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Modeling and Analysis of a 2-Stage Turbocharger

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Description of Task

Bergen Engines is offering the following project-/master thesis subjects for the year 2013/14:

Two-stage turbocharging:

Two stage turbo charging offers possibilities for increasing power density, increasing engine efficiency and lowering emissionss at the same time. The project shall investigate different solutions for two stage turbocharging with regards to process parameters. A simulation model shall be built in a suitable software system and used to evaluate different configurations. The most promising solution shall be chosen and optimized with evaluation of operating parameters.

Possibility for summer internship in 2013.

NTNU contact: Associate Professor Eilif Pedersen

Preface

The subject of this master thesis is 2-stage turbocharging of a medium speed diesel engine. The thesis was proposed by Espen Seeberg, Manager Performance and Process, Rolls-Royce Bergen Engines AS, RR BEAS. He also acted as my contact person at Rolls-Royce Bergen Engines AS, (RR BEAS). My supervisor at NTNU has been professor Eilif Pedersen at the institute of marine technology, IMT. RR BEAS has provided me with a GT-Power model of their C25:33L6 engine and test data for the real engine to enable verification. They have also provided the necessary compressor and turbine maps for building the 2-stage turbocharger. The objectives of this thesis have seen some minor adjustments over the semester, all according to or with consent from both RR BEAS and the IMT.

Abstract

The 2-stage turbocharger concept used in this master thesis is two separate turbochargers coupled in series, the air goes through a low pressure, LP, compressor and a high pressure, HP, compressor before it arrives at the intake valves. The exhaust passes through both a HP and LP turbine before it reaches the tailpipe. This concept allows charge air cooling between the two compressor stages in addition to the intercooler between HP compressor and the engine.

Important concepts like Miller cycle, timing and loss are introduced in the beginning for reference later on. Also expected effects of 2-stage turbocharging with Miller timing is introduced. The literature review looks at what the concept of 2-stage turbocharging is and what has been done in this field. Then as a starting point the matching procedure for a 2-stage turbocharger is treated to some detail. The importance of a good splitting ratio between the pressure ratio of LP and HP compressor is found to be around 2. A simple mathematical model of a 2-stage turbocharged engine is built in Matlab code using simple thermodynamic relations. The code is used for calculations, and can be found in the appendix. By setting many constants and making qualified assumptions the model is used to illustrate a 2-stage turbocharger and the results are presented in a compact format meant to provide basic information needed for turbocharger family and model selection.

In the verification chapter the RR BEAS engine is introduced. It is a 6-cylinder in-line, turbocharged diesel engine capable of running on both HFO and MGO. RR BEAS has provided test results from the real engine as well as a GT-Power model of the same engine, equipped with a 1-stage turbocharger. GT-Power software is powerful and user-friendly. The engine model was in good shape but a new NO_x measurement was established in the tailpipe using a moving average template. In addition the NO_x calculations were changed from equilibrium calculations in the cylinder at exhaust valve open, EVO, to using a EngCylNOx template that then was tuned until satisfactory NO_x measurements were reached. Then the engine model was verified against the test data from the real engine and the results found satisfactory. The next step was building the 2-stage turbocharger in GT-Power. This was done by using the available templates in GT-Power, RR BEAS provided the necessary compressors and turbine maps in the format of SAE files, these were successfully integrated in the model. The intercoolers where modeled as “black boxes”, meaning that they are not made as an exact replica of a real one. A wall temperature close to the desired charge air temperature is imposed on the intercoolers and then the pipes of the intercooler are given a very high heat transfer multiplier value, this forces the charge air temperature close to the imposed wall temperature very fast. New inlet valve lift curves were made for Miller timings from 40-100, crank angle degrees, CAD. An extra 2-stage turbocharger model was made, identical to original only with increased efficiencies by use of efficiency multipliers in the compressor and turbine templates. This was made to be used a reference of what can be achieved with even higher boost pressures than provided by the original 2-stage turbocharger.

The first simulation ran the 2-stage turbocharger at 100 % of maximum continuous rating trying Miller timings 40-100 with 10 CAD increments. Miller 70 was established as an optimal timing for the combined reduction of both BSFC and NO_x. Then a comparison was run between the 1-stage turbocharger, 2-stage turbocharger and the ideal 2-stage turbocharger. They were run through 6 different loads, 110, 100, 75, 50, 25 and 10 % of MCR. The two 2-stage turbochargers were both run with Miller 70, while the 1-stage turbocharger was run with its standard Miller 40. The results showed that the 2-stage turbocharged model reduced NO_x emissions by around 1 g/kWh and BSFC by around 2 g/kWh. The boost pressure was above 7 bar for 100 % MCR. However the maximum cylinder temperature was not reduced and this probably hindered further NO_x reduction. This led to the belief that the turbochargers are maybe not a perfect match for this engine. The HP turbine were underperforming at an efficiency around 70 %, 10 % less than the compressors and LP turbine. When looking at the ideal 2-stage turbocharger it achieved a NO_x reduction around 2g/kWh and BSFC was reduced by almost 6 g/kWh. This confirms the suspicion that the turbochargers and engine is not a perfect match. What happens is that they are not able to deliver high enough boost pressure and then the AF-ratio is reduced and the cylinder temperature goes up, limiting NO_x reduction. When looking at the energy balance it reveals that heat transfer can be substantially reduced with lower temperatures. The engine friction is reduced with lower cylinder pressures and this is directly tied to cylinder temperatures. The conclusion is that further work must be done to find a good match of engine and turbochargers that enables higher boost pressures and lower process temperatures.

Sammendrag

2-trinns turbolader konseptet som brukes i denne masteroppgaven, er to separate turboladere koblet i serie. Luften går gjennom en lavtrykkskompressor, og en høytrykkskompressor, før den kommer inn på innsugingsventilene. Eksosen passerer gjennom både en høytrykks- og en lavtrykksturbin før den når eksosrøret. Dette konseptet tillater ladeluftkjøling mellom de to kompressortrinnene, i tillegg til ladeluftkjøling mellom høytrykkskompressoren og motoren.
Viktige begreper som Miller syklus, timing, og tap blir introdusert i begynnelsen for senere referanse. Også forventede effekter av 2-trinns turboladning med Miller timing er presentert. Litteraturgjennomgangen ser på hva konseptet 2-trinns turboladning er, og hva som er gjort tidligere innen dette feltet. Som et utgangspunkt er matching prosedyren for en 2-trinns turbolader presentert i korte trekk. Betydningen av et godt spalteforhold mellom trykkforhold på lavtrykks- og høytrykkskompressoren er funnet å være omkring 2. En enkel matematisk modell av en 2-trinns turboladet motor er bygget i Matlab-kode, ved hjelp av enkle termodynamiske relasjoner. Koden benyttes for beregninger, og kan finnes i vedlegget. Ved å gjøre kvalifiserte antakelser og sette mange konstanter, kan modellen brukes til å illustrere en 2-trinns turbolader, og resultatene blir presentert i et kompakt format ment å gi grunnleggende informasjon som er nødvendig for å bestemme turbolader familie og modell.
I kapittelet om verifisering presenteres RR BEAS motoren. Dette er en 6-sylindret turboladet rekkemotor som kan kjøre på både tungolje (HFO) og marin gassolje (MGO). RR BEAS har skaffet testresultater fra den virkelige motoren, samt en GT-Power modell av samme motor utstyrt med en 1-trinns turbolader. GT-Power programvaren er kraftig og brukervennlig. Motormodellen var i god stand, men en ny NOx-måling ble etablert i eksosrøret som regner ut gjennomsnittsverdier. I tillegg ble NOx beregningene endret fra likevekts beregninger i sylinderen til å bruke en EngCylNOx mal som deretter ble tunet inntil tilfredsstillende NOx målinger ble oppnådd. Deretter ble motormodellen kontrollerert mot testdata fra den virkelige motoren, med tilfredsstillende resultater. Det neste skrittet var å bygge en 2-trinns turbolader i GT-Power. Dette ble gjort ved å bruke de tilgjengelige malene i GT-Power. RR BEAS har anskaffet de nødvendige kompressor og turbinkart i form av SAE-filer. Disse ble benyttet med hell i modellen. Ladeluftkjølerne ble modellert som "black boxes", noe som betyr at de ikke er laget som en eksakt kopi av en ekte en. Veggtemperaturen på rørene i kjøler blir satt nært ønsket lufttemperatur etter kjøler. Deretter er rørene i ladluftkjøleren gitt en svært høy varmeoverføringskapasitet. Dette tvinger ladelufttemperaturen ned mot den satte veggtemperaturen meget raskt. Nye løftkurver for inntaksventiler ble laget for Miller timinger 40-100 veivakselgrader. En ekstra 2-trinns turbolader modell ble også laget. Den var identisk med originalen, bare med økt effektivitet ved bruk av effektiviserings multiplikatorer i kompressor og turbin maler. Dette ble gjort slik at den kunne brukes som en referanse for hva som kan oppnås med enda høyere ladetrykk, enn det som følger av den opprinnelige 2-trinns turboladete modellen.

Den første simuleringen kjørte 2-trinns turboladeren på 100% av maximum continuous rating, og testet Miller timingene 40-100 med trinn på 10 veivakselgrader. Miller 70 ble funnet som en optimal timing for kombinert reduksjon av både BSFC og NOx. Deretter ble en sammenligning kjørt mellom en 1-trinns turbolader, 2-trinns turbolader, og den ideelle 2-trinns turbolader. De ble kjørt gjennom seks forskjellige belastninger, 110, 100, 75, 50, 25 og 10% av MCR. De to 2-trinns turboladerene ble begge kjørt med Miller 70, mens den 1-trinns turboladeren ble kjørt med sin standard Miller 40. Resultatene viste at 2-trinns turbolader modellen reduserte NOx-utslippene med rundt 1 g / kWh, og BSFC av rundt 2 g / kWh. Turbotrykket var over 7 bar for 100% MCR. Men den maksimale sylindertemperatur ble ikke redusert, og dette hindret sannsynligvis ytterligere reduksjon av NOx. Dette førte til troen på at turboladerne kanskje ikke er en perfekt match for denne motoren. Høytrykksturbinen fungerte sub-optimalt på en virkningsgrad rundt 70%, 10% mindre enn kompressorene og lavtrykksturbinen. Når man ser på den ideelle 2-trinns turboladeren, oppnås det en NOx reduksjon rundt 2g / kWh, og BSFC ble redusert med nesten 6 g / kWh. Dette bekrefter mistanken om at turboladerne og motoren ikke er en perfekt match. Det som skjer er at de ikke er i stand til å levere et høyt nok ladetrykk, derfor blir AF-forholdet redusert, og sylindertemperaturen går opp, begrenser NOx reduksjonen. Når man ser på energibalansen avslører den at varmeoverføringen kan reduseres betraktelig med lavere temperaturer. Motorfriksjonen reduseres med lavere sylindertrykk, og dette er direkte knyttet til sylindertemperaturer. Konklusjonen er at videre arbeid må gjøres for å finne en god kombinasjon av motor og turboladere, som muliggjør høyere ladetrykk, og lavere prosesstemperaturer.

Acknowledgements

I would like to direct a special thanks to my fiancée for her patience and support during my writing of this master thesis.

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And thanks to my supervisor at NTNU, professor Eilif Pedersen, for providing guidance.

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Nomenclature

Abbreviations

ABB	Asea Brown Boveri
BDC	Bottom Dead Center
BSFC	Brake Specific Fuel Consumption
CIMAC	International Council On Combustion Engines
KBB	Kompressorenbau Bannewitz
K2B	Knowledge to Boost
DF	Dual Fuel, refers to an engine capable of running both on gas and diesel
FPP	Fixed Pitch Propeller
IC	Intercooler
IVC	Inlet Valve Closure
SOI	Start Of Injection
SOC	Start Of Combustion
ST	Stage
TC	Turbocharger
VVT	Variable Valve Timing
BMEP	Brake Mean Effective Pressure
IMEP	indicated Mean Effective Pressure
MDO	Marine Diesel Oil
MGO	Marine Gas Oil
IFO	Intermediate Fuel Oil
HFO	Heavy Fuel Oil
CA	Crank Angle
CAD	Crank Angle Degrees
IVS	Inlet Valve Short
IVL	Inlet Valve Long
NO_x	Nitrogen Oxides
SO_x	Sulphur Oxides
EVO	Exhaust Valve Open
AF	Air-to-Fuel
FA	Fuel-to-Air
MCR	Maximum Continuous Rating
ppm	parts per million
IMO	International Maritime Organization
ECA	Emissions Control Area
SCR	Selective Catalytic Reduction
EGR	Exhaust Gas Recirculation
LP	Low Pressure
HP	High Pressure
CAE	Computer Aided Engineering

LHV	Lower Heating Value
RR BEAS	Rolls-Royce Bergen Engines AS
MARPOL	International convention for the prevention of pollution from ships, IMO
ANNEX VI	Prevention of air pollution from ships
IMO	International Maritime Organization
VCM	Valve Control Management
T	Temperature
IMT	Institute of Marine Technology
p	Pressure
K	Kelvin
i	Number of cylinders

Miscellaneous

T_{cw}	Temperature of cooling water
p_{me}	Mean effective pressure
$c_{p,a}$	Specific heat of air, for constant pressure
$c_{p,e}$	Specific heat of exhaust, for constant pressure
$\eta_{LP,C}$	Efficiency low pressure compressor
$\eta_{HP,C}$	Efficiency high pressure compressor
$\eta_{LP,T}$	Efficiency low pressure turbine
$\eta_{HP,T}$	Efficiency high pressure turbine
$\eta_{HP,TC}$	Efficiency high pressure turbocharger
η_{vol}	Volumetric efficiency
η_{th}	Thermal efficiency of engine
$\eta_{mech,TC}$	Mechanical efficiency turbocharger
ρ_a	Density of air
ζ_{HL}	Fraction of fuel energy of in-cylinder heat loss
ζ_{exh}	Fraction of fuel energy in exhaust
λ	The relative air-fuel ratio
k_a	Specific heat ratio of air
k_e	Specific heat ratio of exhaust gas
$p_{L,IC1}$	Pressure loss intercooler 1
$p_{L,IC2}$	Pressure loss intercooler 2
$p_{L,eng}$	Pressure loss engine
$p_{L,int}$	Pressure loss intake
$p_{L,exh}$	Pressure loss exhaust
ε_{IC1}	Efficiency of intercooler 1
ε_{IC2}	Efficiency of intercooler 2
$\dot{m}_a$	Mass flow of air
$\dot{m}_f$	Mass flow of fuel
$\dot{m}_e$	Mass flow of exhaust

$\dot{m}_{corr}$	Corrected mass flow
V_{sw}	Swept volume of cylinder
$V_{sw,\,Miller}$	Swept cylinder volume for a given Miller timing
$\dot{V}_a$	Volumetric flow rate of air
n_{cyl}	number of cylinders
E_b	Energy balance
Mr	Miller degree factor, number between 0 and 1
N	RPM, revolutions per minute
n_a	Engine speed, revolutions per second
$\dot{\omega}_a$	Angular speed, radians per second
π	Pressure ratio
$\pi_{LP,C}$	Pressure ratio low pressure compressor
$\pi_{HP,C}$	Pressure ratio high pressure compressor
$\pi_{LP,T}$	Expansion ratio low pressure turbine
$\pi_{HP,T}$	Expansion ratio high pressure turbine
F	fuel-air equivalence ratio
P_e	Brake power
T_e	Brake torque
BS_{NOx}	Brake specific NO_x production, [g/kWh]
EC_{NOx}	Concentration of NO_x in exhaust gas, [ppm]
M_{NOx}	Molecular weight of NO_x, [kg/kmol]
M_{exh}	Molecular weight of the exhaust gas, [kg/kmol]
h_n	Lower heating value of fuel
F_g	Gas forces
$Miller_{CAD}$	Miller timing in crank angle degrees
P_{fuel}	Fuel Power
P_{exh}	Exhaust Power
P_{fr}	Friction Power
P_{shaft}	Shaft Power

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1 Introduction

1.1 Background

There is a finite amount of fossil fuels available in the world and the remaining reservoirs are increasingly difficult to explore. There is also an increasing energy demand especially from emerging economies. These factors are likely to increase the price of fossil fuels in a long term perspective. In this thesis the fuel considered is diesel oil of different viscosities. Already now, with the fuel prices of today, bunker oil accounts for a big portion of the shipping companies operational expenses.
In the Emission Control Areas, ECA`s, defined by IMO, new and stricter regulations of sulphur, SO_x, emissions will take effect from January 2015. These demands will be tough to meet for a big portion of the commercial fleet which runs primarily on HFO. They will have four choices. They can install expensive exhaust aftertreatment equipment, typically scrubber units which can remove a sufficient amount of SO_x from the exhaust. They can run the engines on low sulphur MGO which meets the SO_x emission regulation but retails at almost twice the price of HFO, (Bunkerworld, 2014). Switching to natural gas as a fuel will satisfy emissions regulations but involves a high investment cost coupled with substantial infrastructure issues. The last option is somewhat uncertain but considers low sulphur IFO or HFO when this becomes widely available. If it does however become available, the price will most certainly increase compared to the same fuels with higher SO_x content. The scrubber and natural gas alternatives will increase investment costs. Low sulphur fuels will increase operational costs. This serves to illustrate that the importance of operating costs in shipping is only increasing.

The upcoming and stricter regulation also covers NO_x emissions. In ECA the reduction from the limit of today is around 80 % but depends on the engine RPM, see figure 4. To meet the new regulations will be tough. The origin of NO_x is primarily connected to two sources. Thermal NO_x is formed in the cylinder and is strongly dependent on the combustion temperature. Especially heavier fuels can also have a NO_x content in the fuel itself. The former is the dominant source at least for medium speed engines, (Heywood, 1988). In addition the engine speed affects thermal NO_x directly through the time available for NO_x formation. There are several options for reducing NO_x emissions. Internal EGR retains more of the exhaust gas during the scavenging sequence. The retained exhaust gas acts like an inert gas in the combustion process and its chemical composition has a higher specific heat value. This forces the combustion temperature down, reducing the production of thermal NO_x. External EGR works by the same principles, but extracts the exhaust gas after the cylinder, it then cool it in a heat exchanger before it is introduced into the cylinder. The cooled exhaust gas is even more effective at reducing the combustion temperature and hence the thermal NO_x. Water injection into the cylinder serves the same purpose, the high specific heat of water along with the high phase change energy demand cools down the combustion process which reduces thermal NO_x. All of these options reduce thermal NO_x production by reducing the combustion temperature, (Johnson, 2006), (Register, 2012).

To increase the power density of an engine means that you will produce the same amount of power from an engine with a smaller cylinder volume. There are two key benefits to this strategy, reduction in engine size, which means lower net weight and reduced spatial requirement, which in turn leads to reduced fuel consumption. A smaller engine providing the same output will operate at higher specific loads enabling higher efficiency. The friction losses are reduced when then engine size goes down which also lowers fuel consumption.

The purpose of this master thesis is to investigate the effects of 2-stage turbocharging on engine performance. The main advantage of a 2-stage turbocharger over a conventional one is the ability to produce a significant increase in boost pressure. However, only increasing the boost pressure will not reap many benefits. In 1957 an American engineer, Ralph Miller, patented a concept with a supercharged engine equipped with variable inlet valve timing and increased intercooling, hence the often used name Miller timing or Miller cycle. It is the combination of increased intercooling and variable valve timing with high boost pressures that enables the potential to be fully exploited.

The reason behind looking at 2-stage turbocharging for delivering the high boost pressures is the limitations of the single stage turbochargers. The limiting factor for boost pressures that can be achieved using a single stage turbocharger is mostly the compressor, (Watson and Janota, 1982). The reason is mainly comprised of the following factors. The efficiency at very high pressure ratios is reduced. The increased pressure ratio lowers surge margin which reduces the possible map width. The temperature of the air rises with increasing pressure ratio which at some point will pass the temperature limit of cheap cast aluminium impellers. This forces the use of heavier and more expensive materials to be used for impellers. A 2-stage turbocharger uses two turbochargers in series which means two compressors and two turbines. Since the pressure ratio is split between two compressors each of them only needs to provide moderate pressure ratios. The more favourable conditions results in higher surge margins providing greater map width. The lower temperature enables aluminium impellers and higher efficiencies can be achieved. The use of two stages enables the use of intercooling between the compressors. The increased density makes it possible to shrink the size of the HP compressor. The reduced temperature means less power is required for the HP compressor to achieve the same pressure ratio. The speed range each compressor must operate in is reduced compared to a single stage and this helps efficiency at low speeds. There are also some benefits regarding the turbines. If the HP turbine is approaching its choking point at high expansion ratios it runs at nearly constant conditions and can`t provide additional power. However, any additional exhaust energy can be utilized by the LP turbine for further expansion, increasing pressure ratio of the 2-stage turbocharger more. The point being, energy lost at the HP turbine can still be recovered in the LP turbine as opposed to a single stage turbocharger where this energy is lost to the environment,(Baines, 2005).

Variable inlet valve timing refers to the concept of adjusting the closing time of the intake valve. This makes it imperative to increase the boost pressure to maintain the AF-ratio at the given load. If you close the intake valve before BDC you need a higher pressure to get the same mass of air into a smaller cylinder volume. If you close the intake valve after BDC you need a higher boost pressure because the same amount of air needs to be delivered as with normal valve timings. Modern application of the latter method is among others in the engine of Toyotas hybrid car, Prius. The former method is used for diesel engines with a high degree of Miller timing and is the concept used in this thesis.

When combining 2-stage turbocharger with Miller timing and additional intercooling it provides a potent package for engine improvements. It has the potential to improve all the three key factors mentioned in the beginning of this chapter, reduced fuel consumption, reduced NO_x emissions and increased power density.

The work done by the pistons during the cylinder compression stroke is negative in the sense that it reduces power output or increases fuel consumption. If mass of fuel and AF-ratio are kept constant by increasing boost pressure, the Miller timing can be increased gradually by closing the inlet valve earlier. This moves more of the compression work from the pistons to the turbocharger which harnesses the energy of the exhaust gas to provide higher boost pressures. In effect, you take advantage of the otherwise wasted energy or free energy if you like. Closing the inlet valves early can be described as shortening the effective compression stroke. In addition to the mentioned effect, the air expands on the pistons way down to BDC which cools the air down and the pistons gets an air spring effect on its way up from BDC. The cooling of the air during the intake stroke also affects the compression and expansion stroke with lower temperatures and therefore also lowered pressures. In addition you have some secondary effects of lower temperatures. The heat loss is reduced. Lowered specific heats at the start of compression means less energy is needed to increase pressure and temperature. Thermal NOx emissions are exponentially dependent on maximum temperatures in the cylinder, so any reduction in temperature gives reduction in NOx emissions.

When the turbocharger performs more of the compression work coupled with reduced heat loss and specific heats, these are factors that reduce BSFC. The lowered temperature forces NOx emissions down. Lower temperature gives lower pressures for the same power output which can be used for increasing power density. Either by increasing the mass of fuel injected to increase power output of the existing engine. Or by reducing the engine size and maintain the original pressures to achieve the same output power.

Reducing NOx and BSFC at the same time is possible up to a certain point of increasing Miller but then BSFC will start to increase while NOx emissions are reduced further. If lowered temperatures are used to improve power density it will reduce the potential of the other two key factors. Like previously stated it is possible to improve all three key factors, however it is not so easy to achieve all three at the same time. Which effect that is strongest depends on how the engine is tuned.

1.2 Objectives

The main objective of this master thesis is to investigate the potential benefits of combining a 2-stage turbocharger with the Miller cycle on a marine diesel engine. Two main parameters are to be evaluated, the potential for lowered BSFC and the potential for lowered NO_x emissions. A third parameter could be the potential for increasing power density, but this is of less importance and will only be briefly mentioned.

- Explain important concepts
    - 2-stage turbocharging
    - Miller cycle
- Conduct a literature review on 2-stage turbocharging
- Do a turbocharger matching procedure using a mathematical model
- Verify the accuracy of the GT-Power engine model against test data
- Build the 2-stage turbocharger
- Design Miller timing lift curves for the intake valves
- Run simulations with the 2-stage turbocharged engine model and judge the success mainly on the two parameters BSFC and NO_x

1.3 Important Concepts

Here some concepts that are to be used in the thesis will be introduced. This is to provide the reader with a better overview. It can also function as a reference where it is possible to look up a concept encountered later in the text.

Miller Cycle

Ralph Miller was an American engineer that developed the concept called Miller cycle for which he holds a US patent from 1957. The Miller cycle is defined by the following traits:

- Increased turbocharging for higher boost pressures: Can be achieved by better and more efficient turbochargers or by using a 2-stage turbocharger.
- Increased charge air cooling: With higher pressures comes higher temperatures, when using a 2-stage turbocharger it opens the possibility of a second turbocharger between the two compressor stages.
- Variable valve timing, VVT: This is an important part of the Miller cycle, the inlet valves are closed before the cylinder piston reaches bottom dead center, BDC. This provides several benefits, lowered pressure and temperature inside the cylinder are examples of this.
- Effective compression stroke is shortened while expansion stroke is unchanged: When the intake valves are closed earlier than BDC, the piston still continues down to BDC expanding the gas causing the temperature to decrease. The piston acts like a gas spring and will use little energy for the compression from BDC and up to the point where the intake valves closed, this saves energy, the pumping work is reduced.

If a Miller cycle is to be used on an engine it is very important that the boost pressure is increased, if not it would reduce the amount of air available for combustion, effectively decreasing the AF-ratio. The original Miller cycle concept included a mechanism that changed the inlet valve timing automatically as a function of boost pressure. This was important to avoid an insufficient amount of air at low engine loads.

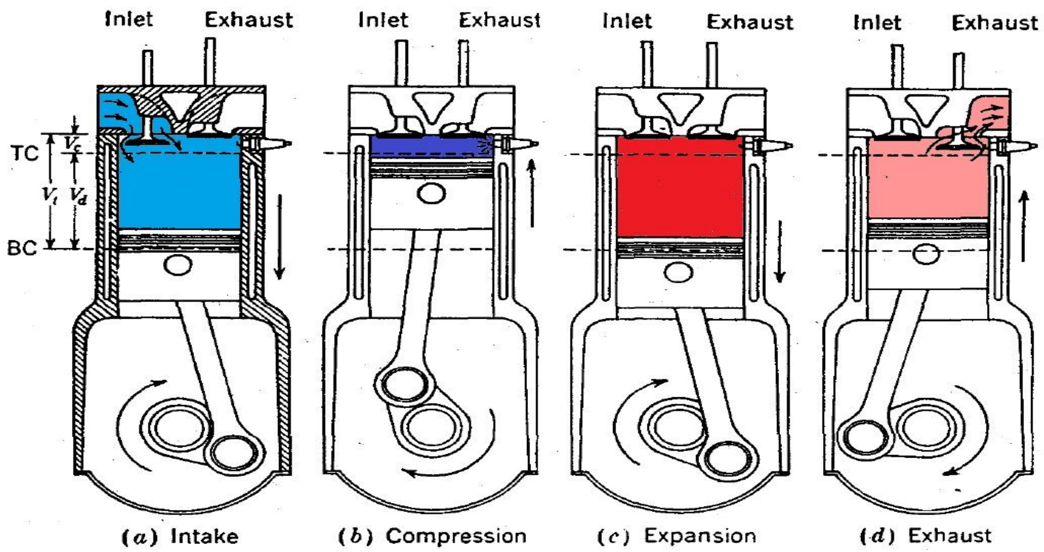


Figure 1: 4-stroke cycle

Figure 1 shows the 4-stroke engine cycle, it makes it easier to see how the compression stroke would be shorter while the expansion stroke remains unchanged. And how cylinder volume would shrink making increased boost pressure a necessity to maintain the same mass flow of air. When referring to a Miller cycle in this thesis, it means the concept explained here in this chapter.

Miller Timing

The concept of Miller timing refers to the variable inlet valve closing which is part of the Miller cycle. The normal way to present Miller timing is in crank angle degrees, CAD. Figure 2 illustrates the connection between Miller timing, CAD and the piston inside the cylinder. On the right side, the piston stroke is defined from top dead center, TDC, and down to BDC. At the bottom is an illustration of the cross section of the crankshaft, showing the crank throw and marked with CAD. From this figure it becomes obvious that the intake stroke would take 180 CAD. As an example it could be stated that Miller 40 gives this and that effect. Miller 40 means that the inlet valve closes 40 CAD before the piston reaches BDC of the cylinder. This means that the inlet valve would close at, 180 – 40 =140 CAD, after TDC. One can easily convert the Miller timing into degree of swept cylinder volume by using formula (1) defined below.

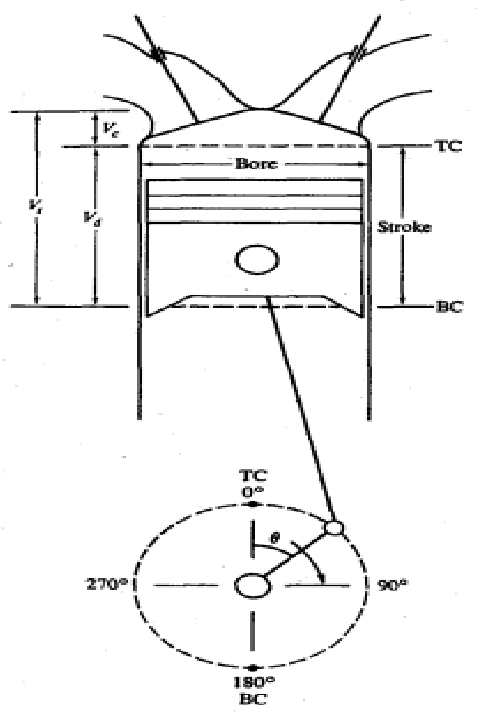


Figure 2: Miller timing

$$V_{sw,Miller} = \frac{Miller_{CAD}}{180} \cdot V_{sw} \tag{1}$$

Formula 1 can be used for calculating the mass flow of air in an engine with Miller timing.

