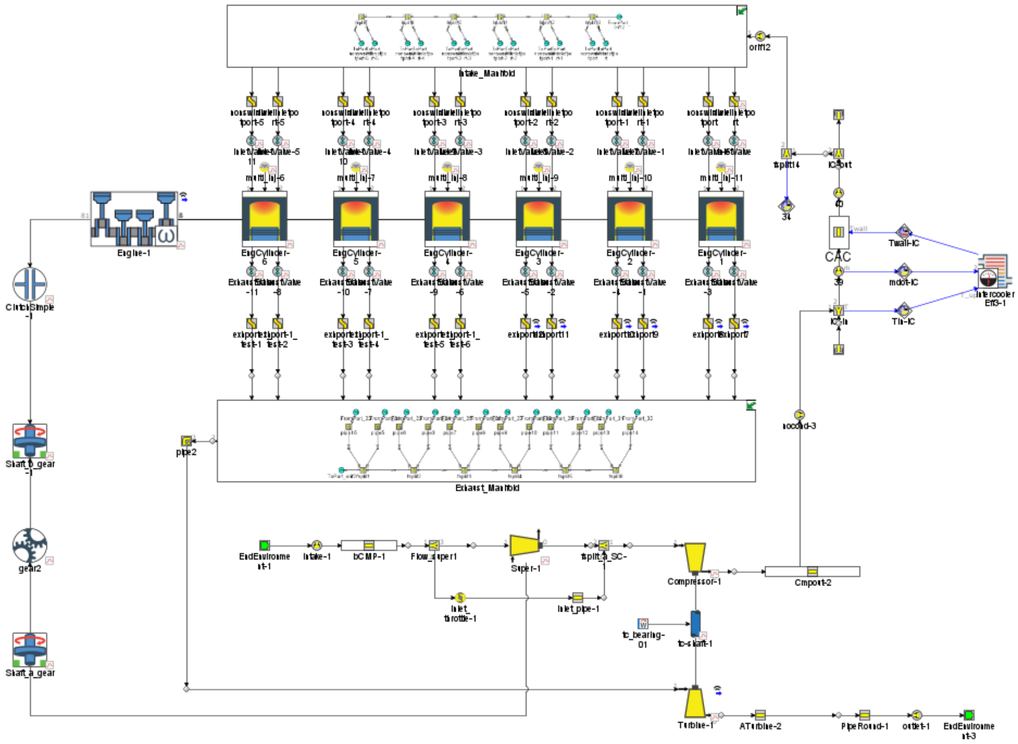

Design and Optimization of Exhaust Manifold For Volvo Penta D6

Master's Thesis in Mobility Engineering

Andreas Winbo
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Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
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Master's thesis 2025

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GT-Power Simulation of Exhaust Manifold

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Cover: Picture of Volvo Penta D6s GT-Power Model.

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Abstract

The Volvo Penta D6 is a high-performance diesel engine specifically designed for marine applications. Its efficiency relies heavily on the turbocharger, making it crucial to retain as much energy as possible from the exhaust gases. However, the current water-cooled exhaust manifold reduces the available thermal energy by cooling down the exhaust gases, which affects turbocharger efficiency. The water-cooling of the exhaust manifold is necessary due to regulations of 220 °C maximum temp for engine surfaces. The limit is important because engines often operate in tight compartments, sometimes with multiple units nearby, where there is a risk of diesel fuel leaking directly onto hot engine parts. The water-cooling of the exhaust manifold is also problematic for implementation of future after-treatment systems such as SCR, as the current exhaust gas temperatures are to low.

This thesis presents a 1-D simulation study in GT-Power to explore how reducing heat loss in the exhaust manifold could enhance engine performance. An engine model of the D6 engine was analyzed and then used to simulate different scenarios such as changes in manifold geometry, levels of heat loss and turbo chargers. A few important conclusions are listed below:

- Only changing the geometry of the exhaust manifold while keeping the cooling unchanged is **beneficial**.

- Reducing the cooling of the original exhaust will not improve the performance, but rather make the engine perform **worse**. However, combining reduced cooling with a more capable turbocharger is **beneficial**.

The simulations showed that changing from a single exhaust pipe to a two-pipe system, without making any changes to the water cooling, improved performance and fuel efficiency. It resulted in a 4.5% increase in peak power and a 2.5% reduction in Brake Specific Fuel Consumption (BSFC).

Further simulations indicated that reducing the heat lost in exhaust manifold with 78% from the original watercooled exhaust to a insulated dry exhaust system led to a worse performance, with peak power decreasing by 1.7% and a less desired torque curve with a large dip around 2400 rpm. This occurred because the increased exhaust pressure and mass flow pushed the turbocharger out of its optimal efficiency zones. Based on those discoveries, a new larger turbocharger was simulated with the same insulated system, showing increase in performance of +4.3% and BSFC reduction of 2.7%, just by supplying more air efficiently.

Further simulations explored how waste-gating could successfully increase exhaust gas temperature, and how changes to injection timing and amount of fuel complimented the benefits of increased exhaust enthalpy. A final configuration of the engine that combined a two bank system with exhaust flow from three cylinders in each bank to get a pulse divided exhaust manifold with less cooling together with a

GT45 turbo, advanced timing and a increase in fuel it became possible to successfully meet Volvo Penta's target of 550 hp without increasing the NOx emissions.

This thesis demonstrates the possibility to unlock performance potential by reducing the heat being currently lost in cooling. It mentions the technical modifications required to harness that energy while maintaining emission levels and safety requirements.

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

1-D	One dimension (used for simulation in one dimension)
ASC	Ammonia Slip Catalyst
BMEP	Brake Mean Effective Pressure
BSFC	Brake-specific Fuel Consumption
CAC	Charge Air Cooler
CAD	Computer Aided Design
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
EATS	Exhaust Aftertreatment System
HP	Horsepower
NOx	Nitrogen Oxides
POC	Proof Of Concept
RPM	Revolutions per minute
SCR	Selective Catalytic Reduction
T_B_T	Temperature Before Turbo
TDC	Top Dead Center
WLTP	Worldwide Harmonised Light vehicles Test Procedure

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1 Introduction

The Volvo Penta D6 diesel engine series uses a water-cooled exhaust manifold to keep lower the surface temperatures and reduce the risk of a fire. However, this cooling consumes energy which could otherwise be used to boost the turbocharger. In addition the current exhaust manifold fails to properly utilize the engines pressure pulses effectively. Instead of having the pulses working together, they instead interfere with each other as they leave the cylinder. They are clashing inside the exhaust manifold and reducing the energy they can deliver to the turbine. A more optimized design could harness these pulses by given the gases more time to travel between the engine and block before the next pulse arrives. This will improve both performance and efficiency.

The goal of this project is to investigate the existing exhaust manifold through 1-D modeling and create a manifold design that is better in performance and efficiency. This involves making better use of pressure pulses and improving flow, while ensuring the manifold remains safe and can fit in the allocated space without making the engine package larger.

1.1 Motivation

Volvo Penta's D6 engine series is designed for marine applications and is used in both recreational and commercial vessels. The engine is a 5.5 liter inline six-cylinder diesel engine featuring common-rail Direct Injection (DI), turbocharging, and max output of 480 hp. Originally introduced in 2003, it is available in several propulsion configurations like inboard, Aquamatic sterndrive, and the Volvo Penta IPS. The D6s design prioritizes reliability, fuel efficiency, and performance.

Figure 1.1: Volvo Penta D6 [1]

Model	Power Output (Hp)	Rated Speed (Rpm)
D6-300	300	3,300
D6-340	340	3,400
D6-380	380	3,500
D6-400	400	3,500
D6-440	440	3,700
D6-480	480	3,700

Table 1.1: Volvo Penta D6 Engine Variants

While the D6 is a versatile engine used in various marine applications, it is most commonly used in pleasure crafts that often have compact engine compartments with restricted airflow. These typically require high performance engines which operate only for few hours each year. Volvo Penta is interested in increasing the engine's power output in a way that delivers a noticeable increase in performance for customers upgrading from the current 480 hp version. As a result, the company is exploring what would be required to raise the power significantly within the mentioned constraints.

Based on internal assessments, Volvo Penta has identified that a considerable part of energy is lost through exhaust manifold cooling. Reducing this loss presents an opportunity to enhance performance and efficiency.

Currently, the D6 meets existing emission standards without the use of after-treatment systems. However, growing customer interest in cleaner technologies and possibility of stricter future regulations motivate Volvo Penta to exploring use of exhaust after-treatment systems, such as Selective Catalytic Reduction (SCR) in future engine designs.

Initial evaluations have indicated that exhaust gas temperatures in the D6 could be too low for optimal after-treatment performance. The exhaust manifold has been identified as a important factor contributing to these lower temperatures. Modifying this component could be a solution for successful implementation of emissions reducing systems such as an SCR mentioned in the previous section.

This initiative aligns with Volvo Group's sustainability roadmap. The company has pledged to achieve net zero greenhouse gas emissions by 2040 and is investing in technologies that can help them reduce their environmental impact, like cleaner combustion, electrification and renewable fuels.[4] Specifically Volvo Penta aims for an absolute 37.5 % reduction in use phase emissions from its products by 2034. These goals push Volvo Penta to make its engines more efficient and add more after treatment systems.

This Master's Thesis aims to investigate how reduced cooling in the exhaust manifold could improve overall engine performance. This study will also explore the needed engine modifications to be able to utilize the energy currently lost in cooling the exhaust manifold. The 1-D simulation tool GT-Power is selected as the primary tool to mode and analyze the engine performance and behavior to the applied changes.

1.2 Problem definition

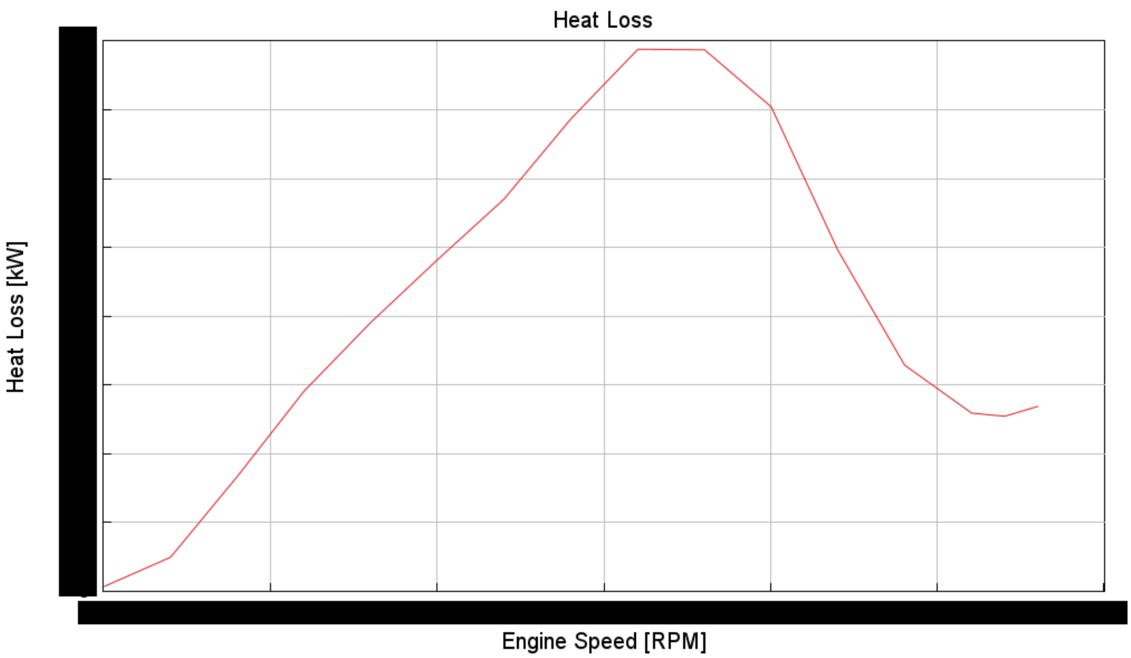


Figure 1.2: Heat transfer of the Exhaust Manifold

A big part of this project was to look into how reduced cooling would or could affect engine performance. In other engine applications outside the marine sector a exhaust manifold normally isn't cooled. The main reason behind this engine application being cooled is its use case, where the engine often is mounted/located in a engine compartment with minimum air flow, space and could be multiple units next to each other. As can be seen in the graph above the heat loss can be up to 47 kW at around 2750 rpm.

Because the engine rooms are a confined small area with increased temperatures and often with multiple units the risk of a failure and fire is a big threat. A fire out at sea could be devastating, so this is something that must be minimized. Since diesel self-ignites at around 225 °C, regulations requires that that fuel isn't allowed to reach any hot surface above 220 °C and that areas exceeding 220 °C shall be well insulated. This is the main reason for watercooling the exhaust manifold. [5][6].

If, however, the cooling of the exhaust were minimized, the enthalpy (energy content) of the exhaust gases would increase. This gives an increased potential for the turbo to use this energy and deliver more air to the engine. With more air to engine it becomes possible to increase performance and efficiency. Penta has expressed interest in if, this could reduce emissions.

1.3 Problem approach

Volvo Penta has built a 1-D GT-Power model of the D6-480, allowing quick changes to parameters like injection timing or fuel rate and to components such as the exhaust manifold or turbocharger. This means that changes to the system can be quickly evaluated and their impact to the system can be understood. The rough methodology is to simulate different parameters and component changes, and understand their impact on the system. These changes can then be combined to achieve performance goals, or goals of increasing exhaust gas temperature for after-treatment systems.

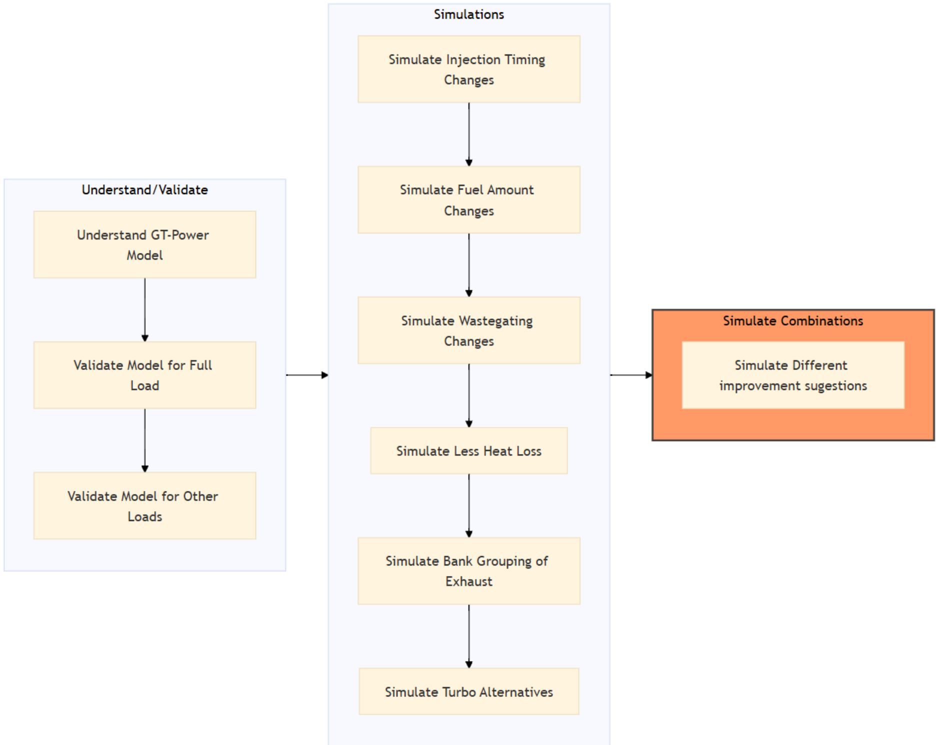


Figure 1.3: Flowchart of Simulation strategy

1.4 Background study

This section reviews earlier work aimed at insulating exhaust manifolds or otherwise reducing heat loss in downstream components. We found no studies that deal specifically with the watercooled exhausts of marine diesel engines, but several investigations exist for automotive and heavyduty engines. Those engines have dry exhaust systems cooled by ambient air or fans, and are not housed in the tight spaces typical of marine installations. Their exhaust manifolds run hotter and lose less heat, yet manufacturers still try to investigate insulation to boost performance, cut fuel consumption, and reduce emissions. It therefore seems likely that limiting heat loss in a watercooled marine diesel exhaust would deliver even greater gains.

