbines are highly complex engines where components determine the operation parameters. Outline in component matching is that the compressor increases pressure by given mass flow rate and rotational speed, and turbine determines the mass flow rate by pressure ratio. The operator of gas turbine cannot directly affect to mass flow and pressure ratio since they are greatly the dependent on component design and their matching compatibility. Pressure ratio and mass flow rate can be changed by modifying rotational speed and turbine inlet temperature.

Gas turbines are essentially designed in a way, that the highest efficiency is reached at design operational point. However, small gas turbines are commonly used in applications which require variable output. Example of this type of application is, for example CHP, where industrial processes requiring on-site electricity or heat, demand may vary tremendously [40]. Jaatinen-Värri *et al*. [40] compared gas turbine control schemes in part load conditions and how those affect the efficiency. These control schemes were 3 types, which are used with small gas turbines: Fixed speed machines, where the operation happens with changing TIT, Variable speed machines, where operation is done by changing rotational speeds of spools and TIT and two-spool variable speed machine, same which was used to create performance simulation tool. In this machine changing operational point is done also with rotational speed and TIT.

Fixed speed machines are controlled by decreasing TIT, and as described earlier, one-spool small gas turbines are usually recuperated, and in recuperated cycle efficiency, Figure 13, turbine inlet temperature is heavily affecting to gas turbine efficiency. During part loads in Fixed speed machines, efficiency is then reduced drastically. In variable speed machines, TIT can be kept the same during part load operation and allowing smaller efficiency decay than fixed speed machines. But in variable speed machines, decreasing rotational speed decreases pressure ratio, which leads to higher turbine exit temperature. In some point, turbine exit temperature reach is maximum allowable, and further need in power output requires lowering TIT. With two-spool variable speed machine spool speed can be modified independently and higher efficiency can reach further in part-loads.

In their study Jaatinen-Värri *et al*. calculated these 3 types of control method efficiency in part-loads and results are shown in Figure 23. In the fixed speed gas turbine, a recuperated process was used, and the efficiency of this process is shown in Figure 23 light dash line. Efficiency starts to decrease as soon operation point is in part-load conditions, and the decrease is almost linear. As variable speed machine, single spool gas turbine was used, and as can be seen, efficiency remains almost constant until 65 % loads. After this point, the turbine exit temperature starts to limit operation and TIT must be decreased. In Figure 23 variable speed is shown in solid line. In the figure, a two-spool variable speed engine is studied Aurelia® A400 turbine, and as can be seen,

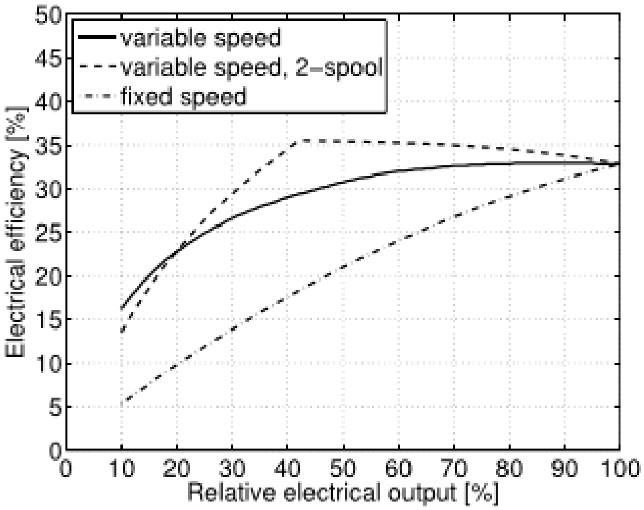


Figure 23. *Electrical efficiency at part-load*

efficiency can be hold on longer, while two variable speed spool allows greater operational range for nominal efficiency. After the point, in figure 40 %, efficiency starts to drop rapidly, while the turbine exit temperature reaches its maximum. Efficiencies are normalized in the figure so that comparison is made easier.

5. SIMULATION METHODS

Gas turbines are highly nonlinear and complex machines, and simulation of these machines are as well highly complex and multiple models are needed over the course of engine lifetime. Designing gas turbine starts with rough models which are determining the main parameters of the engine, to more detailed component modeling all the way to performance modeling using complex time variate thermodynamic models and Computational Fluid Dynamics (CFD) models. The rough models are used mainly to iterate as many iteration rounds as possible, to find simplified main parameters like mass flow rate through the gas turbine, pressure ratio and temperature ratio to fulfill the power requirements. [41]

After the component dimensions are chosen, models simulating gas turbines, are performance models and they describe detailed behavior of the engine. Now the problem in the simulation is to determine how well predefined components operate together in changing ambient conditions and operational parameters. These models are a mathematical description of gas turbine operation, normally using component thermodynamic relations to resolve operation condition. In the literature many kinds of simulation tools are developed with many simplifications and approaches to describe gas turbine performance, these models are discussed later in this chapter.

These models are used to predict the performance of gas turbine in different ambient conditions and power requirements as well as gas turbine operation monitoring and fault detection. Gas Path Analysis (GPA) is a common way for gas turbine performance monitoring and fault detection. GPA measures engine operation with health parameters and any degrading in health parameters can be identified as a faulty component, leading to maintenance procedure. Tsoutsanis *et al*. [14] used GPA in their CCPP to proceed from prescheduled maintenance to condition-based maintenance in their gas turbine. Zhu and Saravanamuuttoo [6] had another way to describe a deteriorated engine. They created a performance model from the investigated gas turbine and used the model to create nominal point data as well as data from 5 different deteriorated engine case. With these cases, they created fault matrix, which could be used to find out which component is deteriorated.

Transient models are used to simulate and optimize gas turbine start-ups and load increases. Those are time variant and in addition to thermodynamic analysis, they need to take account of component heat capacity and mass flow variation through the process.

Several studies are focused start-ups investigation in combined cycle gas turbines [11] [12], where the flue gas from gas turbine is used in heat recovery steam generators. In these applications, start-up creates high stresses in steam turbine and evaluation of start-up schedule is necessary. J. Bausa [13] investigated startup methods in terms of fuel economy and stress evaluating as well as in load increases. This type of simulation brings huge savings in the operational life of gas turbine, while start-ups and shut-downs are occurring quite frequently. Start-up schedule has also been investigated to be able to execute such a way, that no surge or stall are happening in the compressor to cause abnormal shutdowns [42].

In this chapter only steady-state performance models are discussed and firstly component characteristics in modeling point of view is discussed, secondly iteration method to reach equilibrium in the models are explained and lastly different methods for steady state performance modeling techniques from literature is described and related simplifications in each model is discussed and modification to fidelity with each practice.

5.1 Component characteristics in simulation

Accuracy of the component maps means how well they can describe actual components of studied gas turbine. While compressor and turbine are determining most of the gas turbine operation and process parameters, accuracy of mathematical gas turbine models is mostly affected by the accuracy of its component maps. Difficulties in the simulation are usually relating to difficulties having these accurate component maps. Component maps are property of compressor and turbine manufacturers and these are usually not accessible for gas turbine users. Also, early stages of gas turbine designing when component parameters are not necessarily chosen, component maps are not available. This causes gas turbine manufacturers and operators to estimate component operation on their whole operation range. There are different methods to estimate component maps and some of them are discussed in this section.

5.1.1 Estimation methods

Using component maps, which are calculated and validated by testing is the most accurate way to predict the operation of the gas turbine in its full operational range and having accurate maps is crucial for precise performance prediction. Haglund and Elmegaard [10] made a comparison between models using component maps and models using constants to define the operation of components. They created two cases for both methodologies: a simple model, where gas turbine is assumed to consist of a compressor, combustor, compressor turbine and power turbine, and a complex model, where also inlet, exhaust pressure losses, cooling and bleed flows are considered in the modeling. Component maps for calculation were based on GasTurb simulation software data since no manufacturer maps were available. GasTurb has a library of public domain component maps and Haglund and Elmegaard were using maps having similar characteristics to studied components and scaling them to corresponding engine. Constants defining components were flow capacity, pressure ratio and inlet temperature relating to turbine operation. During off-design performance calculation, load ranging from 20 % to 100 %, results indicated that complex models give slightly better, almost negligible, agreement to manufacturers data in both cases. On the contrary, comparing complex model using component maps and constants showed that constants cannot be predicting performance in full range of operation, especially results at loads under 60 % were highly imprecise.

Common methods used to predict component maps are scaling and stacking and modifications of these methods. More precise methods to estimate component maps are CFD-calculation or neural network using data from testing. Scaling method, the one which was used also in Haglund and Elmegaard's study, is used with existing compressor map and scaling map to correspond actual component. There are freely available component maps, like GasTurb libraries, for multiple components, which can be used to estimate studied components. Haglund and Elmegaard used linear scaling, where both compressor and turbine maps were chosen from components having similar characteristics to estimated components. Scaling was done by linearly scale pressure ratio from whole operating are to match design point data. Zhu and Saravanamuuttoo [6] estimated compressor characteristics by generalized compressor maps. Those maps were created by deriving several compressor maps by relative scaling to meet studied compressor characteristics. Compressor maps were functioning very well after slightly modifying efficiency contours and scaling speed lines up to meet the design point data. Scaling works well in applications, where almost similar component characteristics can be found as the studied component. Also, cases where the component is scaled geometrically, i.e. all dimensions are factored by the same value, works well and only overall performance of component is represented. However, the problem arises with multistage components, which would need more detailed lookup concerning variable inlet guide vanes and where pressure ratio requires scaling. These kinds of processes have typically unique shape in every stage and scaling requires more detailed overview. These types of applications are not successful with linear or generalizing scaling, but Kurzke and Riegler [43] introduced a method for scaling to overcome these problems. Their scaling method took concern also change the topology with pressure ratio, and their method could also more accurately describe more complex components and their applications.

Another often used method, especially with axial multi-stage components, for estimating component maps is stage stacking method. Muir *et al*. [8] described stage stacking method where all stages of the component are considered separately. Characteristics of each stage are calculated with known compressor variables, as tangential blade speed, axial velocity and temperature and pressure ratios, using a set of non-dimensional parameters. These parameters are compared to reference non-dimensional parameters, which are referring to the most efficient running line and each stage pressure rise and temperature rise is obtained. Data from each stage is then put together, “stacked”, and created full component characteristics. Song *et al*. [9] introduced a more advanced stage stacking method, which could accurately estimate the compressor map also for a compressor having variable inlet guide vanes and variable stator vanes. Their method used same stage characteristic parameters than normal stage stacking method, but their method used continuity of mass, energy and momentum to create simultaneous solving for all interstage parameters. Casey and Robinson [44] used stage stacking method to create centrifugal compressor map. In their method it is possible to create fast compressor map, using non-dimensional parameters, flow coefficient, work coefficient, blade tip Mach speed and efficiency, and created maps could be calibrated to match a wide variety of stage types.