2-Stage Turbocharging

The 2-stage turbocharger concept in this master thesis is called exactly that because it uses two turbochargers coupled in series. The result is 2-stages of compression and 2-stages of expansion. See figure 3 for the general layout of the 2-stage turbocharger connected with the engine.

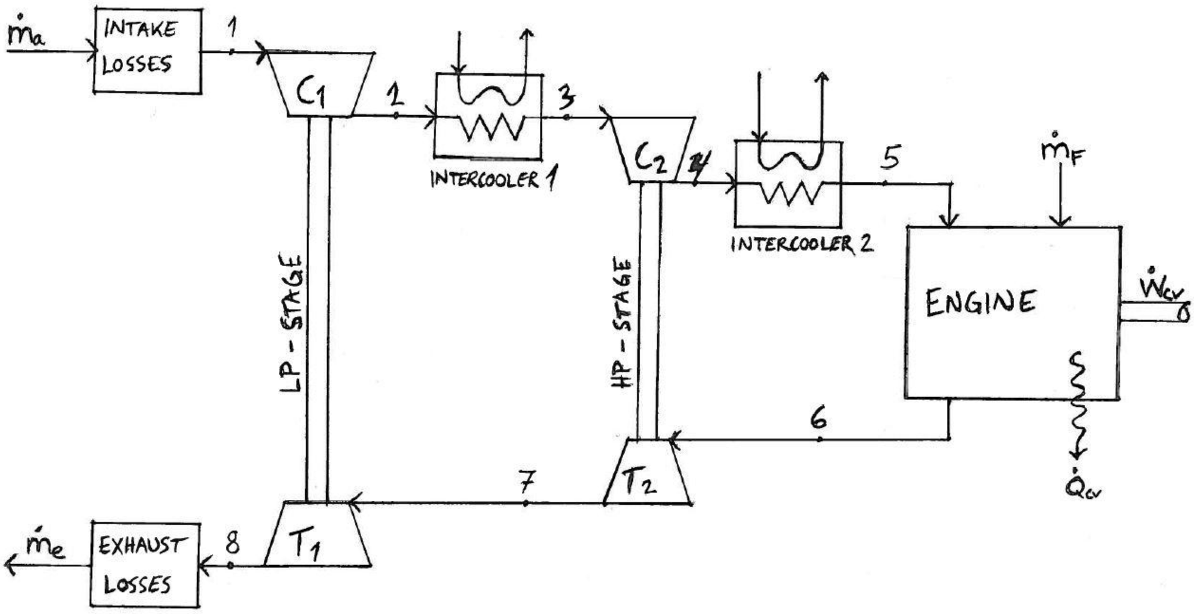
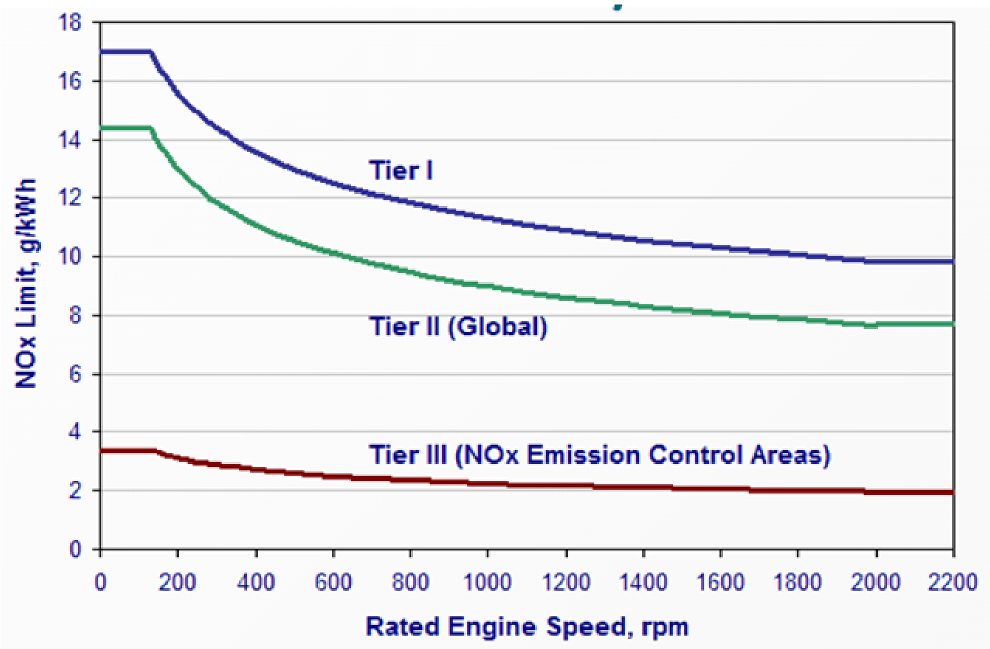


Figure 3: Layout of 2-stage turbocharger

Both turbochargers are of the conventional type where compressor and turbine are connected by a shaft inside the turbocharger housing. This is practical as you in theory could couple any two commercially available turbochargers together in this way. In real life the spatial and weight requirements for on-engine applications would hinder the use of many turbochargers. In this master thesis the high pressure, HP, turbocharger is customized for 2-stage applications. So the basic design of the 2-stage turbocharger is, 2 turbochargers coupled in series, intercooler between the two compressors, a second intercooler after the HP compressor, intake and exhaust losses are defined in case there are any. So what can 2-stage turbocharging offer that is better than using a conventional 1-stage turbocharger:

- Increased boost pressure
- Possibilities for increased intercooling
- Higher efficiencies for each of the two turbochargers since the pressure ratio each has to deliver is relatively small
- Possibility for greater compressor and turbine map widths because of the low individual pressure ratio of each turbocharger

Emissions Regulation



Tier	Date	NOx Limit, g/kWh		
		$n < 130$	$130 \le n < 2000$	$n \ge 2000$
Tier I	2000	17.0	$45 \cdot n^{-0.2}$	9.8
Tier II	2011	14.4	$44 \cdot n^{-0.23}$	7.7
Tier III	2016†	3.4	$9 \cdot n^{-0.2}$	1.96

† In NOx Emission Control Areas (Tier II standards apply outside ECAs).

Figure 4: IMO regulation on NOx emissions

The plot and table in figure (4) is taken from IMO`s MARPOL 73/78 Annex VI on air pollution and sets the limits of allowed NO_x emissions from ships. The limit is a function of RPM because the slower engine speed allows more time for NO_x production. For all cases and simulations in this master thesis, the engine model will be run at a constant speed of 1000 RPM. According to figure (4) this gives the following requirement for the NO_x emissions of tier II: NO_x emissions < 9 g/kWh

Maximum Continuous rating, MCR

Every engine has a defined MCR. It is the highest load at which the engine can be run continuously for longer periods of time. In table (1) the defined engine loads used in the simulations chapter are given as percentage of MCR.

Case	BMEP	Power, kW	% of MCR
1	27.16	2200	110
2	24.69	2000	100
3	18.52	1500	75
4	12.35	1000	50
5	6.17	500	25
6	2.47	200	10

Table 1: Percentage of MCR

Miller Loss

When Miller timing is increased more and more the potential gains can be reduced because of Miller loss. In figure 5 the area marked by the number 1 is cut from the closed cycle. The area marked by the number 2 is cut from the gas exchange cycle.

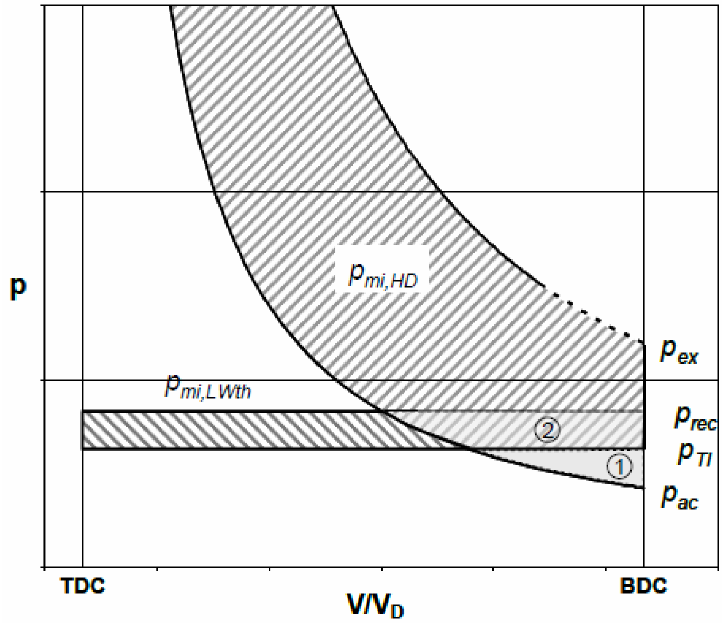


Figure 5: Miller loss 1

In figure 6 the flow losses are also included and the theoretical gas exchange work is shown, this is the total gas exchange work that is available before the losses are included. This last figure is somewhat better explained. What one must remember is that the area marked by number 2 in figure 5 is not removed from the closed cycle, only from the gas exchange cycle, this can be a bit confusing when looking at it for the first time.

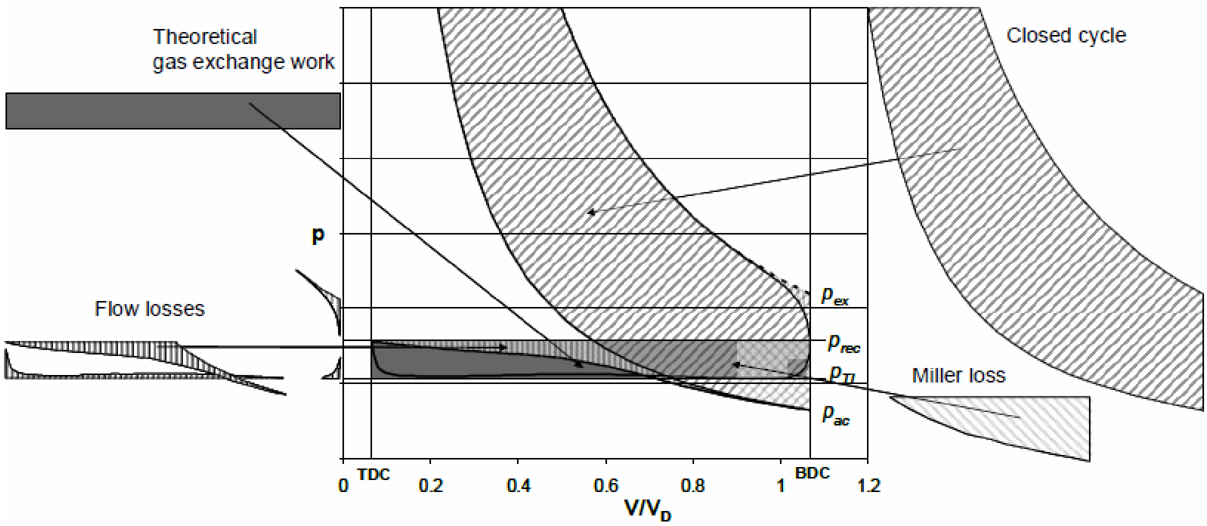


Figure 6: Miller loss 2

The Miller loss demonstrates that nobody is perfect. It is not possible to increase Miller timing indefinitely without any penalty. If running a simulation and the pumping work has a nice curve rising with increased pressure into positive values and then starts to decrease for even higher pressures, it can be the effect of Miller loss.

Expected Results of Combining 2-Stage Turbocharging with Miller Cycle

- Increased boost pressure
- Lower maximum cylinder temperature
- Lower maximum cylinder pressure
- Lower brake specific fuel consumption, BSFC
- Lower NO_x emissions as a function of the lowered maximum temperature in the cylinder
- Possibility for increased power density due to the lowered maximum pressure in the cylinder

1.4 Structure of the Thesis

This shows the layout of the master to provide the reader with a nice overview from the beginning.

1. Introduction
2. Literature Review
3. Turbocharger Matching
4. Verification of the Engine Model
5. Engine Model With 2-Stage Turbocharger
6. Simulations
7. Conclusion

2 Literature review

The literature review will present the different key concepts needed in this master thesis. It will include the reference literature for the different concepts.

2.1 Background

The sources listed in this chapter can be read as an introduction to the concept of 2-stage turbocharging. They contain information in different forms about 2-stage turbocharging, the theory of it, simulations and engine tests. This chapter also covers an overview of what others have done already in this field.

There have been several studies on 2-stage turbocharging in recent years, both on an academic level, by engine manufacturers and turbocharger manufacturers. There have been studies comparing 2-stage turbocharged engine models with a real 2-stage turbocharged engine. Wärtsila has been active in this way, publishing several CIMAC conference papers on the matter. When combining 2-stage turbocharging which provides high boost pressures with Miller timing, good results have been achieved, Reduction in BSFC and NO_x emissionss have been documented, (Christer Wik, 2007) Another study started out by verifying the accuracy of a 1-stage turbocharged Wärtsila W20:28L6 engine against a simulation model of the same engine. This was done to predict the general accuracy of simulation models. Then a 2-stage turbocharged model was built in the same software. This model was then used for extensive testing of valve timing, both intake and exhaust valves, different valve overlap was also investigated. Miller timings up to Miller 100 was tested and when tuned for maximum NO_x reduction, it achieved 50 % NO_x reduction with only moderate BSFC increase, (Frederico Millo, 2010). Wärtsila has also taken the next step and investigated different designs of 2-stage turbochargers in order to integrate them onto the engine block of a W20 engine without unacceptable vibration levels. Several options were explored and connecting two complete turbochargers were the chosen solution. Some of the air ducting was strengthened and the design was optimized with regards to flow. Targets for vibration levels, BSFC and NO_x reduction were all reached, (Tero Raiko, 2010). A study of the important concepts of 2-stage turbocharging was carried out by ABB. They looked at requirements to compressors and the pressure ratio split between them, size of turbines and interaction between the two of them, turbocharger design and turbocharger efficiency. The main conclusion was that 2-stage turbocharging was a viable solution for many engine configurations, (Adrian Rettig, 2010).

ABB has developed a second generation of its 2-stage turbocharging Power2 system. It has four system sizes, delivers pressure ratio up to 12, new extended cartridge design gives down times that match or beat those of conventional turbochargers, new turbine stages design and new LP compressor design, (Thomas Behr, 2013). MAN has developed its own solution for 2-stage turbocharging, called TCX. It is built in a similar way as the ABB Power2 system with a very compact design consisting of two separate turbochargers one LP and one

HP coupled in series, with intercooling between compressors and after HP compressor. The design and performance targets were reached, (Jiri Klima, 2013)

ABB has conducted several studies on the use of 2-stage turbocharging with Miller timing on gas engines and also Dual Fuel, DF, engines. The results have been very promising. For DF engines it was possible to increase the power density so much that the enlarged bore diameter could be reduced to the level of diesel engines of the same size, and improving efficiency while doing so. For the gas engines it showed how the process temperature could be lowered enough to shift the knock limit significantly. This in turn will enable high mean effective pressures and compression ratios, improving efficiency, (Claudio Christen, 2013), (S. Vögelih, 2009). A study conducted by Tokyo Gas confirmed that using the Miller cycle on gas engines has the potential to improve either power density or efficiency significantly, IMEP could be raised to 20 bar and still avoid engine knocking because of the lowered temperatures, (Satoshi Shimogata, 1997).

A few smaller articles published online are also worth mentioning as they represent a readable and quick introduction to the ongoing development. The new Power2 design from ABB is capable of delivering pressure ratios up to 12 and efficiencies over 75% on medium speed engines. Coupled with extreme Miller timings, this system has actually been able to achieve a 60 % reduction of NO_x emissionss in comparison with the 1-stage turbocharger systems. Kompressorenbau Bannewitz, KBB, is developing its own 2-stage turbocharger to meet IMO Tier III, with pressure ratios of 6-10, the program is called Knowledge to Boost, K2B. BorgWarner has admitted that up until now 1-stage turbocharging has been enough, but now the needs for higher boost pressure and power density will make it necessary to look at 2-stage turbocharging systems, (Motorship, 2014). In a special report called dancing with the dragon, which highlights ABB`s cooperation with the Chinese shipbuilding industry, a nice review of the advantages of 2-stage turbocharging is presented. The emphasis is on how the maximum cylinder temperature is a very important factor for improving engine efficiency, through the lowered specific heats and lowered heat loss, (Codan, 2013)

2.2 Miller

The inventor of the Miller cycle, Mr. Ralph Miller himself has published a conference paper on his patented Miller cycle. This paper treats the supercharging of engines combined with variable valve timing and additional charge air cooling. It explains how this concept has allowed up to 20 % increase in power output of both compression and spark ignition engines without increasing the thermal and mechanical loads or changing the AF-ratio, (Ralph Miller, 1957). The Miller cycle was put to the test by several manufacturers after it was introduced, Brown Boveri ran an extensive testing of the concept, finding it possible to increase mean effective pressures without increasing either thermal load nor fuel consumption noticeably. They found the potential for improvements when running a fixed-pitch propeller 4-stroke engine at part load to be even more important than the improvements at full load. This improvement was obtained by avoiding air shortage at part load because variable valve timing was used, (E. Meier, 1977).

There are two very good articles from ABB Turbo Systems that dive into the details and the more technical aspects of the Miller cycle. The first article looks at the basic principles of the Miller cycle, treating turbocharger efficiency, early and late closing of inlet valves, scavenging on a Miller cycle engine, looks at Miller on both 1-stage and 2-stage turbochargers, looks at the pressure ratio split between the compressors of a 2-stage turbocharger and it includes some test data, (E.Codan, 2006). The next article treats some of the same concepts and some new, but is very detailed. It looks at, the thermodynamics of the turbocharged engine, the closed cycle, the gas exchange process, very detailed explanation of the Miller loss, the charging process, the exhaust process, pulse turbocharging and turbocompounding. It also describes different applications and looks at the requirements on the turbocharging system, pressure ratio split between the two compressors of a 2-stage turbocharger, compressor development and availability, (E. Codan, 2012).

The possibilities offered by Variable Valve Timing, VVT, regarding engine performance, transient behavior and emissions are many. ABB has a VVT design under development that can be used on both diesel and gas engines and it is even possible to retrofit, the system prototypes are ready for testing, (Mathey, 2010). The fully developed VVT system called Valve Control Management, VCM, is an electro-hydraulic variable valve timing system which allows the full potential of 2-stage turbocharging to be reached, (Mathey, 2013)

3 Turbocharger Matching

It is essential to find a good match of turbocharger and engine and this is even more complicated with a 2-stage turbocharger since it has two separate turbochargers. This chapter will address the matching procedure with the following layout:

- How to split the pressure ratio between two compressors
- Calculations using mathematical model
  - How the model works
  - Assumptions
  - Formulas used in model
  - Layout of 2-stage turbocharger
  - Constants used in the model
  - Case setup with variables
  - Critical values
  - Tabulated results
  - Discussion of results
- Turbocharger selection
  - Formulas
  - Parameters for turbocharger selection
  - Choosing the turbocharger model
  - Choosing the turbocharger build
  - Discussion

The chosen setup of turbochargers is to connect two turbochargers in series. In this way it is possible to match any two independent turbochargers with each other. Each turbocharger is complete and could be used on its own for a different application. The layout is presented in figure 8.

In lack of a complete and detailed turbocharger catalogue the matching procedure will establish important parameters of a functional 2-stage turbocharger. This will be done by making several assumptions. The result will be a guideline towards choosing the correct turbocharger build. By establishing the pressure ratios and volumetric flow it will be possible to select the appropriate turbocharger family. Corrected mass flows will also be calculated. But without detailed compressor and turbine maps from the manufacturer, the final selection of individual compressors and turbines is not treated in this master thesis.

The calculations are performed using a Matlab code provided in appendix 9.1. The reason for presenting the calculations in tabulated form is to make the matching procedure as compact as possible and easier to read.

Splitting the Pressure Ratio Between Two Compressors

The pressure ratio split between the two compressors must be set. This is an important step when utilizing a 2-stage turbocharger and must be presented here to increase understanding on the subject. A good starting point for obtaining an optimal turbocharging efficiency is to distribute the pressure ratios evenly between the two compressors.

$$\pi_{HP,C} = \pi_{LP,C} = \sqrt{\pi_{system}} \tag{2}$$

While this strategy does ensure an optimum efficiency it also requires quite large turbocharger stages. This can be a problem in many applications. A possible solution is to increase the pressure ratio of the low pressure compressor. This will have the following effects:

- LP compressor shrinks in size and delivers the same mass flow at a higher pressure
- LP turbine shrinks in size and produces more energy from the same mass flow
- HP compressor might shrink a little bit in size but is already small
- HP turbine may increase a bit in size to deliver less energy from the same mass flow but is still small

This illustrates that the standard approach to achieve high efficiency must be adjusted in some way, (E.Codan, 2006)

The optimum ratio between LP compressor and HP compressor is dependent on the degree of intercooling between them. But how do we find the optimal ratio that also minimizes the size of the LP and HP turbine? Figure 7 illustrates this by showing the turbocharging efficiency on the left y-axis, η_T, the effective turbine area on the right y-axis, S_{effT}, and the ratio of the LP/HP turbine pressure ratios on the x-axis.

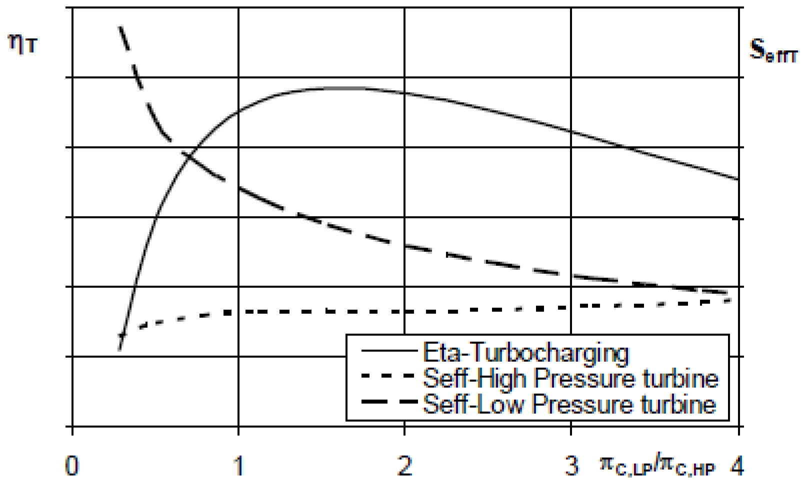


Figure 7: Optimal pressure ratio split between LP and HP compressor for high TC efficiency

An optimum split in pressure ratio between the two compressors can be found around a value of 2, with only a slight reduction in turbocharger efficiency but a significant reduction in the necessary effective turbine area. This ratio ensures a reduction in turbine size, (E. Codan, 2012)

$$\frac{\pi_{LP,C}}{\pi_{HP,C}} \approx 2 \tag{3}$$

3.1 Mathematical Model

The Matlab code in appendix 9.1 represents a mathematical model that uses equations to calculate pressure, temperature, density and mass flow. This model follows the representation of a 2-stage turbocharger found in figure 8.

- It is a first step in a matching procedure where it is possible to use assumed values and check the results
- It is used to make a match at 100 % MCR only, not for all loads, but if a good enough surge margin is selected the match can be sufficient for all loads
- It follows the numbered route through the turbochargers and engine which is shown in figure 8.
- It is necessary to assume efficiencies and losses of the turbochargers, intercoolers and engine.
    - If good assumptions are made the model values can be close enough to predict trends correctly
- It uses simple thermodynamic relations to calculate the temperatures and pressures in each step.

Brief explanation of how the model works

- Efficiencies are set for both compressors and intercoolers
- Cooling water temperatures are set
- Pressure and temperature is calculated all the way from intake and across the engine
- The mass flow of fuel is set and the mass flow of air is calculated
- HP turbine efficiency and HP TC mechanical efficiency is set
- Then it uses an equation for the energy balance of the HP TC to determine the pressure out of the HP turbine, afterwards the temperature can be calculated
- For the LP TC the necessary pressure out is known because it is simply ambient pressure plus the losses in the exhaust piping, the mechanical efficiency of the LP TC is assumed and the variable becomes the LP turbine efficiency, afterwards the temperature can be calculated
- In short this means you assume necessary values and use the efficiency of the LP turbine as a form of quality control
- The model enables a quick analysis of what performance you need from the different components to achieve your targets

Assumptions

Several assumptions had to be done before calculating this mathematical model.

- The matching procedure is performed for the fuel flow at 100 % MCR in the original 1-stage turbocharged engine and the fuel flow is constant for all cases
- Values inspired by results from 2-stage turbocharged engine model in GT-Power, chapter (6): $c_{p,e}$, ζ_{HL}, $p_{L,IC}$, $p_{L,eng}$, η_{th}, ε_{IC}, Mr, $\eta_{LP,C}$, $\eta_{HP,C}$, $\eta_{HP,T}$
- The values for k_a and k_e are average values taken from a ABB training document (Bernard, 2006)

- The exhaust losses are taken as the maximum allowed backpressure for exhaust in the RR BEAS engine project manual
- The AF-ratio in the model uses the air trapped in the cylinder, not the total air flow into the cylinder, it accounts for the trapping ratio
- The Miller degree, Mr, defines the percentage of cylinder filling based on when the intake valve closes. It is defined for Miller 70 which means the intake valve closes 70 CAD before BDC, this is done to avoid excessive AF-ratios and to be more realistic.

Formulas of the model

$$T_2 = T_1\left\{1+\frac{1}{\eta_{LP,C}}\left[\left(\frac{p_2}{p_1}\right)^{\frac{(k_a-1)}{k_a}}-1\right]\right\} \tag{4}$$

$$p_3 = p_2 - p_{\mathrm{L,IC1}} \tag{5}$$

$$T_3 = T_2(1-\varepsilon_{IC1})+\varepsilon_{IC1}T_{cw} \tag{6}$$

$$T_4 = T_3\left\{1+\frac{1}{\eta_{HP,C}}\left[\left(\frac{p_4}{p_3}\right)^{\frac{(k_a-1)}{k_a}}-1\right]\right\} \tag{7}$$

$$p_5 = p_4 - p_{\mathrm{L,IC2}} \tag{8}$$

$$T_5 = T_4(1-\varepsilon_{IC2})+\varepsilon_{IC2}T_{cw} \tag{9}$$

$$\dot{m}_a = \frac{N}{2}\eta_{vol}\rho_a V_{sw} n_{cyl} Mr \tag{10}$$

$$\dot{m}_e = \dot{m}_a + \dot{m}_f \tag{11}$$

$$\zeta_{exh} = 1-\zeta_{HL}-\eta_{th} \tag{12}$$

$$T_6 = \left[\frac{\frac{m_a}{m_f}}{\left(\frac{m_a}{m_f}\right)+1}\right]\left(\frac{c_{pa}}{c_{pe}}\right)T_5 + \frac{\frac{LHV\cdot\zeta_{exh}}{c_{pe}}}{\left(\frac{m_a}{m_f}\right)+1} \tag{13}$$

$$p_6 = p_5 - p_{\mathrm{L},eng} \tag{14}$$

$$\eta_{HP,TC} = \eta_{HP,C}\eta_{HP,T}\eta_{mech,TC} \tag{15}$$

$$p_7 = p_6\left[\left(1-\frac{\left[\left(\frac{p_4}{p_3}\right)^{\frac{(k_a-1)}{k_a}}-1\right]c_{pa}T_3}{c_{pe}T_6\left(1+\frac{m_f}{m_a}\right)\eta_{HP,TC}}\right)^{\frac{k_e}{(\mathrm{k}_e-1)}}\right] \tag{16}$$

$$T_7 = T_6\left(1-\eta_{HP,T}\left[1-\left(\frac{p_7}{p_6}\right)^{\frac{k_e-1}{k_e}}\right]\right) \tag{17}$$

$$\eta_{LP,T} = \frac{\left[\left(\frac{p_2}{p_1}\right)^{\frac{(k_a-1)}{k_a}}-1\right]c_{pa}T_1}{\left[1-\left(\frac{p_8}{p_7}\right)^{\frac{(k_e-1)}{k_e}}\right]c_{pe}T_7\left(1+\frac{m_f}{m_a}\right)\eta_{LP,C}\eta_{mech,TC}} \tag{18}$$

$$T_8 = T_7\left(1-\eta_{LP,T}\left[1-\left(\frac{p_8}{p_7}\right)^{\frac{k_e-1}{k_e}}\right]\right) \tag{19}$$

$$\pi_{LP,C} = \frac{p_2}{p_1} \tag{20}$$

$$\pi_{HP,C} = \frac{p_4}{p_3} \tag{21}$$

$$\pi_{split} = \frac{\pi_{LP,C}}{\pi_{HP,C}} \tag{22}$$

$$\text{AF-Ratio} = \frac{\dot{m}_a}{\dot{m}_f} \tag{23}$$

$$\lambda = \frac{(A/F)_{actual}}{(A/F)_{stoichiometric}} \tag{24}$$

$$F = \frac{1}{\lambda} \tag{25}$$

Layout

There are two separate and complete turbochargers with compressor, turbine, bearings, shaft and housing. There are two separate intercoolers, one after each compressor. The numbers in figure 8 show the route through the system, this is the sequence used in the calculations of the matching procedure.