1. Luján et al. Modeled a turbocharged passenger diesel engine in a GT-Power to study passive thermal insulation of the exhaust ports and turbocharger. By reducing heat losses, the simulation showed higher exhaust temperatures entering the turbo and aftertreatment, improving turbo efficiency and raising aftertreatment inlet temperature. The insulated simulations gave slight gains in fuel economy alongside positive impact on warm up processes for the aftertreatment system, indicating that preserving exhaust energy can boost overall engine efficiency without additional fuel[7]
2. Galindo et al. Investigated ways to retain exhaust thermal energy and reduce interference from pressure pulses during load transient and steady operations

of a turbocharged DI diesel engine. A manifold with an airgap to insulate and a 4-2-1 pulse manifold were modeled using 1-D and then built and tested on a rig at 1500 rpm. The insulated manifold reduced time to reach 90% max torque by around 15% and increased max torque by around 3%. The pulse manifold actually performed worse than the original manifold due to increased heat losses from greater surface area and volume, despite reducing pressure pulse interference. A final improved pulse divided and air gap insulated manifold, were simulated in 1-D, predicting 14% faster response time, 6.6% higher maximum torque, and increased exhaust gas temperature by around 50°C.[8]

3. Fricke et al. (2016): Investigated an air gap insulated sheet metal exhaust manifold and turbine housing and compared it to a cast iron version, for a 2 L turbo Diesel Car engine. A 1-D model predicted that the insulated design halved exhaust heat loss over a drive cycle. Tests confirmed temperatures after the turbo to be 32 °C higher on average during the drive cycle, which reduced tail-pipe HC, CO and NOx by 20 – 50 % in cold WLTP and FTP-75 phases. Due to higher exhaust temperatures benefiting the after treatment system and making it more efficient, engine management strategies could allow engine out NOx to rise, decreasing fuel consumption by around 2 percent, while still keeping the same tailpipe NOx levels. Weight of the exhaust manifold was also reduced by 50 % and surface temperatures on the exhaust dropped by around 300 °C, showing that passive insulation can boost efficiency, emissions control and save weight at once.[9]
4. Volvo Group insulates the exhaust manifolds on some of its truck engines. These castiron manifolds are wrapped in a sheet-metal–fiberglass–sheet-metal sandwich. The insulation retains exhaust heat, improving efficiency while keeping the manifolds surface substantially cooler than it would be without the heat shields.

1.5 Research Questions

This project explores how changes to the exhaust-manifold design affect a marine diesel engine's performance. Specifically, it seeks to answer:

- **RQ1**: What performance gains can be achieved, according to one-dimensional (1-D) modelling, by operating a turbocharged six-cylinder marine diesel engine with a less-cooled exhaust manifold?

- **RQ2**: What geometric, material, and insulation choices should be considered when designing an exhaust manifold that requires less cooling?

1.6 Deliverables

- A 1-D model in GT-Power of current system, and improved system.
- Design guidelines for improving the exhaust manifold based on the 1-D modeling results.

- Engine management strategies for taking advantage of improved exhaust manifold.
- 3D(CAD) models that incorporate one or more of these guidelines while considering placement and packaging.
- Planning report, Midterm presentation, Preliminary reports, Final report, Final presentation.

1.7 Limitations

- No FEM anlysis of the models
- Only D6 engine
- Simulation will be done in GT-power
- Use provided model from Volvo Penta

2

Engine and Exhaust Manifold

2.1 Engine introduction

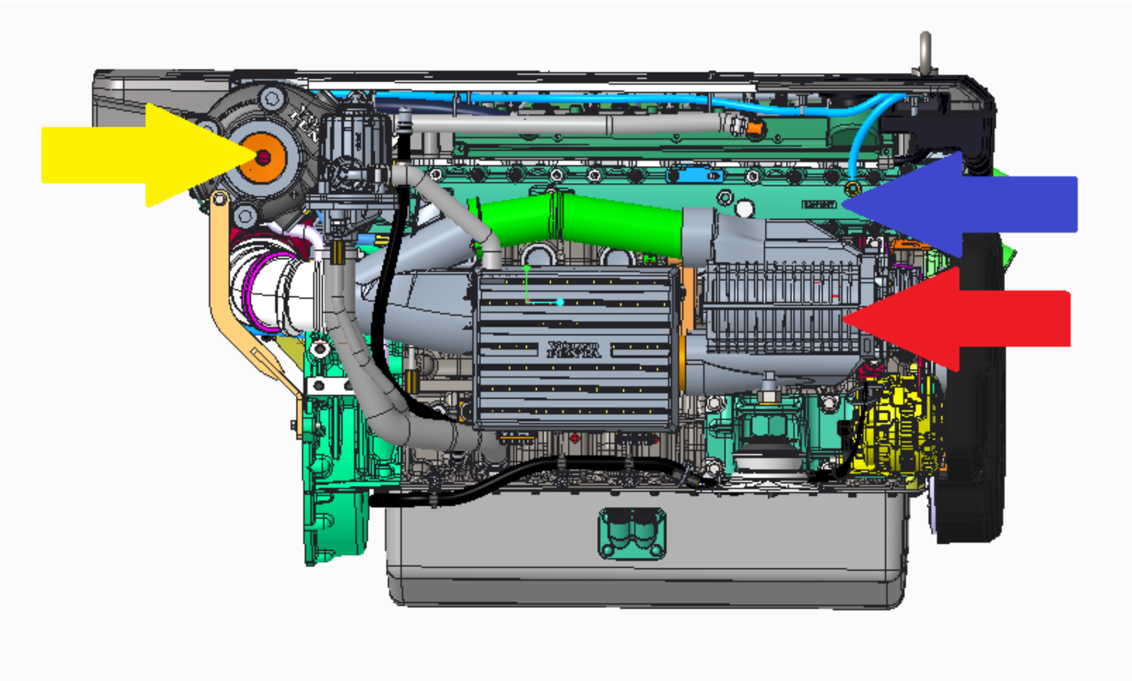


Figure 2.1: CAD model of D6 Engine

- Yellow arrow: Turbo
- Blue arrow: Exhaust Manifold
- Red arrow: Supercharger

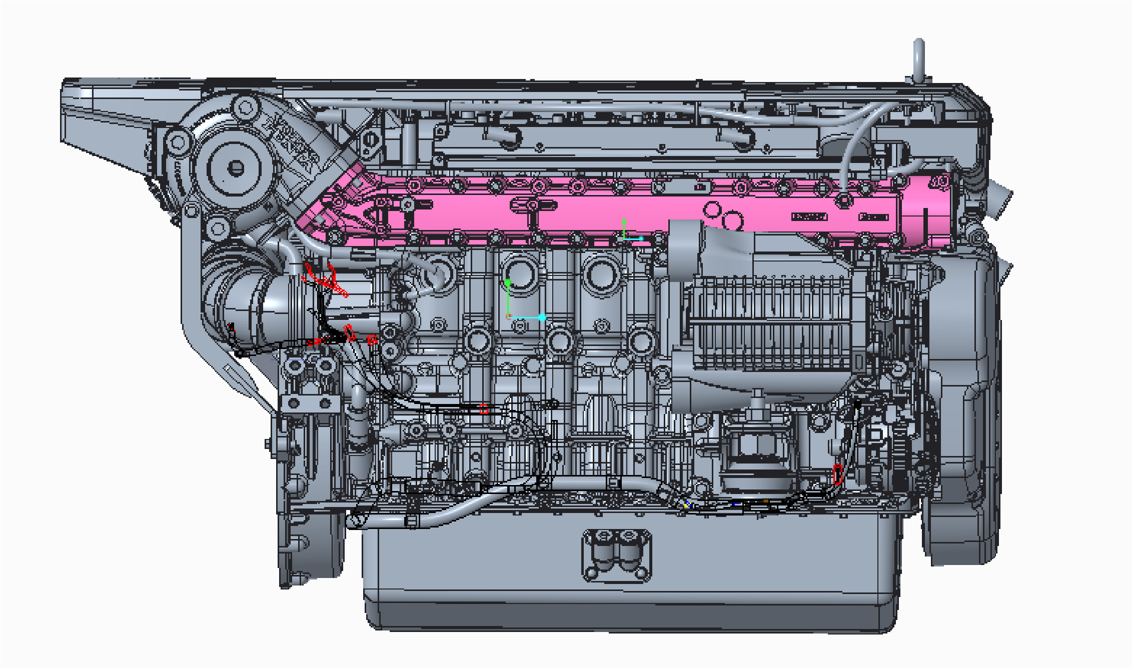


Figure 2.2: Exhaust Manifold in CAD

2.2 Technical Specifications

Engine table for Volvo Pentas D6 480 hp.

Parameter	**Specification**
General description	4-stroke, direct-injected, turbocharged & aftercooled diesel engine
Engine Rating	5 (pleasure craft running <300 h yr^{-1} [10])
Number of cylinders	6
Valves per cylinder	4
Common rail fuel injection	2000 bar
Displacement	5.5 L
Firing order	1-5-3-6-2-4
Bore	103 mm
Stroke	110 mm
Compression ratio	18.0 : 1
Max crankshaft power	353 kW
Max crankshaft torque	1102 N m
Rated speed (R5)	3700 rpm
Governed speed (R5)	3830 rpm
Dry weight	645 kg

Table 2.1: Key specifications of Volvo Penta D6-480.

2.3 GT representation

The GT-power model is of the Volvo Penta D6 engine with the highest power output. The key components of the model are discussed in the subsections below:

2.3.1 Exhaust Manifold

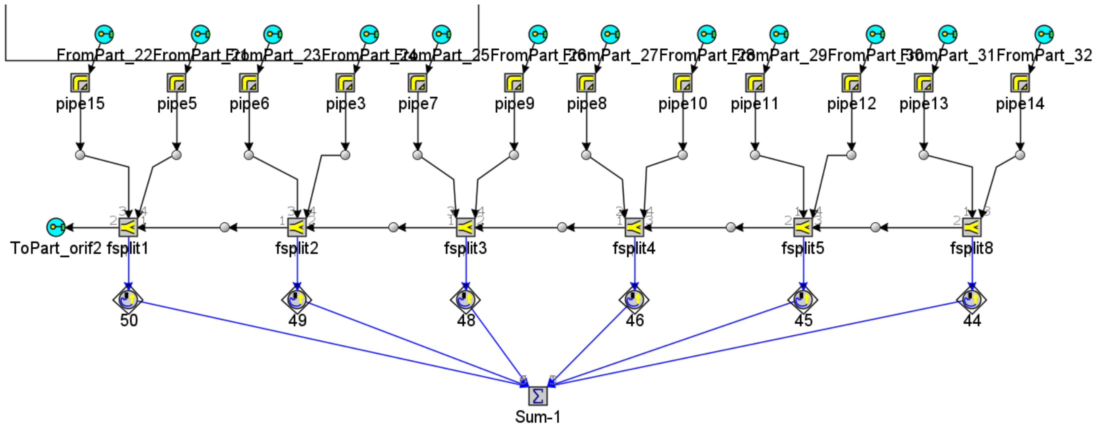


Figure 2.3: Exhaust Manifold in GT-power

The exhaust manifold in GT-Power divided into multiple pipes that acts as the pipe sections and flow splits that acts as the connections. Hot exhaust gases from the exhaust ports of the engine flow in the manifold until they can reach the turbo.

2.3.2 Supercharger

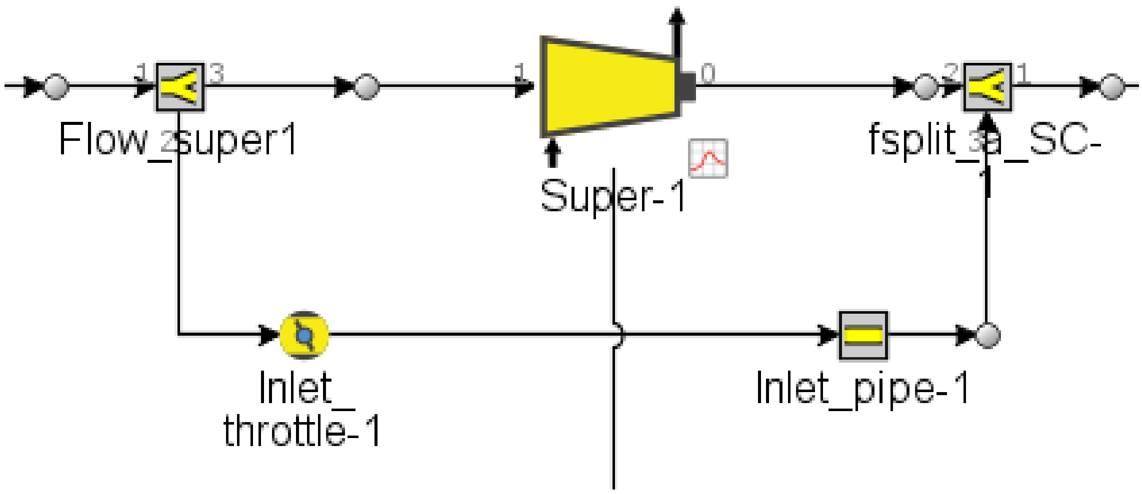


Figure 2.4: Exhaust Manifold in GT-power

The supercharger is driven by a belt connected to the engine and uses a magnetic clutch to engage or disengage. It draws in outside air, compresses it and sends it to the turbocharger compressor rpm, which then compress the air even further and supplies it into the engine. Above a certain rpm, the supercharger is bypassed. In the figure above, this is shown in the two separate airflow paths. One path includes the supercharger when it is active, and the other bypasses it when it is not. The reason that the supercharger only works up until a certain rpm is because the main purpose is to assist the turbo. In this case the turbo has problems in lower rpm to deliver enough air to the engine, so the supercharger assists untill the turbo can provide the needed air by itself.

2.3.3 Turbocharger

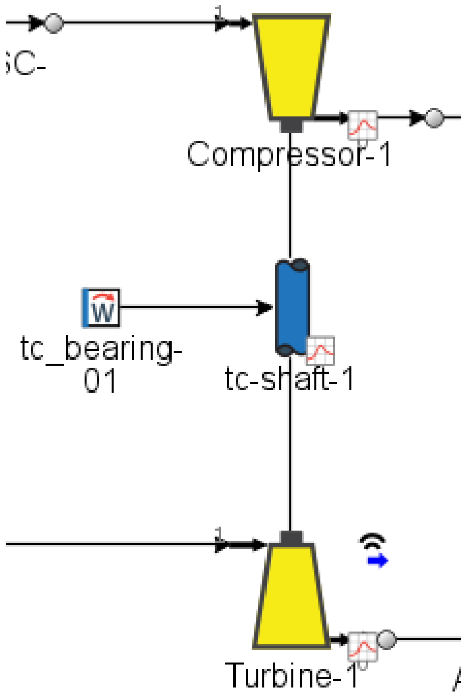


Figure 2.5: Exhaust Manifold in GT-power

The Turbo is modeled as 4 objects. Compressor, turbine, bearing and shaft. Turbine and compressor object have their respective maps which bascially is a performance chart with speed, mass flow, pressure ratios and efficiencies at different stages. Bearing is modeled as a loss increasing with rpm and the shaft has a moment of inertia with a inertia multiplier that makes it harder to spin at lower rpm. All these together model how a turbo works with turbo lag and efficiencies at their respective speeds and mass flows.

The turbine side of the turbo is supplied with hot exhaust gases from the exhaust manifold and is connected to the compressor side that sucks air from supercharger below 2600 rpm and from the environment above that. It then feeds the CAC with air.