Other method to obtain component maps with higher accuracy, is CFD calculations, which was used by Song *et al*. [45] and to further assure accuracy, they calibrated characteristics with scaling method to correspond test data. When actual accurate operational data is available, a neural network model can be used to obtain component maps. Neural network models use test data to find out how components are behaving in each condition and creating characteristics based on these data point. This requires lots of testing and data, but accurate characteristics can be created [15]. Also, there are methods which are concentrating on a rapid creation of component maps [46] or creating maps with limited available data [47].

5.1.2 Reading methods

When accurate component maps are in use, they also must be expressed in a way, that computer can read it. Usually, component characteristics are stored in a file with datapoints, describing pressure ratio, mass flow rate and efficiency in multiple points for every constant rotational speed. Getting data from component maps cannot be done with simple interpolation method since compressor maps have 3 dependent values, so 2 values determine the output of third This leads to need for complex interpolation method, when requested data point is between known data point in component maps. For interpolation, one method to get wanted value is to use table form for data points of the maps. With table form, interpolation is done first with two values and after the right value is found, it is interpolated with the third value. Another commonly used method is to use built-in function in the programming language, but this is only possible in more advanced simulation platforms. Using built-in interpolation function to interpolate values from 3D surface might be accurate during are of data points but can be extremely unstable during extrapolation, and models which are operating during off-design performance and near choke or surge, modeling might become unstable with built-in functions.

A problem can arise when interpolating parameter which has multiple values, for given entry parameter. With low-pressure compressors, speed lines can have multiple mass flow rate values with one pressure ratio and rotational speed, as can be seen from Figure 20, where higher speed lines go horizontal and for both sides, one pressure ratio equals to multiple mass flow rates. Another problem comes with high-pressure compressors. Then speed lines will go vertically near choke and this time the opposite is true: with one mass flow rate and rotational speed, multiple pressure ratios can be obtained. In gas turbine models both, mass flow rate and pressure ratio, are used to interpolate steady state operating point in whole engine models and this brings problems when one input multiple values are received. This can be overcome to introduce so called beta-lines. Beta lines are either straight lines or parabolic, and collinear to surge line. These lines are numbered and now with beta-line and rotational speed, both mass flow rate and pressure ratio are unambiguous.

5.2 Iterating steady state operation point

During off design calculation component parameters are not predefined and efficiency, pressure ratio and mass flow rate of each component varies when conditions are changed. This results that operating point cannot be solved directly with given input parameters, but they need to be calculated iteratively. The number of iteration variables depend upon the complexity of gas turbine: Single shaft gas turbine has 2 iteration variables, while more complex gas turbine process can have up to 15 iteration variables [48].

Iteration can be done with nested loops or matrix iteration, whereas nested loops become inefficient and slow after number of an iteration variables grow more than 2. Nested loop iteration means that each variable is iterated separately one at a time in a loop until the error of output is in feasible within the margin. After that second variable is changed if needed, and with second variable first iteration is run again, until converged. Matrix iteration is more commonly used with gas turbine simulation because it allows multiple iteration variables to be iterated at the same time. Most common matrix iteration is multivariate Newton-Raphson (N-R) method. There was a lot of experience and advice in literature, to advocate N-R method usage in the gas turbine calculations.

5.2.1 Newton-Raphson method

Newton-Raphson is iteration method, where wanted values are sought with a linear approximation of the function. The equation for N-R method is

$$x_{n+1} = x_n - \frac{f(x_n)}{f'(x_n)}. \tag{30}$$

Geometrically Newton's method seeks the root of the function f by drawing tangent of the arc in the position of x_n and suggest its root as a next candidate for the solution of function f. Loop is continued until the accuracy of suggested root candidate is in the expected range. Meaning that when tangent approximation, the right term of equation (30), becomes as close to zero as requested by the wanted margin and change of the result x_n is negligible.

Newton Raphson is a very effective method, when initial guess is close enough to the actual root and convergence is done quickly. Tolerance of initial guess depends on the complexity of function f near the root. [49]

Newton-Raphson method can be used also with multivariable problems. When solving a group of functions f, and size of variable group equal to the size of function group, derivates can be expressed in form of partial derivate matrix, Jacobian matrix. Implementing vectors and matrices to equation (30) Newton-Raphson becomes

$$x_{n+1} = x_n - J_n^{-} f(x_n), \tag{31}$$

where x_n is a solution vector, J_n^{-} is inverse of Jacobian matrix, $f(x_n)$ is error vector and subscript n denotes to iteration round. Size of error vector equals the number of iteration variables, i.e. solution vector. [50]

5.3 Gas turbine models

The most common way to create a gas turbine model is to use matrix iteration with a set of iteration variables. Let's call this as a state variable method [51] state variables being iteration variables, which are used to define the operation of the whole gas turbine. State variables are gas turbine parameters, which can define the whole operation of the gas turbine and they are used to calculate other parameters involving gas turbine. In state variable method, those parameters are given to gas turbine operation calculation as an input and gas turbine operation is calculated. State variable accuracy is tested with error function, one function correlating each state variable, between the state variable and calculated value using other parameters. Using error functions, state variables are adjusted iteratively using for example Newton-Raphson algorithm to find steady state point, where each state variable corresponds to the calculated value from other sources. The number of state variables depends on the complexity of gas turbine. Singh *et al*. [51] created a model for simple cycle jet engine by using state variable method. Their gas turbine included an inlet, compressor, combustion chamber, turbine and exhaust nozzle. In this process, 3 state variables were needed to fully describe the operation of all components. In their model, state variables were chosen as compressor exit pressure, turbine exit pressure and shaft speed and error function which were determined to be mitigated were mass flow rate error between compressor and turbine and mass flow rate error between turbine and nozzle and work compatibility error between compressor and turbine.

Basic schedule in performance modeling is following. As N-R iteration requires initial guesses, the calculation is beginning with an initial guess of state variables as input. While N-R is strict by the accuracy of initial condition in a complex calculation, the order of magnitude of steady-state operation parameters should be known. These state variables and initial condition parameters are given to calculation and used to calculate first round. When there is an error between 2 values, calculated using state variables, the error function can be calculated. Length of the error function must equal to the length of state variables. Generated error is in form of general error formula and in the model created by Singh *et al*. mass flow rate error between compressor and turbine can be expressed as

$$e = \frac{\mathrm{m}_c - m_t}{m_c}, \tag{32}$$

where m_c is mass flow rate got from the compressor map, using state variable rotational speed and compressor pressure ratio as input and m_t is mass flow rate calculated from the turbine map using turbine pressure ratio. General error formula takes account the magnitude of error by dividing error $\mathrm{m}_c - m_t$ by their respectful quantity m_c [49]. Other errors used by Singh *et al*. are mass flow error between turbine and nozzle and work compatibility between turbine and compressor. Errors are now iterated to zero with multivariate Newton-Raphson, equation (31), until iteration converges, and steady-state operational point is obtained.

One aspect of state variable method is that calculation is broken down into series of discrete components and calculation is done by starting at the inlet, using input values and state variables to calculate each component and using parameters of the previous component to calculate next. In model created by Singh *et al.*, mass flow rate and compressor efficiency were calculated using state variable rotational speed and compressor outlet pressure and inlet pressure. Component maps were approximation maps. With mass flow rate and efficiency, outlet temperature and work required by the compressor are calculated. Using outlet pressure, the combustor pressure loss is used to obtain turbine inlet pressure and compressor outlet temperature and mass flow rate is used to calculate turbine inlet temperature. State variable turbine outlet pressure and combustor outlet pressure are used to calculate mass flow rate through compressor and temperature and work output are calculated with pressure ratio and turbine inlet temperature. Nozzle mass flow rate is obtained with pressure ratio from nozzle map, close to turbine map, and the temperature is obtained with pressure ratio is obtained from pressure ratio and turbine outlet temperature. Now mass flow rates between compressor and turbine and turbine and nozzle are calculated as an error for iteration as well as power balance between compressor and turbine.

Nasa has created The Toolbox for Modeling and Analysis of Thermodynamic Systems (T-MATS) for MATLAB Simulink and the source code of the model is publicly available [52]. Toolbox can be used as a gas turbine modeling and as an example, they had created model simple cycle jet-engine, similar to engine modeled by Singh *et al.* T-MATS model is using state variable method, and in that model compressor maps use beta-lines. State variables are mass flow rate through the engine, rotational speed, beta-line number and pressure ratio of the turbine. Errors are used as a mass flow rate error in nozzle, compressor and turbine compared to state variable mass flow rate and torque balance between compressor and turbine.

State variable method has been in use with the model created by Song *et al.* [45]. Their model concerned three-spool gas turbine with turbine cooling. In this model, blade cooling is at the main point, while it is not studied much in literature. Component characteristics in this model have been obtained from CFD calculations and calibrated with the test data. Model is created using FORTRAN programing language. State variables which were used to calculate model, are rotational speed of the low-pressure shaft, pressure ratios of compressors and turbines. Calculated errors are mass flow rates and power balances. Iteration is made with multivariate Newton-Raphson. Jacobian being calculated by linearizing the system in the small neighborhood of the approximate solution at each iteration step.

Haglund and Elmegaard [10] compared two methodologies to approach simulation with design point performance analysis and part load performance analysis. Models were made using Dynamic Network Analysis (DNA), which is a simulation program used for energy systems analysis created by Technical University of Denmark. It is normal practice to use so called beta lines with compressor maps. With concerning the off-design performance, ranging from 20–100 %. With maps agreement with the data was very good for both models, except in the very lightest loads. Compared to simple and complex models, deviation from results were negligible, the complex model having slightly better agreement with manufacturers data. During turbine constant calculation, mass flow and pressure ratio can be calculated accurately, but exhaust temperature and thermal efficiency are really misleading after 60 % load, in down side. This could be due to assumption that isentropic efficiency is assumed to remain constant in the operational range.

Al-Hamdan *et al*. [7] introduced a new component matching method, by superimposing both turbine and compressor to the same coordinates. This requires transforming both characteristics abscissas to identical. It is done by introducing a new dimensionless matching parameter. Gas turbine used in the calculation was a simple gas turbine with a centrifugal compressor and a radial turbine, and the same shaft linked generator. Compressor calculation was made using beta lines, explained earlier and compressor maps were given to calculation program in table form, where given beta lines and shaft speed determines the pressure ratio and mass flow rate of each operation point. Iteration of steady-state operation point is done by loop iteration, iteration variables being combustor temperature and beta lines, of the compressor.