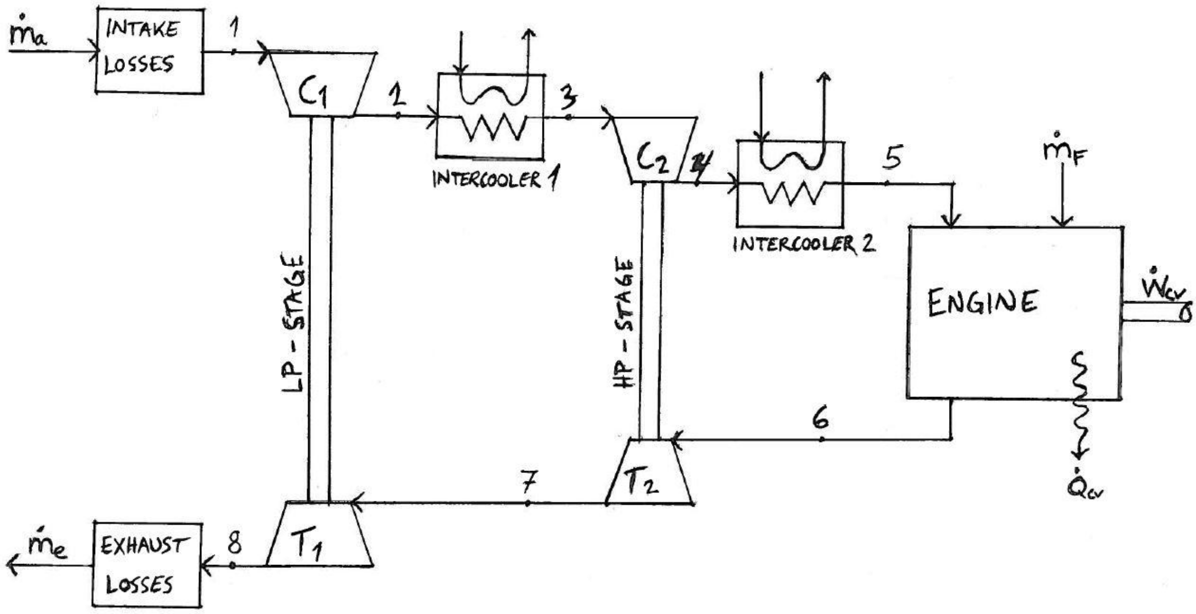


Figure 8: Layout of the 2-stage turbocharged engine, reference project report

Constants

The assumed values of the model are presented as constants in table 2. The constants control the model. To change the behavior of the model the constants must be changed, it is not a dynamic model but a quick way to make a prediction of performance for a given load.

	Value	Unit		Value	Unit
$\eta_{mech,TC}$	0.97	-	$\dot{m}_f$	397	kg/h
k_a	1.4	-	LHV	42.7	MJ/kg
k_e	1.34	-	$p_{L,int}$	0.001	bar
p_8	1.03	bar	$p_{L,exh}$	0.03	Bar
η_{vol}	0.927	-	$c_{p,a}$	1.008	kJ/kgK
ζ_{HL}	0.08	-	$c_{p,e}$	1.16	kJ/kgK
ζ_{exh}	0.483	-	M_a	28.97	Kg/kmol
η_{th}	0.437	-	T_{cw}	37	°C
$p_{L,eng}$	1.6	bar	ε_{IC}	0.80	-
Mr	0.61	-	$p_{L,IC}$	0.043	bar

Table 2: Constants for mathematical model

Case Setup

The case setup is shown in, table 3. The first three cases use constant efficiencies for HP compressor, LP compressors and HP turbine. The total pressure ratio is varied to obtain the necessary critical values. The three last cases keep the pressure ratio found in case 3. Then the efficiencies of the LP compressor, HP compressor and HP turbine are changed. The goal is to lower the efficiencies in order to force the LP turbine efficiency up and closer to the other efficiencies.

	$\eta_{LP,C}$	$\eta_{HP,C}$	$\eta_{HP,T}$	p_2	p_4
Case 1	0.82	0.82	0.80	3.5	6.5
Case 2	0.82	0.82	0.80	3.7	7
Case 3	0.82	0.82	0.80	3.8	7.4
Case 4	0.80	0.80	0.80	3.8	7.4
Case 5	0.79	0.79	0.79	3.8	7.4
Case 6	0.77	0.77	0.77	3.8	7.4
	Variables unchanged between cases				
	Variables changed between cases				

Table 3: Case setup showing variables

Critical Values

- **T_6:** This is the exhaust temperature before the HP turbine. It has a upper critical value defined by RR BEAS to avoid excessive wear on the turbine and increased maintenance intervals
- **$\eta_{LP,T}$:** The efficiency of the LP turbine is the quality control of the model and should not be too high or too low. It will reveal if the calculations are sound and should be around the interval of 60-90%.
- **AF-ratio:** This should be equal to or higher than the value of the 1-stage turbocharged RR BEAS C-engine model this master thesis is based upon, see chapter 4. This is to ensure that the ideal conditions for combustion are kept intact. The AF-ratio uses the trapped air, not the total air inducted. This provides the most precise calculation of exhaust temperature since it accounts for the volumetric efficiency of the engine. Even though the fuel-air equivalence ratio, F, is the most common way to describe the ratio between fuel and air it is not used as a critical value in these calculations. The reason being that it is based on the total air flow into the cylinder and doesn't account for the trapping ratio of the engine.

Results

In table 4 the results are presented. It shows the calculations of formulas (4)-(25) which represents the mathematical model of the 2-stage turbocharged engine. The formulas describe the points 1-8 of figure 8. The results are tabulated to achieve a compact and easy to read presentation. The variables and critical values are marked with their own colors to make the table easier to read. If a critical value is too high or too low its number is given the color red.

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Units
T_1	25	25	25	25	25	25	°C
p_1	1.00	1.00	1.00	1.00	1.00	1.00	bar
T_2	182	190	194	198	200	205	°C
p_2	3.50	3.70	3.80	3.80	3.80	3.80	bar
T_3	66	68	68	69	70	71	°C
p_3	3.46	3.66	3.76	3.76	3.76	3.76	bar
T_4	148	152	157	161	162	166	°C
p_4	6.50	7.00	7.40	7.40	7.40	7.40	bar
T_5	59	60	61	62	62	63	°C
p_5	6.46	6.96	7.36	7.36	7.36	7.36	bar
$\dot{m}_a$	11170	12001	12652	12627	12614	12588	kg/h
$\dot{m}_f$	397	397	397	397	397	397	kg/h
$\dot{m}_e$	11567	12398	13049	13024	13011	12985	kg/h
T_6	658	619	592	593	594	596	°C
p_6	4.86	5.36	5.76	5.76	5.76	5.76	bar
T_7	587	545	515	514	514	513	°C
p_7	3.28	3.49	3.61	3.56	3.52	3.42	bar
T_8	452	402	368	364	362	357	°C
p_8	1.03	1.03	1.03	1.03	1.03	1.03	bar
$\eta_{LP,T}$	61.8	65.6	68.4	70.7	72.3	75.7	%
$\eta_{LP,C}$	82.0	82.0	82.0	80.0	79.0	77.0	%
$\eta_{HP,C}$	82.0	82.0	82.0	80.0	79.0	77.0	%
$\eta_{HP,T}$	80.0	80.0	80.0	80.0	79.0	77.0	%
$\pi_{LP,C}$	3.50	3.70	3.80	3.80	3.80	3.80	-
$\pi_{HP,C}$	1.88	1.91	1.97	1.97	1.97	1.97	-
$\pi_{LP,C}/\pi_{HP,C}$	1.86	1.93	1.93	1.93	1.93	1.93	-
AF-ratio	28.13	30.23	31.87	31.80	31.77	31.71	-
λ	2.122	2.280	2.404	2.399	2.397	2.392	-
F	0.471	0.439	0.416	0.417	0.417	0.418	-
	Variables: These values are changed to reach the necessary critical values						
	Critical values: | $T_6 \leq 600$°C | $\eta_{LP,T} \leq \eta_{HP,T}$ | AF-Ratio ≥ 31.3 |						

Table 4: Results from mathematical model

Discussion of Results

Case 6 represents the 2-stage turbocharger with the minimum compressor and turbine efficiencies needed to satisfy the critical values for the chosen parameters. The three variable efficiencies have been lowered until they are close to the value of the LP turbine efficiency which is more realistic. Case 6 can be used for turbocharger selection based on the minimum requirements regarding efficiencies. Of course if there are compressors and turbines available with higher efficiencies it will only be an improvement.

3.2 Turbocharger Selection

Usually every turbocharger model can be delivered with a few different builds, meaning different compressors and turbines which all fit the same turbocharger. But a detailed turbocharger catalogue containing corrected compressor and turbine maps is not available. However, it is possible to calculate the parameters needed for selection in said catalogue. Chapter 3.1 demonstrated how to calculate an energy balance for the 2-stage turbocharger with chosen parameters. Based on the results of case 6 in table 4, one can extract the available information for detailed turbocharger matching procedure.

Formulas

$$\pi_C = \frac{p_{out}}{p_{in}} \tag{26}$$

$$\pi_T = \frac{p_{in}}{p_{out}} \tag{27}$$

$$\eta_{TC} = \eta_C \eta_T \eta_{mech} \tag{28}$$

$$\dot{V}_a = \frac{\frac{N}{2}\eta_{vol}\rho_5 V_{sw} n_{cyl}}{\rho_{amb} \cdot 60} \tag{29}$$

$$\dot{m}_{corr} = \dot{m}\left(\frac{\sqrt{T_{in}}}{p_{in}}\right) \tag{30}$$

$$n_{corr} = \frac{N}{\sqrt{T}} \tag{31}$$

$$\text{surge margin} = \frac{m - m_{surge}}{m_{choke} - m_{surge}} \tag{32}$$

Parameters for Turbocharger Selection

	Value	Unit
$\dot{V}_a$	2.99	m^3/s
$\eta_{LP,C}$	77.0	%
$\eta_{HP,C}$	77.0	%
$\eta_{LP,T}$	75.7	%
$\eta_{HP,T}$	77.0	%
$\eta_{LP,TC}$	56.5	%
$\eta_{HP,TC}$	57.5	%
$\pi_{LP,C}$	3.80	-
$\pi_{HP,C}$	1.97	-
$\pi_{LP,T}$	3.32	-
$\pi_{HP,T}$	1.683	-
$\dot{m}_{corr,LP,C}$	2.175	$kg \times \sqrt{K} \times m^2/h \times N$
$\dot{m}_{corr,HP,C}$	0.621	$kg \times \sqrt{K} \times m^2/h \times N$
$\dot{m}_{corr,LP,T}$	1.064	$kg \times \sqrt{K} \times m^2/h \times N$
$\dot{m}_{corr,HP,T}$	0.665	$kg \times \sqrt{K} \times m^2/h \times N$

Table 5:Parameters for turbocharger selection

The values in table 5 are calculated using the same constants and variables as in case 6 of table 4. The surge margin is not calculated, but is listed in formulas since it is needed in a matching procedure. With this information it is possible to check if there are turbochargers available that satisfies the target values.

Choosing the Turbocharger Model

The first step in a matching procedure is to identify the correct turbocharger model for the engine at the specific load. In the case of a 2-stage turbocharger this has to be done for each of the two turbochargers. This can be done by using publicly available information from turbocharger manufacturers. On the left side of figure 9, the correct turbocharger model can be selected by using the compressor pressure ratio and the volumetric flow rate. Then the turbocharger efficiency against compressor pressure ratio can be checked on the left side of figure 9.

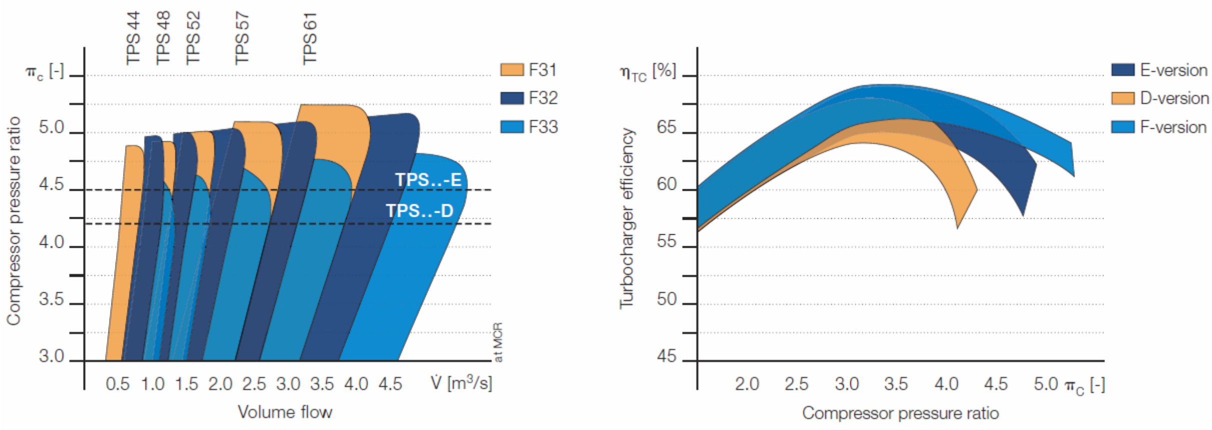


Figure 9: ABB TPS..-F turbocharger family

Choosing the Turbocharger Build

Each turbocharger model usually has several different builds, meaning different compressors and turbines that all fit in the same turbocharger housing. Once a turbocharger model is selected the next step is to acquire information from the manufacturer about the different model builds. Then corrected compressor and turbine maps will be provided and it is possible to make the final selection of turbocharger model and build. This information however is not publicly available and only given to customers. Hence in this master thesis the final turbocharger build selection cannot be performed. But for illustration purposes a

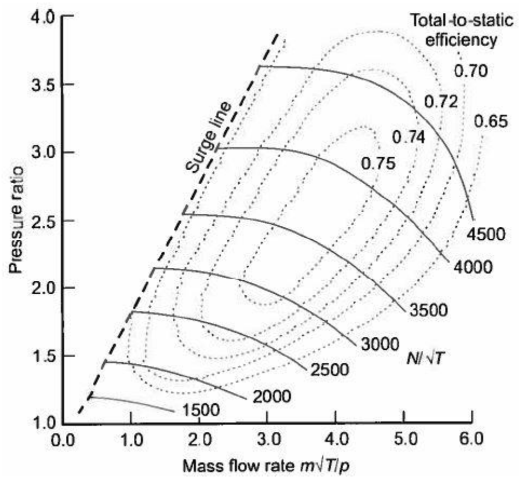


Figure 10: Corrected compressor map

corrected compressor map can be seen in figure 10. By using formulas 26, 30 and 31 or the results of table 5, it is possible to check that the efficiency is sufficient for a given corrected mass flow and pressure ratio. If the efficiency is sufficient the corrected turbocharger RPM will be known. When using the same procedure for the corrected turbine map there will be an extra requirement. The corrected turbine RPM must match the corrected compressor RPM since they share the same shaft. This procedure must be carried out for both turbochargers of the 2-stage turbocharger. When performing the matching procedure for top load only it is important to establish a sufficient surge margin such that the compressor will not surge at lower loads.

Discussion

By using the mathematical model in chapter 3.1 it is possible to obtain the values in table 5. This will provide enough information to make selection of turbocharger models. And if corrected compressor and turbine maps are available the final turbocharger build can be decided by using the values in table 5 plus establishing a sufficient surge margin. However, it becomes an iteration loop to satisfy the requirements of efficiencies, mass flows, pressure ratio and RPM for the different compressors and turbines. This iteration loop will not be treated in this master thesis, but a loop like that can converge relatively quickly if original assumptions are good.

4 Verification of the Engine Model

Modeling is cost-effective and can be a quick way to test different designs and theories. However, before using the results of your model you need to verify its accuracy or at least quantify the inaccuracy. I will ascertain the accuracy of the 1- stage turbocharged GT-Power model provided by RR BEAS by comparing it with measurement data from the same engine. Then I will use the results as my base for accuracy when I expand the turbocharger system to a 2-stage turbocharger. Essentially the engine model will be the same, only the turbocharger will change. Any tuning of the model will be documented. This is the best option for verifying the accuracy of the 2-stage turbocharged model since RR BEAS don't have an engine with 2-stage turbocharger and therefore no measurement data either.

Strategy

- Description of the engine
- Description of simulation software
- Description of the engine model
- Tuning of the engine model
- Comparison of results
- Discussion

4.1 The Rolls-Royce C25:33L6 Engine

Bergen Engines have been making diesel engines since 1946. Their design is a result of nearly 70 years of experience. The engine used for this master thesis is one of the latest designs, their C-engine. It is intended for power generation, marine generator sets and marine propulsion. It is available in 6, 8 and 9 cylinder configuration with a power range of 2000-3000 kW.

Bergen Engine Type C25:33L6	
Power	2000 kW
Bore	250 mm
Stroke	330 mm
Number of cylinders	6
Number of intake valves	12
Number of exhaust valves	12
Cylinder configuration	In-line
Intended use	Generator
Turbocharger	1-stage ABB
Intercooling	2-stage
Governor	Electronic
Fuel type	MDO/HFO
Weight	19650 kg

Table 6: Description of C25:33L6 engine

The engine treated in this master thesis is the 6 cylinder C-engine intended for running on diesel oil, important characteristics of the engine can be found in Table 6. It uses a 2-stage charge air cooler with one high temperature circuit in the first stage that starts the cooling process, the air then enters the low temperature circuit which brings the charge air temperature down to the target value. It basically has two different cooling water circuits, one with high temperature water and one with low temperature water.

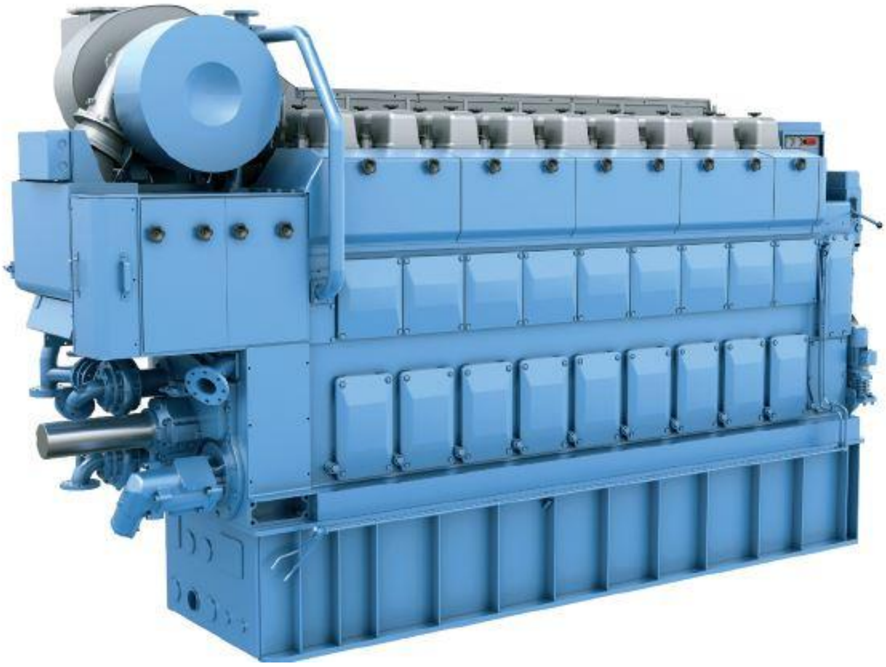


Figure 11: Rolls-Royce C25:33L9P engine

The engine has two settings for inlet valve closing. For startup and the two lowest loads it closes 10 CAD before BDC, for all other loads it closes 40 CAD before BDC, see table 10 for load range. This is the equivalent of Miller 10 and Miller 40. It can change between the two Miller timings while running. The exhaust valves don't have variable timing. There are separate fuel injection pumps for each cylinder and they are camshaft driven. The pressure loss over the cylinders is calculated as the difference between intake pressure and the exhaust pressure just before the turbine. This pressure loss can be calculated relatively accurate with the following formula:

$$\frac{p_{\text{int}}}{p_{exh,b.turb}} \approx 1.3 \qquad (33)$$

This ratio has been confirmed by RR BEAS to hold true for boost pressures at all loads except the two lowest where the intake valves close at Miller 10, see table 10 for load range. For the lower loads where Miller 10 is used, the ratio is closer to unity, (Seeberg, 2015),(Rolls-Royce, 2012).

The engine meets the MARPOL ANNEX VI requirements and is IMO Tier II compliant. The engine has an electronic governor with hydraulic actuator and electronic speed setting. Pulse turbocharging is employed to maximize the use of exhaust energy, this is done by using a twin entry turbine where each entry is fed by three cylinders. This configuration is optimal with regards to overlap between pulses and minimal effect on scavenging, (Baines, 2005).

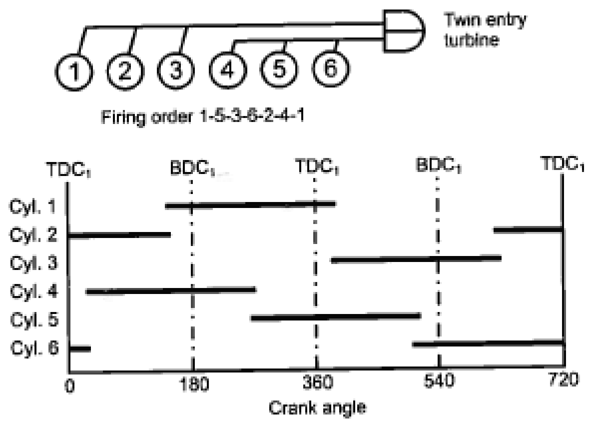


Figure 12: Exhaust valve timing, 4-stroke

4.2 Simulation Software

Dr. Thomas Morel founded the company Gamma Technologies Incorporated (GTI) in 1994. It is an American company with headquarter in Westmont, Illinois, USA.

Gamma Technologies Inc. (GTI) develops and licenses a "virtual vehicle and powertrain" CAE software GT-SUITE. This tool is specifically designed for applications in the engine, powertrain and vehicle industries. In addition to supplying software products, GTI provides user support and training and carries out general consulting projects using its proprietary CAE tools,(GTI, 2015)

GT-SUITE is a multi-physics platform for constructing general system models which can handle both steady state and transient simulations. Flow and heat transfer in the system components is based on one-dimensional gas dynamics. It is a template based program which makes it user friendly and models can be built fast. Models are built by connecting templates and objects to each other and then controlling the initial and boundary conditions, the necessary formulas for calculations are built into the templates and objects. The templates are organized into several libraries:

- Flow library
- Acoustics library
- Thermal Library
- Mechanical library
- Electric and Electromagnetic library
- Chemistry Library
- Controls library
- Built-in 3D CFD and 3D FE

There exist different licensing options, GT-SUITE license gives access to all libraries while the GT-Power license used in this master thesis doesn't include all libraries. The GT-Power license focuses on engine performance, gas acoustics and aftertreatment, so the necessary libraries for these applications are provided.

The interface in GT-Power where the models are built is called GT-ISE and contains a work space for building the model, a mini-map containing the templates, objects and parts used in the model plus the template library. See figure 13.

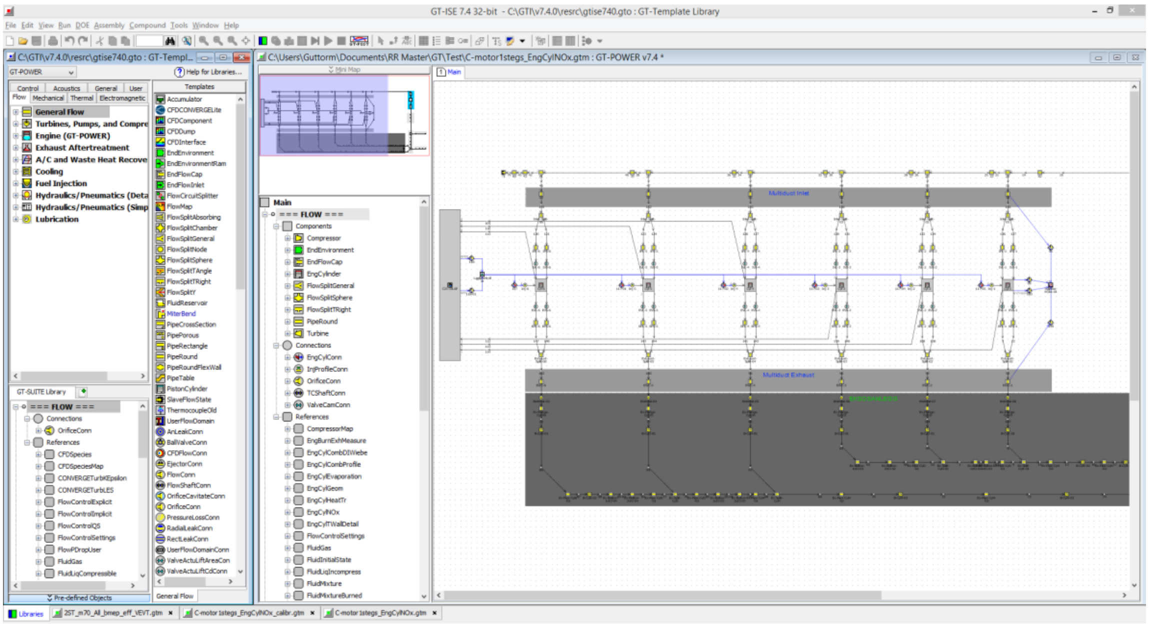


Figure 13: GT-ISE user interface

In Run Setup, simulation time, convergence settings, solution method, solver type and more can be controlled. Each model can be run with multiple cases, this is controlled in case setup where parameters are referenced from objects in the model and can be changed for different cases.

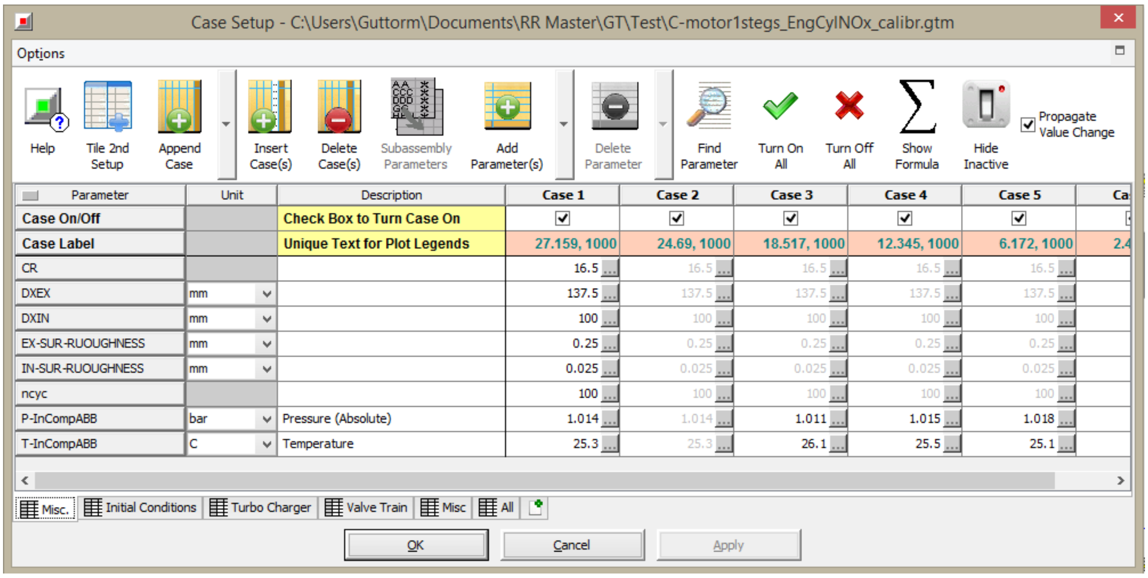


Figure 14: GT-Power case setup

A status window will appear when running a simulation, it contains a lot of information about the ongoing calculations, progression of simulation, case number and so on. See figure 15.