2.3.4 CAC

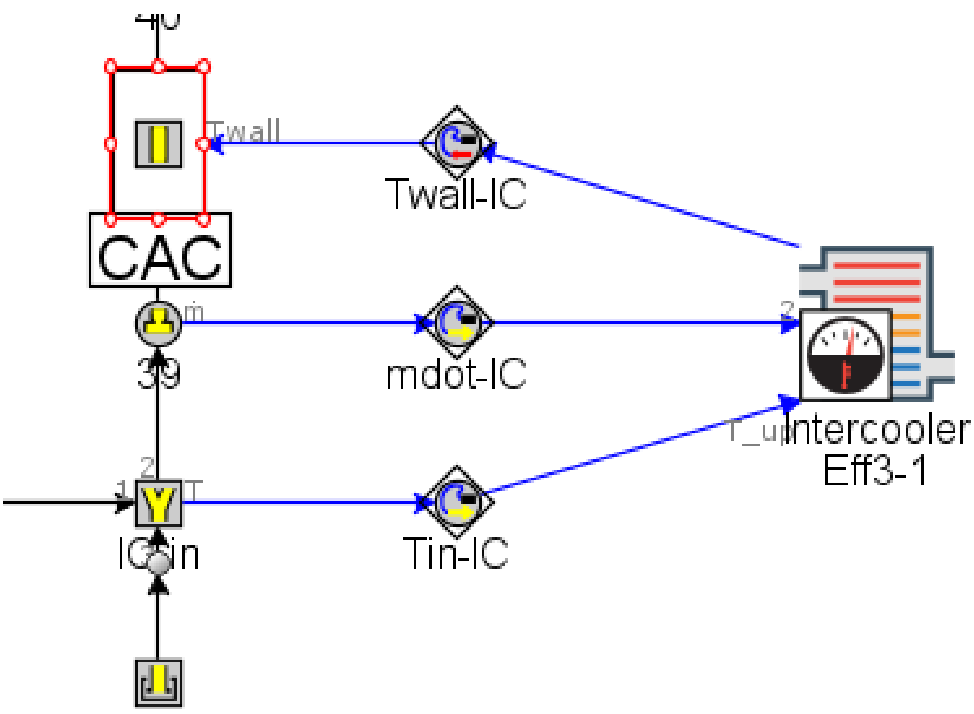


Figure 2.6: Exhaust Manifold in GT-power

Takes the high pressure air delivered by the compressor side of the turbo and cools it with a closed loop cooling system that is externally cooled by seawater, showed in figure 2.9. The purpose of this system is as previously mentioned cool down the air which increase the density.

In GT-power it is modeled as a CAC with an intercooler. The intercooler has a coolant temperature and an efficiency and the CAC is a pipe that applies the temperature from the intercooler on its walls to cool down the fluid.

2.4 The current Exhaust Manifold

The current manifold is of "log" or "bathtub" design, where all the exhaust ports from the engine cylinders flow into a single, larger pipe, resembling a log or a bathtub. This design has no utilization of the engines exhaust pulses but has a compact design. To be able to meet the required surface temperature of 220 °C, the manifold is cooled with coolant from a closed loop system. The coolant is then cooled in a heat-exchanger with seawater(See figure 2.9). Coolant flowing into the manifold is connected in series with the engine and, controlled with a water pump that varies in speed with the engine rpm. When the engine is cold, the flow of coolant is halted by a thermostat that opens at around 85 °C.

The engine has two separate exhaust ports and one coolant channel per cylinder. When the coolant enters the manifold, its pressed down and runs on the underside of the manifold. The coolant "turns around" in the turbo, which is also watercooled.

The manifold is made in cast aluminum of AlSi10Mg. Its manufactured through casting, using sand cores.

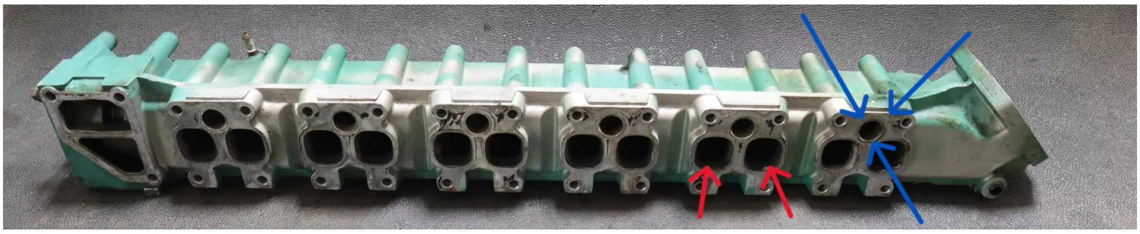


Figure 2.7: Exhaust manifold for the D6(Blue=coolant channel. Red=exhaust gas channels)

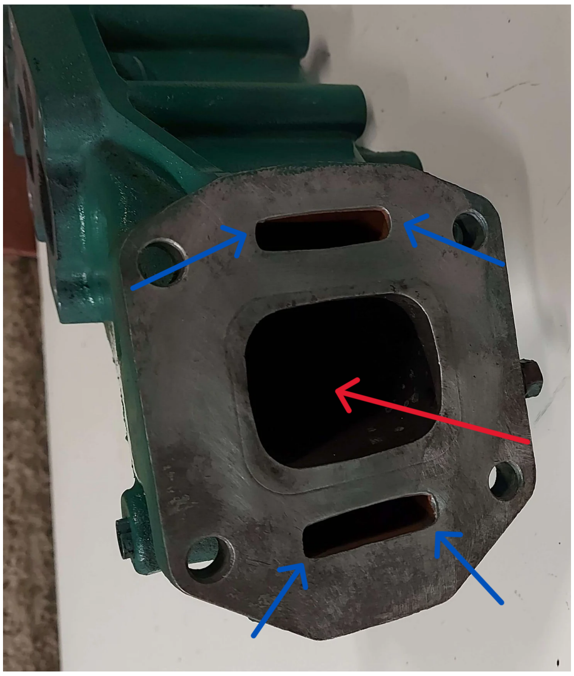


Figure 2.8: End of manifold where turbo attaches. (Blue=coolant channels. Red=exhaust gas channel)

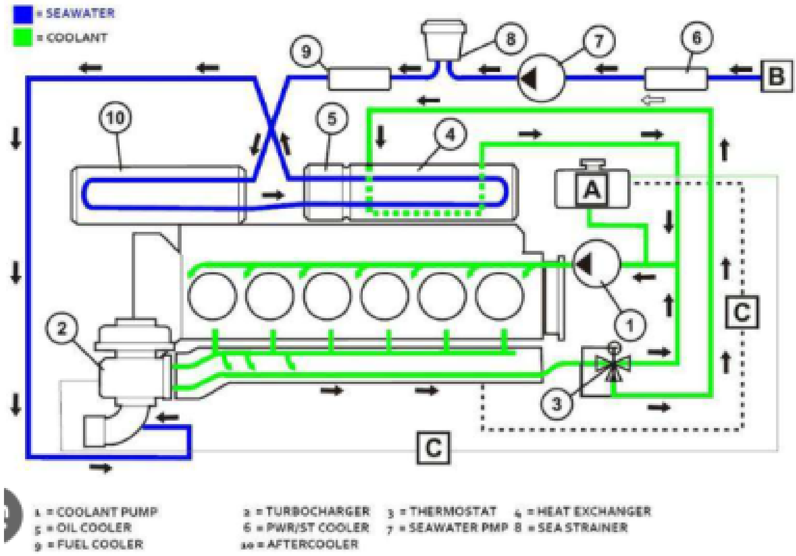


Figure 2.9: Coolant flow in a D6-350

Exhaust temperatures and pressures ranges between engine configurations and load. The D6 exhaust temperatures after the turbo ranges very roughly from 300 °C to around 500 °C at full load. Roughly temperatures before the turbo are 100 °C-150 °C higher than those after the turbo.

2.4.1 Cooling system

The cooling system consists of two circuits. The closed coolant circuit uses a 40–50% glycol/water mixture that flows through the engine block, cylinder head, exhaust manifold, turbocharger housing and a heat exchanger. This circuit is sealed and pressurized to prevent boiling and reduce the risk of corrosion inside the engine. Seawater is corrosive and could contain sea life or other debris, therefore should be avoided in the engine itself.

The second circuit runs cold seawater through the heat exchanger to cool the other circuit and then is fed into the exhaust pipe after the turbo and mixed with exhaust gases before being sent out.

Each circuit has its own mechanically driven pump. Pump speed increases with engine rpm. The system is designed to meet cooling demands at maximum engine load for each rpm, which means it often provides more cooling than needed at lower engine loads.

2.4.2 EATS

The current D6 engine does not have any aftertreatment system. However, due to the increased regulations on emissions, Volvo Penta thinks that its likely that regulations will change in the future. Some costumers are also interested EATS (Exhaust

Aftertreatment System) systems. Volvo Penta therefore want to keep EATS in mind when designing a new manifold, as EATS systems have high demands on temperature, in order to preform optimally.

Due to marine diesel engines being required to be able to run on high sulfur diesel, SCR (Selective Catalytic Reduction) is the only EATS system Volvo Penta wants to consider. The SCR will be further discussed in the theory part, but in summary it can operate at 210 °C-512 °C. However there is a risk of crystallization below 350 °C if the engine is run for a long time under that. The 512 °C is so specific as its an absolute limit, as temperatures exceeding that risks breaking down the Vanadinium in the SCR, which can result in catalyst degradation and system damage. In other marine diesel which have SCR systems, the engine output will be restricted if there is a risk of exceeding 512 °C.

Some of the current lower output versions of the D6 have very low exhaust gas temperature at low/mid load/rpm. This means that the SCR can't dose any urea and therefore not reduce NOx. Its therefore extra interesting to investigate the gain of reducing exhaust gas cooling at those points. Its should however also be considered that some of the higher output versions of the engine at high load might get too high exhaust gas temperatures for SCR, if the cooling is reduced, which is also worth investigating.

2.5 Competitors

The competion has a wide range of different engine sizes and configurations in the around the same horsepower range as Volvo Pentas D6. When comparing Volvo Pentas engine with other manufacturers, we try to take a similar engine in horse power and size. This is done so it could be used in similar cases to the Volvo D6 engine. As well as reaching similair emission strandards are reached which also would allow them to be used in similair cases.

Finding detailed descriptions of the competitors engines and coolant system is not easy since most of them don't want to give everything away. The information we were able to find also differed in each source but the similarites we manged to find will be covered below. The engine we will compare them with is Volvo Pentas D6 engine which is a six cylinder 5.5 L engine with between 300 and 480 hp.[11].

	Power(hp)	engine size(L)	Exhaust manifold cooling	Turbo Cooling
Volvo Penta	300-480	5.5	water cooled	water cooled
Cummins	230-550	6.7	water cooled	water cooled
Yanmar	394-434	5.8	water cooled	water cooled
Mercury	480-550	6.7	water cooled	water cooled
Scania	220-400	9.3	water cooled	dry

Table 2.2: Comparison of the competition sourced from: ([12])([13])([14])([15])([16])([17])([18])([19])

From the Table above we can clearly see that the competition mostly has water cooled exhaust manifold and turbo with only Scania having a dry turbo. We can also see that Volvo has the smallest engine but still being very competitive with their power figures.

3

Theory

This chapter provides the theoretical background needed to better understand the project. It covers topics and theory related to energy, heat transfer, EATS and introduces multiple Insulation techniques that could be used to reduce temperatures.

3.1 1-D Simulations and 3D CFD

3D CFD and 1-D(one dimensional) simulations uses computer models to predict and analyze fluid motion, heat transfer, and related transport phenomenons through simulation. It does this by spiting geometries into a smaller mesh of finite volumes, and running numerical methods that solve the flow equations in each volume. 3D CFD follows variations in the x, y and z directions. It resolves vortices, separation, wall effects etc in all three dimensions. This often require a very fine mesh of elements which needs lots of computational power and can take a long time to run.

1D simulations like those made in this thesis assumes each cross section is perfectly mixed, so variables only change along the main flow axis(x-axis) or 1 dimension. The mesh have fewer elements than 3D CFD. A pipe can for example have one element every 20mm. This reduced amount of elements allow for faster simulations and full engine cycles can be simulated in minutes. The tradeoff is that phenomenons like secondary flows and swirl etc, can not be calculated and are instead approximated with loss factors.

In practice, 1-D models are suitable for system-level simulations, such as engines, where many different parts work together, and there is a need for fast calculations. 3-D models are more often used where local details are important and a high degree of accuracy is required, often in specific parts like a combustion chamber or a propeller blade.

To achieve the goals of this thesis, simulations were done in the software GT-power. GT-power is a 1D software developed by Gamma techonoligies. Volvo Penta have been using this software for early development and system level changes for a long time. Most of Pentas engines have GT-power models. Like previously mentioned, CFD relies on the application of various numerical methods to calculate the dynamics of a fluid. In GT-power the Naiver Stokes equations for continuity, momentum and energy are main equations that are applied. These equations are applied in 1 dimension by averaging them around the direction of the flow.[20]

3.1.1 GT-power objects in Exhaust manifolds

A exhaust manifold is constructed through 3 different GT-Power objects. Pipes, flow splits and orifices. Pipes are used when there is only one entry and exit, while a flow-split is needed when a finite volume has multiple entry or exit port[20].

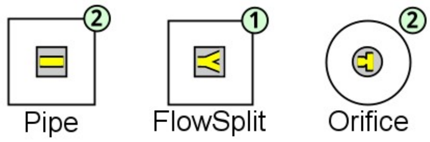


Figure 3.1: The three building-blocks of an exhaust manifold i GT-Power

Pipes are the preferred object type for accuracy as GT-SUITE turns every pipe into a series of short control volumes according to set discretizations length while every flowsplit (T-, Y-piece etc) are treated as a single volume.

For a pipe the pressure, temperature and density is calculated for every slice along its length, while the mass-flow rate and velocity are calculated on the faces between slices. Because the parameters are recalculated in every slice, a pipe can capture pressure waves, temperature gradients and friction along the pipe.

A FlowSplit, on the other hand, is always one lumped control-volume no matter how large the physical junction is. It stores just a single pressure, temperature and density for the whole volume. Momentum is like a pipe also handled separately at the faces, at each port using the port's orientation, expansion diameter and characteristic length.

A orifice is a connection, not a part. An OrificeConn is a plane that joins two flow volumes. It has zero volume but is the place where the solver applies the one dimensional momentum equation. GT-POWER inserts a default OrificeConn every time you draw a line between flow components, ensuring that the solver can always apply this equation and include a boundary loss element. The orifice automatically takes the smaller of the two port areas and assumes a simple sharp shape. If the connecting geometries are different or there is a flow restriction, the orifices can be changed to account for that.

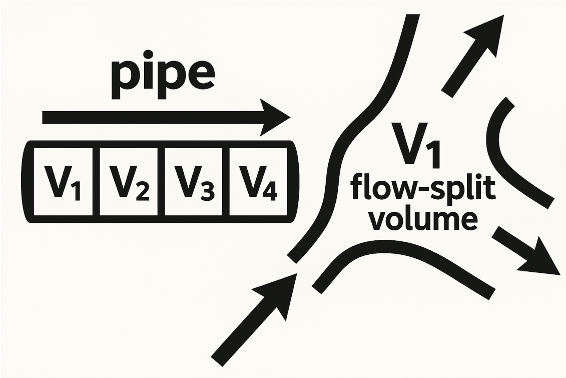


Figure 3.2: Pipe vs Flowsplit

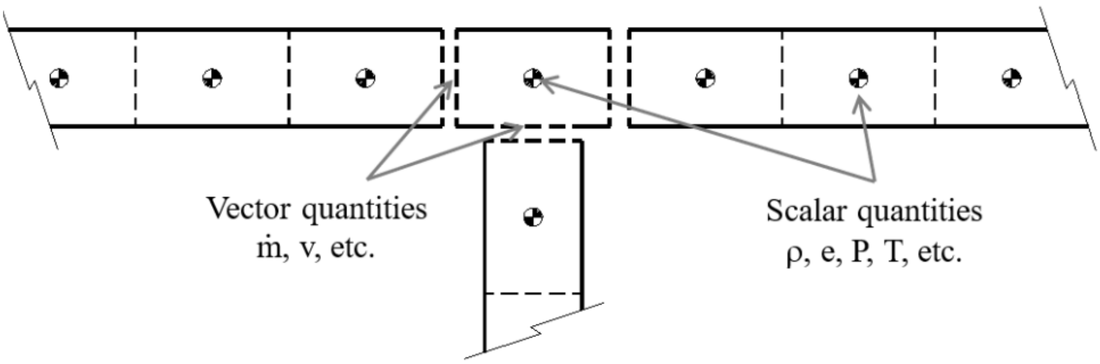


Figure 3.3: Scalars stored at center, Vectors stored at faces.

- **Scalars** — pressure, temperature, density, energy — are stored at the center of each control-volume.
- **Vectors** — massflow,speed etc — are calulated at the faces that connect two volumes.

3.2 Combustion modeling in the D6 1D-model

In 1-D engine modelling, what happens inside the cylinder is very important to the simulation results. A combustion model determines how quickly the fuel-air mix transitions from unburned to burned, and GT-POWER has three different types.

- **Non-predictive**: fixed burn profile (e.g. Wiebe curve). Does not change with speed, boost or injection timing.
- **Semi-predictive**: Same base profile, but shape factors stretch with inputs such as speed, λ, or pressure.
- **Predictive**: burnrate curve generated from spray and flame physics each cycle. Changes in injection timing or air density etc directly affect pressure, torque,emissions etc.