Another matching procedure was introduced by Zhu *et al*. [6]. This method was so called Hot end method, where calculation was started form turbine end rather than determining compressor operation point. Component maps were estimated with generalized component maps from multiple compressors similar to studied one. Generalized component map was scaled to correspond the data, provided by a sales brochure from the manufacturer, at the design point. Generalized compressor maps were noticed to function very well in the engine model. The engine Zhu *et al*. simulated was twin-spool gas turbine with power turbine. In Hot end method, the calculation is started with assuming specified power turbine pressure ratio and with this assumption, operation conditions of power turbine, LP turbine and HP turbine is determined by turbine characteristics and mass flow continuity. Rest of operation parameters will be specified by work compatibility in spools, and compressor operation point becomes a second-order problem. With this method using work compatibility and power turbine operation point specification, complex matrix iteration can be avoided and accurate results can be obtained.

Kurzke has listed common simplifications in modeling in his article [48] some simplifications which are commonly used in performance calculation. It is normal, that gas turbine modeling uses such simplifications which creates the simplest model to solve given task in accuracy needed. As told before constant component efficiencies are not sufficient for industry to describe gas turbine operation in needed accuracy. Also, nested loop iteration is usually deployed to the calculation for simplification reasons and while multiple iteration variables shows up, this simplification brings unnecessary complications to the calculation. Instead of using static temperature during pressure and temperature calculations stagnation properties are used. This is sufficient for the accuracy of the model and this leads to simplifications that not flow areas and velocities up- and downstream of components are not necessary to know. Also during component temperature and pressure calculations specific heat capacity is simplified usually to constants for the hot and cold sides or is calculated by mean of temperatures for both sides. An accurate version would require integration of entropy function. Humidity is usually left from calculations. Humidity would bring the third coefficient for non-dimensional analysis in component characteristics, gas constant R. This simplification does not bring a huge error to the performance model, but should be taken in to account when the gas turbine is used in humidity conditions, in the sake of condensed water and affect to R in large quantities. Also, with combustor calculations temperature increase is not directly proportional with a fuel-to-air ratio as described in (23), but in reality stalls in larger ratios, at stoichiometric conditions. But while most of the combustors are operating near low fuel-to-air ratios, given equation correlates to real phenomena sufficiently.

6. PROCESS SIMULATION TOOL DEVELOMENT

In this chapter creation of process simulation tool for Aurelia® A400 is described. First requirements and needs to build the model was discussed and afterward methods used for modeling are explained. Finally, the creation process of the model is discussed, and the operation of the tool is shown.

6.1 Requirements

In the company, performance model created by Lappeenranta University of Technology (LUT) has been in use. This model is created in Microsoft Excel and it has good accuracy at and near of the design point, but at off-design and partial loads model doesn't correspond to real engine well enough. This might be due to inadequate input arguments and nested loop iteration, and it has been noticed that Excel suffers from instabilities when input arguments are changed in big chunks. The company also had requirements for performance simulation tool which couldn't be fulfilled with the current model because of the nature of excel and origin (LUT) of the model. These accumulated the demand to create new performance simulation tool inside the company.

The new process simulation tool was desired from commercial, testing, and R&D departments. Naturally, many departments with different purposes create numerous needs and versatility requirements for the model. The commercial department was interested in what was gas turbine output and efficiency in changing ambient conditions, while R&D and testing were interested in more detail how gas turbine behavior will change with the change in operation parameters.

The commercial department had following need for performance prediction tool

- predict performance of the gas turbine with changing external conditions, like ambient condition, back pressure or intercooler parameters,
- the model had to be designed to be used only by persons deemed competent by the company,
- it must be running on a computer with a standalone program, preferably program without the need for installation,
- the output of the program must be in a printable format. Main outputs were power output and efficiency.

Simulation model requirements for R&D and testing were

- able to run without external software,
- operation parameters for all components should be expressed, as well as component characteristics and operation point in those,
- changing external conditions, like ambient conditions, back pressure, intercooler parameters, efficiencies
- changing operational parameters like rotational speeds and fuel mass flow.

Both user groups were requesting the ability to change inlet conditions and standalone program, but in calculation point of view, user groups wanted opposite approaches. As described in chapter 4.3, the operation of a gas turbine is done with changing TIT or rotational speeds, and those parameters eventually determine the power output and efficiency. Now commercial was requesting input variables of power output and R&D and testing were requesting input variables of rotational speeds and TIT. While all other steps are calculated independent inputs, the model was necessary to build the way that iteration was possible to be done in two ways, depending which was requested by the user.

6.2 Platform

The company had no specific requests for the used platform. More important in the scope of the model was to create a working model to fulfill all the requirements or required features could be added to the model easily in the future. Chapter 5 introduced some commonly used gas turbine simulation software and which were also considered to be modeling languages of the created model. In addition to these Scilab, GNU Octave, Modelica and Python were examined as a candidate for the modeling language.

The most important requirements for the model as a platform perspective was able to support the accurate thermodynamic calculation of a steady state performance model, existing libraries and available support, the external program must be able to be provided and source code could be hidden in the external program. In previously listed platforms, all are capable of creating an accurate thermodynamic model of the gas turbine. When considering executable standalone program and hiding of its source code, which was an important aspect in this model, while persons outside of the company could be able to use the model. This discarded many of the platforms: Python and Modelica are both good modeling languages but the nature of openness forbids the easy source code hiding and drops out them in the platform inspection.

MATLAB was chosen as the simulation platform, while MATLAB is diverse language, current steady state thermodynamic model can be fully described as well as the model can be enhanced in the future with all required needs. MATLAB has also built-in function “App Designer” which allows creating graphical user interface with figures and inputs/outputs to a standalone program. This type of function was missing for example FORTRAN. One requirement was to create a program, which was allowed to be used without installation of platform or function libraries, but this was not possible with all the available simulation platforms in the scope of this master’s thesis. MATLAB executable programs can be used without installing MATLAB in the computer but require installing the MATLAB run time environment to be able to execute the program correctly. Anyway, this was thought as a minor setback in all positive sides of MATLAB, thus it was decided to be used as a simulation platform. Also, there was enough support and help available in the coding with MATLAB, so previous in-depth knowledge of programming and programming language was not a block creating performance model.

6.3 Simulation method

It was clear from the start, that model was going to be built by thermodynamic relations, explained in chapter 3, and to assure detailed and accurate performance prediction, component characteristics had to be used. There were compressor characteristics and turbine efficiency characteristics available from the Excel model. These characteristics were estimations from LUT performance calculations during component design.

6.3.1 Component maps

With MATLAB as a simulation platform, it permitted built-in calculation functions libraries to assist programming as well as calculation power to allow the use of complex multivariate iteration algorithms. In MATLAB built-in functions also provided interpolation functions to seek operation point from component characteristics. Data provided in LUT performance estimations were in the form of a set of points for every constant speed curve. In Figure 20 each constant speed curve has around 10 datapoints, leading to an operation point in half way of provided speed curves having accuracy issues. While interpolation operation point from characteristics depends on three variables, mass flow rate, rotational speed and pressure ratio, two values must be known, and interpolation is done in nearby those two known values. Interpolation methods which were compared were box interpolation explained in chapter 5.1 and MATLAB built-in functions “scatteredInterpolant” and “griddedInterpolant”. These methods scatteredInterpoland needed the smallest amount of modification to data, griddedInterpolant bit more modification and box iteration needed the most modification, iteration methods were examined in respective order. ScatteredInterpolant could use component characteristics data points in their native form, just modifying data vectors of mass flow, pressure ratio and efficiency to matrixes, each row referring to one constant speed line. Figure 24 shows 3D map from map created with both interpolation functions as it is used in the calculation. Figures are not in the same scale, while only shape of the surfaces are important for this examination. As can be seen in the middle of the scatteredInterpolant surface, during normal operational area scatteredInterpolant created surface which interpolated operational data points accurately and the whole operational area was smooth curved surface, but problem became when operation was done outside of the data points. Near surge or choke lines, extrapolation was linear and not smooth continuation for curved surface, while calculating operational point, interpolation could be operating momentarily out of the component maps, with such linear approximation steady state point could not be accurately extrapolate complex 3D surface and problem became crucial when interpolation was happening near maximum or minimum rotational speed. During this area error was in the magnitude which crashed the whole calculation.

Next griddedInterpolant were investigated. This method required bit more modifications to characteristics data, while all constant speed curves had to be expressed as curves. MATLAB fitting method was used and the 3rd order function could accurately express every speed curve. The right side of Figure 24 shows surface created by griddedInterpolant. With speed curves, griddedInterpolant also created a smooth surface and because curves were extending speed curves, similar to Figure 22, also surface outside of the data points were smooth and correspond to actual behavior. This leads to accurate interpolation as well as extrapolation. Because griddedInterpolant worked well there was no need to test box interpolation. With turbines, pressure-mass flow characteristics was interpolated with data fitting function, while only one speed line was used and turbines efficiency similar to the compressor with griddedInterpolant function and found out that it accurately interpolated turbine data.

6.3.2 Iteration

With the new gas turbine process, N-R methodology from literature could not be straight use with this process, and parameters to set as state variables had to be studied again. Iteration method was multivariate matrix iteration which allowed multiple variables to be iterated simultaneously. Jacobian matrix, which is a partial differential matrix of investigated function, was calculated using the same method than Nasa's open source gas turbine model T-Mats. In this calculation, each input was increased and decreased

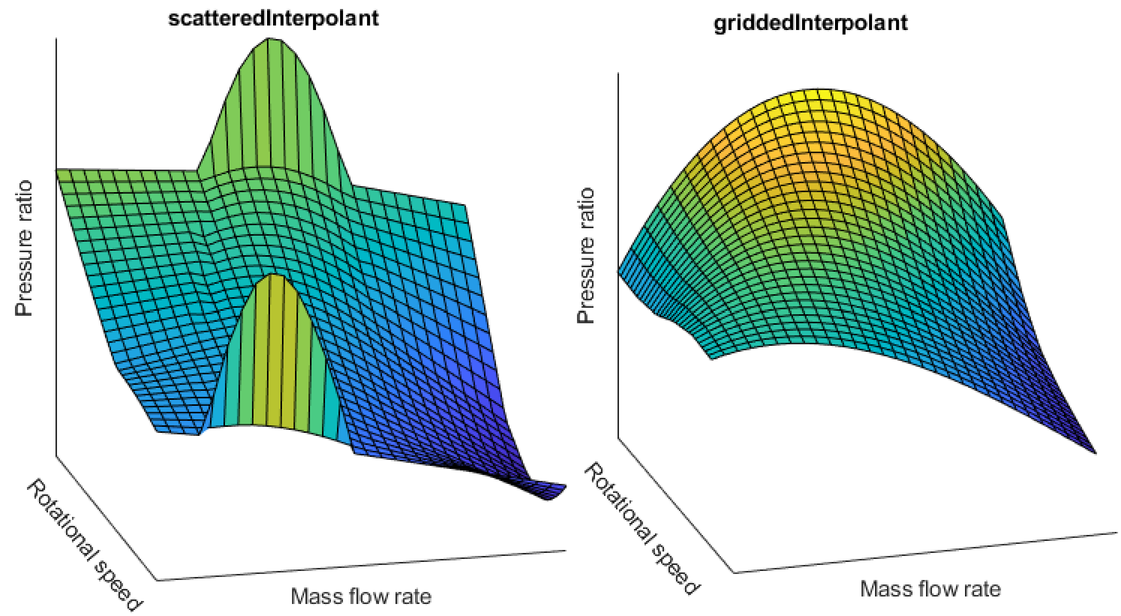


Figure 24. *Interpolation method comparison*

fractionally in turn, and each time running calculation and storing changed error function. Finally, when calculations were done, errors were normalized to found out the significance of each perturbation and how each input changes the output and by this Jacobian was created. This type of gas turbine process, with intercooler, recuperator and two spools, could be solved with 4 state variables if rotational speeds were determined at the start, or with 5 state variables with power input. In choosing the state variables, it is important that with chosen state variables all other parameters of gas turbine operation can be solved and for each state variable, error function must be made. This error function needs to be correlating to the input variable as close as possible, so iteration can effectively converge. In both calculating methods following state variables were chosen:

- mass flow rate,
- LP turbine pressure ratio,
- HP turbine pressure ratio and
- recuperator inlet gas temperature i.e. turbine outlet temperature.