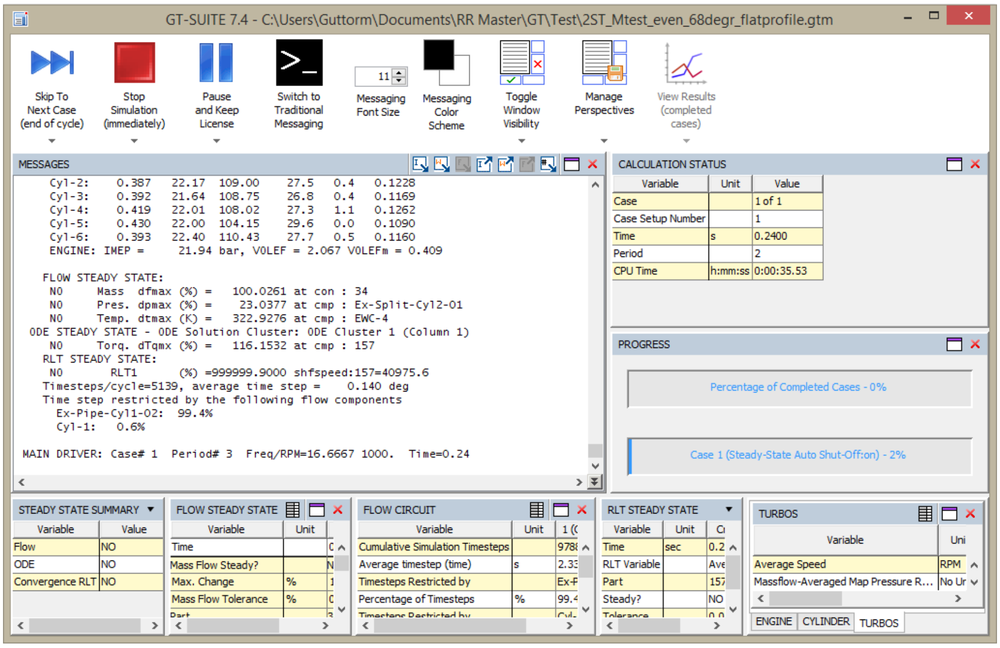


Figure 15: GT-Power simulation status window

When the simulation is finished the results can be accessed in a separate program called GT-POST, through a file created for this purpose. The GT-POST file contains all results and can be used to view and manipulate data, the file is also updated if you rerun the simulation. In the main window of the interface the same model representation as in GT-ISE can be found, by clicking on the different components the results are presented at the bottom of the screen. There are advanced plotting options, regular graphs, box plots, combined datasets, contour plots, imported data can be compared with results from the simulation. It is also possible to present results in tables. All in all it is a powerful data analysis tool.

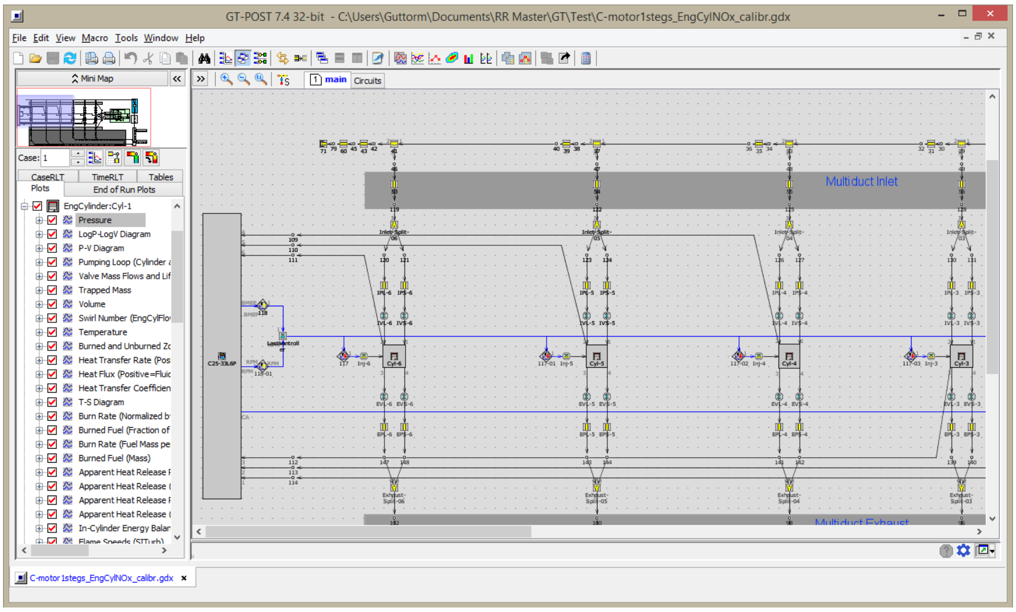


Figure 16: GT-POST screenshot

4.3 The GT-Power Model of the Engine

A GT-Power model of the engine has been provided by RR BEAS, it is the digital version of the physical engine. How the model works and is controlled will be explained.

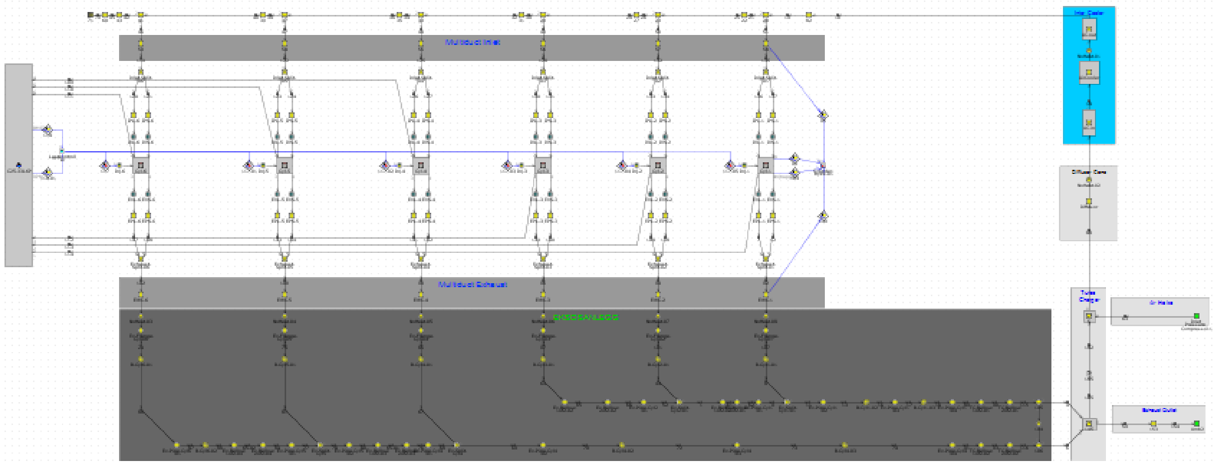


Figure 17: Engine model with 1-stage turbocharger

To the left in figure 17 is the engine cranktrain which controls the movements of the pistons through setting the RPM, it is connected individually to each piston. In addition cylinder geometry, firing order, bearing loads and engine inertia are set in the cranktrain. The load controller sets the desired power output using RPM input from the cranktrain, the load controller is connected to the injector of each cylinder. The injectors govern the injected mass of fuel to the cylinders. The cylinders are the heart of the model where power, friction, heat loss, pressure, temperature, energy, combustion and so on are calculated. The intake manifold is at the top and the pulse exhaust manifold at the bottom. The turbocharger is found on the right side, the turbine with its double inlet driving the compressor through a shaft and an air cooler before the intake manifold. The compressor and turbine maps are both defined by an imported SAE file in tabulated form. The ambient conditions are controlled through the "EndEnvironment" template, with one object for the air intake and one for the exhaust outlet.

For the real engine, torque is the measurement used for power calculations. The model however has more options since all values are calculated. Through the load controller template you can select the desired type of input for power calculations. In this thesis BMEP is the chosen input.

Controller input	Unit
Bmep	Bar
Brake Torque	Nm
Brake Power	kW
Brake Power	HP
Imep	bar

Table 7: Controller input for power calculations

Both the engine speed and BMEP are parameters which can be controlled from the case setup. An advantage with the parameters is that when you change a parameter in case setup it will change in all the objects of the model that reference the parameter. Power and torque are calculated through the following formulas:

$$P_e = iV_{sw} p_{me} n_a = T_e \dot{\omega}_a \tag{34}$$

$$T_e = \frac{P_e}{2\pi\dot{\omega}_a} \tag{35}$$

In the real engine, combustion is controlled by injection timing, after injection the fuel will ignite at sufficient temperature and pressure. Early injection gives a larger portion of pre-mixed fuel and air before ignition. In the engine model the injection timing is in a sense detached from combustion. There is one parameter that sets injection timing, which governs the start of fuel injection. The start of combustion, SOC, is governed by the “EngCylCombProfile” template, this template contains a tabulated combustion rate which is derived from a Wiebe combustion function used by ABB. Every load needs its own combustion rate table so in that sense it is not dynamic since combustion is not calculated directly from a function. The injection timing and combustion are detached because the combustion will happen according to the tabulated values as long as there is sufficient fuel present. So, in theory fuel could be injected several CAD earlier and the combustion would still be the same. Changing the SOC must be done inside the “EngCylCombProfile” template, injection timings only function is to ensure sufficient fuel at the SOC.

There are piping templates for bends, straight pipes, flow splits and so on. Each template has attributes which control, dimensions, discretization length, surface roughness, heat transfer, pressure drop, elevation change and so on. The discretization length is an important parameter because it determines the step length between each calculation and directly affects simulation time. Once a template is imported into the model from the library it can produce multiple objects where the attributes can be different. Each object can have multiple parts in the model, all the parts from a specific object shares the objects attribute values, one can however easily override an attribute in a certain part if needed. In figure 18 the top of the root structure called PipeRound is the template, the level under contains the objects made from the template and the lowest level contains the parts made from the “Ex-Bellow-1of2” object. This creates an organized overview of the different parts in the model.

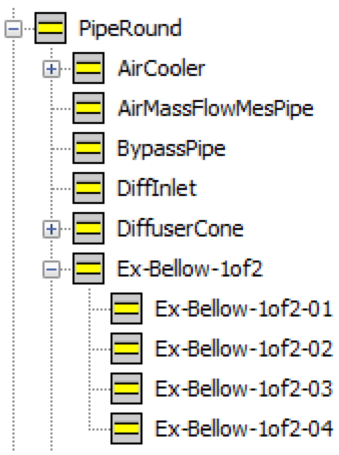


Figure 18: Template-Object-Part

The charge air cooler is not built as an exact replica of a real one. It is built around a “PipeRound” template that is constructed as 242 separate small diameter pipes all with the same length. In both ends there is a endcap in form of a “FlowSplitGeneral” template defined with a volume, these collect the flow from all the small pipes. Then the heat transfer multiplier for the 242 small pipes is set to an artificially high value, and the imposed wall temperature is set to the expected temperature of cooled air after the charge air cooler. In effect what happens is that the temperature of the air is forced down to the imposed wall temperature very fast because of the high heat transfer multiplier. Then the boost air temperature can be easily controlled but it requires knowledge of realistic values for the cooled air temperature. In short the intercooler is not dynamic, it simply delivers what is defined. The performance of the intercooler is based on the real one and the available cooling water temperature, through formula 6 the efficiency of the IC can easily be calculated and checked against the real numbers.

Like in the real engine the model also has variable intake valve timing with two settings, Miller 40 and Miller 10. Both Miller 40 and Miller 10 share the same lift curve, this lift curve is found inside a “ValveCamConn” template. Under the Lift Array menu inside the template, the lift curve is imported in tabulated form, one column for CAD and one column for lift height in mm. This lift curve can also be referenced with a parameter inside the “ValveCamConn” template, it is then possible to reference different lift curves in the case setup. To change the miller timing a parameter is made that controls the starting crank angle for the lift curve, it is set to 0 CAD for Miller 40, when Miller 10 is needed the parameter simply moves the whole lift curve 30 CAD. The lift curve itself is unchanged. Figure 19 shows both Miller 40 and 10 for the lift curve, the exhaust valve lift curve is included and makes it possible to see how Miller 10 changes the valve overlap dramatically.

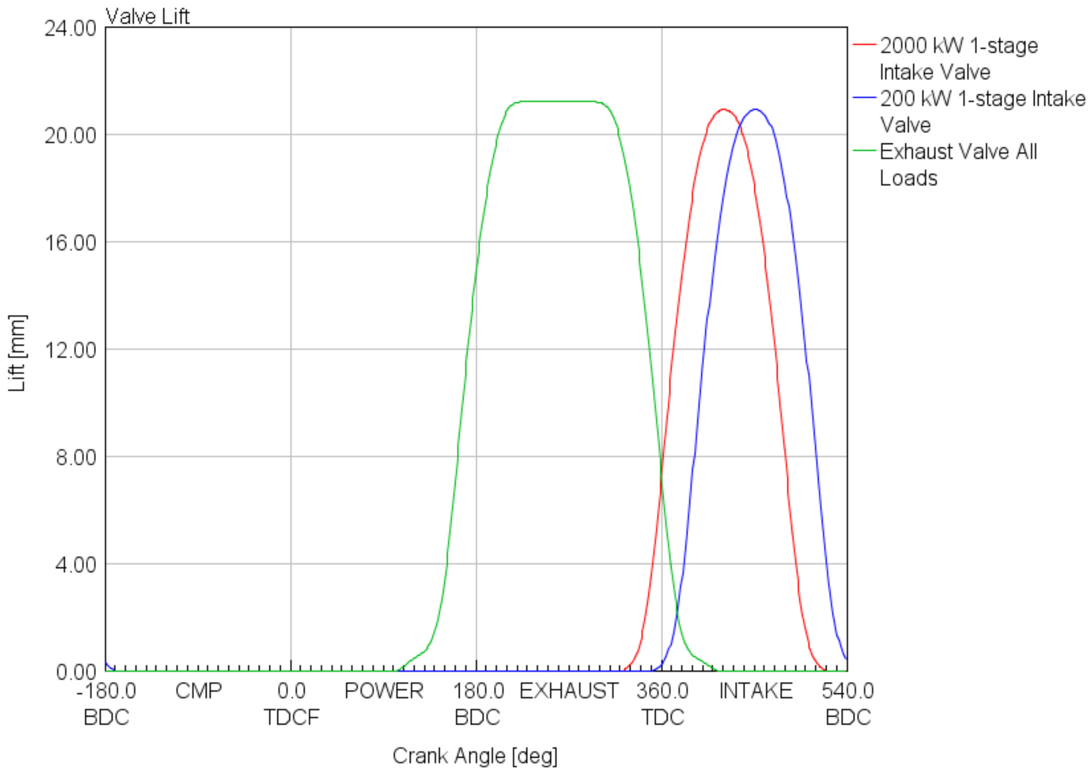


Figure 19: Engine model VVT

4.3.1 Establishing a Reliable NO_x Measurement

The production of NO_x in a medium speed diesel engine is mainly formed during combustion and is strongly dependent on the maximum temperature in the cylinder. The engine speed determines how much time is available for formation. Heavier fuel oils can have some NO_x content, but if so it will be a small percentage of the total NO_x production. Figure 20 demonstrates the connection between rate of NO_x production, temperature and AF-ratio, (Heywood, 1988). Initial simulations with the RR BEAS engine model showed that NO_x levels were diverging, in fact they were twice as high as the test data. As a result it was necessary to tune and calibrate the NO_x emissions to approximate the values of the test data.

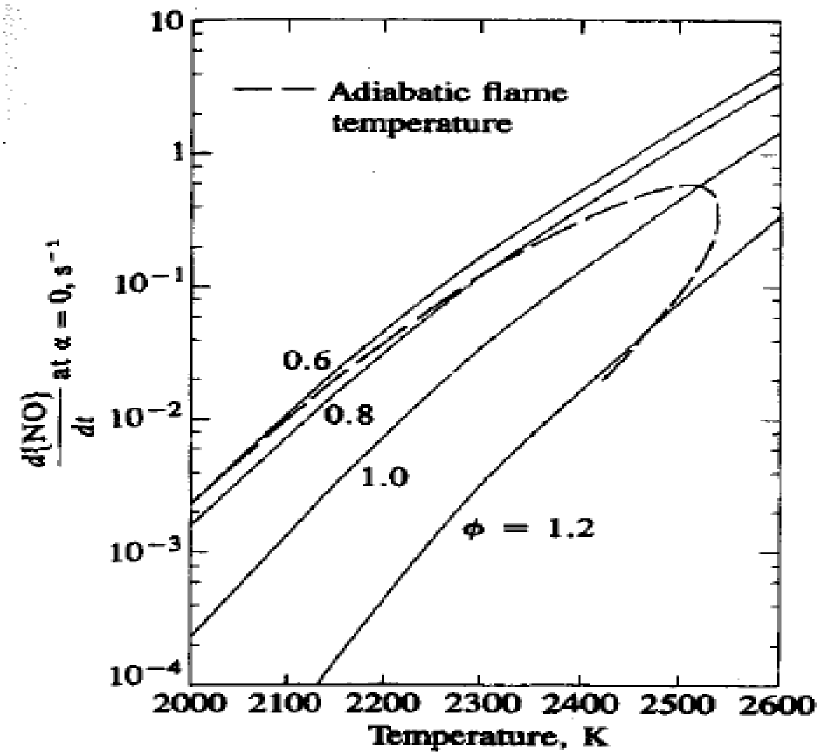


Figure 20: NOx dependency on temperature and AF-ratio

Cycle Averaged NO_x Measurement

In GT-POST the NO_x concentration is given in ppm. The NO_x concentration is calculated in each cylinder and also as an average value of all the cylinders in the cranktrain. For the cranktrain value the individual cylinders NO_x production is averaged, this value is found in the emissions drop-down menu. There will be some error with this method because of variations in flow rate and outlet temperature between cylinders. To get the most accurate result NO_x should be measured in the tailpipe.

To measure NO_x in the tailpipe it is necessary to connect a "SensorConn" part to the tailpipe and then connect this to a "MovingAverage" part which enabled a calculation of the cycle averaged value. See the setup in figure 21. The "SensorConn" measures the species mass fraction from the exhaust pipe in ppm, this data is sent as input signat to the "MovingAverage" where the output signal is smoothed by averaging the value over 5 cycles.

This cycle averaged value is used exclusively as reference for the NO_x concentration in the model, given in ppm. The brake specific NO_x production, g/kWh, is calculated using this cycle averaged value through formula 36, (VDMA, 2008).

$$BS_{NOx} = EC_{NOx} \frac{M_{NOx}}{M_{exh}} \frac{\dot{m}_e}{P_e} \tag{36}$$

The following values were used for molecular weights in the formula:

- $M_{NOx} = M_{NO2} = 46.00$ [kg/kmol]
  - NO_x treated as NO_2, standard convention used by GT-Power
- $M_e = 28.82$ [kg/kmol]
  - Value for wet exhaust

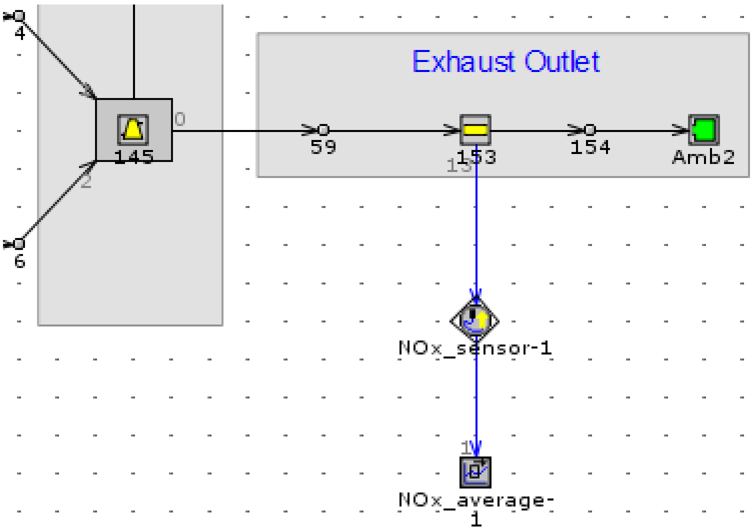


Figure 21: Tailpipe NOx measurement

In the model the NO_x level is given as a species mass fraction, measured in parts per million, ppm. This unit is not the best as it is not that commonly used. The IMO MARPOL 73/78 Annex VI which stipulates the allowed NO_x emissions inside and outside of ECA, uses the unit, g/kWh, to define the limits. This makes it necessary to convert the NO_x values from ppm to g/kWh. The value ppm is a measure of concentration, in the metric system ppm can be expressed as 1 ppm = 1 mg/kg.

Tuning the Model

As mentioned the NO_x levels were very high, in addition they didn't follow the expected trends either. In the GT-Power manual it says that the NO_x calculation is very sensitive to the following factors: trapped cylinder mass, AF-ratio, combustion rate and maximum cylinder temperature. The initial simulations revealed a strong dependency on AF-ratio and not much dependency on the other factors. It resulted in quite an opposite effect of the expected. With high maximum temperature and a low AF-ratio the NO_x levels dropped significantly, demonstrating a lot stronger dependency on AF-ratio than maximum temperature.

To solve this problem it was necessary to change the "NO_x reference object" inside the combustion object. The "EngCylNOx" reference object was selected instead of setting the value to ignore. This reference object calculates NO_x production based on the extended Zeldovich mechanism and is strongly dependent on AF-ratio and temperature. It is essential to check that engine airflow, trapping ratio, AF-ratio and combustion rate in the model are within realistic values before the NO_x calculations can be trusted. After checking this, new simulations were performed, revealing a substantial decrease in NO_x levels and the expected trends were followed. To further fine tune the NO_x levels to be more in sync with the test data it was necessary to adjust the "NOx calibration multiplier" attribute inside the "EngCylNOx" object. The final value of the NO_x multiplier was set to 0.71. Then the model and test data values were close enough, see table 8.

Comparison of NO_x levels						Unit
Power	2000	1500	1000	500	200	kW
Original model	1414	1428	1442	1696	1820	ppm
Calibrated model	692	715	754	610	375	ppm
Test data	710	700	690	720	420	ppm

Table 8: Comparison of NOx levels

4.4 Verification

In this chapter the real engine and the simulation model will be compared. Important parameters will be plotted against each other and the full comparison in tabulated form can be found in appendix 9.2. The load range that will be tested is defined by the test data provided by RR BEAS and can be found in table 10.

4.4.1 Factory Test Setup

At Rolls-Royce Bergen Engines they mainly use two techniques when testing engines, for generator sets they use the generator as resistance and for propulsion engines they use a waterbrake for resistance.

Data from a test run of the engine described in chapter 4.1 has been provided by RR BEAS. Although there are a lot of possibilities for control of the engine the most important for the verification is how engine speed and load is controlled. In the test stand they connect to the engine control system, this gives control of the engine to the control room in the test stand. First step is to obtain the correct engine speed. This is done by giving an input speed signal to the engines governor, this signal is checked against a speed reading from the camshaft. The governor controls the fuelrack which adjusts the mass flow of fuel from the fuel pumps. It automatically adjusts the fuel rack to reach the target speed. The load of the engine is controlled by a waterbrake. In the control room the target torque for the given RPM can be set and then the waterbrake automatically increases resistance until the target value is reached.

The principle of the waterbrake is that the shaft of the engine is connected to a turbine or propeller in an enclosure filled with water. Running the propeller in water requires energy and the amount of water determines how much energy is required. There is one inlet of cold water and one outlet for the water heated by the rotation of the propeller. The waterbrake itself is mounted on bearings, this will allow it to turn around its own axis. The natural response of the waterbrake will be to turn in the direction of the propeller. On the side of it there is mounted a torque arm which stops it from turning and at the same time measures the force required to stop it. This force multiplied by the distance from where it is measured and in to the center of the shaft is the torque produced by the engine. See figure 22.

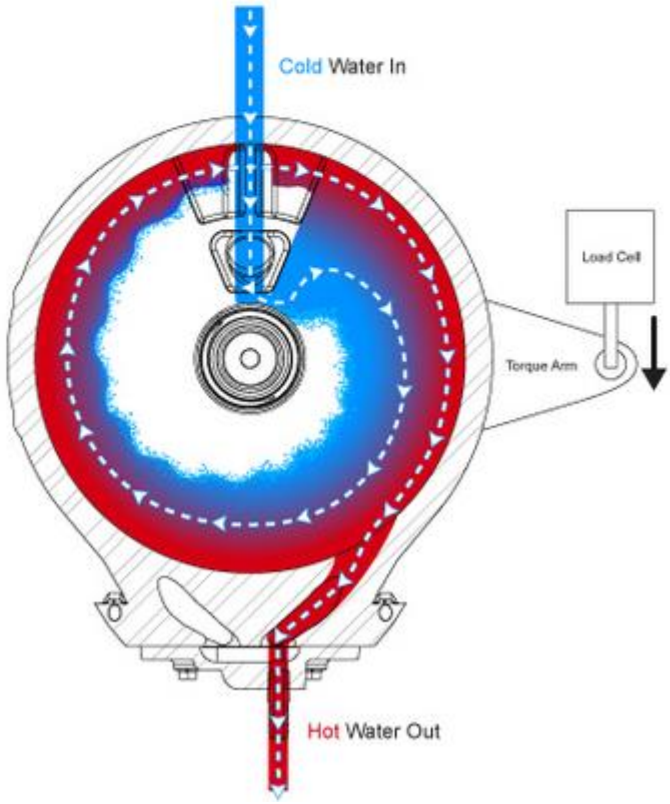


Figure 22: Illustration of a waterbrake

4.4.2 Calculation of Parameters

In an engine test some parameters are measured and put directly in the results, some parameters are calculated with the help of other selected values. In table 9 is a list of which parameters are measured and which are calculated.

Values	Unit	Engine	Formula	Model	Formula
Speed	RPM	Measured	-	Measured	-
Brake Torque	Nm	Calculated	37	Calculatd	38
Power	kW	Calculated	39	Calculated	40
BMEP	Bar	Calculated	41	Measured	
Air Flow	kg/h	Measured		Measured	
Fuel Flow	kg/h	Measured		Measured	
BSFC	g/kWh	Calculated	42	Calculated	42
BSAC	g/kWh	Calculated	43	Calculated	43
AF-Ratio	-	Calculated	44	Calculated	44
Ambient Pressure	Bar	Measured		Measured	
Ambient Temperature	K	Measured		Measured	
Temperature after compressor	K	Measured	-	Measured	-
Temperature air receiver	K	Measured	-	Measured	-
Cylinder maximum pressure	Bar	Measured		Measured	
Exhaust Manifold Pressure	Bar	Measured		Measured	
Exhaust manifold temperature	K	Measured		Measured	
Turbocharger Speed	RPM	Measured		Measured	
Turbocharger Pressure Ratio	-	Calculated	45	Calculated	45
Temperture after compressor	K	Measured		Measured	
Temperature raise compressor	K	Calculated	46	Calculated	46
Temperature exhaust receiver	K	Measured		Measured	
Temperature exhaust after turbine	K	Measured		Measured	
Thermal Efficiency	-	Calculated	47	Calculated	47
Pressure difference across cylinder	-	Calculated	48	Calculated	48

Table 9: Measured values for verification

The necessary formulas for the calculations in table (9) are listed below:

$$T_{wb} = \mathrm{F}_{wb}\mathrm{a}_{wb} \tag{37}$$

$$T_{eng} = F_{cg}a_{ct} \tag{38}$$

$$P_e = T_e\dot{\omega} \tag{39}$$

$$P_e = iV_d p_{me} n_a \tag{40}$$

$$p_{me} = \frac{P_e}{iV_d n_a} \tag{41}$$

$$\dot{m}_f = BSFC \cdot P_e \tag{42}$$

$$\dot{m}_a = BSAC \cdot P_e \tag{43}$$

$$AF - Ratio = \frac{BSAC}{BSFC} \tag{44}$$

$$\pi_{TC} = \frac{p_{\text{after comp}}}{p_{\text{before comp}}} \tag{45}$$

$$\Delta T_{comp} = T_{\text{after comp}} - T_{\text{before comp}} \tag{46}$$

$$\eta_{th} = \frac{P_e}{h_n \dot{m}_f} \tag{47}$$

$$P_{\text{across cyl}} = P_{b.cyl} - P_{a.cyl} \tag{48}$$

4.4.3 Results

The strategy for presenting the results of the engine verification is to compare some key parameters plotted for both the engine and the model. The complete comparison is available in tabulated form in appendix 9.2. Both the engine test data and the simulation model share the same load and speed conditions in this verification process, see table 10.

Loading conditions						Unit
Case	1	3	3	4	5	-
Speed	1000	1000	1000	1000	1000	RPM
BMEP	24.69	18.52	12.34	6.17	2.47	Bar
Power	2000	1500	1000	500	200	kW

Table 10: Engine speed and load conditions

The results are quite good the model follows the engine values closely. But the air receiver temperature and exhaust temperature before turbine deviates noticeably. The brake specific NO_x emissions also deviate but mostly at low loads. It is very difficult to get a correct NO_x prediction when using 1-dimensional program code, this was passed on to me by RR BEAS, (Seeberg, 2015).