This project uses the EngCylCombDIPulse model. The model is predictive and therefore adapts combustion when changes are made to the model. However, it doesn't use measured cylinder pressure traces, as those are not available at in the test protocols that the model is based on. This means that calibration multipliers for entrainments, igintion delay and combustion rates are calibrated and hardcoded into the model. The model is physically consistent, but accuracy can drop at operating points far from those calibration cases.[20] As this model is calibrated for maxload, lowload cases have less accuracy due to this. Supplying a matrix of pressure traces would increase accuracy.

3.3 NOx modeling in the D6 1D-model

The GT-POWER model works out NO_x emissions with the three extended Zeldovich reactions shown below[21]:

$$\begin{aligned} &\text{R1} \quad N_2 + O \longrightarrow NO + N \\ &\text{R2} \quad N + O_2 \longrightarrow NO + O \\ &\text{R3} \quad N + OH \longrightarrow NO + H \end{aligned}$$

All of the above chemical reactionsc produce some form of NOx. Depending on what the fuel-air mixture contains during the burn, NOx are formed by a combination of these three equations. If the environment is very O2 rich, alot of NOx is formed by equation R2, and if the enviroment is rich in OH alot is formed by equation R3.

The composition is very complicated and depends on a variety of factors like mix ratios, fuel/air ratios etc, what important to know is that GT-POWER predicts and the amounts of O,N_2, N O_2 and OH at every crank-angle step, so it can feed them into these three reactions and predict the in-cylinder NO_x level.

NOx formation is highly temperature-dependent. Heywood has done test, raising the flame temperature by adding oxygen to the intake or lowering it by diluting with nitrogen and showed that NOx emissions climb almost exponentially with increasing flame temperature [21]. However, a well known challenge in diesel engines is that lowering combustion temperatures to reduce NOx typically leads to higher soot emissions. This tradeoff, often called the NOx/soot dilemma or diesel dilemma, and must be carefully managed.

It should be noted that when simulating engines the NOx emissions are very sensitive, as the growth is exponential to temperature, and availability of O_2 to combust largely affects this. Therefore its especially sensitive to trapped cylinder air mass, fuel-to-air ratio, and combustion rate[22].

3.4 Energy

One of the goals of this project is to raise the exhaust gas temperature. By insulating or cooling the exhaust less, more heat is retained, so the exhaust arriving at the turbocharger has higher temperature. This higher T will result in a higher enthalpy (energy content) according to the formula for enthalpy.

$$h = c_p \cdot T$$

- h: specific enthalpy (J/kg)
- c_p: specific heat capacity at constant pressure (J/(kg K))
- T: temperature (K)

This higher enthalpy/energy could then under the right circumstances be harnessed by the turbo, increasing energy efficiency and performance of the engine[23].

3.5 Heat transfer

Heat transfer is a subject within thermal engineering and refers to how physical objects use, transfer, generate and converts heat. When studying exhaust manifolds, we aim to keep the inner pipe temperature high while ensuring the outer temperature stays below 220°C. This makes heat transfer between engine parts an important focus.

The heat generated by an internal combustion engine can be elegantly described using the first law of thermodynamics, also know as the principle of conservation of energy.

$$\Delta U = Q - W$$

where:

- Q is the heat that is generated by the combustion of the injected fuel inside the engine.
- W is the work done by the piston as the ignited fuel/air mixture expand and push it downwards.
- ΔU is the change in internal energy of the system.

This means that the heat generated by combustion, minus the work done, results in a change in internal energy, part of which will be transferred as heat to the engine components. The first law of thermodynamics does not state how much of this internal energy that is transferred or in what direction or to what extent. This is instead determined by the second law of thermodynamics[24].

3.5.1 How Heat Moves

Heat can be transferred in three different ways: Conduction, Convection and Radiation. This section will contain a brief introduction of each of them, to better understand how heat is transferred. Its important to understand that the transfer of heat requires a temperature difference between substances and that heat flows from the substance with higher temperature to that of the lower.

3.5.1.1 Conduction

Conduction is the transfer of energy between particles through contact, where the more energetic part of a substance transfers to the less energetic, through interaction. Think of it as your hand touching a cold can of soda, where your hand quickly becomes cold as the heat leaves your hand through conduction with the wall of the aluminum can. Conduction of heat can occur in all states of a material, solid, fluid and gaseous. In solids, conduction occurs due to a combination of vibrations of the molecules and energy transport by free electrons. In fluid and gaseous form it occurs due to the collision of molecules during their random motion[25].

3.5.1.2 Convection

Convection involves a combined effect of conduction and fluid motion, where a solid surface transfers energy to an nearby fluid that is in motion. The faster the motion, the larger is the convection heat transfer. If there is no motion in a fluid, the transfer of heat is purely by conduction. The motion of the fluid complicates the calculation of transfer rates as it enhances the heat transfer. The heat from the coffee transfers to the nearby air through conduction. Meanwhile, convection occurs as cooler air moves in towards the cup as warmer air moves away, helping the coffee cool faster[25].

3.5.1.3 Radiation

Radiation is the energy emitted from a object through electromagnetic waves (or photons). Radiation, unlike conduction and convection does not require a present of an intervening medium. This means that it doesn't get affected by vacuum, and its how energy of the sun is transported to earth. All objects over absolute zero emit thermal radiation to varying degree.

3.6 EATS

EATS or Exhaust after treatment systems purpose is to take care of emissions in the exhaust system. This is done by using a catalyst to accelerate the reaction between the elements so the emmisions are confined inside the exhaust instead of releasing them to the atmosphere. EATS consist of different parts that take care of stages during the cleaning of the exhaust gases.
Harmful gas and particles in the exhaust:

- Particulate matter (PM)
- Carbon monoxide (CO)
- Hydrocarbons (HC)
- Nitrogen oxides (NOx)
- Ammonia (NH_3)
- Carbon dioxide (CO2)

Particulate matter are soot particles usually in the size of 0.04 μm to 1 μm. Created by unburnt fuel and is harmful to human health.

Carbon monoxide is usually created during a rich combustion where there is too much fuel but can also be created during lean combustion where you have excessive amounts of air. Because diesel engines operate with a lot of air flow the CO emission is lower than a gasoline engine. CO is very dangerous to human health and in high doses fatal, it binds to the hemoglobin in the blood but is a lot heavier and therefore lowers the oxygen level in the blood. Permanent damage to cognitive functions can be caused by small concentrations of just above 9 ppm[26].

Hydrocarbons are made out of different chemical structures and therefore the impact they have on the environment and human health depends which structure it has. They are mostly formed when the combustion isn't hot enough which usually occurs at lower rpm and loads. They are in many cases not very harmful to humans but they are important factors in forming NO_2 and O_3 (ozone) which have negative impact on humans and the environment[26].

Nitrogen oxides which is considers to be the biggest problem with diesel engines. The higher combustion temperatures and excess of oxygen is when the diesel engine becomes as efficient as possible and less of the other emission are created but the more NOx will form. To reduce NOx being produced it require lower engine temperatures and less oxygen which would as mention earlier increase other emission and lower all efficiency. NOx react with the atmopshere forming a nitrogen acid with fine particles. That can cause various lung disease since they can penetrate into the lungs tissue. NOx when reaction with the atmosphere break up into NO and O (oxygen) that can later form O_3 [26].

Ammonina is inject into the exhaust in SCR to handle the NOx but exessive amounts are either handled by the ASC if the system has one or it is sent out the exhaust as additonal emissions. In normal amounts found in the atmopshere is not dangerous but high exposure can hurt your skin, eyes, throat and lungs and cause burns and coughing[27].

Volvo pentas EATS for off road engines consist of:

- DOC: The Diesel Oxidation Catalyst funtion is to by oxidation remove/change PM, CO and HC to CO_2 and also turning HC to H_2O
- DPF: Diesel Particulate Filters task is to captured the PM in the filter.
- SCR: Selective Catalyst Reduction is removing NOx by reduction to N_2
- ASC: Ammonia Slip Catalyst handles the excess NH_3 also knows as urea from the SCR.

The final product out of the exhaust is that most of these harmful emissions are removed and mostly CO_2 is left.

4
Methodology

This chapter outlines the methodology used in improving and evaluating the exhaust manifold design for the Volvo Penta's D6 Engine. A design process combining GT-Power 1-D simluations, CAD modeling, and thermal calculations was used. The goal was to explore how different changes in gemotory, cooling, and turbocharging affected the engines behavior and performance. A high focus was put on reducing heat loss and increasing performance while still maintaining safety requirements, packaging constrains and emission requirements. The methods were chosen to allow quick iteration and analysis that gave a insight into valuable design aspects and directions that later could be looked into and worked on.

4.0.1 Design methods

Product design can be approached in various ways, ranging from an intuitive "just do it" mindset to following detailed frameworks and guidelines. However, when faced with a large and complex problem, especially within a masters thesis, it is essential to follow a structured design methodology. This ensures a systematic approach to problem solving and that it meets the academic standards.

Mital Anil [2] introduces a five step methodology called Basic design process, and claim it is generalizable to almost all types of design. The process, illustrated in figure [4.1], includes the following steps:

- Understand the need: Identify the problem, understand needs and wishes, conduct research and observations.
- Explore Concepts: Explore concepts that solves the problem. Keep as many on the table as possible
- Define the Design: Add details to selected concept, develop models and prototypes
- Test the Design: Assess the solution through testing.
- Refine the Design: Improvise the design using the results and iterating it until the requirements are satisfied.

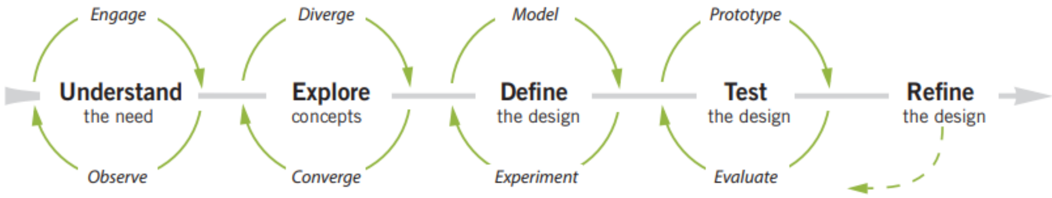


Figure 4.1: Basic design process from Mital,Anil(2008)([2])

4.0.2 Design research

The design research conducted during the project aimed to understand the current design of the exhaust system and identify areas for improvement. Additionally, it was important to analyze how competitors were approaching their designs and to explore emerging technologies in the field.

4.0.2.1 Current solution

The initial problem was to explore how to improve the exhaust manifold and turbocharger setup in Volvo Penta's D6 marine diesel engines. The exhaust manifold is cast aluminum and watercooled, which Volvo Penta thought could be improved. According to the Volvo team who developed the thesis, the existing manifold design is inefficient, costly, and difficult to manufacture. This problem definition provided the initial direction for the project. However, during the first week, it became clear that there was a need to support the claim of inefficiency in the current manifold with data and information, which the Volvo team initially lacked.

Therefore a plan for data collection, which contained two parts. Interviews and data collection from Volvo internal document library phoenix. Interviews were held with:

- Design team
- CFD team
- Turbo specialist
- 1D simulation team
- Product owner
- Thermal management
- Installation team
- After-treatment specialist
- Engine optimization
- Exhaust manifold specialist

The meetings were done in person at Volvo Penta and Trucks in Lundby. The structure for the interviews were to first introduce the project, then let the interviewee talk about their thoughts and views about the current exhaust manifold, especially to the areas of improvement. Further on, we asked questions that had come up during our process. Many interviewees recommended other people, which proved very useful.

Data collection from the internal document library was conducted by searching for information related to the exhaust manifold of the D6 engine. The documents obtained varied in subject and detail, but they greatly enhanced understanding.

4.0.2.2 Competitor benchmark

A competitor benchmark was done to better understand how competitors solve similar challenges in the marine market. While benchmarking competitors, it was essential to focus on specific attributes to ensure a fair and relevant comparison. The benchmarking was limited to competitors with engines who had:

- Diesel engine
- Marine application
- Over 4 Liters of engine displacement
- Under 7 liter of engine displacement
- Currently available for purchase
- Similar power output range

4.0.2.3 Emerging technology research

The emerging technology research was done by literature studies. This was done in an fairly unstructured way initially, to get further refined at the end, when relevant subjects and search words were more apparent. Searches were done through sites like google, google scholar and various school and organizations databases like Chalmers ODR. Some of the important search words were

- Diesel engine, marine exhaust, water cooled exhaust, etc etc
- Ceramic coating, etc etc

4.1 How to simulate less heat losses

The main goal of this study is to explore how changes in insulation and heat loss affect engine performance and the overall behavior of the system, rather than focusing on highly detailed heat transfer calculations.

We recognize that accurately modeling heat transfer involves complex interactions, detailed boundary conditions, and exact material properties, that go beyond what we're aiming for in this initial study. So, we're intentionally using simplified assumptions and approximate thermal properties to demonstrate general system effects rather than precise values.

Heat loss changes in the exhaust manifold were simulated in GT Power using two different approaches. The first approach uses the object "HeatMultiplier" to reduce the rate at which heat transfers through the pipes and flow splits, effectively slowing it down. How changes in the heat multiplier affects the overall heat loss can be seen in figure 4.3.

The second approach involves modeling an insulated pipe with separate layers and defined boundary conditions. This method is more challenging, as it is difficult to know how the air flows around the pipes and what cooling effect the surrounding air has on the outermost layer. GT Power also does not calculate the temperature of this outer layer, making it hard to determine if the 220 °C limit is exceeded.

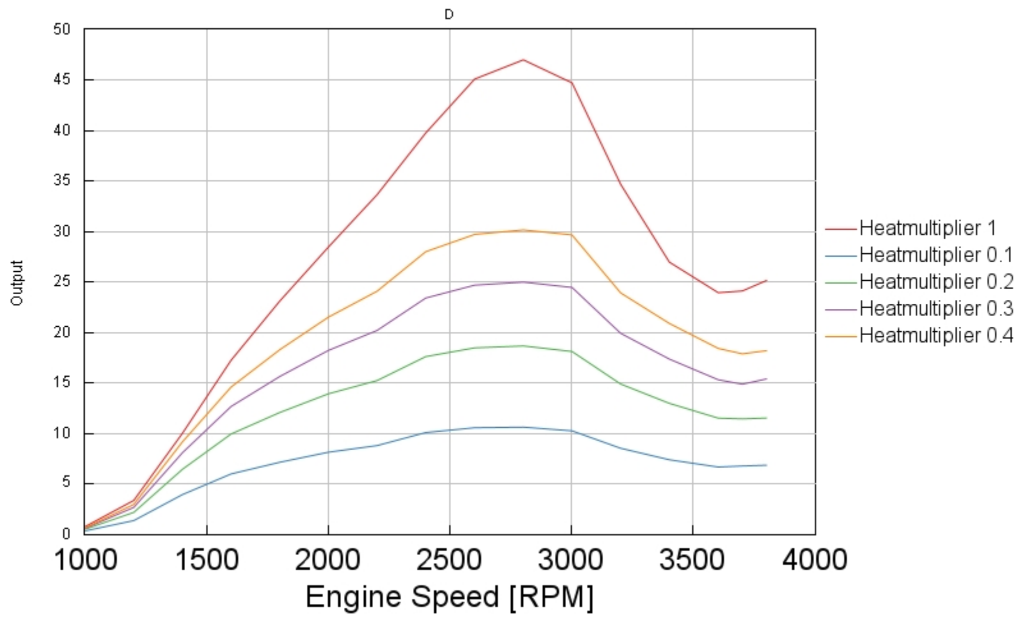


Figure 4.2: Total heat loss due to cooling in exhaust with varying heat mulipliers

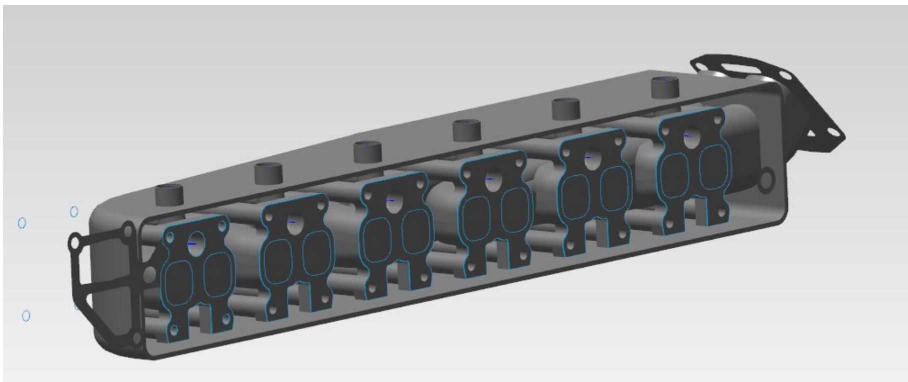


Figure 4.3: A aluminum heat cover for the dry 2 bank system was modeled in CAD

We first tried adjusting the HeatMultiplier. An important question to answer was: **What total heat loss will the new insulated system have?** This is difficult to determine, as many factors are unknown.