Error functions to these variables were

- mass flow error between state variable and value obtained from HP turbine characteristics using pressure ratio
- similar mass flow error to LP turbine
- pressure error between upstream and downstream of combustor and
- recuperator temperature error between effectiveness and state variable.

With these variables and error functions, iteration was robust and converged quickly. Recuperator temperature convergence was the slowest one of these, but not significantly. Reason for slower convergence was, that it was most affected by other state variables. With power input, one more state variable was added, and it was LP shaft rotational speed, when rotational speed control scheme was applied and TIT, when power was controlled with temperature. With each power, the correlation between speed of both shafts was given to calculation. This correlation was calculated with changing rotational speeds individually in the whole operational range and storing efficiencies and output power to matrices to find out the most efficient running line throughout the power range. For rotational input, power error between requested and actual value was the final error function. This was found out to be quite slow to converge especially in lower power requirements, and it might need more inspections in the future if the calculation is required to be more rapid. However, calculation was made to converge in less than 20 iteration rounds in every tested ambient conditions in lower power range, hence it did not require further inspection in the scope of this master’s thesis.

6.3.3 Simplifications

During simulation, the specific heat capacity was calculated by mean temperature of up- and downstream gas temperature of each component and it was used as a constant value for each component. In the gas turbine, hot and cold gas are composing differently, while cold gas is pure air and air is mixed with fuel in the combustor. To assure accurate results, separate specific heat capacity function must be used for both streams. In this model own specific heat capacity function was used with compressors and turbines, but simplification was made, as earlier discussed, by taking specific heat capacity of mean temperature and not using integration of enthalpy.

Calculating combustor temperature rise, fuel temperature was assumed to be at the same temperature than the air coming from recuperator. In a real case, fuel temperature is usually lower with recuperator, but mass flow rate being negligible compared to mass flow of the air, heating power required to increase fuel temperature to recuperated air temperature is not significant. As defined in the earlier chapter, using equation (23), temperature rise is not accurate when approaching stoichiometric burning in the combustor, but can be used with sufficient accuracy with a small fuel-to-air ratio. Also, in combustor all the fuel is assumed to burn completely, and all fuel heat value is assumed to transform to temperature.

This model was using turbine map independent of rotational speed, and as mentioned earlier, error in this situation is negligible and the map can be calculated with such assumption. Also, data for compressor maps are not obtained from testing or CFD calculations, but they are estimations of radial compressor characteristics. Initial values and efficiency constants for example drive train and internal usage was kept constants, but made available for the user to define, and these constants were the same which were used in Excel-model. But in the scope of this master's thesis, this was sufficient to compare these two models. As specific heat function, for both air and gas including air and methane, these were same which were used in the Excel model. Source was an old thermodynamic book, but they were valid compared to values from National Institute of Standards and Technology (NIST) thermodynamic tables [53] and VDI heat atlas [54].

6.4 Design point calculation

Performance model was created in two stages: first design point calculation was made using efficiencies and parameters predefined in the design phase. Design point calculation means that all components are operating at highest efficiency at operation point which was predefined as an ideal spot, including component parameters as well as ambient conditions. This was created with thermodynamic relations and the only iteration with recuperator temperature was needed. No component maps were used in this method, while efficiencies and pressure ratios of components were given in the calculation. In design point calculation could be seen that MATLAB corresponded accurately to Excel model in thermodynamic calculations. After design point calculation, off-design simulation was made. In this phase, performance was simulated also with part load conditions, and most of the earlier discussion has been relating to this part. During off-design simulation, the model used the same thermodynamic framework than design calculation but this time operating parameters of components were obtained from component characteristics and iteration of operation point was done in Newton-Rapson iteration.

Design point calculation was done with predefined compressor and turbine pressure ratios and polytropic efficiencies, turbine inlet temperature, ambient condition and other component pressure losses. The goal in this phase was to make sure, that basic thermodynamic calculation was corresponding to the Excel model. With predefined values, mass flow rate and rotational speed of shafts were calculated and compared to the Excel model in given power output. For each component calculation was done using the thermodynamic equation explained in chapter 3 and more detailed flow of calculation is discussed next.

With readily available pressure ratios, ambient temperatures and polytropic efficiencies, compressor temperature and enthalpy rise were calculated with equation (18).

Simplification with heat capacity was used, and it was assumed to be constant in temperature range of turbines and compressors with an average of two temperatures. The equation for specific heat was the same used in Excel. Obtaining average of inlet and outlet temperatures required iteration, and it was made with loop iteration. There were two values used for gas constant R: for dry air $R = 0.287\ \frac{kJ}{kgK}$ in compressor calculation and for air-fuel mixture $R = 0.29\ \frac{kJ}{kgK}$ in turbine calculations. These values were the same, which were used in Excel to see accurate of models, but data from VDI heat atlas [54] corresponded to these values.

Intercooler pressure loss was assumed to be a percentage of incoming pressure and temperature decrease was calculated with heat exchanger effectiveness, equation (22). Pressure loss in recuperator and combustor was calculated similarly to intercooler pressure loss, with a given percentage of incoming pressure. Recuperator air outlet temperature was also calculated with given effectiveness, but this required hot gas inlet temperature, and it had to be iteratively solved. This was done with loop iteration, where the turbine outlet temperature was solved after combustor inlet temperature was constant.

Turbine calculation was made with equation (21). Using inlet temperature, polytropic efficiency and pressure ratio to determine temperature and enthalpy drop in the turbines. Constant heat capacity was calculated using the same simplification than with compressors, using an average of minimum and maximum temperature and calculating heat capacity in given equation. Also, gas constant R was given to calculation.

With calculated enthalpies, required specific work was obtained and required mass flow rate was iterated with mass flow rate for fuel to correspond required power output. Efficiency of process was calculated using fuel heat power and required power output.

6.5 Off-Design calculation

In the off-design simulation, component parameters are not predefined at the start of the simulation and they must be solved from component maps. Efficiencies of components as well as pressure ratio and mass flow rate changes with changing operational parameters, namely process maximum temperature and rotational speeds of the shafts. The calculation is not as straight forward as in design point and performance parameters of components need to be iteratively solved. Operational point is solved using components characteristics and it is called component matching, described in chapter 4.2 where turbine and compressor characteristics determine component behavior in determined rotational speed and prevailing mass flow rate or pressure ratio need. Iteration of operation point is done with multivariate Newton-Raphson method with Jacobian matrix.

Excluding Iteration and compressor maps, the calculation is done according to design point calculation: using thermodynamic relations explained in chapter 3 and calculating pressure and temperature after each component chronologically from inlet to outlet. Flowchart for calculation is shown for two different input cases in Figure 25. Left side shows calculation flow in a case where required power output was given and the right side shows a case where the rotational speeds of the shafts were given to calculation. Meaning of the abbreviations is explained in Table 1 below the figure and the user defined input variables are shown as underlined values. In Figure 25 N-R refers to the iteration method and with each input values thermodynamic relations are calculated, and an error function is created between given values and calculated values. Depending on error magnificence input variables are changed and tried out calculation again hopefully closer to steady state performance point.

Two input cases were explained in the earlier chapter, where marketing and R&D and testing requirements were discussed. As can be seen from the flowchart in Figure 25 calculation is happened similarly in both cases, just values given to calculation from iteration or user varies. Also, error function building is different in both cases.

6.5.1 Error generation

Iteration with this type of process, with two spools, both equipped with generators, could not be done with state variables described in the literature. All showed multi-spool gas turbines had only one turbine giving output power to generator or nozzle. In these cases, an error function can be built with momentum balance as one variable and other error functions are quite straight forward to choose. In this process, neither of spools had momentum balance, or know torque surplus, so state variables and error functions associated with these had to be re-examined. The first step was to made calculation with rotational speeds as inputs. Power input required a relation between rotational speeds, and it could not be calculated before developing optimal or requested speed balance. It was known in earlier studies, that there was a possibility to reach higher efficiencies when operating shafts independently and the first objective was to inspect which would be the optimal path in operational range.