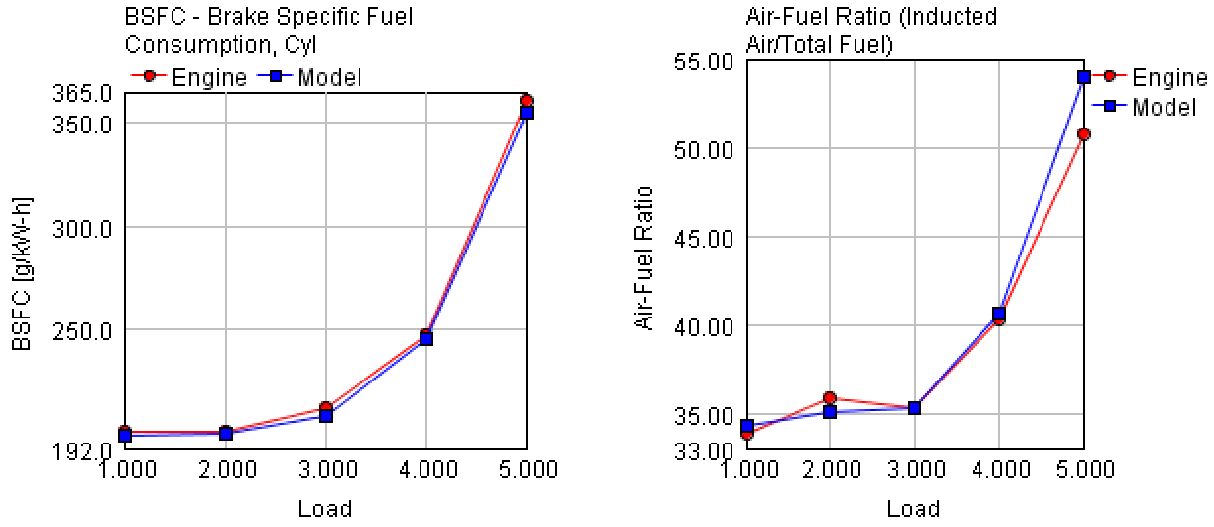


Figure 23: BSFC and AF-ratio verification

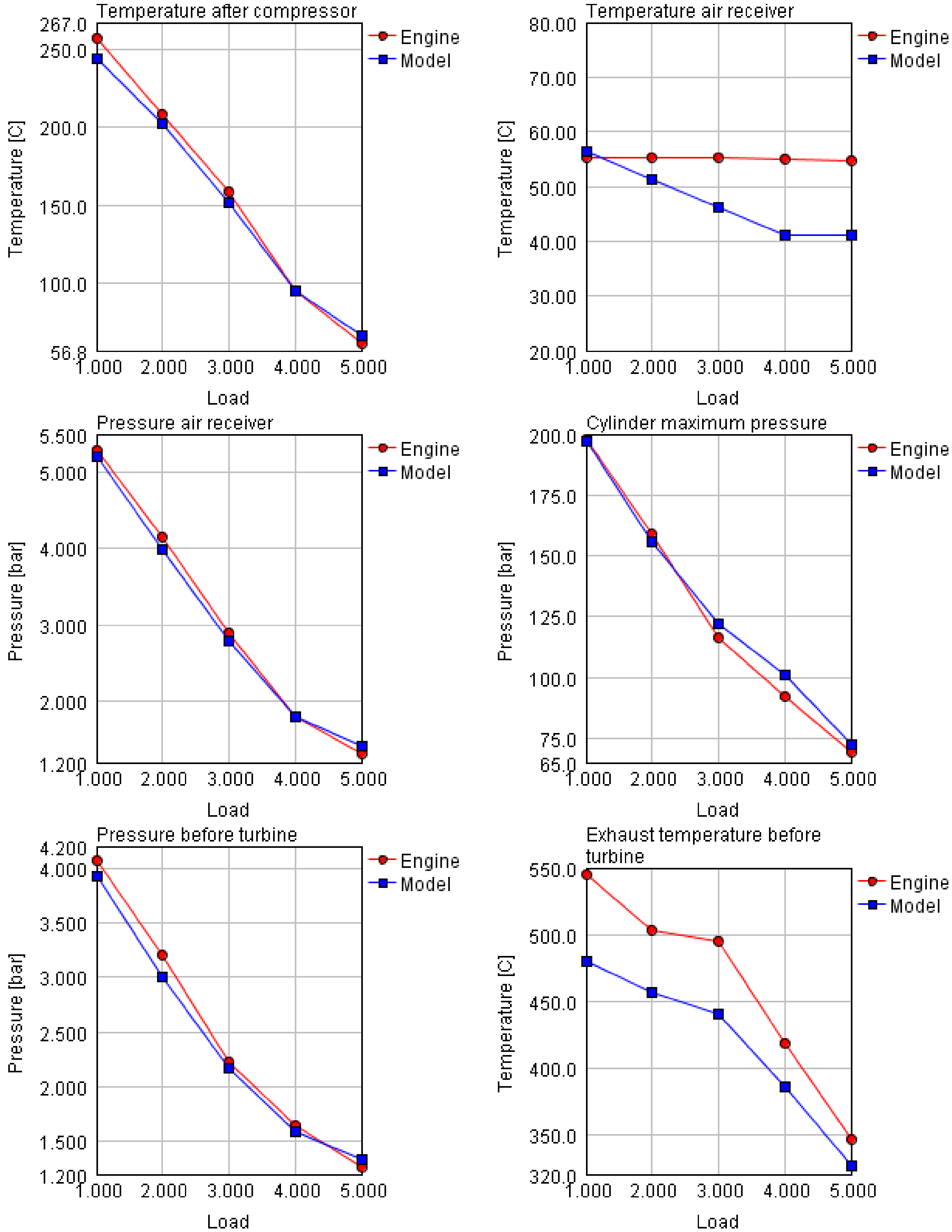


Figure 24: 6 Collected verification plots

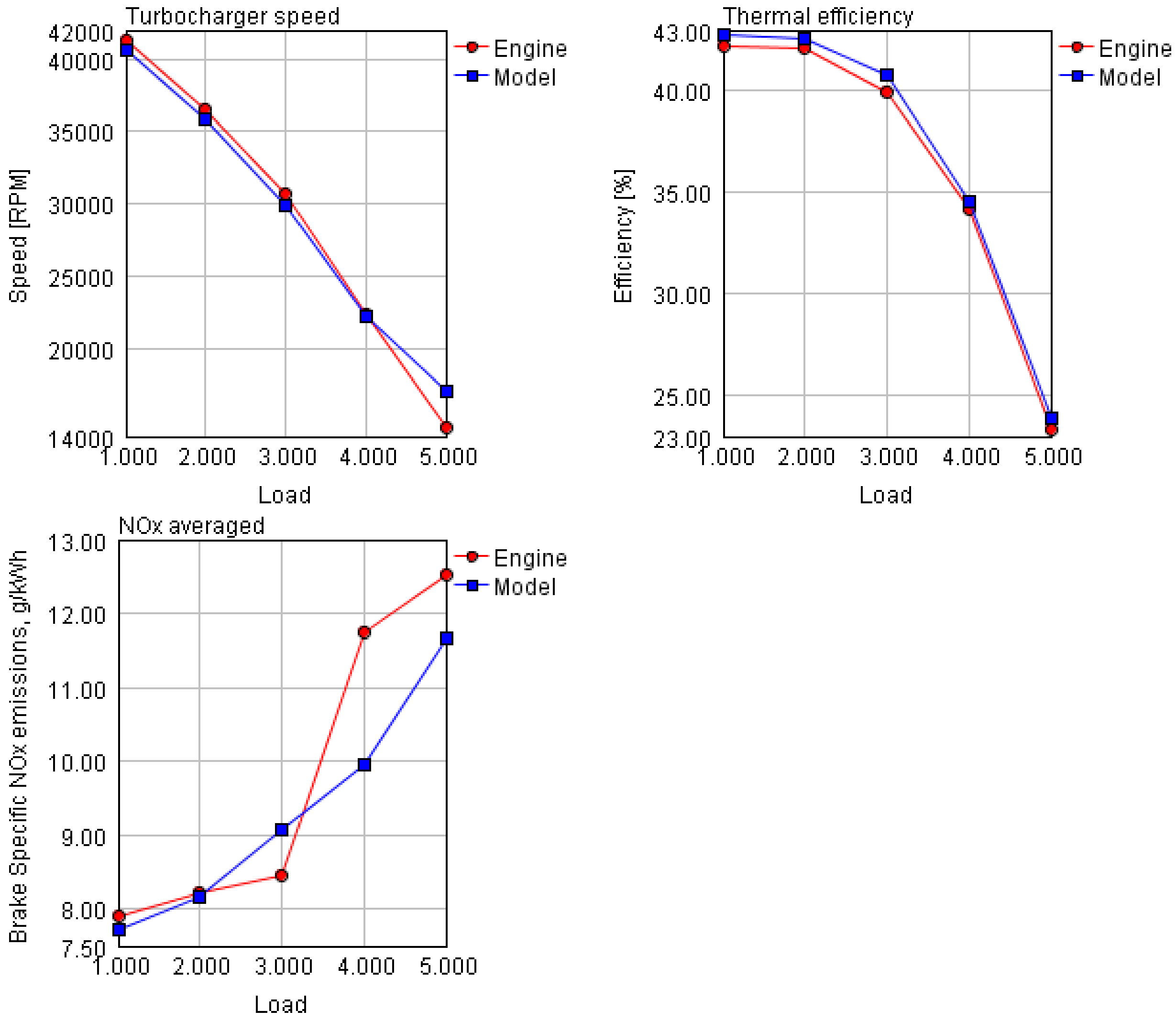


Figure 25: TC speed, thermal eff, NOx, verification

Figure 23-25 show plots comparing test data from the C25:33L6 engine with the GT-Power model of the same engine. The load range from 1-5 on the x-axis of the plots corresponds to the cases in table 10. The results are acceptable. Temperature air receiver, exhaust temperature and brake specific NO_x emissions are all a bit less accurate. In the tabulated version of the verification, found in appendix 9.2, the differences are quantified as percentages.

When making an engine model in GT-Power it is impossible to get 100 % accurate results. The following deviation rating describes a rough view on the matter, (Seeberg, 2015).

- Deviation < 5 % Considered very good
- Deviation < 10 % Considered good
- Deviation < 15 % Considered adequate depending on the parameter

5 Engine Model with 2-Stage Turbocharger

To evaluate the potential benefits of the 2-stage turbocharger the computer simulation program GT-Power will be used. The GT-Power model of the engine provided by RR BEAS will be used and matched with a 2-stage turbocharger. It is the very model used in chapter 4, with the tuning of NO_x emissions for better accuracy.

5.1 The 2-Stage turbocharger

The 2-stage turbocharger will be built along the same lines as the 1-stage turbocharger but with different piping and more components. For comparison, the principal layout of the turbocharger can be seen in figure 8, of chapter 3. Below in figure 26 is a screenshot of the turbocharger from GT-Power, it is difficult to see all the details but it shows the layout inside GT-Power.

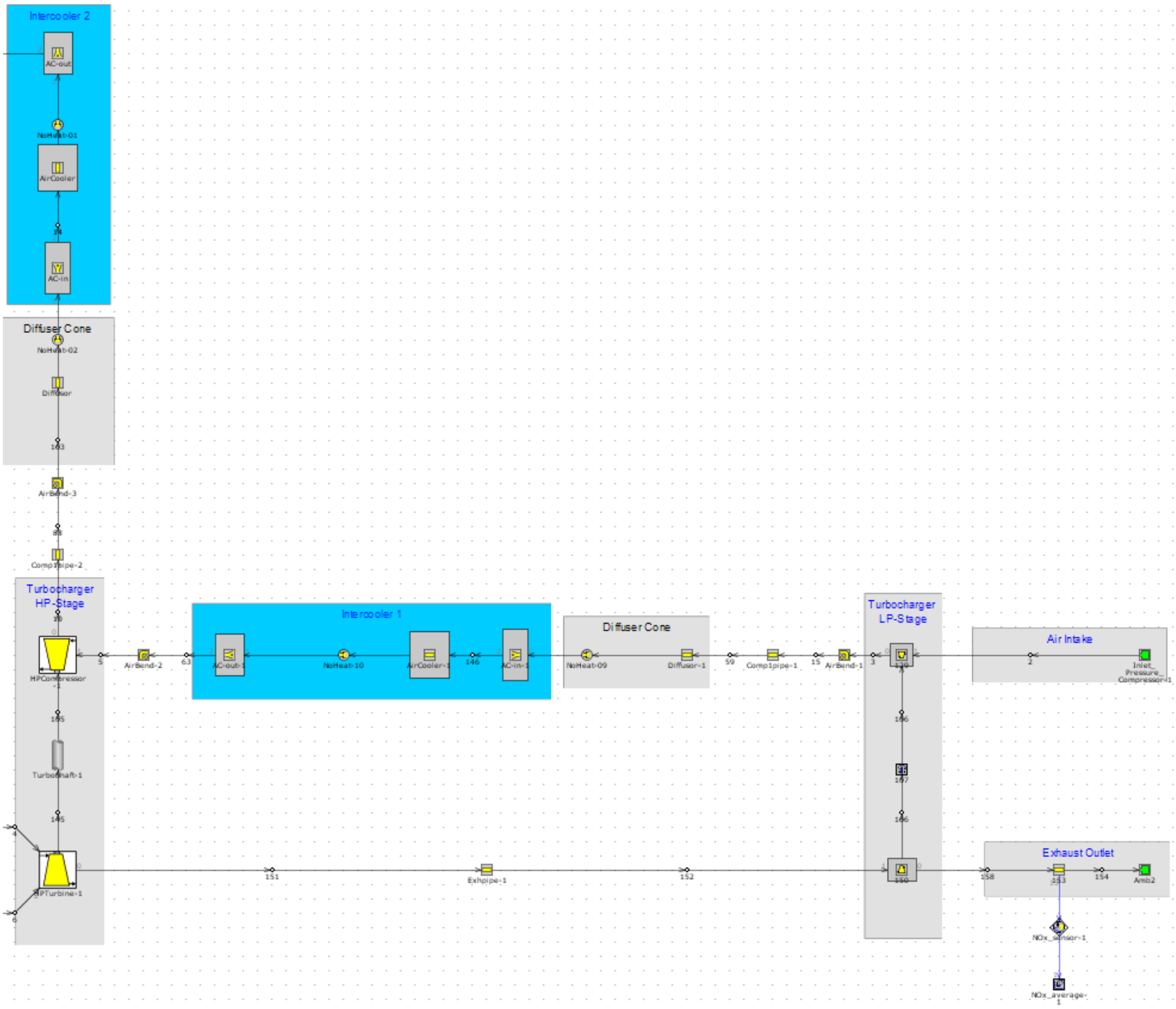


Figure 26: GT-Power 2-Stage turbocharger

Building a turbocharger in GT-Power is done by importing the necessary templates from the template library into your model. The next step is to create objects from these and alter the necessary attributes. When an object is done, one can simply drag and drop it into the workspace and a part is created inhibiting the set attributes of the mother object which in turn is created from the template. An example of this root structure can be seen in figure 18, chapter 4.3. In other words building the empty shell or infrastructure of the model is fairly quick. The demanding part is controlling all the conditions inside the parts to make sure that the model is realistic. In most of the parts in the model several initial conditions need to be set, they are inserted through the attributes of the selected part. As an example the edit menu of the LP compressor part is shown below in figure 27.

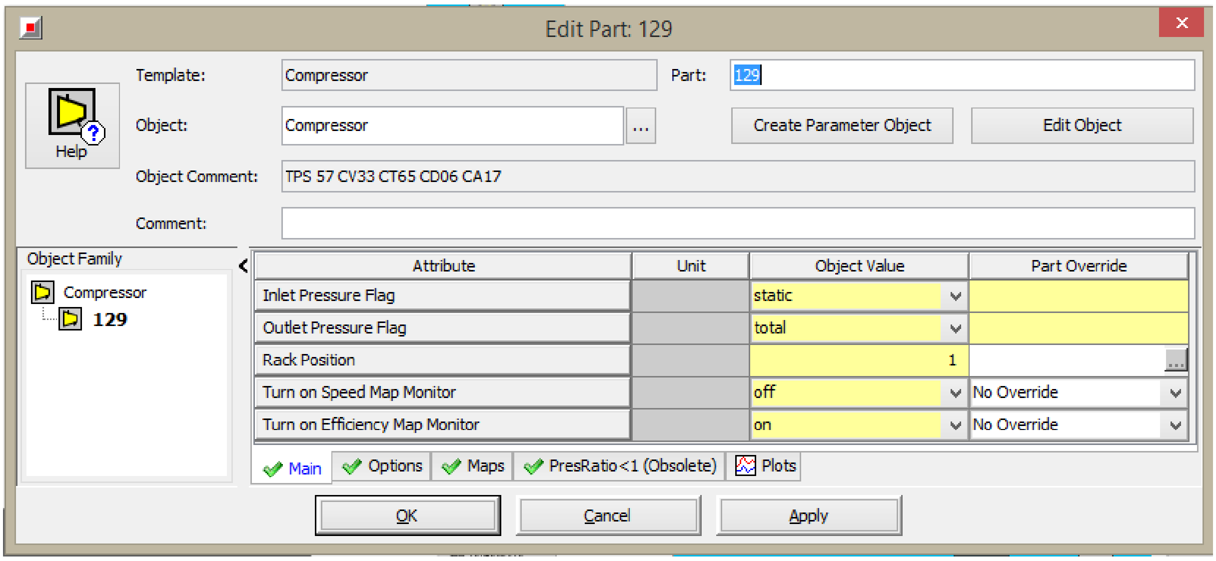


Figure 27: Edit menu LP compressor part

The mentioned structure can be seen, it is part 129, from the compressor object, made from the Compressor template. The different attributes can be seen with both the set object values and part override values if there are any. At the bottom there are several menus which organize all the attributes of the part.

To build the 2-stage turbocharger there is one thing that is not so easy to find good values for, that is the compressor and turbine maps. There are several compressor and turbine templates in the template library that provides a turbocharger representation but they all need compressor and turbine maps either in form of a SAE file or filled directly into the tables of the template. These maps are what control the compressors and turbines and determine their behavior. An example of a compressor map can be found in figure 10, chapter 3.2. The turbocharger manufacturers guard these maps well and they are only available to customers. For this master thesis RR BEAS has provided compressor and turbine maps for both the LP and HP turbocharger through a third party business partner. They are in the form of SAE files. To facilitate a better understanding, the 2-stage turbocharger will be explained in its different parts, turbochargers and intercoolers.

Turbochargers

The compressors are connected with the turbines through a shaft, there is no need to construct a turbocharger housing for simulation purposes. Some important characteristics of the turbochargers:

- LP Compressor inlet pressure attribute set to static pressure, because inlet is ambient conditions
- LP Compressor outlet pressure attribute set to total pressure, both static and dynamic
- Turbine inlet pressure attribute is set to total and outlet to static
- HP and compressor and turbine is total to total
- The compressor and turbine maps are referenced through the “Map Object or File” attribute and must be located in the same folder on the computer as the model. When the first simulation is run the compressor and turbine templates imports and reads the data from the SAE file, they perform some extrapolation of the maps at the extremities following the established trends of the map, this is done because the SAE files provided are often not the complete map.

Intercoolers

The charge air cooler is not built as an exact replica of a real one. It is built around a “PipeRound” template that is constructed as 242 separate small diameter pipes all with the same length. In both ends there is a endcap in form of a “FlowSplitGeneral” template defined with a volume, these collect the flow from all the small pipes. Then the heat transfer multiplier for the 242 small pipes is set to an artificially high value, and the imposed wall temperature is set to the expected temperature of cooled air after the charge air cooler. In effect what happens is that the temperature of the air is forced down to the imposed wall temperature very fast because of the high heat transfer multiplier. Then the boost air temperature can be easily controlled but it requires knowledge of realistic values for the cooled air temperature. In short the intercooler is not dynamic, it simply delivers what is defined. The performance of the intercooler is based on the real one and the available cooling water temperature, through easily available formulas the efficiency of the IC can easily be calculated and checked against the real numbers.

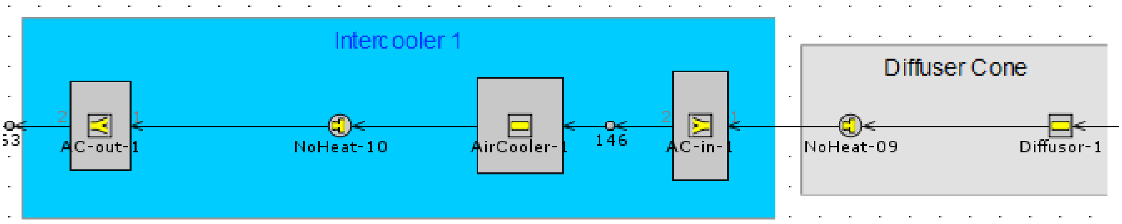


Figure 28: Engine model intercooler

At each side of the intercooler part is a flow-split part which acts as a endcap for all the small pipes of the intercooler, these endcaps are given volumetric size. In the Diffuser Cone of the intercooler there is a diffusor part which increases the pipe diameter going into the intercooler. The other part is from a “OrificeConn” template and its function is to stop any heat transfer between the intercooler and the piping, this is to make sure the cooling happens only in the intercooler.

Regarding the geometry of the piping that connects the different parts, turbochargers and intercoolers, it is loosely inspired by the design of the compact Power2 turbocharger from ABB, see figure 29, be advised the picture doesn't include the intercoolers. Assumptions has been made to get the dimensions right, however this can be something to look more closely at in the future. It will in any cases be necessary to make assumptions when modeling a fictive turbocharging system that doesn't exist in real life.

Figure 29: ABB`s Power2 turbocharger

5.2 Miller Timing

I want to run multiple Miller timings in the same model. To achieve this I have to make a case setup parameter where I can vary the value for each case. In the engine model there are 2 intake valves for each cylinder. They are called intake valve short, IVS, and intake valve long, IVL. This is because on the physical engine one of the intake valves has to be longer because of the shape of the engine where it is seated. Both however have the same diameter and lift height. In the model it is used a ValveCamConn template to make one object for IVS and one for IVL. Each object has 6 parts, one for each cylinder, these parts represent the intake valves of the engine. See figure 30.

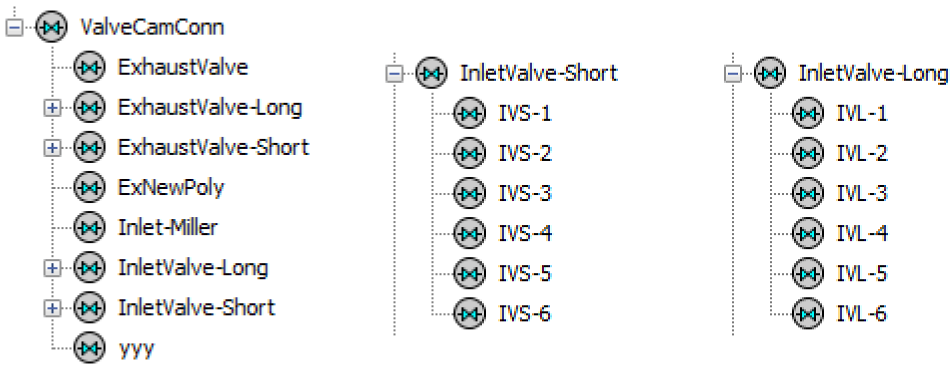


Figure 30: Intake Valve template, object and parts

The lift curves are located in the Lift Array menu of the intake valve objects. They can be listed directly as a numerical representation of the curve itself in the provided table. Since there is a need to change the intake valve lift curve for different Miller timings it is better to insert a parameter reference in the table, one for the crank angle and one for the lift height of the valve. These parameters will appear in the case setup of the model. Here you can easily fill in different values for the different cases you will be running. Since the values we want to fill in essentially are table representations of lift curves we link to a reference array template containing the tabulated values. There is one array template for the crank angle of each Miller timing, representing the x-axis values of the lift curve. There is one array template for the lift height of each Miller timing, representing the y-axis values of the lift curve. In figure 31, you can see the array templates where the tabulated values are stored, in overview to the left and inside to the right. In the middle of figure 31 from the top and down: you see the lift array folder is chosen inside the valve object, then the parameter link inside the lift array table and at the bottom you have the parameter inside the case setup with the link to the appropriate array template in green text.

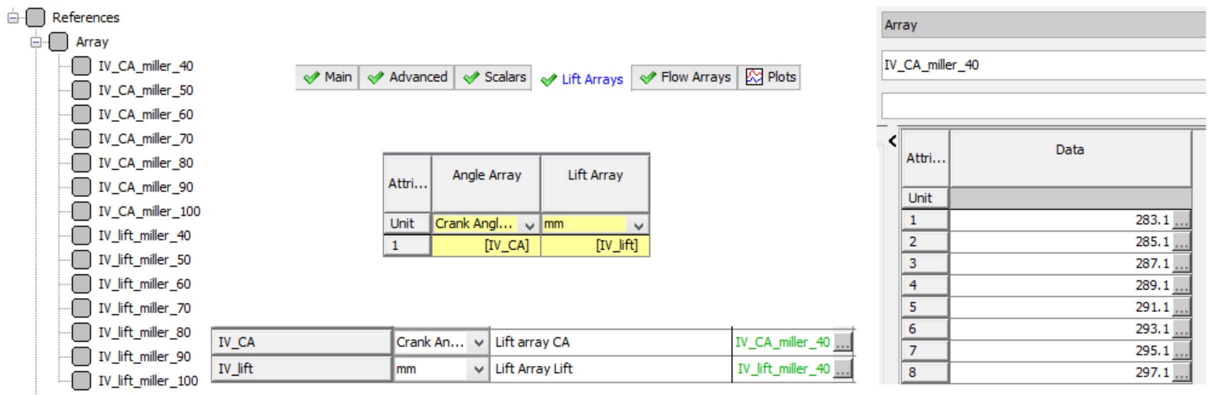


Figure 31: GT-Power lift curve setup

The concept of Miller timing evolve around adjusting the closing of the air intake valve. The bell shaped lift curve of the intake valves must be adjusted to achieve this. If you design a lift curve from scratch there are many considerations. You must account for the acceleration of the valve and especially during the closing. This is because if you get it wrong you risk fatigue problems and worst case failure of the valves if they are not closed gently enough. The solution was to shorten the lift curve at its top center. In this way the established and well functioning lower parts of the lift curve are kept intact.

In figure 32 the lift curves of all Miller timings from Miller 40 to Miller 100 is included. It is clear that the lift curves get more and more pointy at the top for increasing Miller. These lift curves are not realistic for use on a conventional engine with camshaft driven valves. It would be difficult to construct cams capable of dealing with the sharp edges, the result would probably be material fatigue and increased maintenance intervals. But there exists other solutions than the camshaft for driving the intake valves, like ABB`s Valve Control Management, VCM, (Ltd., 2012). The VCM system uses electro-hydraulic technology to achieve progressive variation of inlet or exhaust valve timing, systems like this could handle the unorthodox lift curves. Because of available solutions in the marketplace, confirmation was given form RR BEAS, (Seeberg, 2015), to proceed with these lift curves.

This is an easy solution for altering the already established lift curves while changing them as little as possible. For the engine in question the valves are regarded as opened or closed at a value of 1.8 millimeters. The original lift curves are already at a Miller timing of 40, which means the air intake valve closes 40 CA degrees before BDC. The bell shaped lift curve of the intake valves is quite narrow. This means that for extreme Miller timings you will decrease the maximum lift of the valve. But this is also within acceptable values for the task in hand.

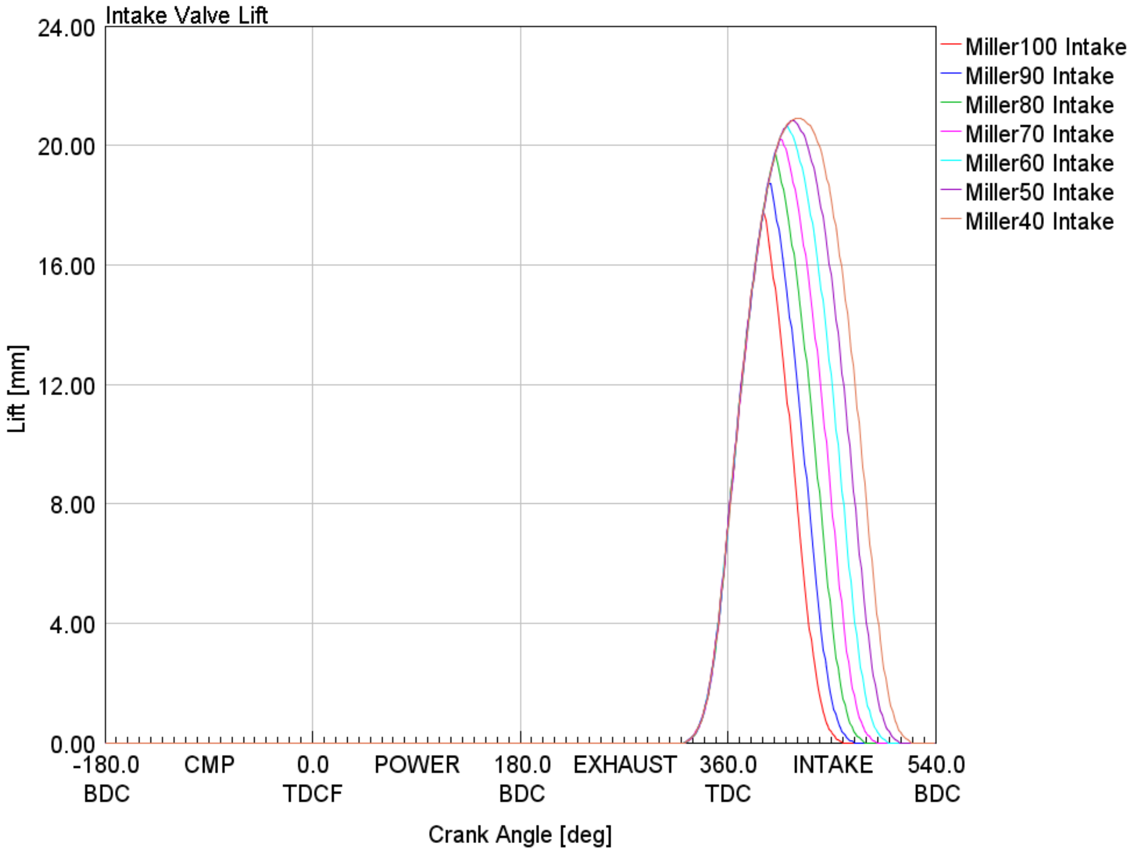


Figure 32: Lift curves for all Miller timings

5.3 Ideal 2-Stage Turbocharged Engine Model

The reason behind looking at an ideal turbocharger is the suspicion that the 2-stage turbocharger model used in this thesis might not be a good enough match with the engine. In that case it will not be able to take advantage of the full potential of 2-stage turbocharging with Miller timing. If so, it will be difficult to establish clear trends and results.