To get an initial idea, we performed a calculation to estimate how much energy an aluminum cover would lose to convection and radiation in a room with still air at 25°C. This example assumes that the exhaust is insulated using some form of insulation material and an outer aluminum cover, such that the aluminum surface is maintained precisely at 220°C.

$$\boxed{\begin{aligned} Q_{\text{rad}} &= \varepsilon\,\sigma\,A\Big(T_s^4 - T_a^4\Big) \;\approx\; 1.28\ \text{kW} \\ Q_{\text{conv}} &= h\,A\Big(T_s - T_a\Big) \;\approx\; 1.07\ \text{kW} \end{aligned}}$$

Equations from [28].

$$\begin{aligned}
\varepsilon &= 0.8 && \text{(surface emissivity, same as original exhaust)}\\
\sigma &= 5.67\times 10^{-8}\ \mathrm{W\,m^{-2}K^{-4}} && \text{(Stefan–Boltzmann constant[29])}\\
A &= 0.548\ \mathrm{m}^2 && \text{(external surface area of a theoretical heatshield)}\\
T_s &= 220°\mathrm{C} && \text{(surface temperature)}\\
T_a &= 25°\mathrm{C} && \text{(ambient temperature)}\\
h &= 10\ \mathrm{W\,m^{-2}K^{-1}} && \text{(convection coefficient for low-speed air over plate[30])}
\end{aligned}$$

Convection and radiation loss was combined with an estimate for heat loss due to contact(chapter 4.1.1), and came up with that 10 kW of total heat loss seemed reasonable

A heat multiplier of 0.1 was applied, resulting in a total heat loss of 10 kW for the original exhaust. The heat multiplier of 0.1 was then used for most simulations, which aimed to observe the engine's behavior under reduced heat loss conditions.

The other option than estimating the total heat loss through hand calculations is trying to detailed modeling of an insulated pipe in GT-Power. This was tested in later simulations of the complete system with the new turbo and two bank setup. This approach presents challenges such as selecting a suitable insulation material and defining its thermal properties and appropriate boundary condition. Similar to the previous method, it also requires an estimation of the correct convection coefficient.

The main issue with this approach is that GT-power doesn't calculate heat loss to the engine block through contact. So that either has to be disregarded, or the contact loss has to be baked into the convection rate.

$$\boxed{k_f \;=\; \frac{L_f}{\dfrac{T_{\text{hot}} - T_s}{q} - \dfrac{t_{\text{Al}}}{k_{\text{Al}}} - \dfrac{t_{\text{CI}}}{k_{\text{CI}}}} \;\approx\; 3.4\times 10^{-2}\ \mathrm{W\,m^{-1}K^{-1}}}$$

$$q = \frac{Q_{\text{conv}}\ (\text{prev}) + Q_{\text{rad}}\ (\text{prev})}{A} = \frac{1.28\ \text{kW} + 1.07\ \text{kW}}{0.548\ \text{m}^2} \approx 4.29\times 10^3\ \mathrm{W\,m^{-2}}$$

$$
\begin{aligned}
L_f &= 25 \text{ mm} && \text{(insulation thickness)}\\
T_{\text{hot}} &= 600°\text{C} && \text{(inner hot boundary)}\\
T_s &= 220°\text{C} && \text{(outer aluminium surface)}\\
t_{\text{Al}} &= 5 \text{ mm}, \quad t_{\text{CI}} = 5 \text{ mm}\\
k_{\text{Al}} &= 130 \text{ W m}^{-1}\text{K}^{-1}, \quad k_{\text{CI}} \approx 43.3 \text{ W m}^{-1}\text{K}^{-1}
\end{aligned}
$$

The space for insulations inside the heat cover was around 25mm at the point were the exhaust manifold itself was closest to the aluminum cover. The simplified calculation concludes that if a iron-insulation-aluminum sandwich of 5mm-25mm-5mm, that had 600 °C exhaust gas,and a 2.3 kW heat loss due to radiation and convection from a 220 °C aluminum cover . Have approximately a insulation material with the properties of 0.034 W/m*K.

4.1.1 Heat loss due to contact

Estimating the heat transfer through the contact between the exhaust manifold and engine block is difficult because there isn't enough detailed information. For hand calculations, we need to simplify the situation and make assumptions.
There is also a gasket between the exhaust manifold and the cylinder head, which will slow heat transfer. The exhaust manifold could theoretically reach temperatures of around 600°C, while the engine block is usually about 100°C. However, the actual temperatures at the surfaces where the materials touch will be somewhere in between, due to heat resistance at the contact points.

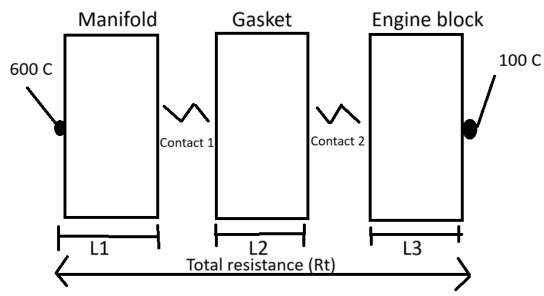


Figure 4.4: How conduction of should ideally be modeled

Total restiance then becomes:

$$
R_t = \underbrace{\frac{L_1}{kA}}_{\text{Manifold}} + \underbrace{\frac{1}{h_{c1}A}}_{\text{contact}} + \underbrace{\frac{L_2}{kA}}_{\text{Gasket}} + \underbrace{\frac{1}{h_{c2}A}}_{\text{contact}} + \underbrace{\frac{L_3}{kA}}_{\text{Engine block}} .
$$

Where:

- L_1, L_2, L_3 are the thicknesses of the manifold, gasket, and engine block sections respectively,
- k_1, k_2, k_3 are their respective thermal conductivities,

- A is the effective contact area,
- h_{c1}, h_{c2} are the thermal contact conductance values at the two interfaces.

Then heat flow can be calulated as:

$$q = \frac{\Delta T}{R_t} \tag{4.1}$$

The thermal resistance equation illustrates that the total resistance to: conduction through the exhaust manifold, contact resistance at faces,conduction through the gasket and conduction through the engine block. A higher resistance results in lower heat transfer.

We know the contact area $A = 375.89$ cm^2, but the thicknesses L of the exhaust manifold and engine block are unknown due to their complex geometries. Since thermal resistance increases with greater thickness, a larger L would reduce the overall heat transfer.

To estimate a worstcase scenario for maximum heat transfer that is possible with the contact area of the manifold, we can simplify the system by assuming:

- Thin layers (i.e., $L \simeq or \to 0$),
- No gasket is present,
- There is direct steel-to-steel contact.

$$\begin{aligned}
A &= 375.89 \text{ cm}^2 = 0.037589 \text{ m}^2 \\
h_c &= 2000 \text{ to } 3700 \text{ W/m}^2 \cdot \text{K} \\
R_t &= \frac{1}{h_c \cdot A} \Rightarrow \begin{cases} R_{t,\max} = \frac{1}{2000 \cdot 0.037589} \approx 0.0133 \text{ K/W} \\ R_{t,\min} = \frac{1}{3700 \cdot 0.037589} \approx 0.0072 \text{ K/W} \end{cases} \\
\Delta T = 500 \text{ K}, \quad q &= \frac{\Delta T}{R_t} \Rightarrow \begin{cases} q_{\min} = \frac{500}{0.0133} \approx 37,594 \text{ W} \\ q_{\max} = \frac{500}{0.0072} \approx 69,444 \text{ W} \end{cases}
\end{aligned}$$

$$\boxed{R_t \approx 0.0072\text{–}0.0133 \text{ K/W} \quad \text{and} \quad q \approx 37.6\text{–}69.4 \text{ kW}}$$

Assuming steel-on-steel contact with $h_c = 2000\text{–}3700\ W/m^2 \cdot K$ [28].

In reality, the materials have thicknesses that slow down heat transfer. However, this example shows that conduction can be significant, and this serves as a worst case scenario. When designing an insulated manifold, heat transfer through conduction should be taken into account.

If we now try to model the entire interface more realistically, we can assume a thickness of 3 cm for both the exhaust manifold and engine block, and 2 mm for the gasket. Using a midpoint of the contact conductance value from the previous calculation, we can estimate the total thermal resistance and resulting heat transfer:

$$
\begin{aligned}
A &= 0.037689\ \mathrm{m}^2, & \Delta T &= 500\ \mathrm{K}, \\
k_{\mathrm{Fe}} &= 50\ \mathrm{W\,m^{-1}K^{-1}}\ [28], & k_g &= 5\ \mathrm{W\,m^{-1}K^{-1}}\ [31], \\
h_c &= 3.0 \times 10^3\ \mathrm{W\,m^{-2}K^{-1}}, & \mathrm{L}_1 &= L_3 = 0.03\ \mathrm{m}, \quad L_2 = 0.002\ \mathrm{m}\ [31]
\end{aligned}
$$

Total resistance (two contacts):

$$
\begin{aligned}
R_t &= \frac{L_1}{k_{\mathrm{Fe}}A} + \frac{1}{h_cA} + \frac{L_2}{k_gA} + \frac{1}{h_cA} + \frac{L_3}{k_{\mathrm{Fe}}A} \\
&= \frac{0.03}{50A} + \frac{1}{3000A} + \frac{0.002}{5A} + \frac{1}{3000A} + \frac{0.03}{50A} \\
&= 0.060\ \mathrm{K\,W^{-1}}.
\end{aligned}
$$

Heat flow:

$$
Q = \frac{\Delta T}{R_t} = \frac{500}{0.060} \approx 8.3 \times 10^3\ \mathrm{W}\ (8.3\ \mathrm{kW}).
$$

The heat transfer through contact could therefore be around 8 kW for the simplified case. However we quickly notice how sensitive this equation is, and that many assumptions are made. But for the goal of studying the behavior of the engine under less heat loss, 8 kW for contact and 2 kW for radiation/convection, a total of 10 kW was choosen.

4.1.2 Boundry conditions

Below are the boundary conditions for the original water-cooled model and for the new insulated manifold.

Keep in mind that the water-cooled exhaust was manually adjusted/calibrated so the gas temperature just ahead of the turbo matches values from engine test. As a result, every heat loss along the pipe has been "baked into" convection and radiation loss. Material properties etc were tuned, to recreate the total testing cooling effect. Because of that tuning it is tricky to design an insulated pipe using the same parameters.

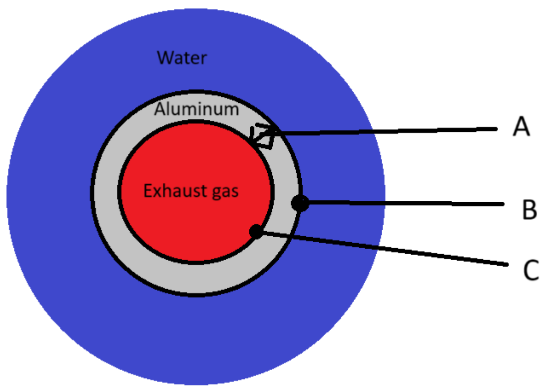


Figure 4.5: Cut through of pipe and boundary conditions for original exhaust

In the provided model, these material properties and boundry conditions were provided and already in the model. A: Thickness, material type, thermal conductivity and surface emissivity B: Boundry temperature and convection rate C: Initial temperature

Table 4.1: Original water-jacketed (single-wall aluminium) exhaust

Label	Thickness (mm)	Material	Thermal conductivity*	ε	T (°C)	Convection rate*
A	5	Aluminium	$\sim$ 130–30	0.8	—	—
B	—	Water	—	—	90	$\sim$ 200–1600

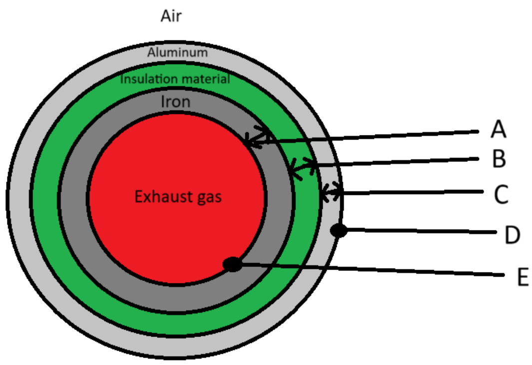


Figure 4.6: Cut through of pipe and boundary condition of insulated exhaust

The following is entered for each part: A: Thickness, material type, thermal conductivity B: Thickness, material type, thermal conductivity C:Thickness, material type, thermal conductivity and surface emissivity D:Convection rate, boundary temperature. E: Initial temperature.

Table 4.2: Cast-iron / Fibrafax / aluminium sandwich (insulated) exhaust

Label	Thickness (mm)	Material	Thermal conductivity*	ε	T (°C)	Convection rate*
A	5	Iron	$k_{\text{orig}}/3$	—	—	—
B	25	**Unknown**	**0.034**	—	—	—
C	4	Aluminium	$\sim$ 130–30	0.8	—	—
D	—	Air	—	—	25	10

*Thermal conductivity in $\dfrac{\text{W}}{\text{m} \cdot \text{K}}$; Convection rate in $\dfrac{\text{W}}{\text{m}^2 \cdot \text{K}}$. All parameters and boundary condition are entered in the thermal object of every pipe and flow-split of the exhaust manifold.

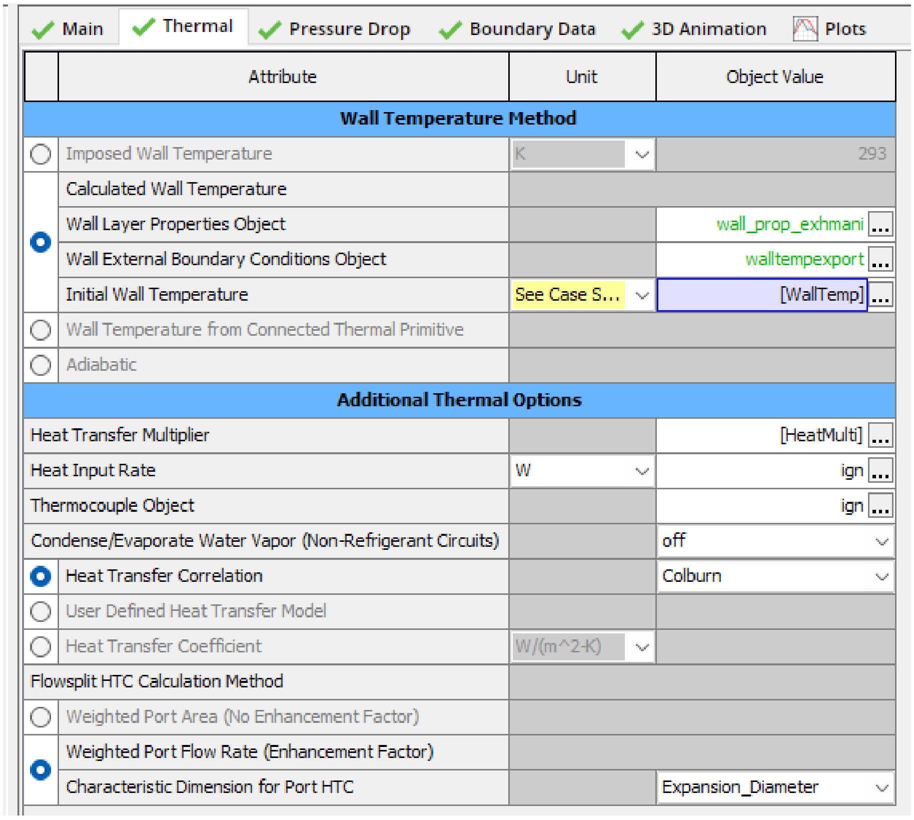


Figure 4.7: Thermal object in GT Power

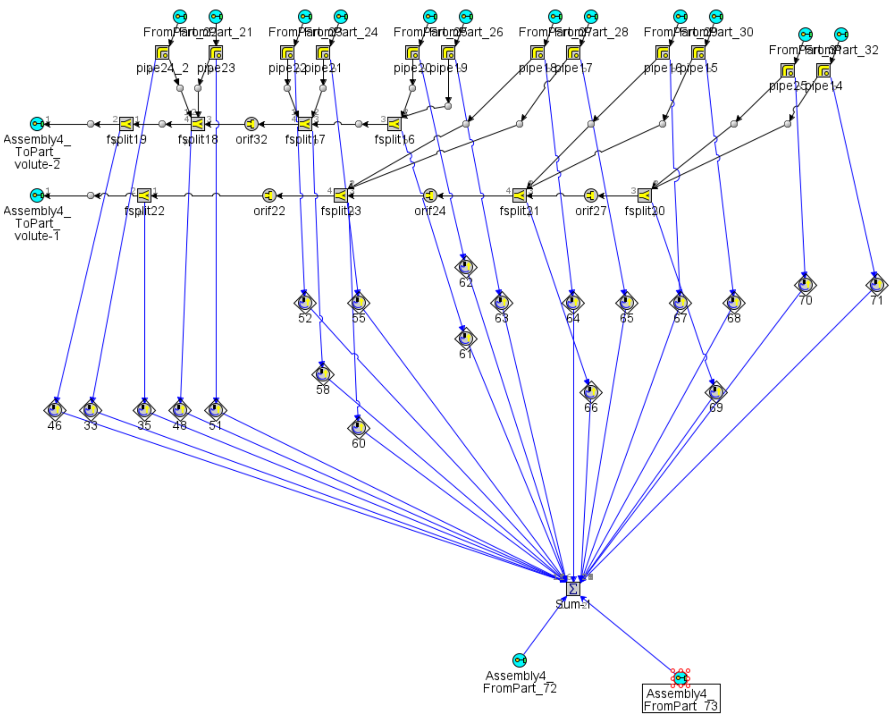


Figure 4.8: The exhaust manifold and heat measurement sensors

In the figure above we place a sensor that takes the heat transfer from each individual objects in the exhaust manifold and takes them into a sum. This will give the total heat loss in the exhaust manifold.