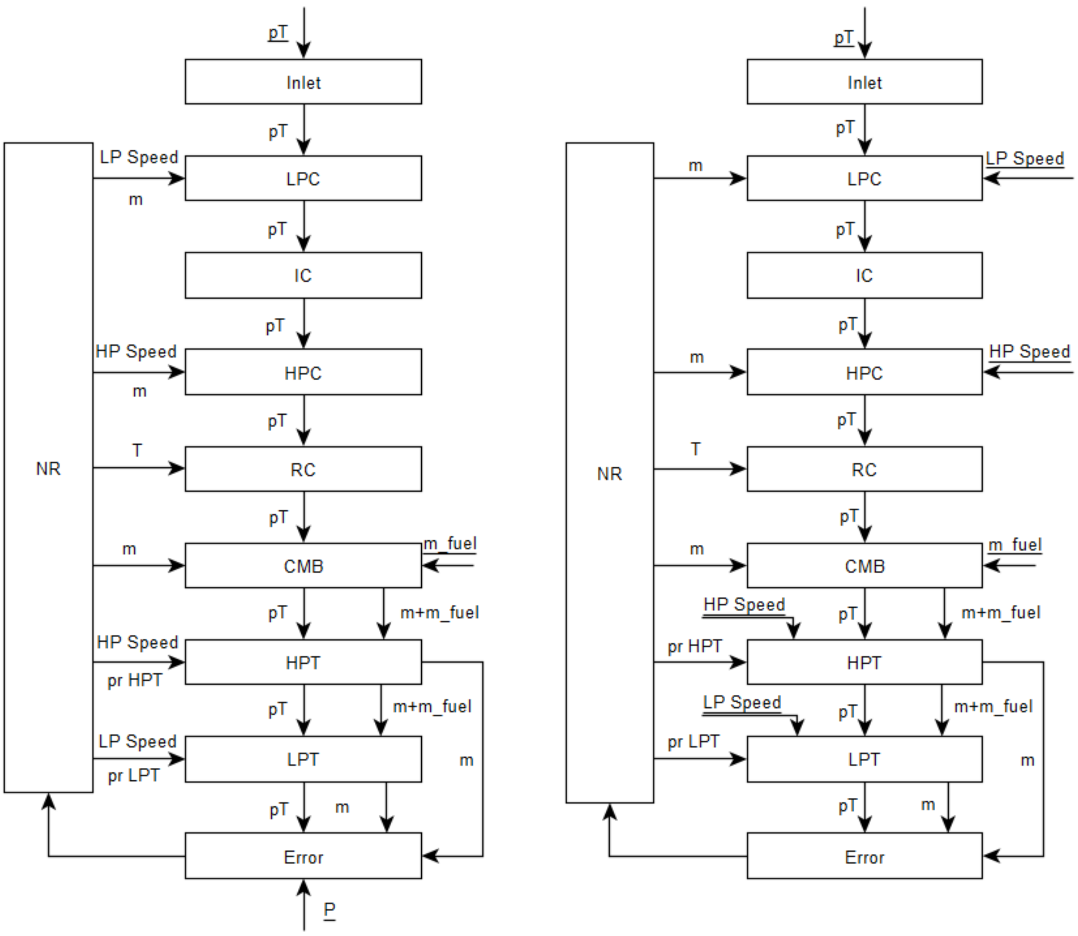


Figure 25. *Flowchart of the model*

Table 1. *Meaning of abbreviation for Figure 25*

LPC	Low-pressure compressor	IC	Intercooler
HPC	High-pressure compressor	RC	Recuperator
HPT	High-pressure turbine	CMB	Combustor
LPT	Low-pressure turbine	NR	Newton-Raphson
pT	pressure/Temperature	m	mass flow rate
LP Speed	Low-pressure spool speed	m_fuel	fuel mass flow rate
HP Speed	High-pressure spool speed	P	Input power
pr HPT	pressure ratio of HPT	pr LPT	Pressure ratio of LPT

When rotational speeds were given to calculation, compressors either needed mass flow rate or pressure ratio as state variable from iteration to be fully defined. If the pressure ratio would have been decided as a state variable, both compressors would have needed their own variable. With mass flow, one state variable would have defined both compressors completely. If beta-lines would have been used in compressor characteristics, they would have been state variables and one variable would have been needed for both compressors. Turbine calculation must be made with pressure ratios as state variables, since mass flow can exceed the choke, and mass flow rate as state variable calculation would have become unstable when state variable became larger than choke limit. Recuperator required iteration variable to solve effectiveness calculation and turbine outlet/ recuperator gas inlet was chosen to be a state variable. Compressor calculation was the only issue choosing state variables. If pressure ratios were chosen as a state variable, in a total of 5 variables would have been needed to fully describe gas turbine operation. Mass flow rate of each component would have been compared to component downstream to create error function, and addition to mass flow inspection, recuperator effectiveness would have been considered as one and final would have been pressure balance between compressors and turbines with total pressure losses. Similarly using beta-lines, would have brought 5 state variables and similar mass flow comparison as well as pressure comparison, but this would have created modifying to compressor characteristics data. With mass flow rate as a state variable, the calculation could have been done with 4 state variables, and with testing, calculation was robust in the whole operational range. Larger state variables make calculation heavier, while Jacobian matrix already takes most of the calculation power and increasing its size shows in a slower calculation. With 4 state variables, error calculation is explained later in more detail.

When using 4 state variables, also 4 errors must be generated. Best practice is to choose error functions such way, that they correlate to state variables as directly as possible. Turbine pressure ratios were giving mass flow rate through the turbines, so with both turbines closely correlatively error function could be made comparing given mass flow rate to state variable mass flow rate

$$e_{1/2} = \frac{\mathrm{m}_{NR} - m_{cha}}{m_{NR}}, \tag{33}$$

where m_{NR} is state variable mass flow rate and m_{cha} is mass flow rate obtained from turbine characteristics with pressure ratio state variable.

As straight correlation couldn’t be found with recuperator temperature and mass flow rate, but mass flow rate was determining pressure ratio of compressors. When considering pressure rise in compressors as total and comparing it to the turbine pressure ratio added with other component pressure losses, the next error function could be made. The calculation was made with downstream pressure from inlet to combustor, compared to upstream pressure from the outlet through turbines to combustor

$$e_3 = \frac{p_{us} - p_{ds}}{p_{us}}, \tag{34}$$

where p_{us} is pressure upstream and p_{ds} is pressure downstream. This could be seen converging almost as fast as turbine mass flow calculation, so no direct correlation was needed.

Last error function compared state variable hot air temperature to recuperator effectiveness equation (22) with the calculated turbine outlet temperature

$$e_4 = \frac{T_5 - T_{calc}}{T_5}, \tag{35}$$

where T_5 is recuperator hot air temperature, number referring to Figure 8, and T_{calc} is calculated hot air temperature from effectiveness equation. This was slowest converging in examined error functions, and it can be explained by that turbine outlet temperature is dependent on multiple variables and it's not straightly correlating to effectiveness function.

With chosen state variables and error functions iteration was converging and steady-state operational point for each rotational speed. Next, the operation of the gas turbine in its whole range could be made with varying speed balance of the shafts. LP shaft speed was incremented with steps of $500\ RPM$, for the whole operational range and calculation was done with HP shaft speed of $\pm 1{,}000\ RPM$ of LP speed in $100\ RPM$ increments. With storing power output and efficiency to a matrix, the most efficient operational line could be found. This relation was used to calculate process when power input was used. When wanted power was inputted, Newton-Raphson required one more state variable and error calculation, and these were decided to be LP spool speed, as well as generated power error

$$e_5 = \frac{P_i - P_{calc}}{P_i}, \tag{36}$$

where P_i is inputted power and P_{calc} is calculated power output with turbine and compressor power consumption/creation. HP spool speed was calculated with a relation of highest efficiency with state variable LP speed. LP spool speed was chosen to state variable since pressure rise from LP compressor had to be sufficient for HP compressor not surging in the calculation. Convergence with this error was slow and perhaps needing more investigation in the future.

6.5.2 Calculations

The thermodynamic calculation is done using equations discussed in chapter 3. The calculation is started with user-given ambient conditions and requested power or rotational speeds. Initial guesses for state variables are calculated in a function of LP speed, ambient pressure and temperature. It was seen during designing, that constant initial guess in lower power outputs, or in extreme ambient conditions were easier to lead non-wanted values and a situation where no convergence was happening. With making initial guesses function of user-given values, the calculation was made much more robust.

The calculation was requested to create with two different input methodology, but in both cases, the basic principle of information flow stays constant as can be seen from Figure 25. With both situations, all critical values were either as inputs to the system or as state variables. This resulted that calculation could be made similarly in both cases, all changes created in “switch” cases, so framework in both cases could be the same. With noticing which phase was in use, program knows which variable it needs to send to N-R and which were results.

Calculation is started from ambient conditions and these values are brought to compressor after pressure losses of inlet. From compressor maps, pressure rise and compression efficiency are obtained with inputted rotational speed and state variable mass flow rate. Compressor maps were using reference values to make characteristics non-dimensional, chapter 4.1.2, which meant that current inlet conditions were normalized with reference point characteristics were made. Values were obtained from compressor characteristics in the following method:

$$\left(\frac{N}{\sqrt{T_{in}/T_{ref}}}, \frac{m\sqrt{T_{in}/T_{ref}}}{p_{in}/p_{ref}}\right) \rightarrow (\pi, \eta),$$

where T_{in} and p_{in} are inlet pressure and temperatures and T_{ref} and p_{ref} are reference temperature and pressure, which were used to create the component map, π is pressure ratio and η is efficiency. With efficiency temperature rise is calculated with equation (15) and power requirement for the compressor is calculated with equation (1). With temperature calculations, specific heat capacity was iterated as in design phase with loop iteration using a mean of inlet and outlet temperatures.

In intercooling, pressure loss is calculated with predefined pressure loss in intercooler, and the output temperature of the air is calculated with effectiveness of the intercooler, using equation (22).

HP compressor pressure and temperature rise were calculated with the same principles than values in the LP compressor. In this point, calculation of recuperation temperatures is skipped, and state variable value of recuperation hot air is used to proceed to combustor calculations. Combustor calculation can be calculated with given TIT or fuel mass flow, depending on how the user is requesting. The commercial department is requiring information about required fuel mass flow rate and it is calculated with equation (23), using predefined TIT in normal operational use. The same equation is used when fuel mass flow is used as an input variable.

In turbine calculation, mass flow rate was obtained in rotational speed independent characteristics from pressure ratio with a following equation

$$(\pi) \rightarrow \left(\frac{m\sqrt{T_{in}}}{p_{in}}\right),$$

where π is pressure ratio and $\frac{m\sqrt{T_{in}}}{p_{in}}$ non-dimensional mass flow. Temperature decrease in the turbines was calculated with efficiency obtained from turbine maps, in function of rotational speed and mass flow rate

$$\left(\frac{m\sqrt{T_{in}}}{p_{in}}, \frac{N}{T_{in}}\right) \rightarrow (\eta).$$

Efficiency was used to calculate temperature decrease in the turbine using equation (20). Again, iterating specific heat with mean temperature. The same equation was used for both turbines.

Lastly, error functions for N-R method were calculated, in earlier described manners. With error functions, Newton-Raphson iteration was used until error functions were inside given margin.

7. RESULTS AND DISCUSSION

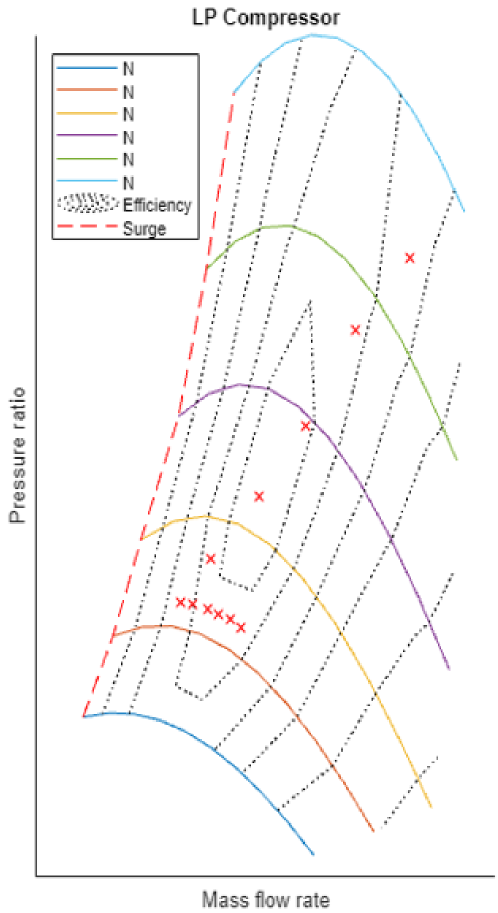
The created MATLAB model gave realistic values in the whole operational area. Also, results agreed well with the Excel model. Both models were using thermodynamic relations of components, but the differences between models were iteration method and the method with which data was obtained from component maps. The iteration method used in Excel was a loop iteration, where in a total 5 nested loop were used. In component characteristics calculation Excel was using flow coefficient and work coefficient, which are coefficients depending on operational speeds and the physical size of components. Results were agreeing well, even if the methodology was changing in the critical part of the modeling.