The easiest way to make an ideal model with enhanced performance is to tune the turbochargers. To create the ideal model the normal 2-stage turbocharger model was used as a base. Inside both the compressor and turbine templates there is an attribute, inside the options menu, which is called “Efficiency Multiplier”. By default this is 1, when setting a value above 1 the efficiency will be improved, see figure (31). The values used for efficiency multiplier were:

- LP Compressor: 1.05 (5%)
- HP Compressor: 1.05 (5%)
- LP Turbine: 1.05 (5%)
- HP Turbine: 1.15 (15%)

The reason why the HP turbine was adjusted a lot more is that it was underperforming, delivering efficiencies 10 % below the other compressors and turbines. These were the only adjustments made to the model, otherwise it is identical with the 2-stage turbocharged model used in this master thesis. The effect of increasing efficiencies of the turbochargers is that they deliver higher boost pressure, which was the goal all along. The extra boost pressure will help to investigate the potential of 2-stage turbocharging combined with Miller timing further.

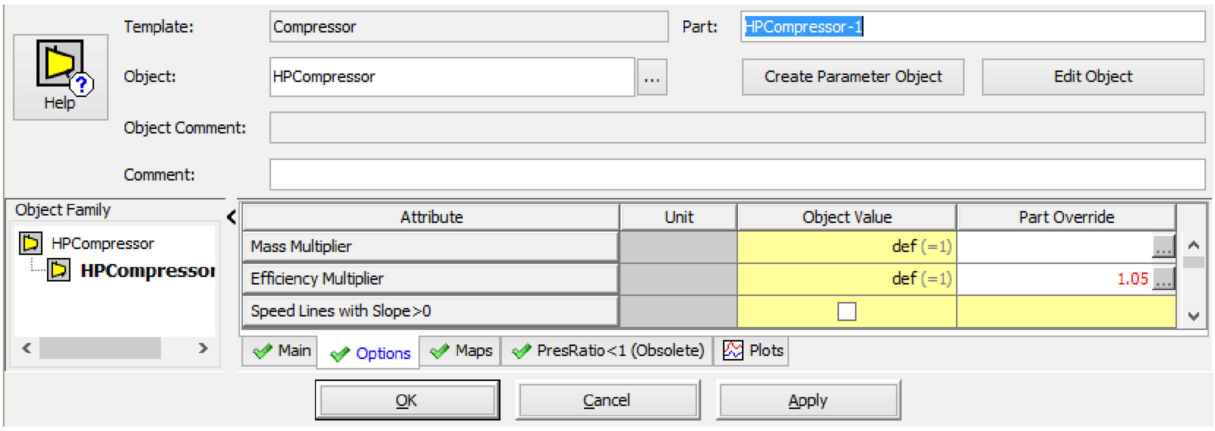


Figure 33: Turbocharger efficiency multiplier

6 Simulations

6.1 Strategy

The strategy for performing simulations is to use the model with single stage turbocharger delivered by Rolls-Royce Bergen Engines as a base model and compare the performance with a model using 2-stage turbocharging. The goal is to quantify the differences and possible improvements. The accuracy of the base model is already established and serves as a quality control for the simulations. The setup for simulations of 2-stage turbocharger in this chapter:

- Evaluate optimal Miller timing by running the 2-stage turbocharged model at 100 % MCR, and try Miller timings from 40-100 CAD with increments of 10 CAD
- Run the 2-stage turbocharger model with the selected optimal Miller timing through the whole load range and analyze the results. The same is to be done with the ideal 2-stage turbocharged model constructed in chapter 5.3 This is not to be analyzed here, only used in comparison
- Perform a comparison of the following models
    - 1-stage turbocharged engine model
    - 2-stage turbocharged engine model
    - 2-stage turbocharged ideal engine model
    - Evaluate the results
    - Look at NO_x reduction
    - Look at BSFC reduction
- Setup an energy balance for the engine and compare the three models, looking at BSFC and determining the size of the different losses

Demand when simulating, temp exh before turbine <600C, optimal lambda 2,3-2.4, but no less than 2.0. The NO_x concentration in ppm is measured at the tailpipe and then averaged for the most accurate result, this is done in all models. It is converted from ppm into g/kWh by using formula (X, in verification chapter). The NO_x emissions limit for the engine running at 1000 RPM is 9 g/kWh. The BSFC of the 1-stage turbocharger engine model at 2000 kW is 198 g/kWh so this is also a target to beat.

6.2 Evaluate Optimal Miller Timing

The goal is to find the optimal Miller timing for reduction of both BSFC and NO_x emissions. All the simulations in this master thesis are run at a constant 1000 RPM. The strategy is to use the 100 % MCR power output as a basis for finding the optimal Miller timing. This Miller timing will later be used for all loads except the two smallest which retain their standard value of Miller 10 defined in chapter 4.1. This way is in tune with how the real engine is built with only two possible settings for variable valve timing.

In table 11 the range of Miller timings used in the simulation can be found. Miller 40 is the setting used for the 1-stage turbocharged C25:33L6 real engine. Miller 100 is a very extreme setting, but is included to reveal the trends of increasing Miller timing. The case numbers of table 11 correspond to the case number on the plots in figures 34-36.

Case	1	2	3	4	5	6	7	Unit
Speed	1000	1000	1000	1000	1000	1000	1000	RPM
Power	2000	2000	2000	2000	2000	2000	2000	kW
Miller	100	90	80	70	60	50	40	CAD

Table 11: Case setup optimal Miller timing

The GT-Power files for this model:

- Name: MCR100_all_Miller.gtm
- Name: Evaluate Optimal Miller.gu
- Location: Restricted zip-file, (Danielsen, 2015b)

Criteria for choosing optimal Miller timing:

- The case number that reveals the lowest combined value of BSFC and NO_x
- Lambda: Preferred, 2.3-2.4, minimum 2.0
- AF-ratio: reference value 1-stage TC model: 31.3 (trapped air/total fuel)
- Trapping ratio: close to 0.91, which is the value of 1-stage TC model at 2000 kW
- Exhaust temperature before turbine: < 600°C or < 873K

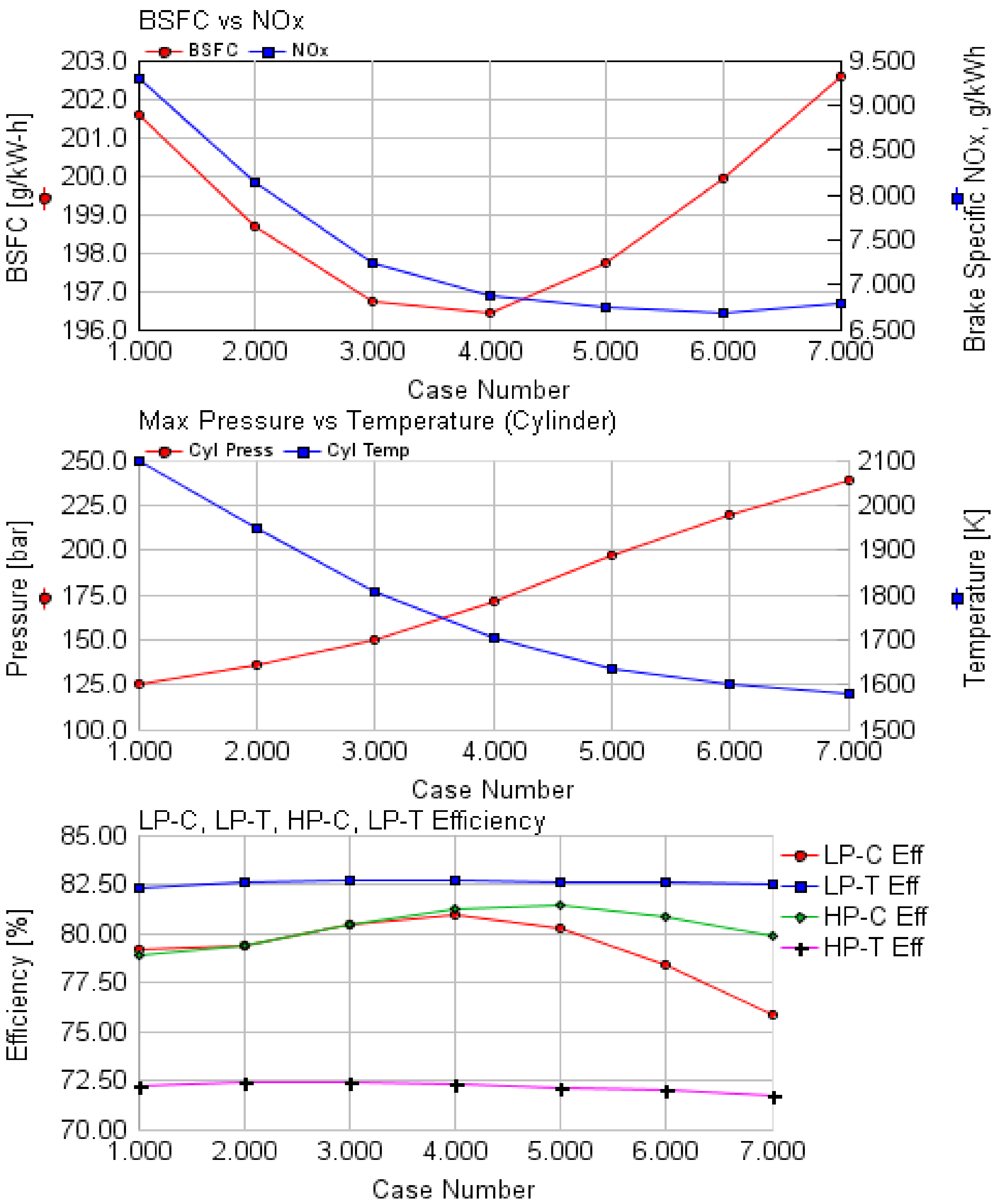


Figure 34: Optimal Miller plot 1

Figure 34: *Top plot:* This shows the BSFC against BS_{NOx} emissions, the optimum is found in case 4, representing Miller 70. *Middle plot:* For case 4 both maximum pressure and temperature are acceptable, although temperature is not that low. *Bottom plot:* Both compressors and turbines are at their maximum efficiency or close to it for case 4, so this looks good.

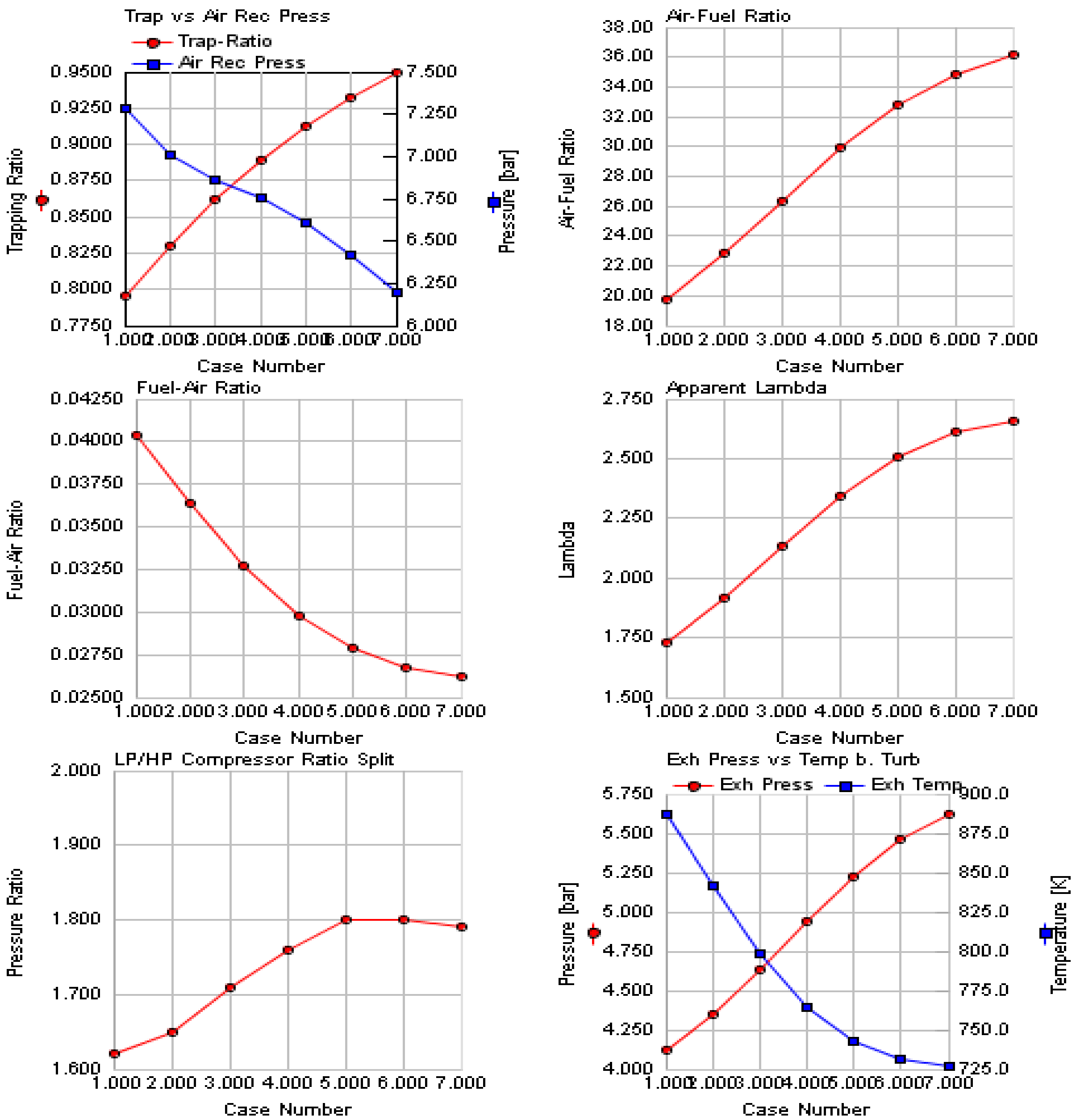


Figure 35: Optimal Miller plot 2

Figure 35: *Top left:* for case 4 the trapping ratio is a bit low but adequate. *Top right:* AF-ratio is a bit low in case 4 (trapped air/total fuel) but acceptable. *Middle left:* Fuel-air ratio is defined as (total fuel/inducted air). *Middle right:* lambda is fine for case 4, the criteria, 2.3 < lambda < 2.4, seems to allow quite more than the AF-ratio and trapping ratio, this indicates a wider acceptance, lambda is defined in formula (21). *Bottom left:* The pressure ratio split between compressors is a bit low for all cases compared to the desired value of around 2, see beginning of chapter 4. *Bottom right:* Both exhaust pressure and temperature before turbine are fine for case 4, but case 4 is fine.

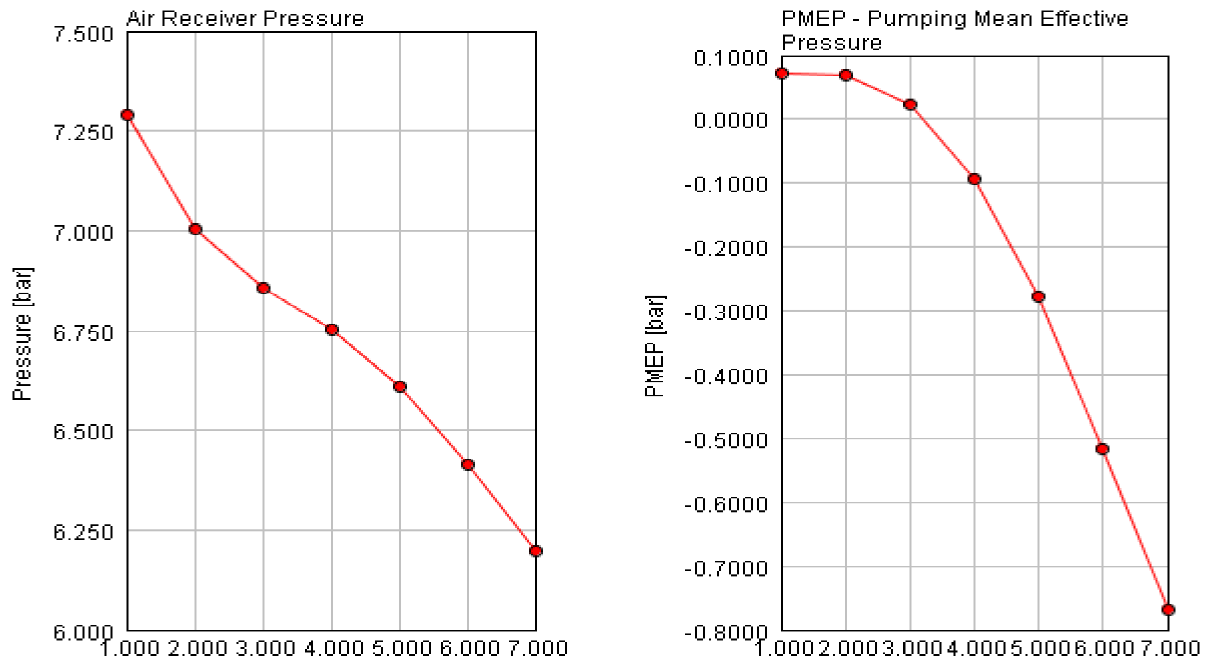


Figure 36: Optimal Miller plot 3

Figure 36: The air receiver pressure is fine for Miller 70, and noted as increasing with increasing Miller. Write the explanation of how the PMEP is affected by Miller and include some reference to 4-stroke picture.

The optimum Miller timing was found in the top plot of figure 34. For case 4, representing Miller 70, there was a nice compromise with both low BSFC and BS_{NOx} emissions. None of the other plots revealed unacceptable values for Miller 70. But for Miller 80-100 some clear trends emerged when looking at the rest of the plots:

- Air receiver pressure increases with increasing Miller, which is an expected result
- Exhaust pressure drops with increasing Miller
- The AF-ratio and lambda drops with increasing Miller
- The Fuel-Air ratio increases with increasing Miller
- Both maximum cylinder temperature and exhaust temperature increase with increasing Miller
- The split of total pressure ratio between LP and HP compressor is a little less than the ideal ratio between them of LP/HP around 2
- The HP turbine efficiency is significantly less than the other compressors and turbines

All these trends put together shows that the 2-stage turbocharger matched with this engine is not capable of increasing Miller beyond Miller 70 with good results.

Optimal Miller Timing

Miller 70 is chosen as the optimum Miller timing and will be used in the further simulations as replacement for the Miller 40 setting of the 1-stage turbocharger model, see chapter 4 for information on the 1-stage turbocharger.

6.3 Analysis of 2-Stage Turbocharger Model

In this chapter the 2-stage turbocharger model will be analyzed. It is run with Miller 70, which was found to be the optimal Miller timing in the previous chapter. The case setup for this simulation can be found below in table 12. The case numbers of the table corresponds to the case numbers in the individual plots.

Case	1	2	3	4	5	6	Unit
Speed	1000	1000	1000	1000	1000	1000	RPM
Power	2200	2000	1500	1000	500	200	kW
Miller	70	70	70	70	70	70	CAD

Table 12: Case setup for analysis of 2-ST TC

The plots in figure 37 and 38 show the trend of the 2-stage turbocharged engine for Miller 70 over the whole load range. Some of the plots are a bit coarse and it is hard to read precise values. The plots show a significant increase in boost pressure with around 7.5 bar in case 1, in case 2 we find the optimal compromise between BSFC and NO_x, maximum cylinder pressure is noticeably reduced, the maximum cylinder temperature is still high. Compressor and turbine efficiencies are all high except the HP turbine which seems to be the weak link in that context. NO_x is quite low for the relatively high maximum temperature in the cylinder, at the same time this correlates with a slightly low AF-ratio at those loads, lambda values are fine. Exhaust pressure and temperature follow the trend of boost pressure and temperature. BSFC and thermal efficiency are reciprocal as expected. The pressure ratio split between LP/HP compressors is close to the optimal value 2, see formula (X) chapter 3. The LP compressor gives the highest temperature rise, which also is expected because it has the largest pressure ratio.

The GT-POST files containing the results and plots used here can be found in the restricted zip-file, (Danielsen, 2015a). Filenames are listed below:

- Miller70_all_loads.gdx
- Comparison.gu

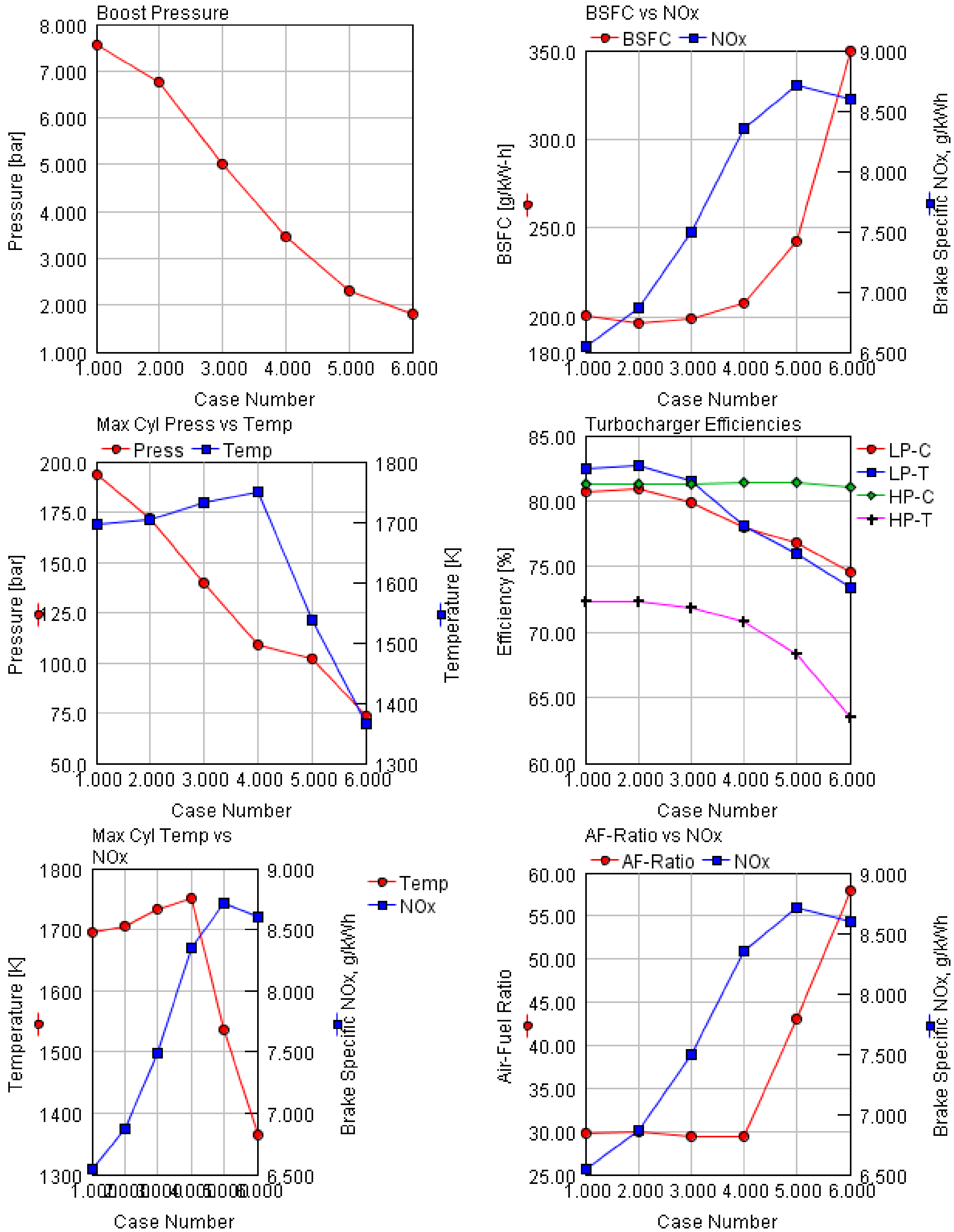


Figure 37: 2-ST TC analysis plot 1

Figure 37: *Top left*: Boost pressure in air receiver. *Top right*: Case 2 is optimal for low BSFC and NO_x. *Middle left*: Max cylinder temperature versus max cylinder pressure. *Middle right*: Efficiencies of the two compressors and the two turbines, HP turbine has a low efficiency. *Bottom Left*: Increase in temperature gives increase in NO_x. *Bottom right*: Very high NO_x emissions and AF-Ratio at low loads.

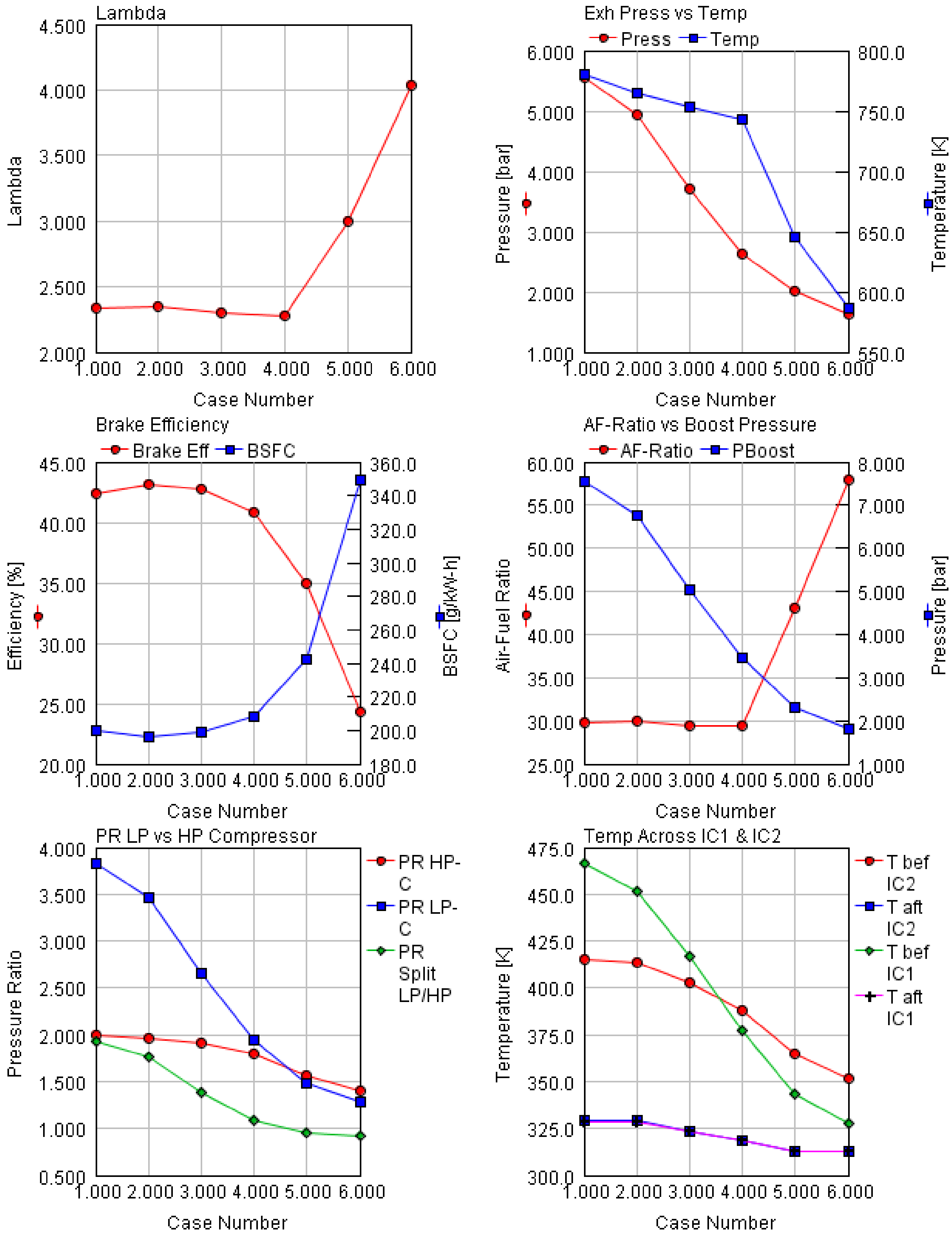


Figure 38: 2-ST TC analysis plot 2

Figure 38: *Top left*: High lambda at low loads. *Top right*: Pressure and temperature correlate. *Middle left*: Reciprocal as expected. *Middle right*: High pressure high AF-ratio. *Bottom left*: PR means pressure ratio, PR split LP/HP has an optimum around 2. *Bottom right*: T = temperature, bef = before, IC1 =intercooler 1between compressors, IC2 = intercooler 2 after HP compressor. Plot shows temperature difference across the two intercoolers.