4.2 Turbo

In GT Power a turbocharger is represented by a turbine and compressor objected connect by a shaft between them. To be able to simulate a turbo these objects have their respective map that includes different operating speeds and the respective operating conditions and behaviors at these speeds.

4.2.1 Compressor Object

The main function of a compressor is to compress air, increasing both pressure and density of the air being delivered to the engine.

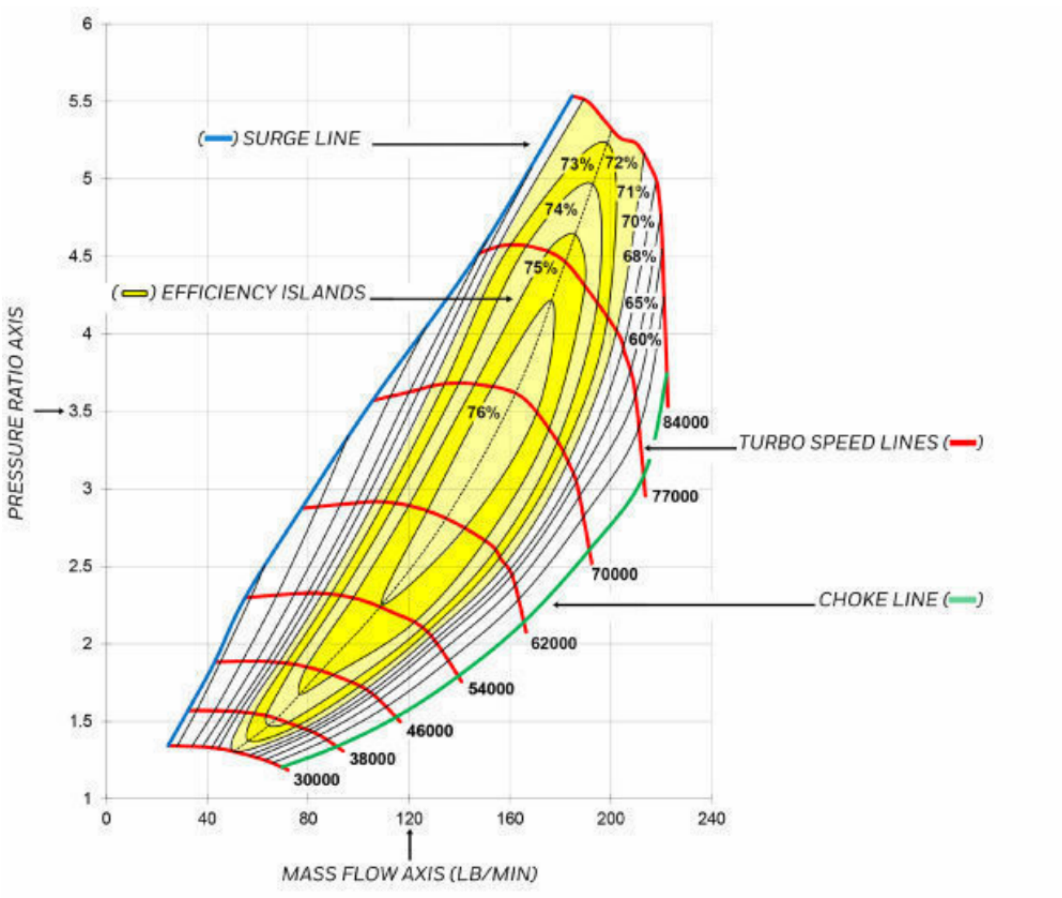


Figure 4.9: Compressor map ([3])

The map above is a graph of the compressor where it shows efficiency, mass flow, boost pressures and turbo speed. The "turbo maps" are provided by the manufacturer and below is a simplification on how they could calculate them.

$$\text{Pressure Ratio compressor} = \frac{P_{\text{outlet-absolute}}}{P_{\text{inlet-absolute}}}$$

When calculating pressure ratios it is important to use the absolute pressure. With absolute pressure means the pressure in relation to normal atmospheric pressure. Normal atmospheric pressure is around 1.01 bar, so a pressure reading at absolute pressure would read 0 if the actual pressure is 1.01 bar.

For a compressor the pressure at the inlet is often less than atmospheric pressure because of restrictive intakes and air filters. This needs to be accounted for and would be done by reducing the inlet pressure by the amount restricted by intakes and filters. The same method to adjust the absolute pressure has to be used if you are not at sea level and since atmospheric pressure likely would be different ([3]).

4.2.2 Map Fundamentals and Operating Limits show cased in Figure 4.9

- **Choke Line** — can be seen as the right-hand boundary for the compressor's operating zone. Past this line, efficiency is reduced and turbo speed will likely exceed its maximum allowed speed. If you are operating past this line you will need a larger compressor wheel [3].
- **Surge Line** — marks the left-hand boundary of stable operation. To the left of this line the flow becomes unstable; prolonged operation here accelerates wear and can lead to component failure [3]. Surge usually occurs when the compressor is still generating boost but mass flow drops sharply—e.g., a very large compressor wheel at low engine load or a sudden throttle closure after boosting. Blow-off or bypass valves can vent excess pressure, reducing boost and shifting the operating point back into the stable region [3].
- **Turbo Speed Lines** — represent loci of constant turbo-shaft speed. As speed rises, pressure ratio, mass flow, or both increase. You can interpolate between the lines to estimate intermediate speeds. When a compressor approaches or crosses the choke line, turbo speed increases rapidly [3].
- **Efficiency Islands** — closed contours that denote equal compressor efficiency. Peak efficiency lies at the center of each island, decreasing outward [3].

4.2.3 Turbine Object

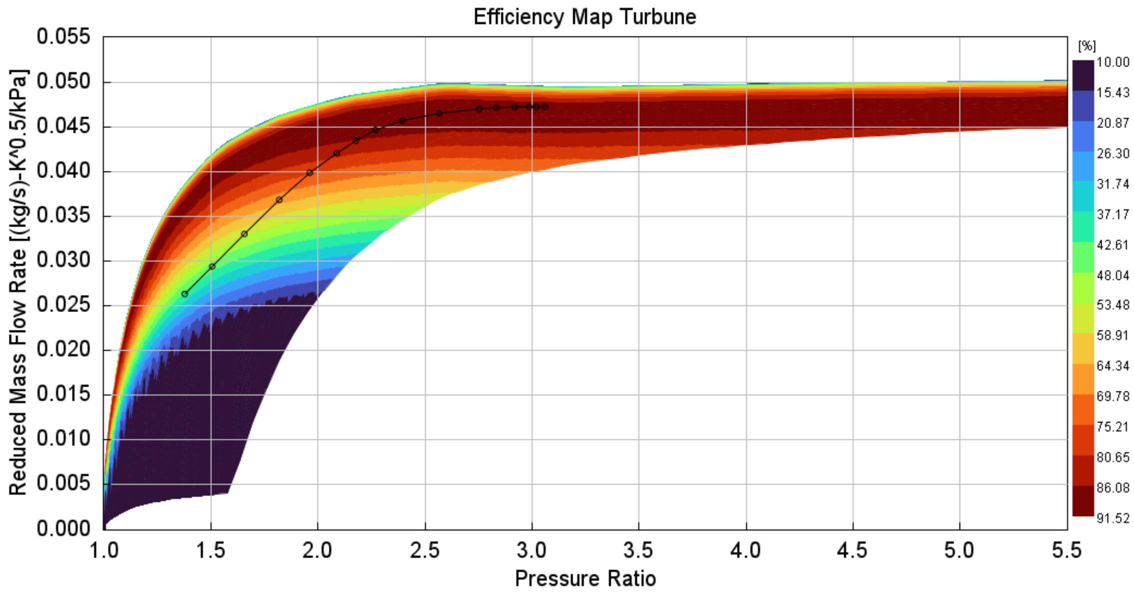


Figure 4.10: Compressor map (sourced from garret motion)

Effciency zones in the graph above is similair to the effciency islands mentioned for the compressor object

4.2.4 Dual Entry

A Dual entry turbo also knows as twin-scroll, is a turbo that has two separte inlets paths leading into the turbine wheel. This makes it able to better utilize exhaust pulses since they can be separated from the engine and go directly into one side of the turbine wheel with less collision and interference from other pulses compared to a single entry turbo. By utilizing more energy of the pulses of the engine it is possible to reduce turbo lag and increase efficiency.

4.2.4.1 Dual Inlet

Since a dual entry turbo has two separate inlets they have to be modeled as two separate units. This gives two flow paths into the turbo with their own respective pressure ratios, mass flow and dimensions.

5
CAD Modeling of the Exhaust Manifolds

5.1 Dual Pipe Exhaust Manifold Rear Mounted Turbo

To be able to utilize a the exhaust pulses better and have a dual entry turbo a dual pipe exhaust manifold was needed. A CAD model was made without mounting brackets since GEM3D (Discretization tool) only wants the volumes that materia can flow in. The turbo would be mounted in the rear to make packaging of the engine easier.

The exhaust system has smooth long bends to allow for a good flow with little disruptions and to minimize friction. The design is also narrow to make sure it doesn't take up much more space than the original design, to make it possible to fit in the current engine without too much modifications.

A dual bank exhaust manifold is supposed to separate the cylinders into two banks with three cylinders in each. With this type of system the exhaust pulses were not influenced by the exhaust pulses of the other bank.

To avoid restricting the flow, the dimensions of the inlet to turbo of Volvo Pentas bigger D8 engine was taken since this one was already producing 600 hp. The cross sectional area of the exhaust manifold were taken from the exhaust ports and then swept to the cross sectional area of the turbos inlet as previously mentioned.

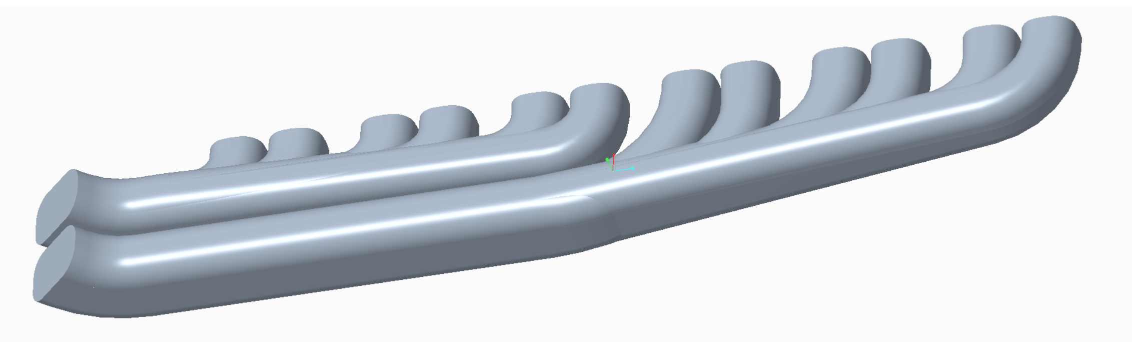


Figure 5.1: CAD of Dual Pipe Exhaust Manifold

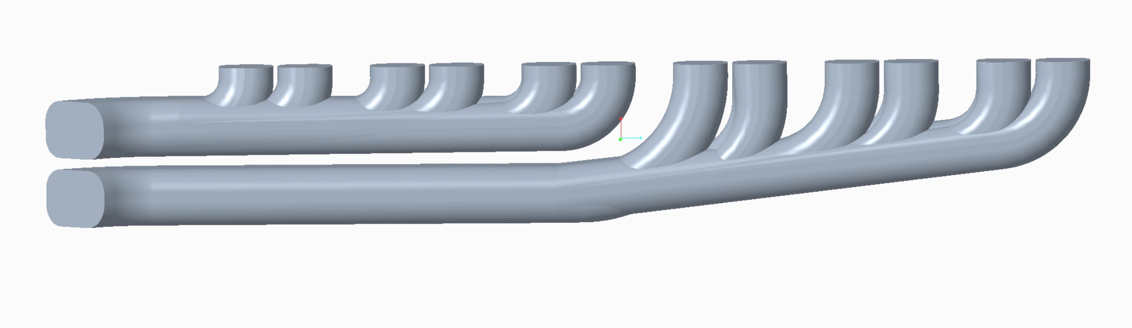


Figure 5.2: CAD of Dual Pipe Exhaust Manifold

Wall thickness of 5 mm was used for the manifold as this was used for the previous manifold. The area of the inlet is 1644.73 mm^2 with 41.89 mm width and 44.15 mm height. The area of the outlet is 2697.41 mm^2 with 44.49 mm in width and 66.5 mm in height.

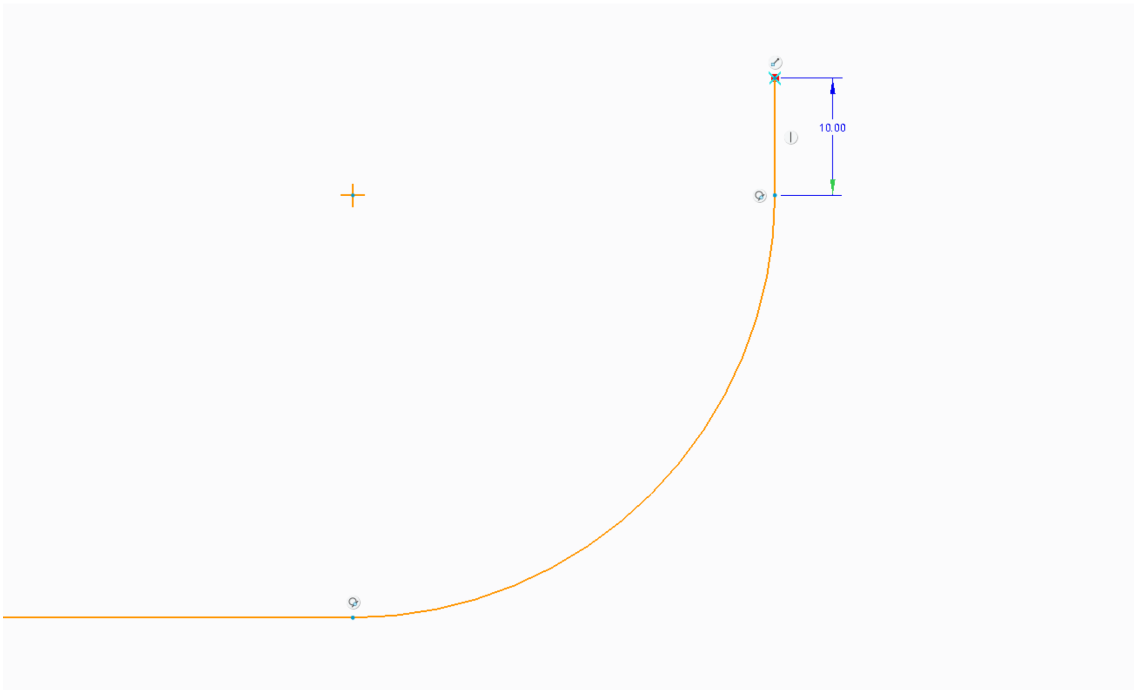


Figure 5.3: Dual Piped Exhaust Manifold placed inside old cooling chamber

The CAD is made up out of multiple sweeps with center points in each inletport swept until the center points of the turbo inlets. The position of these inlet ports is given by the position of the engines exhaust port since they have to be connected. The position of the turbo was taken from the previous turbo position.