In the following chapters first the user interface (UI) of the MATLAB model is discussed, then accuracy of the developed model is described by comparing it with the Excel-model. Parameters which were chosen to compare were running line, total power output, mass flow rate and component parameters. Those parameters are the most critical in describing the accuracy of the model. The last part is focusing on future improvements for the model.

7.1 UI and running line

The user interface was created just for the presentation of this master's thesis to show some results and it will require further development and feedback from users. The current style of the UI is shown in the appendixes with two tabs, one with inputs and the other with outputs. User starts the simulation by changing ambient conditions and input data if needed, appendix page 1. The calculation can be done either by requesting power output or rotational speeds. This is chosen using a drop-down menu initially showing "Power input", and if necessary, it can be changed to rotational speed input. Calculate-button starts the calculation and the results are shown in the UI. When the calculation completes, the compressor operation points are drawn in the compressor maps. In the appendixes, design point is drawn for both compressors. Main parameters, which are thermal efficiency, power output, rotational speeds, mass flow rate, and fuel mass flow rate, are shown below the compressor maps, and the temperature and the pressure as well as the compressor and turbine parameters are listed in the Results-tab, shown in Appendix page 2. If the calculation cannot be executed, because of an invalid input parameter or non-convergence calculation, an error message is shown.

When calculating multiple points, all calculations leave a red cross in the compressor characteristics and the operational running line can be seen. In Figure 26, the running line of the LP compressor is shown for both models, MATLAB compressor map is shown on the left side and Excel on the right side. Running lines are calculated from 0 kW to 400 kW. In Figure 27, HP compressor running line is shown. Both models are using the same compressor maps, but the style is different due to unique plotting style for both platforms, but the operation point in the compressor maps varies since the models use different methodology. Initial conditions were kept constant for both models during comparison. The L-shape of the running line comes from the change from the TIT control to the rotational speed control, at lower power output the gas turbine is controlled by changing TIT and at higher power output with rotational speeds. As can be seen, running lines in MATLAB correspond for the most part to running lines created in Excel. With low-pressure compressors, running line goes across the map during rotational speed operation and with high-pressure compressors, the running line follows the efficiency curves. Both models also correspond s with the TIT control.



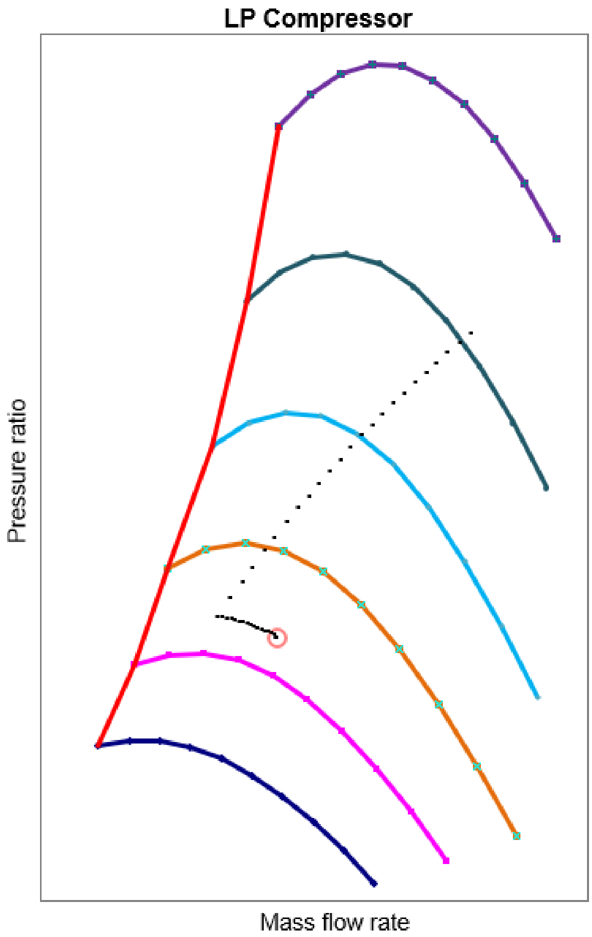


Figure 26. *LP compressor running line, left MATLAB, right Excel*

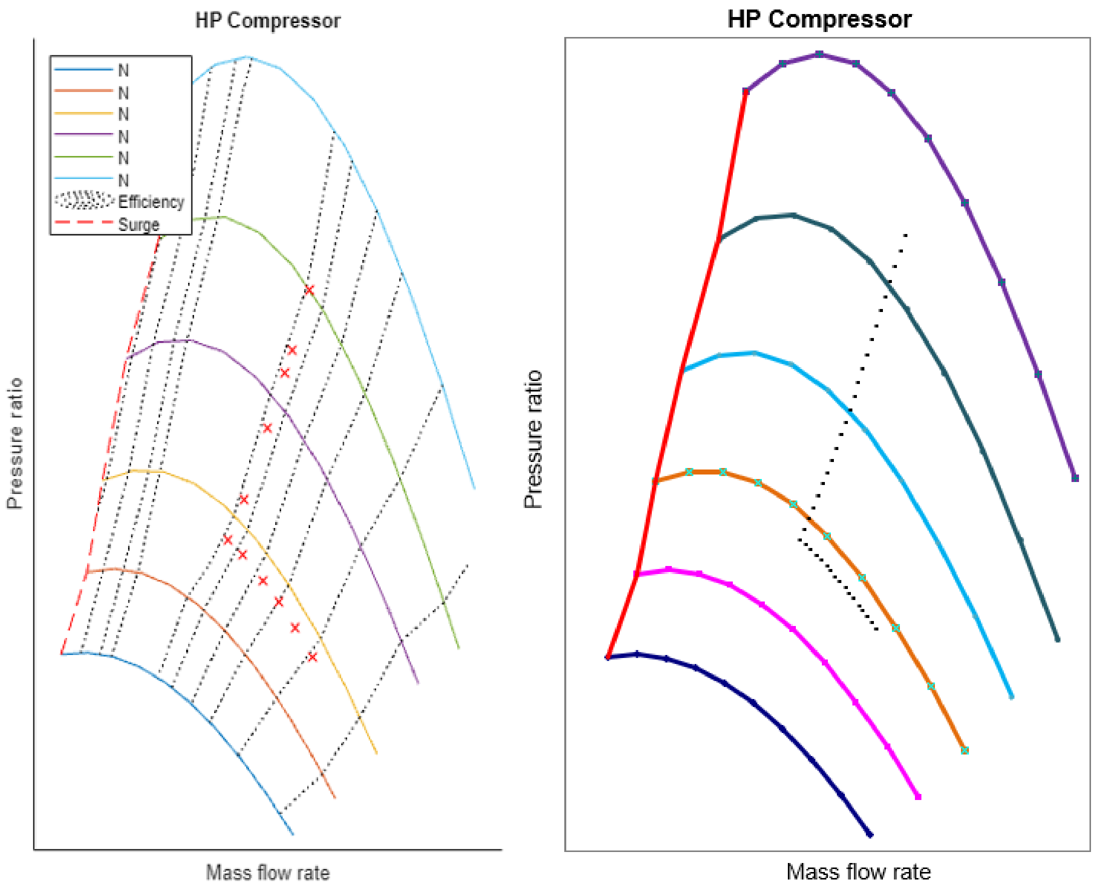


Figure 27. *HP compressor running line, left MATLAB, right Excel*

Difference between the models can be seen in higher rotational speeds. This can be explained by rotational speed determination. In the Excel model, a relation between rotational speeds of the shafts are calculated by determining low-pressure shaft speed and calculating equilibrium operational point with predefined constant rotational speed difference. In this method, ratio has been manually determined and it is constant in the whole operational range. With the MATLAB model, the most efficient operational line has been calculated for every power request independently. As can be seen from figures above, it does not bring a big difference in the shape of running lines. At higher rotational speeds, MATLAB model reaches higher speeds than Excel with LP compressor and vice versa with the HP compressor. Moreover, LP compressor running line is steeper with MATLAB model, to allow LP compressor to operate in higher efficiency area with higher rotational speeds.

7.2 MATLAB compared to Excel

In more detailed comparison between models is done using same rotational speeds in both models. Using same rotational speeds, similarity of the model’s behavior can be best described, and parameters of the component can be compared, while all inputs are kept equal. the chose rotational speeds were Excel running line speeds from 170 kW to 450 kW in seven points 50 kW increment 200 kW to 450 kW and addition 170 kW, where temperature operation has begun. 450 kW was chosen to upper limit to examine model's extrapolation capability. Parameters which were decided to compare, were power output, mass flow rate, compressor and turbine pressure ratios and efficiencies. These parameters define the operation of the gas turbine, and the differences between models can be best described by comparing listed parameters.

In Figure 28 comparison of power output and mass flow rate are shown. The data is plotted against power output of Excel model with used rotational speeds. This was a simpler and clearer way to show operational point, than plotting parameters against both rotational speeds. In Figure 28, the blue line describes the correlation between power output of MATLAB to power output of Excel, Pm/Pe. As can be seen, MATLAB model gives 1.2 % higher power output at lower rotational speeds than Excel, but at higher rotational speeds MATLAB shows 3.8 % lower power output. The orange line, mm/me, is showing correlation between mass flow rate and both models gave similar results error being maximum of 0.8 %.