6.4 Comparison of Models

Here the three different models will be compared with each other to highlight the differences between them.

- 1-stage turbocharged engine model, the original from RR BEAS
- 2-stage turbocharged model, the one constructed in this thesis using the provided engine model along with the compressor and turbine maps.
- 2-stage turbocharged ideal model, in essence the same model as the normal 2-stage but the performance of the turbochargers have been enhanced to boost the potential benefits of combining 2-stage TC with the Miller cycle. The reason for this model is to see what potential benefits could be had, with a better match of engine and turbochargers.

Comment

The 2-stage turbocharged ideal model exhibits strange results for case 5 & 6, the BSFC here is in tune with the load placed by the load controller, but it exhibits some really strange and too high values for power, torque, AF-ratio and so on, the mass of fuel injected is larger than in case 4, but when checking the model it is not possible to find anything wrong which can explain this. All the values are correct, the set BMEP of the load controller, the initial injected mass of fuel, the combustion model is also correct which by the way explains how the BSFC seems to be correct. The results of case 5 & 6 of the 2-ST TC ideal model must be disregarded and is not included. This was discovered just before deadline for delivery of the master thesis and there was no time to correct this error, just to remove the deviating values.

Case Setup

The models will run the same case setup, which is defined in table 13 below. As mentioned for the 2-stage turbocharged ideal model the values of case 5 and 6 will not be included in any plots. That being said, case 5 and 6 are the two lowest loads and not as a strong a focus as on the 50-100 % of MCR load range.

Case	1	2	3	4	5	6	Unit
Speed	1000	1000	1000	1000	1000	1000	RPM
Power	2200	2000	1500	1000	500	200	kW
Miller	70	70	70	70	70	70	CAD

Table 13: Case setup comparison

The plots are made extra big for this last comparison. That is to be able to see the differences between the three models as clearly as possible. Explanation of abbreviated plot names:

- 1-ST: Is the 1-stage turbocharged engine model
- 2-ST: Is the 2-stage turbocharged model
- Ideal: Is the 2-stage turbocharged model with increased efficiency

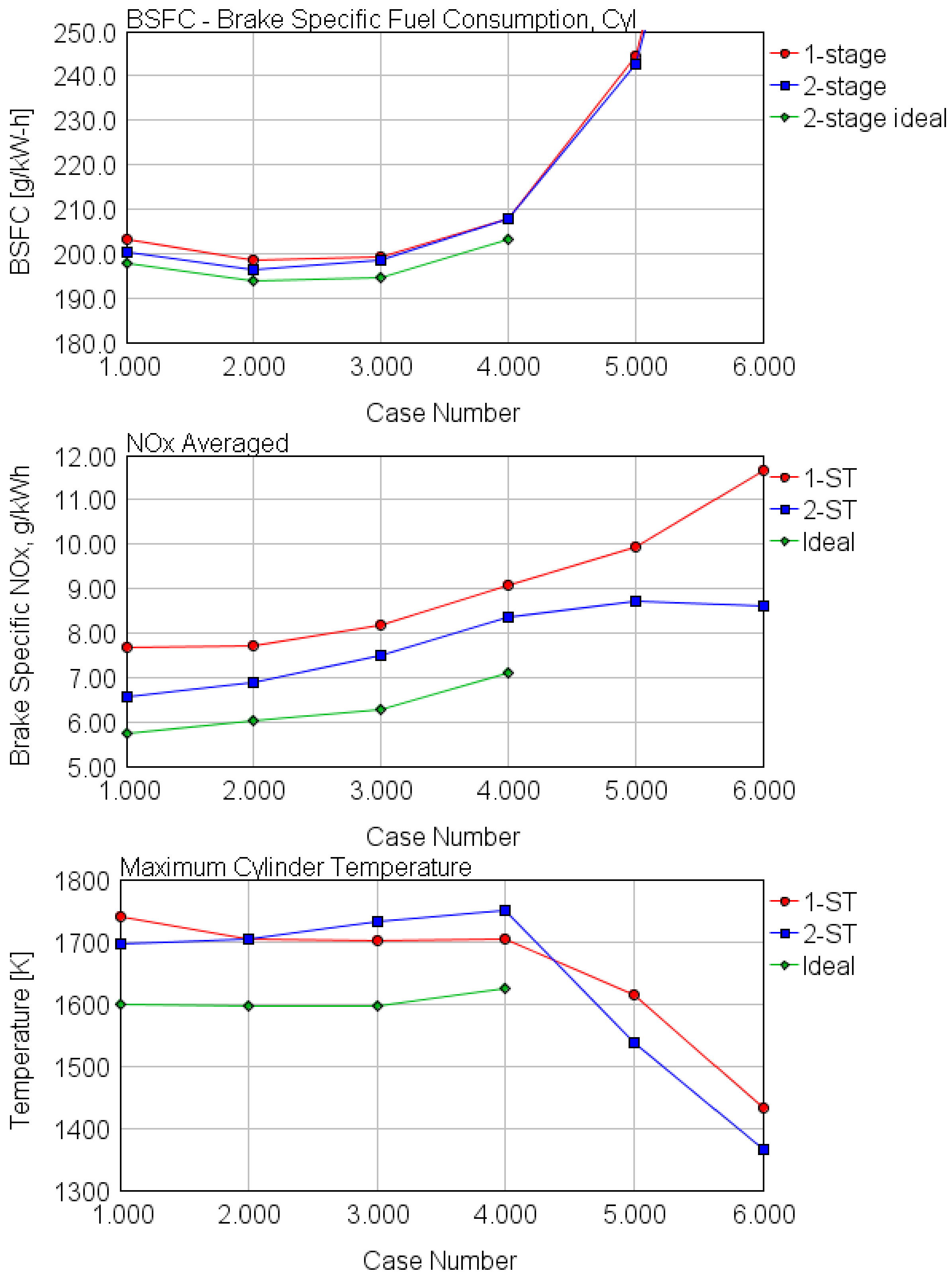


Figure 39: Comparison of models, plot 1

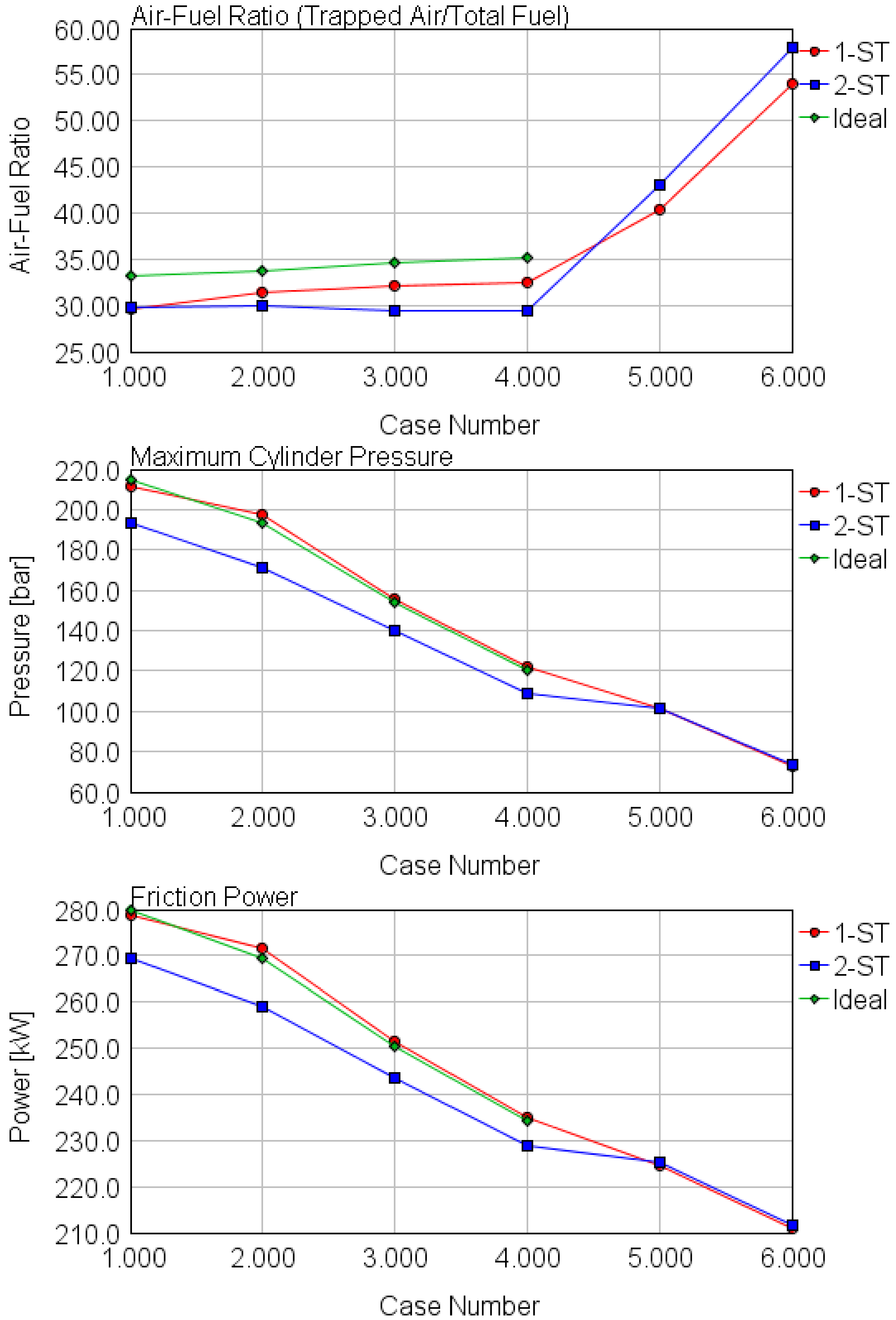


Figure 40: Comparison of models, plot 2

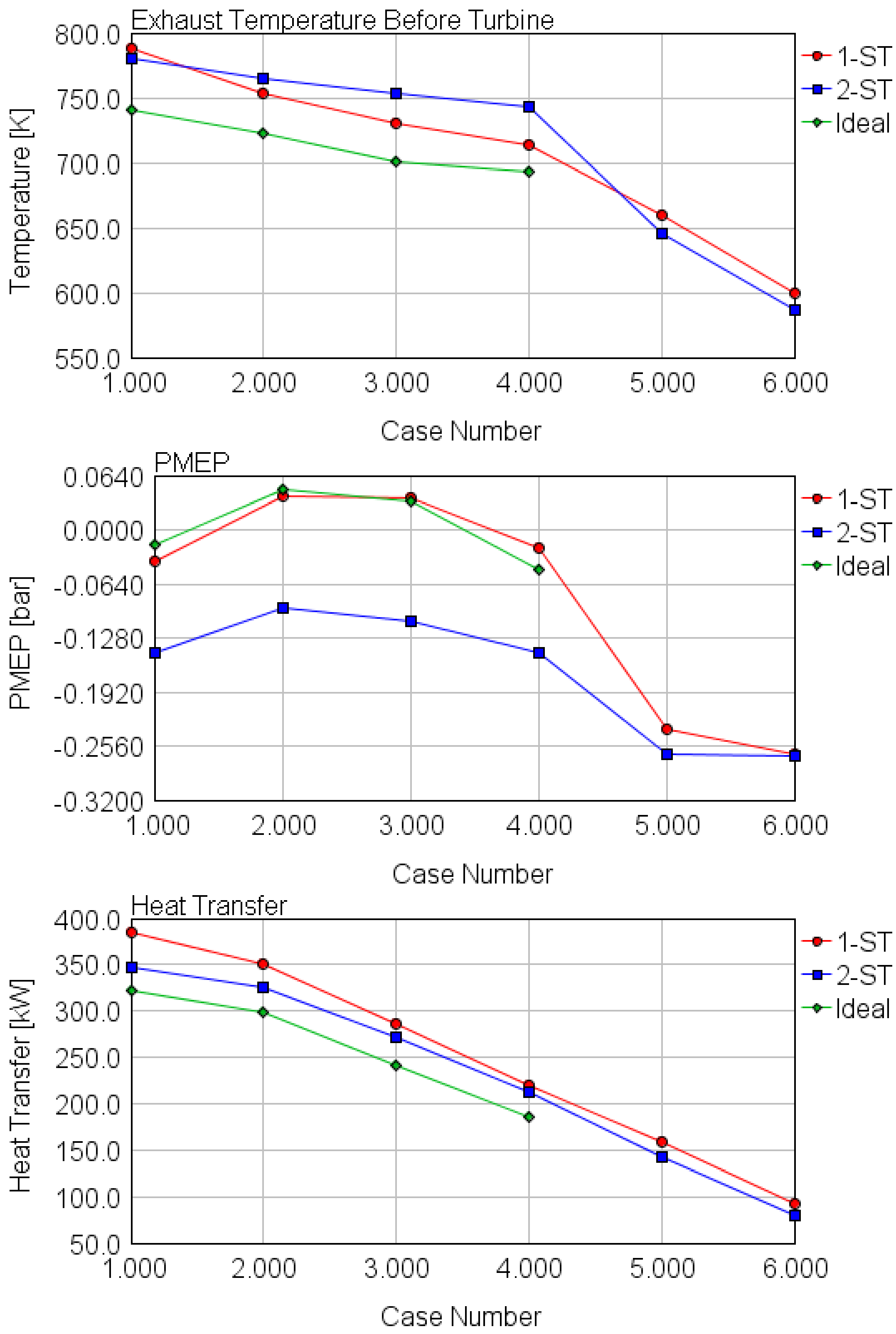


Figure 41: Comparison of models, plot 3

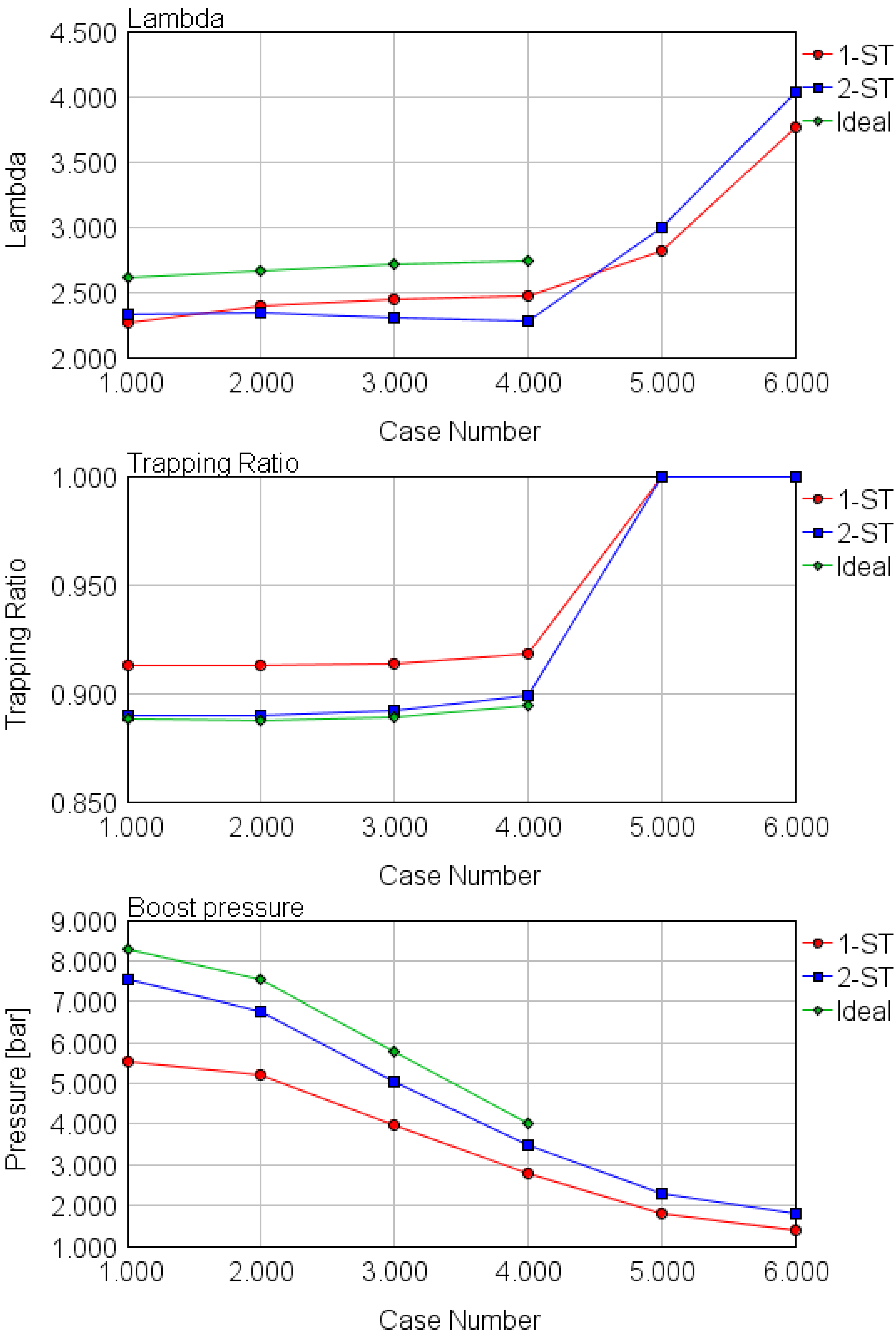


Figure 42: Comparison of models, plot 3

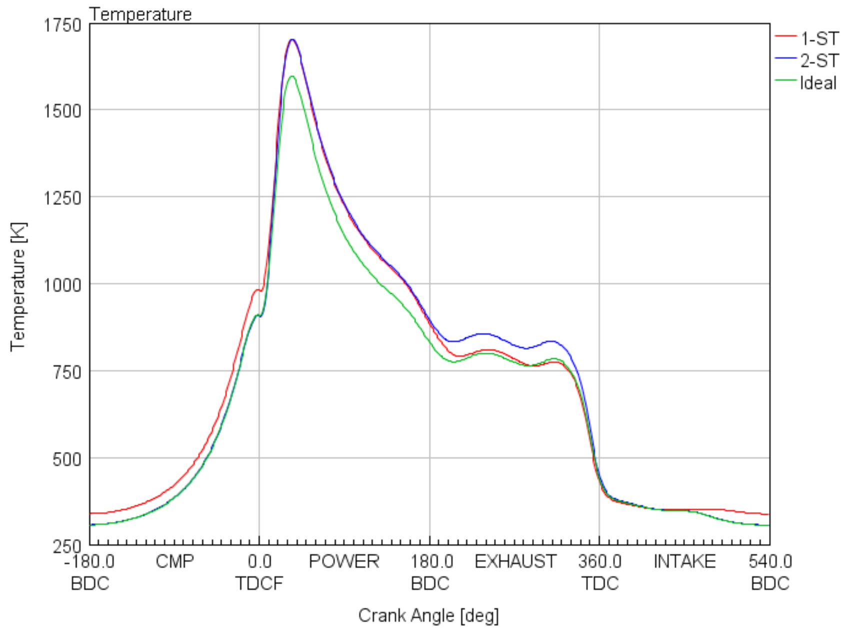


Figure 43: Comparison of max cylinder temperature at 100 % MCR

The files used for this comparison is located in the restricted zip-file, (Danielsen, 2015a). Name of the GT-Post files containing the results :

- C-engine_1st_tc_comparison_6loads.gdx
- Miller70_all_loads.gdx
- Miller70_all_loads_ideal.gdx
- Comparison.gu

Results

BSFC is a bit lower in the 2-ST than the 1-ST model and only for the two top loads. The reduction is more significant in the ideal model and holds true for the 4 top loads. The NO_x emissions is reduced from the values of the 1-ST in both the 2-ST and Ideal model, for the Ideal model this seems logical considering it has a much lower maximum cylinder temperature, around 100 Kelvin. For the 2-ST model it is more complicated, for case 2-4 it has the same or higher maximum cylinder temperature as the 1-ST model, at the same time it has a lower NO_x in all those cases. This indicates that the NO_x production is maybe dependant on more factors than the cylinder temperature alone. In figure 44, (Heywood, 1988), the NO production is plotted as a function of fuel-air equivalence ratio, see formula 25 chapter 3.1. This means that for a fuel-air equivalence ratio of 1, there is just enough air for combustion of all the fuel present, a lower value indicates a surplus of air not used for combustion. In short, the NO production drops for

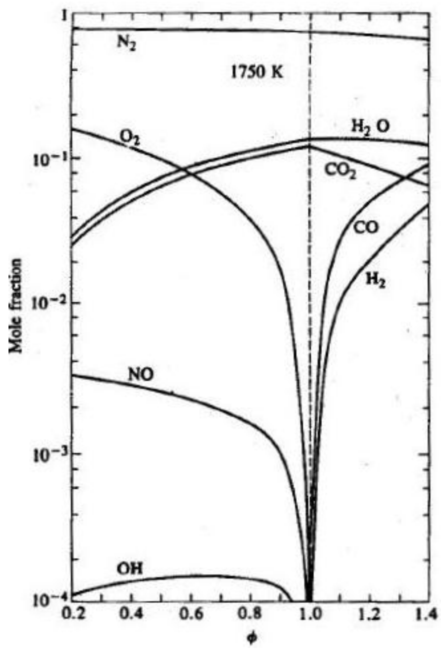


Figure 44: NO vs fuel-air equivalence ratio

reduction in the AF-ratio. Looking at the AF-ratio plot, it is clear that the 2-ST model has a lower AF-ratio than the 1-ST model in cases 2-4. This could be a possible explanation of the results in simulations. However, this is just regarding what happens in the simulation model and can be due to an imperfect NO_x prediction capability in GT-Power. The explanation of increased maximum cylinder temperature for 2-ST model with lower AF-ratio can be found by looking at the adiabatic flame temperature. The lower excess air ratio during combustion the higher temperature of the combustion products, which reflect the cylinder temperature, (Michael J. Moran, 2010).

The 1-ST and Ideal model has higher cylinder pressures than the 2-ST model, this could be explained by them having a higher AF-ratio meaning higher mass flow. The high pressures looks like they increase the friction power, the 1-ST and Ideal models have higher friction than the 2-ST with its lower maximum cylinder pressures. The exhaust temperature looks to reflect the maximum cylinder temperature. PMEP is the pumping work performed by the cylinders during the intake and compression stroke, see figure 1 for picture of the 4-stroke cycle. For the 1-ST and Ideal model it goes from negative to positive before it decreases again, the 2-ST model has the same shaped curve but never enters a positive value. A possible explanation of this is that the 2-ST model doesn't deliver high enough pressures for its Miller timing. The Ideal model has higher boost pressure and can achieve a better PMEP, the 1-ST model has the lowest boost pressure but it also has the smallest Miller timing at Miller 40. It should also be pointed out that the PMEP values are small compares to the size of FMEP and heat transfer.

The heat transfer correlates closely with the maximum cylinder temperature, as expected, but the differences in kW are significant. Lambda is fine for the 1-ST and 2-ST model, but almost a bit high for the Ideal model. The trapping ratio of the 2-ST and Ideal model is lower than for the 1-ST model. This is due to the higher boost pressures of the 2-ST and ideal model while they share the same overlap between exhaust valves and intake valves as the 1-ST model. The higher pressures increase the mass flow of air through the cylinder and out the exhaust valves before they close. This indicates that variable valve overlap should be looked into for 2-stage turbocharging with high pressures. The final plot shows the boost pressure of the three models, as expected the Ideal model has the highest pressure followed by the 2-ST and then 1-ST model. It is interesting to see that even with the significant increase in boost pressure from the 1-ST to the 2-ST model the simulations suggest that this is maybe not enough. The boost pressure of the ideal model actually passes 8 bar for case 1. In figure 43 one can see how the temperature drops for the 2-ST and Ideal model when the intake valve closes earlier than for the 1-ST model. The temperature then stays lower for the whole compression stroke, but during combustion the 2-ST temperature is just as high as the 1-ST temperature while the Ideal model is around 100 K lower. The most probable explanation as to why the 2-ST loses its advantage during combustion is the fact that it has a lower AF-ratio than the ideal model, which could lead to a higher combustion temperature.

This can be explained through the theory of adiabatic flame temperature which shows how a lower excess of air during combustion drives the exhaust temperature up, (Michael J. Moran, 2010). In figure 43 it is clear that the 2-ST model has the highest exhaust temperature of all models, and the exhaust temperature is directly linked to the cylinder temperature.

Reduction of NO_x Emissions

The reduction of NO_x emissions is one of the key factors of this master thesis. The differences between the 2-ST and 1-ST, plus the difference between the Ideal and the 1-ST model are highlighted in green.

NO_x Emissions							
Case	1	2	3	4	5	6	Unit
1-ST TC	7,65	7,72	8,18	9,07	9,94	11,67	g/kWh
2-ST TC	6,54	6,86	7,49	8,34	8,69	8,60	g/kWh
Difference	-1,11	-0,86	-0,69	-0,73	-1,25	-3,07	g/kWh
2-ST TC Ideal	5,75	6,02	6,28	7,09	-	-	g/kWh
Difference	-1,9	-1,7	-1,9	-1,98	-	-	g/kWh

Table 14: NOx reduction

Reduction of BSFC

The reduction of BSFC is one of the key factors of this master thesis. The differences between 2-ST and 1-ST, as well as between Ideal and 1-ST model are highlighted in green.

BSFC							
Case	1	2	3	4	5	6	Unit
1-ST TC	203,1	198,3	199,1	208,0	244,6	354,9	g/kWh
2-ST TC	200,2	196,4	198,6	207,9	242,5	349,3	g/kWh
Difference	-2.9	-1,9	-0,5	-0,1	-2,1	-5,6	g/kWh
2-ST TC Ideal	197,7	193,7	194,5	203,2	-	-	g/kWh
Difference	-5,4	-4,6	-4,6	-4,8	-	-	g/kWh

Table 15: BSFC reduction

6.5 Energy Balance

An energy balance can be used to increase the understanding of what happens in the simulations, it keeps track of all the energy entering and leaving the system. In figure 45 below is an energy balance flowchart for the engine model. It shows the energy division used by GT-Power. The heat transfer is the sum of heat radiation and the heat loss to lubrication oil and cooling water in a normal engine. The energy balance describes all heat losses from the cylinders and engine in the model, it is in a way simplified compared to real life, because of its more coarse divisions of energies, but it is correct for the model in question.

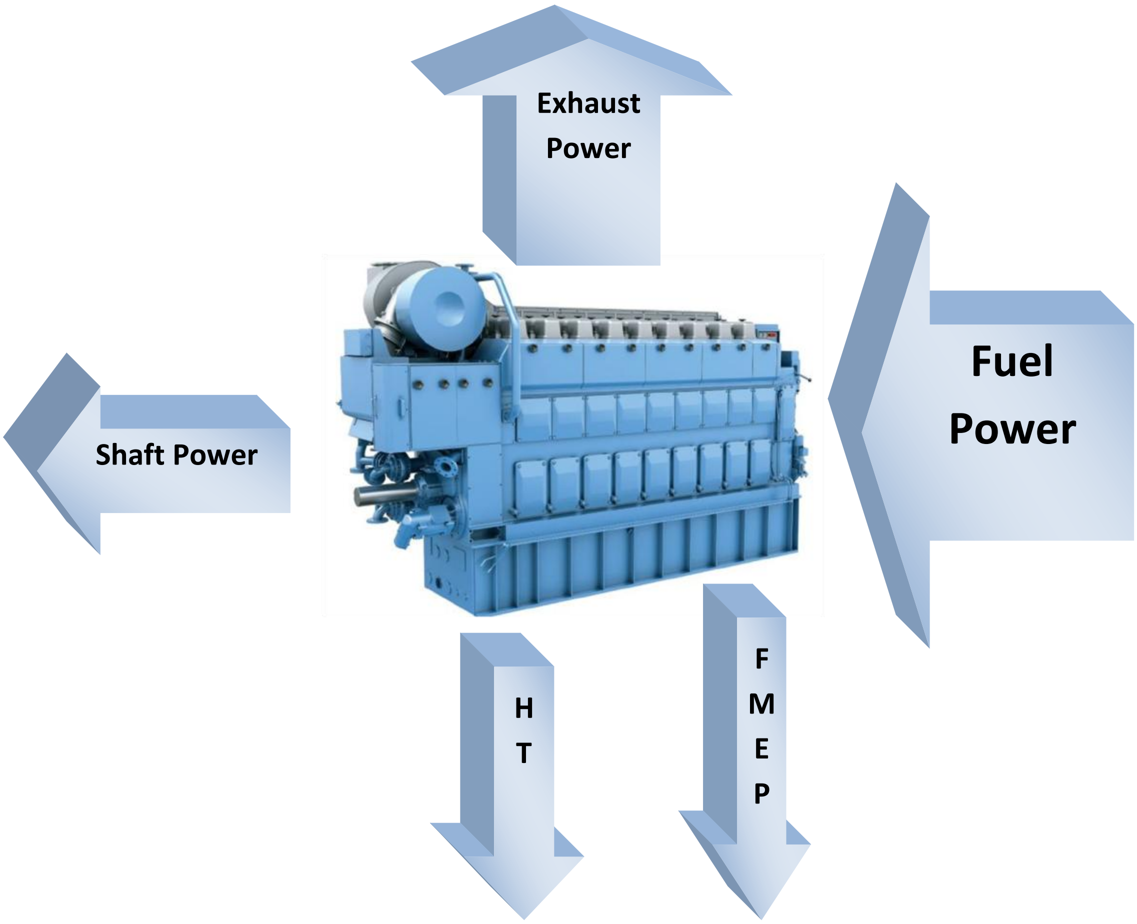


Figure 45: Energy balance

In the engine model the load is controlled by setting the BMEP to a desired value, BMEP can also be written, p_{me}. The model calculates the output power through the following formula:

$$P_e = iV_{sw}p_{me}n_a \tag{49}$$

The formula combines BMEP with the geometry and speed of the engine to arrive at output power in kW. It is nevertheless interesting to take a closer look at what BMEP represents inside the model, to increase the understanding of how the model works.