The bends and pipe themselves are going straight for 10 mm from the inlet port to not hit anything on the engine block. An arc between the line from the manifold inlet to manifold outlet is later made to be as big as possible and have as small as possible angle to make it smooth for both exhaust flow and CREOs sweep function. This arc tangents both lines and creates a smooth sweeping path.

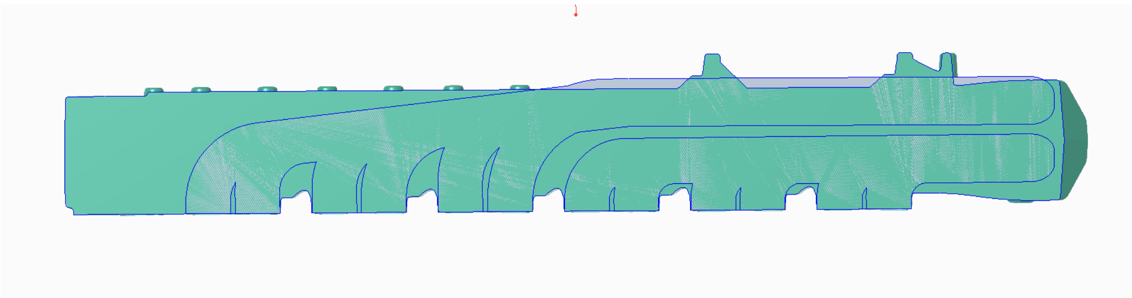


Figure 5.4: Dual Piped Exhaust Manifold placed inside old cooling chamber

The figure above shows our new Exhaust Manifold placed inside the old cooling chamber. As can be seen it is bigger and would require an even bigger cooling chamber than the previous one. It would be possible to extend the cooling chamber towards the two mounting fixtures in the top part of the figure.

5.2 Mid Mounted Turbo Exhaust Manifold

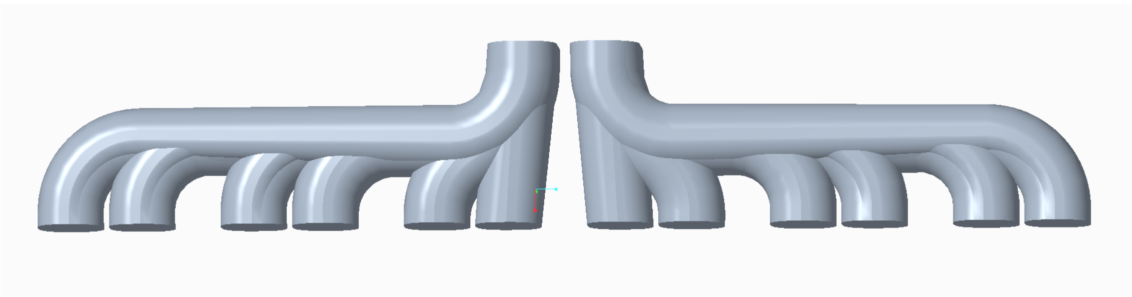


Figure 5.5: CAD of Mid Mounted Turbo Exhaust Manifold

The mid mounted exhaust manifold was designed to reduce the distance, the exhaust pulses have to travel until they reach the turbo. Bends were kept smooth to increase the flow like the rear mounted version. The exhaust manifold has pipes from each exhaust port that connects to a "collector" that connects the first three cylinders into one part of the exhaust manifold and the same goes for the last three cylinders.

The dimension of the inlet and outlet ports of the exhaust manifolds were the same as for the dual piped rear mounted turbo one. To keep the mid mounted setup fairly compact the pipe furthest away from the engine block's turbo inlet was chosen as the exterior limit which came to around 110 mm. The sweep was with 10 mm straight sections in the beginning and the end as well with two 45 mm radius bends for the pipe furthest away from the middle and the closer ones had 45 mm radius bends to connect to the pipe from the cylinder furthest away until the pipe closest to the middle being just an angeld straight line into the "collector" for the rest of the pipes.

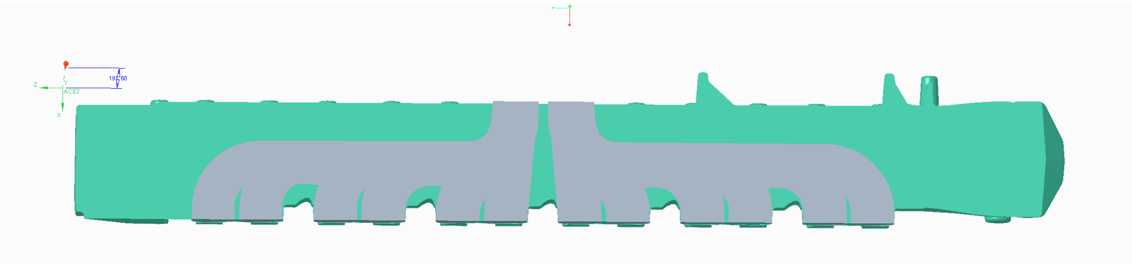


Figure 5.6: CAD of Mid Mounted Turbo Exhaust Manifold

In figure 5.6 we have the new mid mounted turbo in grey and the orginal outer shell with water cooling in green. As can be seen the mid mounted exhaust manifold itself takes up less space than the dual piped exhaust manifold with rear mounted turbo. But the whole setup with having the turbo mid mounted will add the turbos dimensions to the width of the engine package. However when mounting the turbo in the middle, more space will of course be available in the rear where it was previously mounted.

By having the size of the manifold being smaller than the current design the total area to be cooled is reduced and this gives us in theory a lower heat loss.

5.3 Discretizing to GT-power (Gem3D)

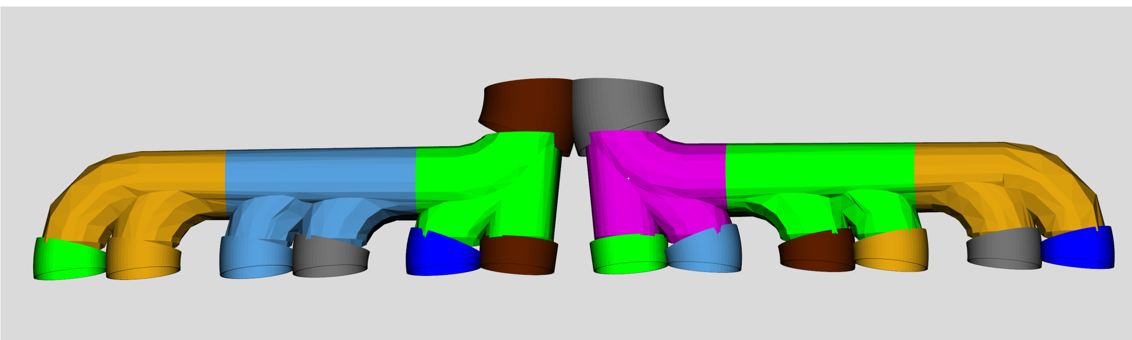


Figure 5.7: Discretization of Mid-mounted exhaust manifold

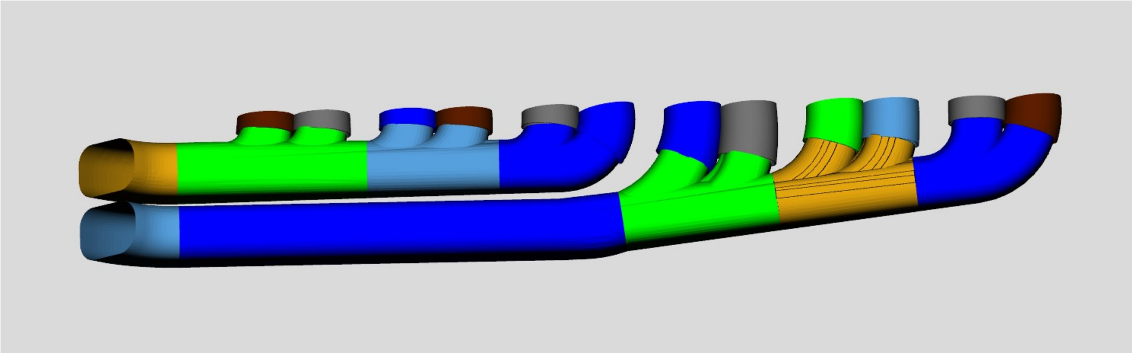


Figure 5.8: Discretization of sidemounted exhaust manifold

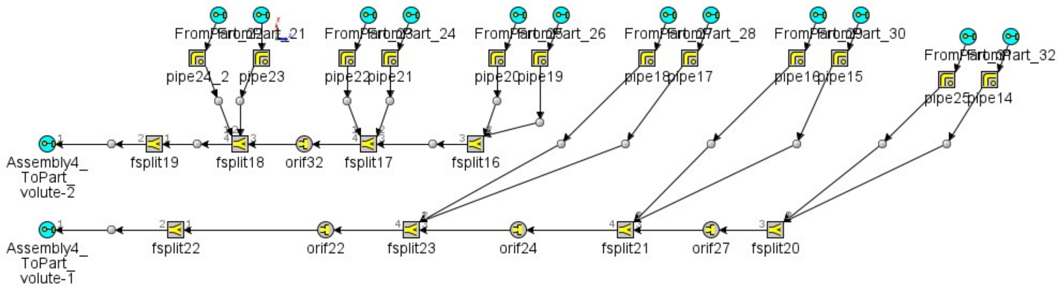


Figure 5.9: Discretized exhaustpipe imported into GT-power

The discretizations of the manifold consists of pipe, flow splits and oriface type objects. The metrics and volumes of the objects are automatically calculated by GEM3D. Input and output ports are selected manually. Directions of flow for each object is displayed in GEM3d (Figure 5.9), and are manually crosschecked if they appear as expected.

Wall layer properties, wall external boundary conditions and initial wall temperature are the same as the original exhaust. The heat transfer multiplier is changed between models to achieve the desired total heat loss.

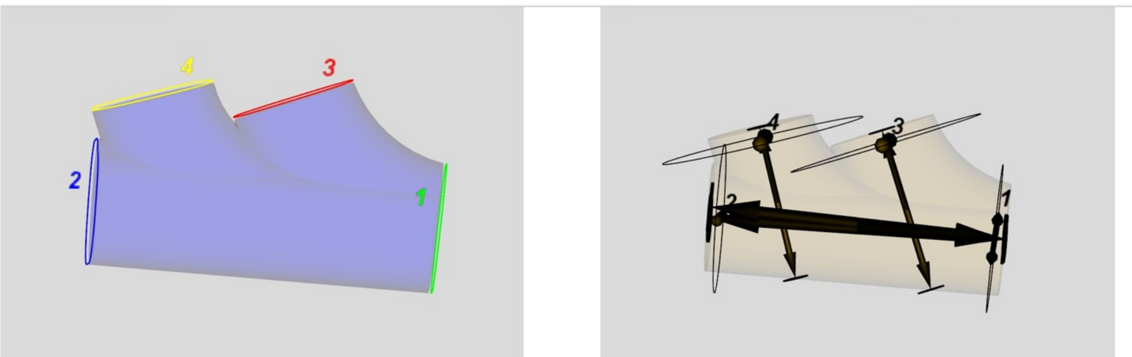


Figure 5.10: Flowsplit in GEM3d

6
Results

6.1 Validation results

Before the GT-Power model can be trusted, it has to be validated against measurements from a real engine. The model needs input parameters like engine speed, rail pressure, injection timing etc, which are extracted from an test log and fed into the model (see Figure 6.1). GT Power then predicts output metrics. By plotting these simulated results alongside the corresponding data in the test protocol, we can evaluate how closely the model replicates actual engine behavior and decide whether additional calibration is necessary.

The GT Power model validation results are separated into two parts: full load validation and part load validation. The part load and full load validations were based on different engine test protocols, meaning that the input parameters differed between the two runs. Consequently, each comparison was made according to its respective protocol.

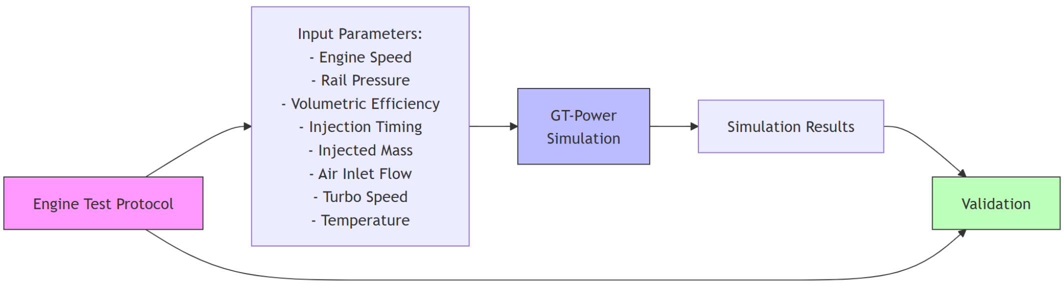


Figure 6.1: Flowchart of the validation

6.1.1 Full load validation

The full load validation include 16 full load operation points between 1000 to 3800 rpm. The results can be seen in table 6.1.

Most of the parameters (Brake Power, Brake Torque, BMEP, BSFC, and TBT) have relatively small average absolute errors, generally in the range of 1-2 percent. This suggest that the model is relatively accurate for those parameters.

NOx stands out as it has a much higher error with a mean absolute error of about 15%. NOx is therefore significantly less accurate than other parameters. This is consistent with other research that NOx predicting is very sensitive and difficult to predict. Arrègle et al. showed that a mistake in the temperature of the air in the intake can inflate the error of the NOx estimate 20x, while an error in the measured intake air mass can inflate the same error 12x. In contrast, getting the in-cylinder pressure slightly wrong has almost no effect on the predicted NOx[32]. Arrègle et al also state that the specific NOx model is largely irrelevant, because the errors arise earlier, when determining the thermodynamic conditions in the cylinder and when estimating the oxygen concentrations[32].

Lambda has a slightly higher mean abs error than most parameters, however the mean error of -2.5 percent shows that lambda is under predicted. The standard deviation of 1.14% is moderate. Therefore lambda is relatively consistent, just shifted negatively.

Table 6.1: Error Metrics for full load (Simulation vs Test protocol).

Parameter	Mean Error (%)	Std Dev (%)	Most Negative (%)	Most Positive (%)	Mean Abs. (%)
T_B_T (Outlet Pipe 2)	-0.383	2.106	-7.621	3.061	1.453
NOx Concentration (ppm)	13.352	13.755	-9.775	32.116	15.033
Brake Power	-0.942	1.538	-3.666	1.852	1.343
Brake Torque	-0.957	1.511	-3.649	1.829	1.341
BMEP	-0.834	1.462	-3.576	1.840	1.310
BSFC	1.274	1.907	-1.816	3.775	1.457
Apparent Lambda	-2.510	1.140	-4.114	0.776	2.515

6.1.2 Varying load validation

The varying load validation include 21 operating points at 2400 rpm, and 18 at 3800 rpm. These points can bee seen as varying the load on the engine, while keeping the rpm constant.