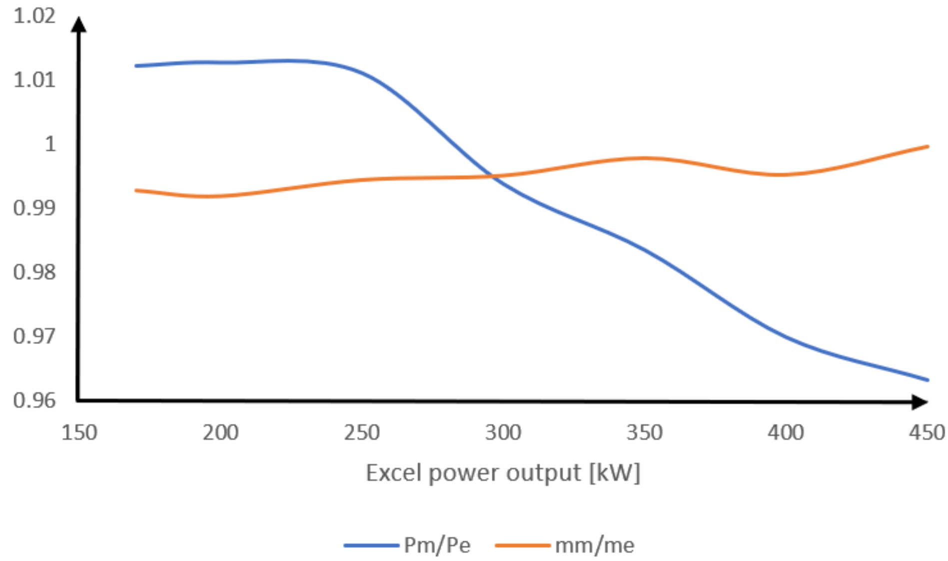


Figure 28. *Power and mass flow correlation*

Trend in component pressure ratios are similar in both models, Figure 29. In Figure 29 data points from Excel are shown in crosses, each color referring to one component and MATLAB data is shown in solid lines. Again, power output of Excel, which used rotational speeds are referring, is at x-axis, while pressure ratio is at the y-axis. Both models give almost identical pressure ratios, while error between models being 2 % at maximum. As can be seen from the figure, both low-pressure compressor and turbine have higher pressure ratios than high-pressure components.

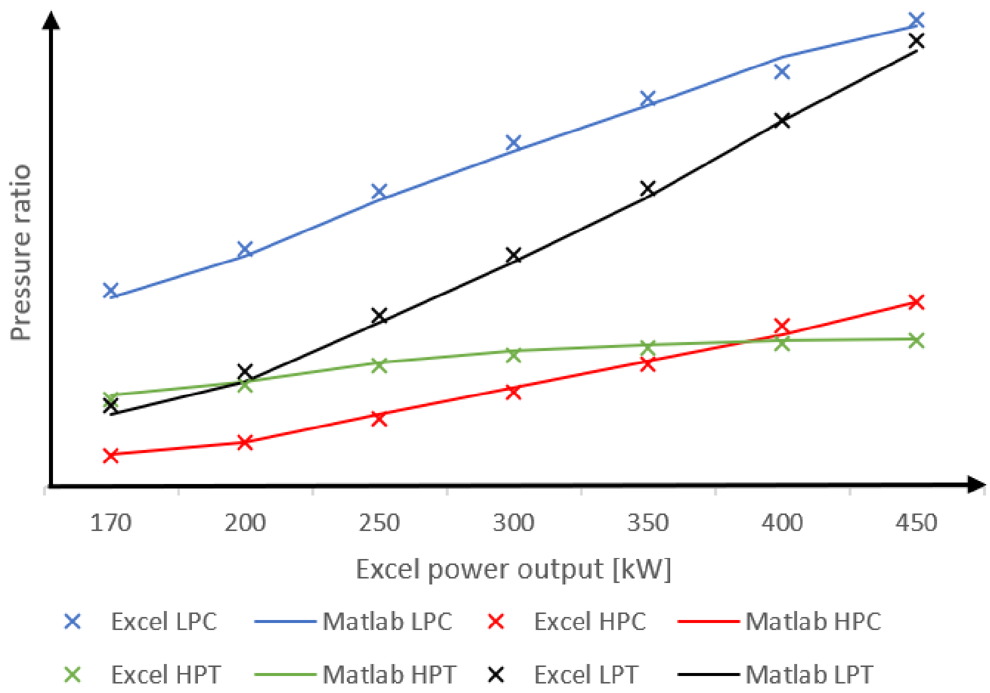


Figure 29. *Component pressure ratio comparison*

With component efficiencies models' results don't correspond as well. Component efficiencies are shown in Figure 30, where again results from Excel are shown in crosses and results from MATLAB in solid lines and each color relating to each component. Excel shows higher efficiency for each component in the whole operational range, except LPC pressure at lower power outputs. This can be explained by looking Figure 26, where the running line goes over the high efficiency area, and these models cross it in different rotational speeds.

Currently the MATLAB model is showing better thermal efficiency in higher power outputs, even if component efficiencies are lower and mass flow rate and pressure ratios are about the same. Thermal efficiency is a ratio between total power output and heating power of supplied fuel. When turbine efficiencies are lower, temperature difference between inlet and outlet of the diminishes, as can be seen from equation (20). This leads to higher turbine outlet temperature and thus more heat can be utilized from recuperator and required fuel decreases. This might be the reason why MATLAB model is giving higher efficiency at the higher rotational speeds, even if the power output and component efficiencies are lower.

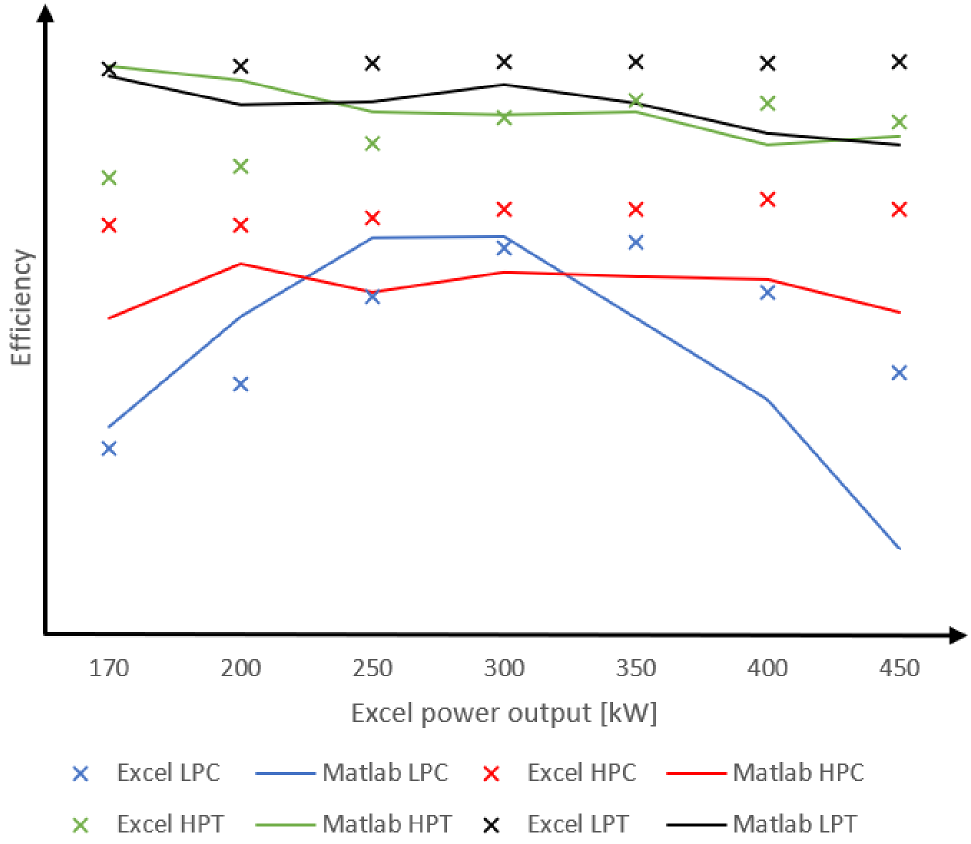


Figure 30. *Component efficiency comparison*

Overall, developed MATLAB model is corresponding well to Excel model, while using the same inputs, greatest errors being power output and component efficiencies and staying below 4 %. These errors can be further decreased with fine tuning component maps and it must be done later with test data from actual engine. Mass flow rate was achieved almost identical in the comparison, while turbine map for pressure ratio and mass flow rate obtained from other calculations were not accurate enough. Used turbine maps were tuned with Excel data to get corresponding mass flow rates. This also leads to equal pressure ratios between each component, while compressor maps are same between models, meaning same mass flow rate with rotational speed results same pressure ratio and turbine pressure ratios were optimized through mass flow rate tuning.

The difference between efficiencies was resulting from different calculation methods between models: Excel was using work and flow coefficients to obtain efficiencies with functions of rotational speeds, while MATLAB was using efficiencies straight from the component maps using pressure ratio and mass flow rates. This results little bit higher than results especially at higher rotational speeds. Big difference can be seen on HPT efficiency data, where at low power outputs, results diverge. Currently MATLAB model is using mass flow rate as determining the efficiency with rotational speed, and these curves are not quite accurate enough. With later development in this model, this must be taken account and find turbine maps using pressure ratio instead of mass flow rate to be used to determine the efficiency.

As described in chapter 3, power demand in compressor and power output in turbine is function of the pressure ratio and the mass flow rate. When both models are giving similar pressure ratios for each component and mass flow rates going through the engine, models should be giving equal power outputs at given rotational speeds if efficiencies are not taken in to account. Adding efficiencies given in Figure 30 to power calculation, with higher LPT and LPC efficiencies in lower rotational speeds would results higher power outputs in MATLAB model as shown in Figure 28 and with increasing rotational speeds lowers component efficiencies in MATLAB model resulting to lower power outputs. This was the case with comparison and scaling component maps, error would have been able to reduce.

Developed model correspond well to old model and current differences can be explained but more important to developed model is to find out, how well it can predict the performance of the actual Aurelia® A400 and users are mostly interested about operation in normal running line, shown in Figure 26 and Figure 27. Comparing and validating developed model to Excel model assured that MATLAB model can be made to predict performance of the actual engine with scaling component maps with test data when suitable data is available, and company can start transition from Excel model to MATLAB model.

7.3 Future improvements and possibilities

The developed model had true demand in the company as told in chapter 6 and in the cope of this master’s thesis all demands were not able to fulfill. The main goal in this master’s thesis was to develop a model as a base for the company so the model can be further developed to precisely correspond demands. Meaning that all further possible updates were kept in mind while designing the platform and creating the model. In this master’s thesis comparison was done only to current model to be certain that model corresponds in the same magnitude during the operational area.

Keeping this in mind, one of the main point in the development and choosing platform was to take into account future use and improvements. Programming was made to be easily understandable and modified in the future regardless of the person in use as well as coding as described and detailed flow chart of the operation of the code. Also, additional features are easy to implement in the model. In the MATLAB model, component maps are the single most important parameters to describe how well the model corresponds to an actual machine, and these maps are easy to update or completely change without needing to additionally change the program. Currently component maps are text files, with multiple datapoints each column referring to one parameter. With these text files, component map updates or new maps, can be brought to model by comparing it to data from testing or whole component testing, to better correspond engine and model. Maps can be either changed completely or just part of the values can be multiplied with predefined coefficients. Biggest accuracy increase can be done with changing turbine mass flow-pressure ratio map to depend also with rotational speed if this kind of map is in the to-do list in CFD calculations. Another improvement is with efficiency turbine maps. These types of maps are quite inaccurate in near edges of the operational area. Currently they are using mass flowrate to determine efficiency, but more accurate way would be using pressure ratio to determine efficiency.