BMEP = IMEP720 – FMEP = IMEP360 + PMEP – FMEP (50)

- IMEP360: Indicated mean effective pressure for 360 CAD of the 4-stroke cycle, representing the expansion and exhaust stroke.

- PMEP: Pumping mean effective pressure, the pumping work of the engine cylinders during the intake and compression stroke, is influenced by turbocharging.
- FMEP: Friction mean effective pressure, represents all the frictional losses in the engine, contains one constant, one variable affected by peak cylinder pressure and two variables affected by mean piston speed.
- IMEP720: Indicated mean effective pressure for 720 CAD, representing the whole 4-stroke engine cycle.

The indicated mean effective pressure, IMEP, is the link between mass of fuel needed to reach the set BMEP giving the desired output power. IMEP represents the mean value of the cylinder pressure during the engine cycle, if this is multiplied by the area at the top of the piston the result represents the gas forces inside the cylinder, acting on the piston.

$$F_g = IMEP \cdot A_{ph} \tag{51}$$

These gas forces are a direct result of the combustion process, meaning more fuel burned gives higher IMEP and less fuel burned gives lower IMEP. Looking at formula (X, BMEP formula), if BMEP is held as a constant, what happens to IMEP when PMEP or FMEP change:

- If PMEP increase, IMEP360 decrease to maintain BMEP
- If PMEP decrease, IMEP360 increase to maintain BMEP
- If FMEP decrease, IMEP360 decrease to maintain BMEP
- If FMEP increase, IMEP360 increase to maintain BMEP

If we look at the energy balance of figure (42) we see that it is easy to formulate this into an equation:

$$E_b = P_{fuel} - P_{exh} - P_{fr} - P_{HT} - P_{shaft} = 0 \tag{52}$$

This formula says the exact same thing as the energy balance flowchart, energy in minus energy out is zero.

Comparison of Models

To explore this further we can set up an energy balance for the three models in chapter 6.4 to highlight the differences between them. The comparison will be made for all three models at 100 % MCR, which in this case means the engines are run at 1000 RPM with 2000 kW power output. It is too time consuming and really not necessary to look at all loads just to demonstrate the energy balance.

	1-ST TC	2-ST TC	Difference	2-ST TC Ideal	Difference	Unit
BSFC	198,33	196,41	-1,92	193,65	-4,68	g/kWh
Fuel Power calc	4705	4659	-45	4594	-111	kW
Fuel Power	4676	4631	-45	4567	-110	kW
Exhaust Power	2054	2048	-6	1998	-56	kW
Friction Power	272	259	-12	269	-2	kW
Heat Transfer	351	325	-26	299	-52	kW
Shaft power	2000	2000	-	2000	-	kW

Table 16: Energy Balance comparison

Table 15 shows the BSFC, energy in and energy out of all three models. It also calculates the difference in kW between the 1-ST TC and 2-ST TC, and between 1-ST TC and 2-ST TC Ideal.
Energy in: Fuel Power, the fuel consumption times the heating value of the fuel
Energy out: Exhaust power, friction power, heat transfer and shaft power.
This is exactly the same energy balance as in figure 45. The difference in BSFC can be calculated into kW and it should end up being the same value as the fuel power. The row called "Fuel Power calc" represents a conversion of the BSFC into fuel power using formula 53, below.

$$P_{fuel} = \frac{BSFC}{1000} P_{shaft} \frac{1h}{3600s} h_n \tag{53}$$

The reason behind including this calculation is to demonstrate how you can trace the savings in BSFC between the models. When employing a 2-stage turbocharger with the result being fuel savings it is nice to be able to track where the savings come from. The next step is to determine where the largest savings are.

	2-Stage Turbocharger		2-Stage Turbocharger Ideal	
Unit	kW	%	kW	%
Fuel Saving	45	100	110	100
Exhaust Power	6	13	56	51
Friction Power	13	29	2	2
Heat Transfer	26	58	52	47
Sum of savings	45	100	110	100

Table 17: Energy balance savings

The 2-stage turbocharger model:
The largest savings are heat transfer and friction. Savings from heat transfer comes from cooling of the whole process achieved by using Miller timing. Savings in friction comes from reduced cylinder pressure, because friction increases with high pressures. The savings in exhaust means that less of the available fuel energy disappears into the exhaust.
The 2-stage turbocharger ideal model:
Largest savings are in heat transfer and exhaust. Again the savings in heat transfer is linked to the Miller timing cooling down the process, in addition the maximum temperature in the cylinder was lower than in the 2-ST TC model. Exhaust savings is how much less energy disappears into the exhaust. This model had higher maximum pressures in the cylinder which explains why friction savings are so small.

This chapter on energy balance explains the differences in BSFC between the different models. The result of changing PMEP between the models is integrated into the fuel savings through formula 50. The PMEP value is relatively small compared to the friction and heat transfer contributions.

7 Conclusion

7.1 Discussion and Conclusion

The turbocharger matching procedure was useful as a tool for quick calculations of turbocharger performance, and the results can be used for turbocharger selection. But exhaust temperatures were a bit higher than expected because the model of the engine didn't account for the reduced process temperature you would get from high boost pressure and Miller timing.

The verification of the engine model was close enough to be accepted but some borderline deviations were noted for air receiver temperature and exhaust temperature. Calibration of the NO_x emissions was satisfactory, bringing it into the correct trend.

When running simulations of the 2-stage turbocharged model the optimal Miller timing was found to be Miller 70. This provided around 1 g/kWh reduction of NO_x and around 2 g/kWh reduction of BSFC at 100 % MCR. It was illustrative to run the 1-stage, 2-stage and 2-stage ideal models together and plot them against each other. This confirmed the suspicion that the 2-stage turbocharger model is an improvement but not as much as was hoped for. The ideal turbocharger acted as nice clue to what might be achieved. In the energy balance, the heat transfer was a major factor in reduced BSFC. The reduction of friction can be helped by reduced cylinder pressures.

Summary of the expected results of combining 2-stage turbocharging with Miller cycle

- Increased boost pressure
  - This was achieved with good margin
- Lower maximum cylinder temperature
  - This was not achieved, only the ideal model had reduced max temperature
- Lower maximum cylinder pressure
  - This was achieved, but probably due to reduced AF-ratio
- Lower brake specific fuel consumption, BSFC
  - This was achieved, down 1 g/kWh, a moderate reduction
- Lower NO_x emissions as a function of the lowered maximum temperature in the cylinder
  - This was achieved, down around 2 g/kWh

The possibilities of increased power density are definitively present because of the lowered cylinder pressures. However, the ideal model which performed better on BSFC and NO_x reduction did not have much of a reduction in cylinder pressure. This indicates the need for accurate control of the AF-ratio in order to manage all the competing effects which are at work when you tamper with the combustion process.

In the literature study emphasis was put on the advantages of lowered specific heats, this is not as easy to track successfully, but that effect is included in the reduced energy need of both compression and combustion that follows from lowered temperatures. The Miller loss has not been accurately determined, but what it really affects is the PMEP, because the Miller loss reduces the positive contribution from the gas exchange process and to some degree the closed cycle work. This should be clear in the case of increasing boost pressure, and in fact all three models show a PMEP reduction in case 1 with the highest boost pressure.

During simulations it became clear that even though an optimal point for reduction of both BSFC and NO_x was established, it would be easy to increase the NO_x reduction further but then at the expense of increased BSFC. This is interesting because it shows the underlying opportunities available, and what direction to take depends on the defined goals.

All this leads to the conclusion that the 2-stage turbocharger model just didn’t deliver high enough boost pressures. If higher boost pressures can be achieved then maybe even more extreme Miller timings can be used and the process temperature will be reduced. The reduction of process temperature is the key to both BSFC savings and NO_x emissions reduction. However, special attention must be given to the AF-ratio to ensure that it`s neither too low or too high. Too low and the maximum cylinder temperature rises. Too high and the cylinder filling goes up, increasing maximum pressures. Get the AF-ratio right in combination with high boost pressure and extreme Miller timings and the documented positive effects will most certainly increase. The whole clue is reduced process temperature, this gives lower heat transfer, lower cylinder pressures which reduces friction and opens possibilities for increased power density, the aforementioned reduces BSFC and last but not least a lower maximum cylinder temperature coupled with a balanced AF-ratio provides the opportunity of greatly reduced NO_x emissions.

All this leads to the conclusion that the provided turbochargers might not be the best match for this engine model. A weak link is the HP turbine which had a relatively low efficiency of around 70 %, that is down 10 % from the LP turbine and the two compressors running in the same conditions.

7.2 Recommendations for further work

- Re-matching the engine with new turbochargers
- Variable exhaust valve timings
    - If we look at how the trapping ratio goes down with increasing Miller timing then it is obvious that the loss could be reduced with reduced valve overlap. When you keep the same overlap and increase the boost pressure it is obvious that you will lose more mass of air in the scavenging process
- Experiment with adjustments of injection timing
- Looking at options for varying Miller timing seamlessly over the whole load range instead of just switching it on and off
- Look into pressure losses from sharp angles in 2-stage turbocharger ducting
- Investigate the necessary size, prize and availability of the intercoolers as modeled
- Make assessment on the intercoolers, does it exist IC that can handle the loads in this task without being too big or needing so much pumping power that the efficiency gains of the engine is lost?

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9 Appendix

9.1 Matlab code

Here are the Matlab scripts used in chapter 3, Turbocharger Matching. When the main script is run it loads the datascript first, then the user must choose the pressure out of the LP compressor, p_2, and the pressure out of HP compressor, p_4. The script then calculates all the equations and then prints the chosen parameters to screen.

Datascript, inputs.m

```
%this data file contains the necessary input to calculate the main script

%GENERAL INPUTS
Ru = 8314; %universal gas konstant, units [J/kmol*K]
Mair = 28.97; %Molar mass of air, units [kg/kmol]
c = 1E5; %constant representing [(N/m^2)/bar] or [pascal/bar]
AFs = 14.3; %stoichiometric AF-ratio for the fuel used
%COMPRESSORS
T1 = 298; %ambient temperature
Pamb = 1; %ambient pressure
Piloss = 0.001; %pressure loss of air intake
Etac1 = 0.77; %efficiency of compressor 1
Etac2 = 0.77; %efficiency of compressor 2
ka = 1.4; %specific heat ratio of air
%INTERCOOLERS
Plic1 = 0.043; %pressure loss intercooler 1
Plic2 = 0.043; %pressure loss intercooler 2
Tc1 = 310; %temperature cooling water, intercooler 1, unit [K]
Tc2 = 310; %temperature cooling water, intercooler 2, unit [K]
eps1 = 0.8; %efficiency of intercooler 1
eps2 = 0.8; %efficiency of intercooler 2
%ENGINE
N = 1000; %RPM
B = 0.25; %piston bore RR C-engine
S = 0.33; %piston stroke RR C-engine
Vsw = (pi/4)*(B^2)*S; %swept volume
etavol = 0.927; %volumetric efficiency of cylinder
ncyl = 6; %number of cylinders
LHV = 42700; %Lower Heating Value, [kJ/kg]
zetaL = 0.08; %fractional heat loss, [Wp/Hv]
etath = 0.437; %thermal efficiency of the engine
zetaE = 1-zetaL-etath; %fraction of total energy available in exhaust
Cpe = 1.16;%1.087; %approx spec heat of exhaust gas from Table A-20 at 750K
PlossE = 1.6; %pressure loss across engine (original value 0.05)
ER = (LHV*zetaE)/Cpe; %energy release to the exhaust gas
%TURBINES AND TURBOCHARGERS
ke = 1.34; %specific heat ratio for exhaust
Cpa = 1.008; %specific heat for air
EtaT2 = 0.77; %efficiency of turbine 2
Etamech2 = 0.97; %mechanical efficiency of turbocharger 2
Etamech1 = 0.97; %mechanical efficiency of turbocharger 1
EtaTC2 = Etac2*EtaT2*Etamech2; %HP turbocharger efficiency
```

Main script, match.m

```
clc
clear all
run inputs;
```

```
p2 = input('set p2: '); %choose pressure out of comp1
p4 = input('set p4: '); %choose pressure out of comp2

%POINT 1
Rair = Ru/Mair; %unit [J/kg*K]
p1 = Pamb - Piloss; %pressure before compressor 1
rho1 = (p1*c)/(Rair*T1); %air density at entry to turbine, unit [kg/m^3]
%POINT 2
T2 = T1*(1+(1/Etac1)*(((p2/p1)^((ka-1)/ka))-1)); %temp out of comp1
rho2 = (p2*c)/(Rair*T2); %density after comp1
%POINT 3
T3 = T2*(1-eps1) + eps1*Tc1; %temperature out of intercooler 1
p3 = p2 - Plic1; %pressure after intercooler 1
rho3 = (p3*c)/(Rair*T3); %density after intercooler 1
%POINT 4
T4 = T3*(1+(1/Etac2)*(((p4/p3)^((ka-1)/ka))-1)); %temperature out of comp2
rho4 = (p4*c)/(Rair*T4); %density after comp2
%POINT 5
T5 = T4*(1-eps2) + eps2*Tc2; %temperature out of intercooler 2
p5 = p4 - Plic2; %pressure after intercooler 2
rho5 = (p5*c)/(Rair*T5); %density after intercooler 2
%ENGINE
mtot = (N/2)*rho5*0.61*Vsw*ncyl*60; %total mass flow of air inducted,
[kg/h]
ma = (N/2)*rho5*0.61*Vsw*etavol*ncyl*60; %mass flow of air, unit [kg/h]
mf = 397; %ma/AF; %mass flow of fuel, [kg/h]
me = ma + mf; %mass flow of exhaust, [kg/h]
AF = ma/mf;
%POINT 6
T6 = (AF/(AF+1))*T5 + (ER/(1+AF)); %temperature after engine
p6 = p5 - PlossE; %temperature after engine
rho6 = (p6*c)/(Rair*T6); %density after engine
%POINT 7
C2S = ((p4/p3)^((ka-1)/ka)) - 1;
mr = (1 + (1/AF));
p7 = p6*(1-((C2S*Cpa*T3)/(mr*EtaTC2*Cpe*T6)))^(ke/(ke-1)); %press after T2
T2S = 1 - (p7/p6)^((ke-1)/ke);
T7 = T6*(1 - EtaT2*T2S); %temperature after T2
rho7 = (p7*c)/(Rair*T7); %density after turbine 2
%POINT 8
C1S = ((p2/p1)^((ka-1)/ka)) - 1;
mr = (1 + (1/AF));
%p8 = p7*(1-((C1S*Cpa*T1)/(mr*EtaTC1*Cpe*T7)))^(ke/(ke-1)); %press after T1
p8 = 1.03;
Etaturbine1 = C1S*Cpa*T1/((1-(p8/p7)^((ke-
1)/ke))*Cpe*T7*mr*Etac1*Etamech1);
T1S = 1 - (p8/p7)^((ke-1)/ke);
T8 = T7*(1 - Etaturbine1*T1S);%temperature after turbine 1
rho8 = (p8*c)/(Rair*T8); %density after turbine 1

%ENERGY BALANCE TURBOCHARGERS
Wc1 = -(T2-T1)*(ma/3600)*Cpa;
Wt1 = (T7-T8)*(me/3600)*Cpe;
Wc2 = -(T4-T3)*(ma/3600)*Cpa;
Wt2 = (T6-T7)*(me/3600)*Cpe;
EBTC1 = Wc1 + Etamech1*Wt1;
EBTC2 = Wc2 + Etamech2*Wt2;

%EFFICIENCIES OF TURBOCHARGERS
EtaLPTC = Etac1*Etaturbine1*Etamech1; %total efficiency of LP turbocharger
EtaHPTC = Etac2*EtaT2*Etamech2; %total efficiency of HP turbocharger
```

```
%EXPANSION RATIO OF TURBINES
ExpRatioHPT = p6/p7;
ExpRatioLPT = p7/p8;

%CORRECTED MASS FLOWS
macc1 = ma*(sqrt(T1)/(p1*c));
macc2 = ma*(sqrt(T3)/(p3*c));
mect2 = me*(sqrt(T6)/(p6*c));
mect1 = me*(sqrt(T7)/(p7*c));

%VOLUMETRIC FLOW RATE
Va = ma/(rho1*3600); %the mass flow of air at ambient conditions, [kg/h]

%PRESSURE RATIO SPLIT
PRLP = p2/p1; %pressure ratio LP compressor
PRHP = p4/p3; %pressure ratio HP compressor
Split = PRLP/PRHP; %ratio between LP & HP compressors

%Fuel-air equivalence ratio
lambda = (mtot/mf)/AFs; %the relative AF-ratio
F = 1/lambda; %the fuel-air equivalence ratio

fprintf('\n');
fprintf('-----------------------\n');
fprintf('POINT 1\n');
fprintf('T1 = %f\n',T1);
fprintf('p1 = %f\n',p1);
fprintf('rho1 = %f\n',rho1);
fprintf('-----------------------\n');
fprintf('POINT 2\n');
fprintf('T2 = %f\n',T2);
fprintf('p2 = %f\n',p2);
fprintf('rho2 = %f\n',rho2);
fprintf('-----------------------\n');
fprintf('POINT 3\n');
fprintf('T3 = %f\n',T3);
fprintf('p3 = %f\n',p3);
fprintf('rho3 = %f\n',rho3);
fprintf('-----------------------\n');
fprintf('POINT 4\n');
fprintf('T4 = %f\n',T4);
fprintf('p4 = %f\n',p4);
fprintf('rho4 = %f\n',rho4);
fprintf('-----------------------\n');
fprintf('POINT 5\n');
fprintf('T5 = %f\n',T5);
fprintf('p5 = %f\n',p5);
fprintf('rho5 = %f\n',rho5);
fprintf('-----------------------\n');
fprintf('POINT 6\n');
fprintf('T6 = %f\n',T6);
fprintf('p6 = %f\n',p6);
fprintf('rho6 = %f\n',rho6);
fprintf('-----------------------\n');
fprintf('POINT 7\n');
fprintf('T7 = %f\n',T7);
fprintf('p7 = %f\n',p7);
fprintf('rho7 = %f\n',rho7);
fprintf('-----------------------\n');
```

```
fprintf('POINT 8\n');
fprintf('T8 = %f\n',T8);
fprintf('p8 = %f\n',p8);
fprintf('rho8 = %f\n',rho8);
fprintf('-----------------------\n');
fprintf('Efficiency LP Turbine \n');
fprintf('Etaturbine1 = %f\n',Etaturbine1);
fprintf('-----------------------------------------\n');
fprintf('Corrected mass flow LP compressor \n');
fprintf('macc1 = %f\n',macc1);
fprintf('-----------------------------------------\n');
fprintf('Corrected mass flow HP compressor \n');
fprintf('macc2 = %f\n',macc2);
fprintf('-----------------------------------------\n');
fprintf('Corrected mass flow HP turbine \n');
fprintf('macc1 = %f\n',mect2);
fprintf('-----------------------------------------\n');
fprintf('Corrected mass flow LP turbine \n');
fprintf('macc1 = %f\n',mect1);
fprintf('-----------------------------------------\n');
fprintf('Volumetric Flow Rate \n');
fprintf('Volume Flow Air = %f\n',Va);
fprintf('-----------------------------------------\n');
fprintf('Pressure Ratio LP Compressor \n');
fprintf('PRLP = %f\n',PRLP);
fprintf('-----------------------------------------\n');
fprintf('Pressure Ratio HP Compressor \n');
fprintf('PRHP = %f\n',PRHP);
fprintf('-----------------------------------------\n');
fprintf('Pressure ratio split between compressors \n');
fprintf('Split = %f\n',Split);
fprintf('-----------------------------------------\n');
fprintf('Expansion Ratio LP turbine \n');
fprintf('ExpRatioLPT = %f\n',ExpRatioLPT);
fprintf('-----------------------------------------\n');
fprintf('Expansion Ratio HP turbine \n');
fprintf('ExpRatioHPT = %f\n',ExpRatioHPT);
fprintf('-----------------------------------------\n');
fprintf('Efficiency LP Turbocharger \n');
fprintf('EtaLPTC = %f\n',EtaLPTC);
fprintf('-----------------------------------------\n');
fprintf('Efficiency HP Turbocharger \n');
fprintf('EtaHPTC = %f\n',EtaHPTC);
fprintf('-----------------------------------------\n');
fprintf('Air-Fuel Ratio \n');
fprintf('AF-Ratio = %f\n',AF);
fprintf('-----------------------------------------\n');
fprintf('Mass Flows in kg/h: \n');
fprintf('Mass flow Air = %f\n',ma);
fprintf('Mass flow Fuel = %f\n',mf);
fprintf('Mass flow Exhaust = %f\n',me);
fprintf('-----------------------------------------\n');
fprintf('The Relative Air-Fuel Ratio \n');
fprintf('lambda = %f\n',lambda);
fprintf('-----------------------------------------\n');
fprintf('Fuel-Air equivalence ratio: \n');
fprintf('F = %f\n',F);
fprintf('-----------------------------------------\n');
```

9.2 Verification Data

This is the natural continuation of chapter X. Here I will present the comparison of results between the measured data from the engine factory and data from the engine simulation. I will compare the results of the points defined in Table 9, from chapter X.

BMEP 24.69 RPM 1000				
Values	Unit	Model	Engine	Difference %
Speed	RPM	1000	999	0,1
Brake Torque	Nm	19094	19098	0,0
Power	kW	2000	1998	0,1
BMEP	bar	24,69	24,69	0,0
Air Flow Rate	kg/h	13572	13509	0,5
Fuel Flow Rate	kg/h	397,0	399,8	-0,7
BSFC	g/kWh	198,0	200,1	-1,1
BSAC	g/kWh	6786	6761	0,4
AF-Ratio (Inducted air/Total fuel)	-	34,3	33,8	1,4
Ambient Pressure	bar	1,01	0,998	1,6
Ambient Temperature	C	25	29	-14,8
Temperature after compressor	C	244	257	-5,5
Temperature raise compressor	C	219	228	-4,4
Pressure raise compressor	-	5,14	5,25	-2,1
Temperature air receiver	C	56,5	55,2	2,3
Pressure air receiver	bar	5,19	5,29	-1,9
Cylinder maximum pressure	bar	197	198	-0,8
Exhaust pressure before turbine	bar	3,91	4,07	-4,1
Exhaust temperature before turbine	C	482	545	-13,1
Turbocharger Speed	RPM	40541	41311	-1,9
Temperature exhaust after turbine	C	295	326	-10,6
Thermal Efficiency	%	42,76	42,17	1,4
Pressure ratio across cylinder	-	1,33	1,3	2,1
NOx emissions	g/kWh	7,71	7,89	-2,3
NOx emissions	ppm	692,00	710	-2,6

BMEP 18.52 RPM 1000				
Values	Unit	Model	Engine	Difference %
Speed	RPM	1000	999	0,1
Brake Torque	Nm	14323	14325	0,0
Power	kW	1500	1499	0,1
BMEP	bar	18,518	18,52	0,0
Air Flow Rate	kg/h	10443	10731	-2,8
Fuel Flow Rate	kg/h	299	300,1	-0,4
BSFC	g/kWh	199,3	200,2	-0,5
BSAC	g/kWh	6962	7159	-2,8
AF-Ratio (Inducted air/Total fuel)	-	34,9	35,8	-2,4
Ambient Pressure	bar	1,01	1,00	1,6
Ambient Temperature	C	26,1	25,9	0,8
Temperature after compressor	C	245	208	14,9
Temperature raise compressor	C	219	182	16,6
Pressure raise compressor	-	3,95	4,13	-4,6
Temperature air receiver	C	51,2	55,1	-7,7
Pressure air receiver	bar	3,97	4,16	-4,8
Cylinder maximum pressure	bar	159	159	0,0
Exhaust pressure before turbine	bar	2,99	3,2	-7,0
Exhaust temperature before turbine	C	458	503	-9,8
Turbocharger Speed	RPM	35759	36522	-2,1
Temperature exhaust after turbine	C	308	325	-5,4
Thermal Efficiency	%	42,59	42,11	1,1
Pressure ratio across cylinder	-	1,33	1,3	2,1
NOx emissions	g/kWh	8,17	8,22	-0,6
NOx emissions	ppm	715	700	2,1

BMEP 12.34 RPM 1000				
Values	Unit	Model	Engine	Difference %
Speed	RPM	1000	999	0,1
Brake Torque	Nm	9548	9544	0,0
Power	kW	1000	998	0,2
BMEP	bar	12,44	12,34	0,8
Air Flow Rate	kg/h	7324	7452	-1,7
Fuel Flow Rate	kg/h	208,1	210,9	-1,3
BSFC	g/kWh	208,1	211,3	-1,5
BSAC	g/kWh	7325	7465	-1,9
AF-Ratio (Inducted air/Total fuel)	-	35,2	35,3	-0,4
Ambient Pressure	bar	1,01	1,00	1,5
Ambient Temperature	C	26	28	-8,2
Temperature after compressor	C	151	158	-4,5
Temperature raise compressor	C	126	131	-3,8
Pressure raise compressor	-	2,77	2,87	-3,8
Temperature air receiver	C	46,3	55,1	-19,0
Pressure air receiver	bar	2,79	2,89	-3,4
Cylinder maximum pressure	bar	122	116	4,9
Exhaust pressure before turbine	bar	2,17	2,22	-2,6
Exhaust temperature before turbine	C	442	495	-11,9
Turbocharger Speed	RPM	29970	30714	-2,5
Temperature exhaust after turbine	C	336	364	-8,3
Thermal Efficiency	%	40,77	39,90	2,1
Pressure ratio across cylinder	-	1,29	1,3	-0,8
NOx emissions	g/kWh	9,07	8,45	6,7
NOx emissions	ppm	754	690	8,5

BMEP 6.17 RPM 1000				
Values	Unit	Model	Engine	Difference %
Speed	RPM	1000	999	0,1
Brake Torque	Nm	4774	4770	0,1
Power	kW	500	499	0,2
BMEP	bar	6,17	6,17	0,0
Air Flow Rate	kg/h	4980	4970	0,2
Fuel Flow Rate	kg/h	122,6	123,2	-0,5
BSFC	g/kWh	245,3	246,9	-0,7
BSAC	g/kWh	9961	9960	0,0
AF-Ratio (Inducted air/Total fuel)	-	40,6	40,34	0,7
Ambient Pressure	bar	1,01	1,00	1,5
Ambient Temperature	C	25	29	-14,7
Temperature after compressor	C	95	95	-0,4
Temperature raise compressor	C	70	66,4	4,7
Pressure raise compressor	-	1,77	1,78	-0,4
Temperature air receiver	C	41	55	-33,2
Pressure air receiver	bar	1,80	1,80	0,0
Cylinder maximum pressure	bar	101	92	8,9
Exhaust pressure before turbine	bar	1,58	1,64	-3,3
Exhaust temperature before turbine	C	387	418	-7,9
Turbocharger Speed	RPM	22246	22444	-0,9
Temperature exhaust after turbine	C	327	348	-6,3
Thermal Efficiency	%	34,58	34,15	1,2
Pressure ratio across cylinder	-	1,14	1,10	3,2
NOx emissions	g/kWh	9,94	11,73	-18,1
NOx emissions	ppm	610	720	-18,0

BMEP 2.47 RPM 1000				
Values	Unit	Model	Engine	Difference %
Speed	RPM	1000	999	0,1
Brake Torque	Nm	1910	1913	-0,1
Power	kW	200	200	0,0
BMEP	bar	2,47	2,47	0,0
Air Flow Rate	kg/h	3829	3664	4,3
Fuel Flow Rate	kg/h	71,0	72,3	-1,8
BSFC	g/kWh	354,8	361,1	-1,8
BSAC	g/kWh	19149	18309	4,4
AF-Ratio (Inducted air/Total fuel)	-	54,0	50,7	6,0
Ambient Pressure	bar	1,02	1,00	1,9
Ambient Temperature	C	25	29	-17,1
Temperature after compressor	C	66	61	7,5
Temperature raise compressor	C	41	32	22,6
Pressure raise compressor	-	1,39	1,31	5,4
Temperature air receiver	C	41,1	54,7	-33,2
Pressure air receiver	bar	1,40	1,32	5,7
Cylinder maximum pressure	bar	73	69	5,0
Exhaust pressure before turbine	bar	1,34	1,26	6,0
Exhaust temperature before turbine	C	327	346	-5,8
Turbocharger Speed	RPM	17145	14619	14,7
Temperature exhaust after turbine	C	291	309	-6,1
Thermal Efficiency	%	23,90	23,35	2,3
Pressure ratio across cylinder	-	1,05	1,05	-0,3
NOx emissions	g/kWh	11,67	12,52	-7,2
NOx emissions	ppm	375	420	-12,0