Iteration method is done by Newton-Raphson algorithm, and parameters which it is using to solve steady state operational point were optimized to the smallest number of iteration variables to obtain similar results in power output and rotational speeds compared to Excel model. As described earlier this could be done with 4 iteration variables, but if it becomes necessary to calculate in more detail for example operational points in compressors, new iteration variables can be introduced to the N-R method. This might be when both compressor operation points are wanted to be described in more detail then both pressure ratios are defined and iterated individually. Currently pressure ratios are handled as a total pressure rise, but by introducing one more variable to the iteration, both pressure rises can be estimated in more detail. This might be due to more depth surge or choke examination or when ambient conditions are in extreme conditions. Currently model operated well when ambient conditions are in normal operation conditions, but iteration is converging slowly and unstable when the model is used at maximum or minimum allowed ambient conditions.

User interface showed at the beginning of this chapter was fully designed to just be a description of the operation of the model during presentation of this master’s thesis. UI was created show data for each operation point and expression of the maps, without concerning user experience. This also needs some future improvements, with conversations and opinions from users, to help make user interface easier to use and all features which were required are available.

One requested feature, which was not included in the scope of this master’s thesis and would be included in the model in the earliest convenient is an analysis which includes the humidity of the air in the calculation. This would require just inspection of the incoming air and its humidity, in compressor maps, non-dimensional values including gas constant R, which would vary with varying humidity. Also deposition of water inside the intercooler would be in the concern.

8. CONCLUSION

Future changes in the energy market allow small gas turbines to play a big part in the next decades' energy production. This is accelerated with the tendency to move towards cleaner fuels, like natural gas, higher efficiency applications, like CHP, and decentralized energy production. With increased gas turbine demand, accurate and versatile simulation models for predicting their performance in changing ambient conditions and whole working life is also required. With simulation, cost and time of the designing phase can be reduced significantly and the maintenance is made much more efficient with precise flaw detection and performing maintenance only on the deteriorated components.

In this master's thesis, a new process simulation model for IRG2 gas turbine process was developed to replace the current model. Methodologies proposed in the literature could not be used directly to predict performance of the IRG2 process, but the simulation tool was created by implementing some of the methods used in the literature. The developed model is using the state variable method with the Newton-Raphson iteration, and component performance is predicted with component maps obtained from calculations done by LUT. The model was also created to easily allow future modifications and improvements.

The results of the developed model were compared with the results from the currently used model in the company. Both models are using different methodologies in iteration and in predicting performance of the components, but results from both were consistent with each other, with error being at maximum 4 % and the difference in the results could be explained.

In the future development, the developed model can be made to predict the performance of the actual engine by scaling component maps and fine-tuning input parameters by data obtained from testing. Overall, the developed model created a good framework for future updates to allow the model to better correspond to the company's need for a process simulation tool.

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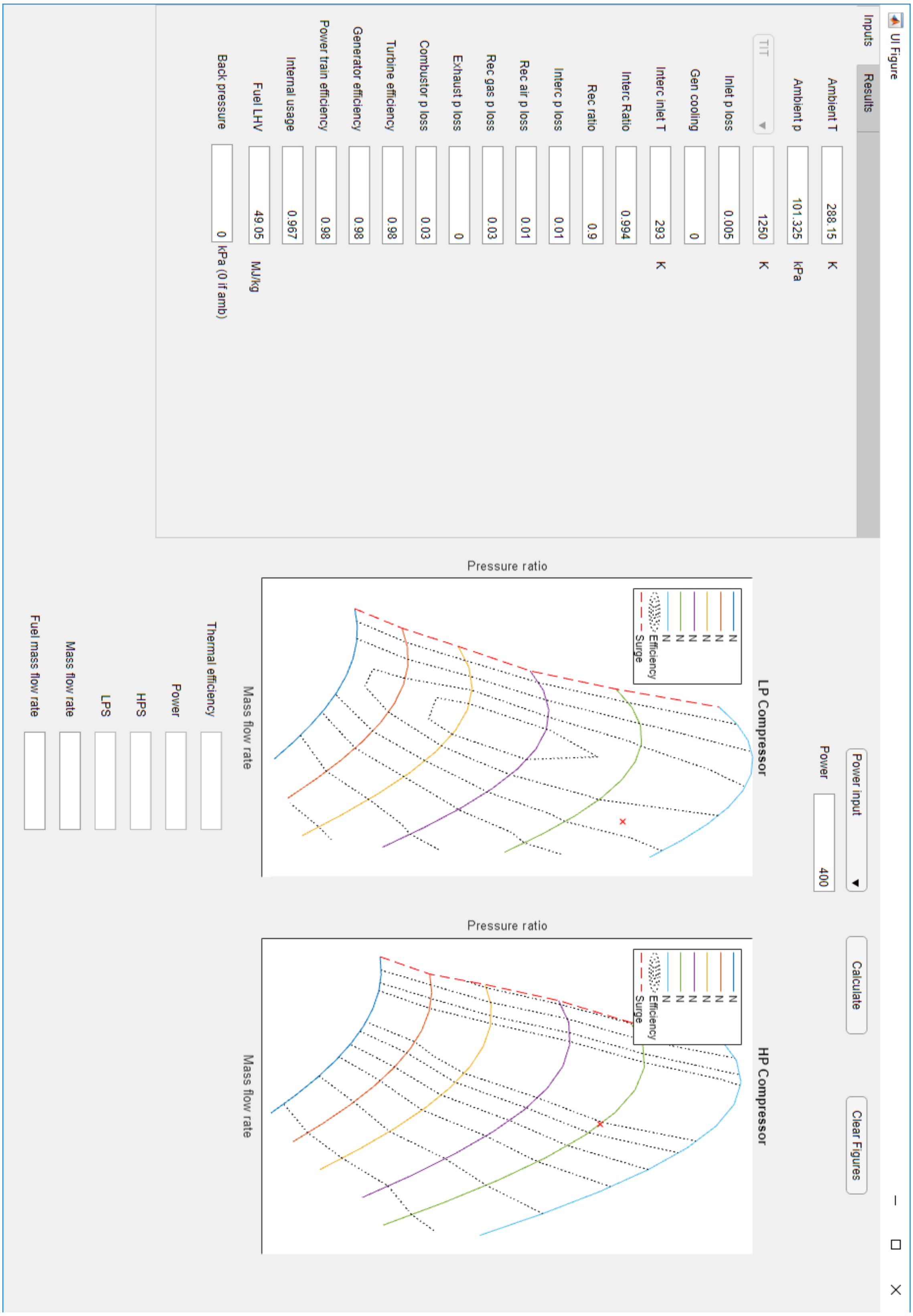
APPENDIXES



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Rec gas p loss
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kPa (0 if amb)
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Power
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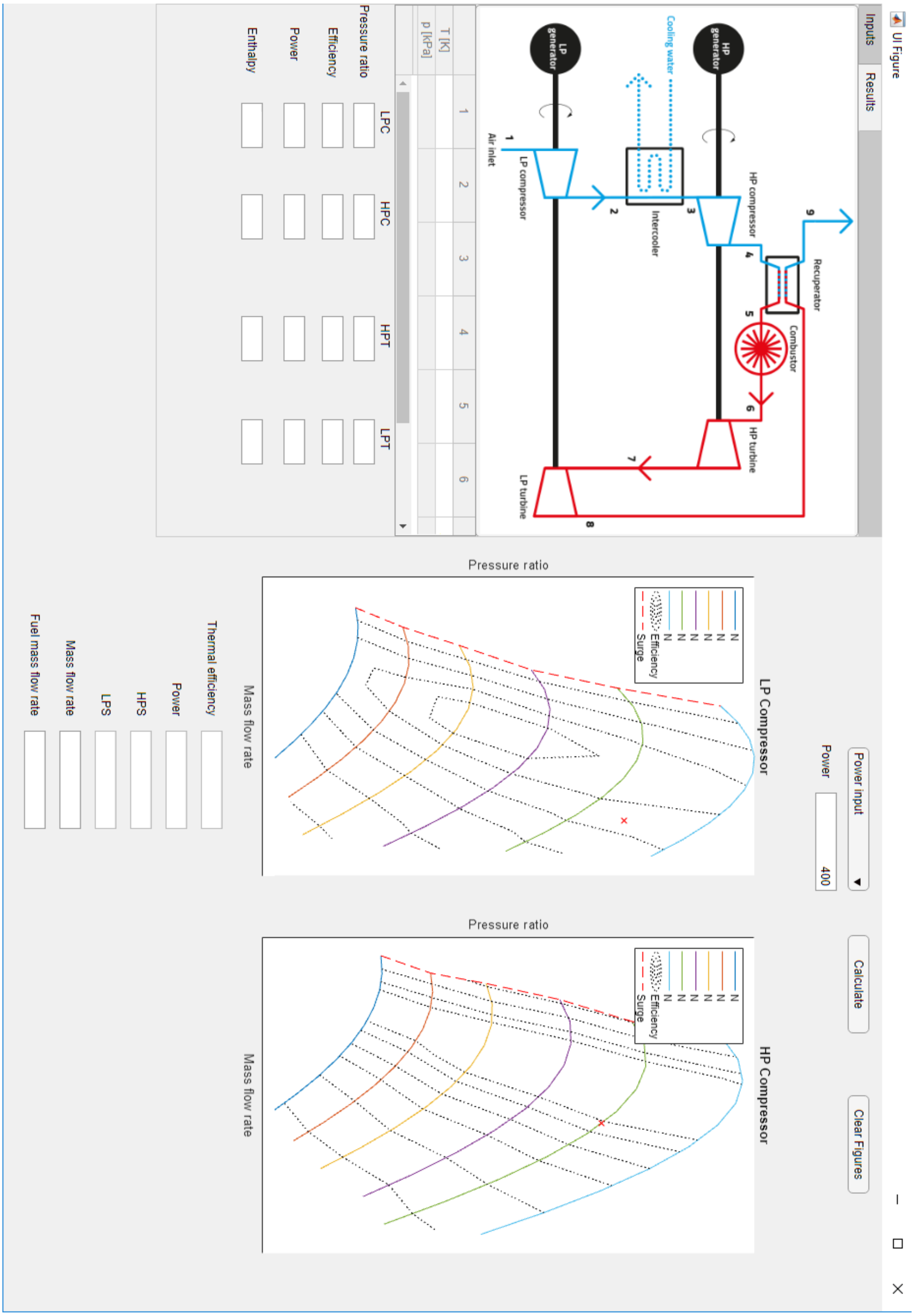
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HPT
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