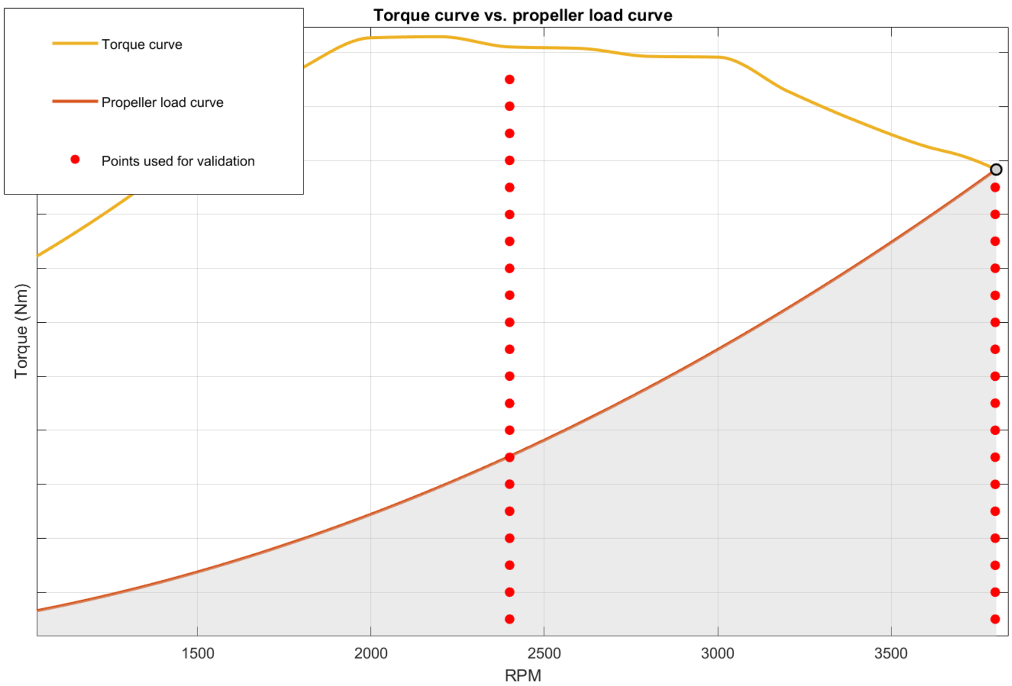


Figure 6.2: Part load validation points

There were some operating points in the test protocol that had negative power output, these were excluded.
The results are shown in table 6.2

Table 6.2: Error Metrics for Part load at 2400 rpm (Simulation vs Test protocol).

Parameter	Mean Error (%)	Std Dev (%)	Most Negative (%)	Most Positive (%)	Mean Abs. (%)
T_B_T	-18.59	6.66	-25.88	-4.63	18.59
NOx Concentration	36.18	49.66	-21.54	199.98	43.29
Brake Power	6.47	149.18	-490.70	389.54	69.11
Brake Torque	6.51	149.13	-490.29	389.83	69.09
BMEP	-89.23	13.42	-134.13	-57.13	89.23
BSFC	-34.40	62.13	-267.83	2.47	35.10
Apparent Lambda	-37.68	16.17	-60.26	-7.70	37.68

In total, the error metrics for part load are greater than those for full load, indicating a need for calibration. Some mean errors are particularly large, such as BMEP at -89%, while others, like brake power and torque, exhibit a large spread (standard deviation of 149%).

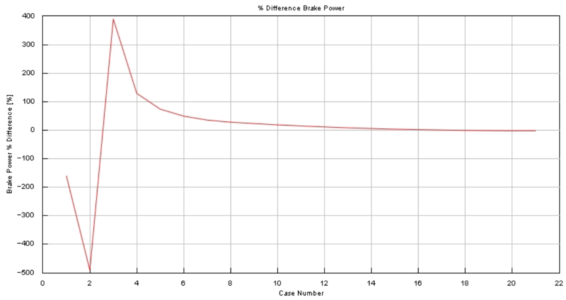


Figure 6.3: Brake power error at part load (2400 rpm)

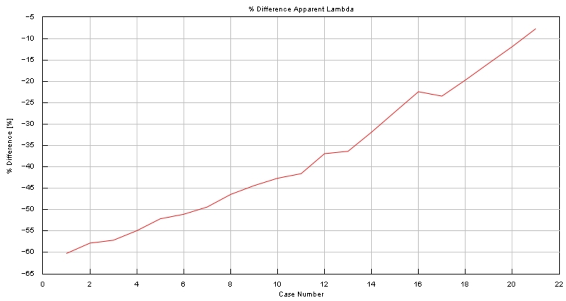


Figure 6.4: Lambda error at part load (2400 rpm)

Figure 6.3 and 6.4 showcase how Lambda and Brake Power has a large spread and a big error margin at lower loads. The error margins decrease as loads increase. This behavior applied for all parameters in table 6.2. This discrepancy between simulated and measured data shows that the Gt-power simulations are inaccurate with error to large to trust.

6.2 One-Parameter-at-a-Time Analysis

Several test simulations of the original model were carried out, with one input parameter changed at a time. The engine's responses to these changes revealed its current tuning and suggested directions for further adjustments.

The parameter changes showed the behavior of the 1-D model of the Penta D6 engine. Its meant to showcase in what direction the change of one engine parameter of the engine will lead to. It is not necessarily applicable to all diesel engines, and it is not certain that the same direction will hold if multiple parameters are changed simultaneously.

6.2.1 Altering injection timing

Earlier injection timing (advancing the timing) was found to shift the main phase of combustion closer to top dead center, thereby increasing in-cylinder pressure and temperature while the piston is still near its highest point. When Looking at 50% and 90% of the fuel had been burned, along with in-cylinder temperature and pressure, the simulations showed that earlier injection burns the fuel hotter, faster, and earlier in the cycle. This results in higher power and improved efficiency (lower BSFC). However, the higher combustion temperature leads to increased NOx emissions, a reduction in lambda, and lower exhaust temperatures upstream of the turbo.

The earlier and hotter burn can be assumed to decrease soot.

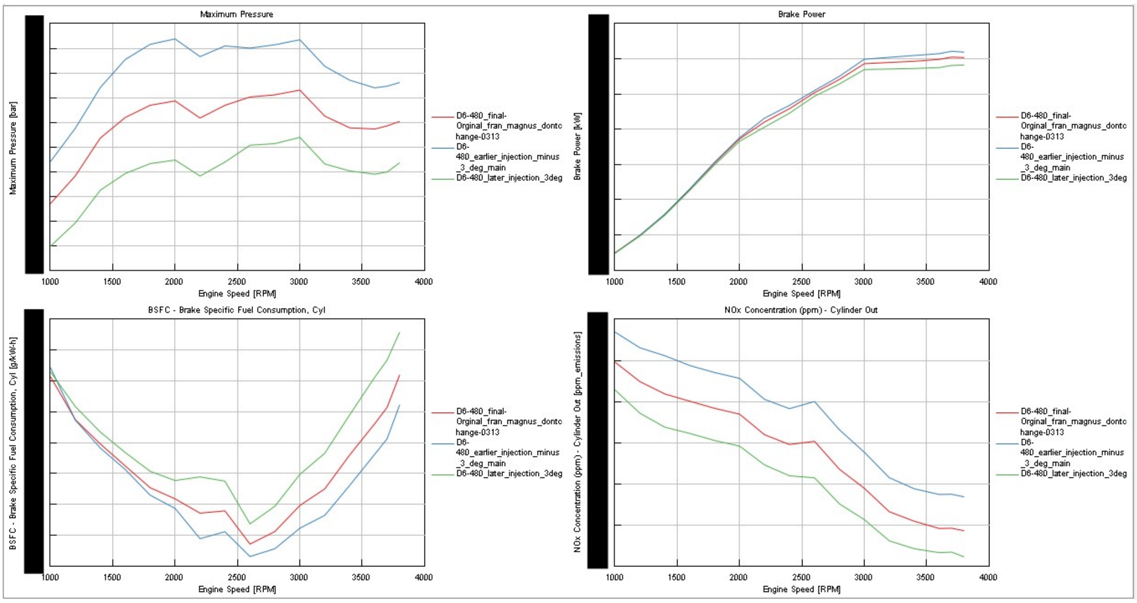


Figure 6.5: Original(Red) compared to 3 CAD earlier(Blue) and later (Green) injection timing

6.2.2 Adjusting fuel amount

Increasing the total amount of fuel injected showed a rise in engine power output, as more chemical energy becomes available for conversion into mechanical work. Because the air mass flow was not proportionally increased, this additional fuel lowered the lambda, creating a richer mixture. The peak in-cylinder pressure and exhaust temperatures upstream of the turbo also increased.

The lower lambda lead to a slight decrease in NOx emission, but since there was less available air for combustion, soot can be assumed to increase.

6.2.3 Wastegating

The current engine is not currently equipped with a waste gate. It is however interesting to experiment with a "theoretical wastegate" to see how the engine would respond when letting some exhaust gas bypass the turbo.

When simulating the engine with a wastegate the air supply to the engine decreased, as less air is propelling the compressor side of the turbo. This reduce lambda in the exhaust gases and more importantly raises exhaust gas temperatures.

Wastegating the current engine showed a decrease in engine power and torque. Waste-gating also increased exhaust gas temperatures before the turbo.

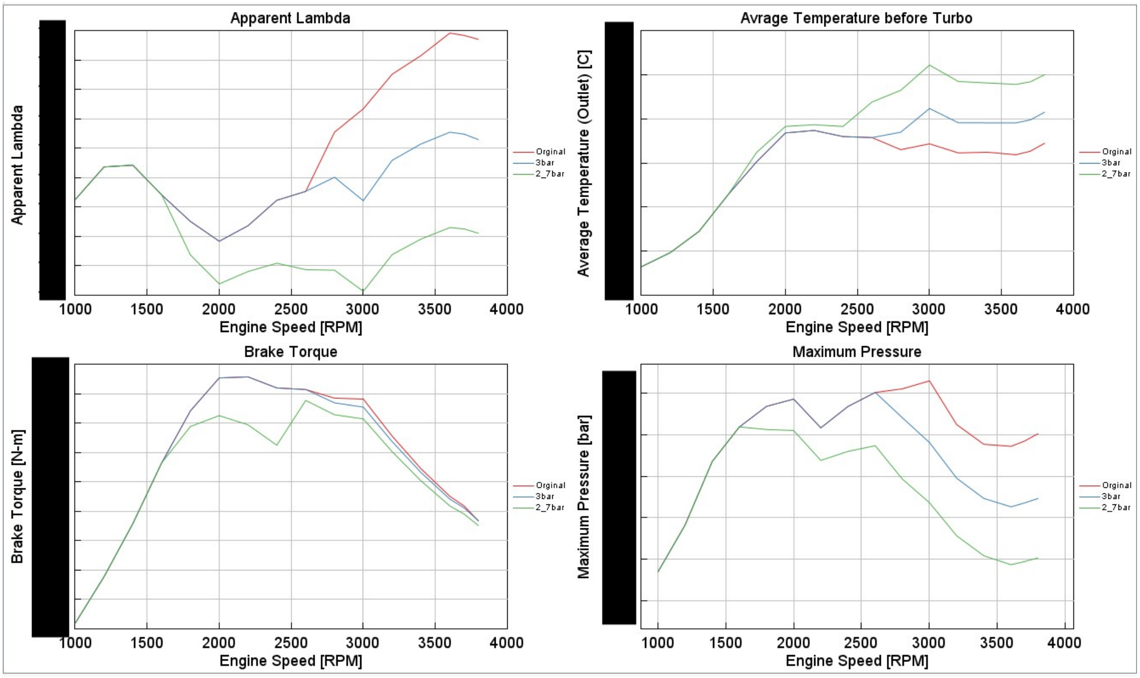


Figure 6.6: Effects of dumping turbo pressure with wastegate. Orginal(Red) wastegate open at 3bar(blue) and 2.7bar (green)

6.2.4 Summery tabular

Table 6.3: Engine response to parameter change

Parameter Change	Power	Peak Press.	NO_x	Lambda	Temp before turbo	Fuel Cons.
Earlier Injection	↑	↑	↑	↓	↓	↓
Later Injection	↓	↓	↓	↑	↑	↑
Increased Fuel	↑	↑	↓	↓	↑	↑
Wastegating	↓	↓	–	↓	↑	↑

6.3 GT-Power simulation results

This section presents simulation results from GT-power, focusing on various combinations of exhaust manifold geometry, cooling strategies, and turbocharger setups. Finally, results from an optimized model with advanced injection timing and increased fuel delivery are detailed.

6.3.1 Pipe modeling strategies

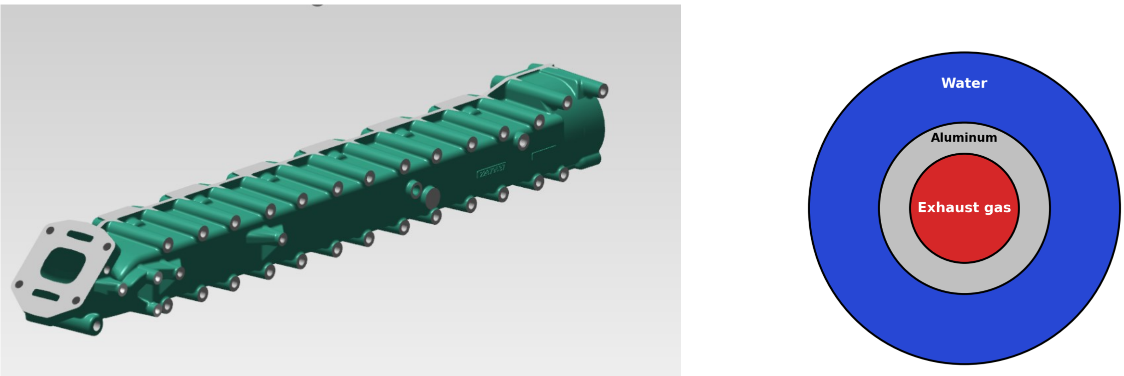
Two modeling techniques for simulating reduced heat loss were discussed earlier in the methods chapter. The first technique involved maintaining the original boundary conditions from the water-cooled exhaust manifold while reducing the heat transfer multiplier (rate of heat transfer) to 0.1. This simplified approach aimed at achieving approximately 10 kW heat loss for the original exhaust. The second technique involved detailed modeling, incorporating each layer of the insulated pipe with specific boundary conditions and accurate material properties.

Both modeling methods were applied to the final system simulation, and the comparative results showed minimal differences. Differences observed were 0.2% for brake power, 0.25% for brake torque, 1.1% for lambda, and 2.1% for exhaust gas temperature before the turbocharger. The total heat loss was 6.69 kWusing the heat multiplier approach, compared to 4.8 kW using the detailed insulated pipe model.

Complexity or specifics of the modeling method is therefore not significant. A noteworthy observation from the detailed pipe modeling was the significantly lower surface temperature, peaking around 60°C, compared to the previously estimated limit of 220°C. This difference suggests either overly effective insulation or differences in convection and radiation assumptions compared to manual calculations.

This variation underscores the challenges in accurately modeling heat loss and determining exact values. Ultimately, the critical goal remains understanding and simulating the effects of reduced cooling, rather than precisely quantifying heat loss or specific surface temperatures.

6.3.2 Original system



(a) Original log type exhaust manifold **(b)** Cut-through

Figure 6.7: Simulated system

The original system with a log type water cooled manifold with stock turbocharger serves as a baseline that will be used to compare the other models with and changes against.

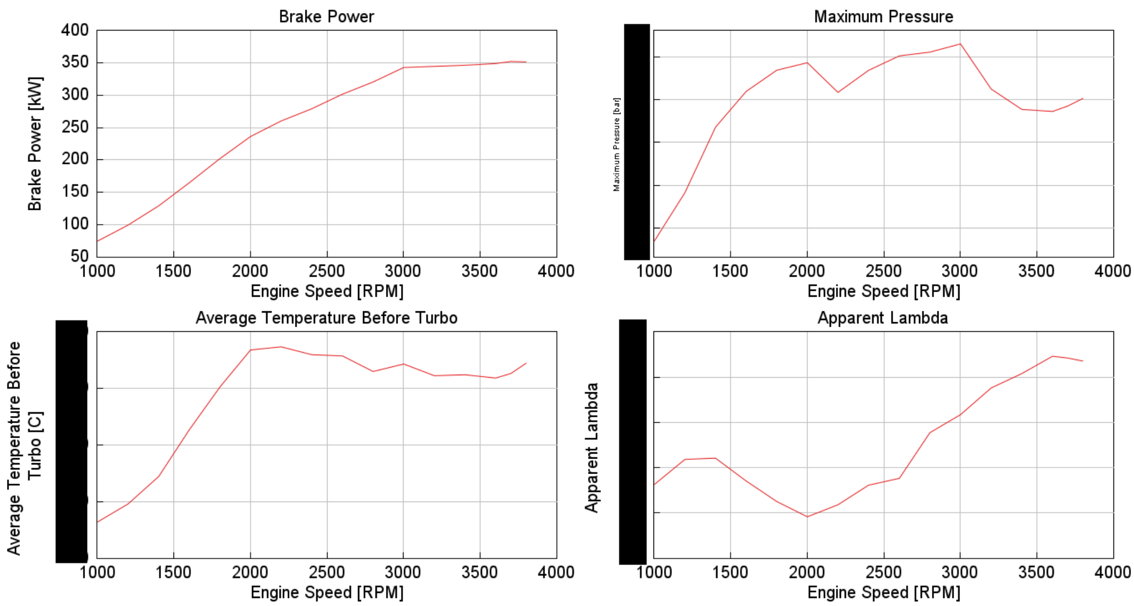


Figure 6.8

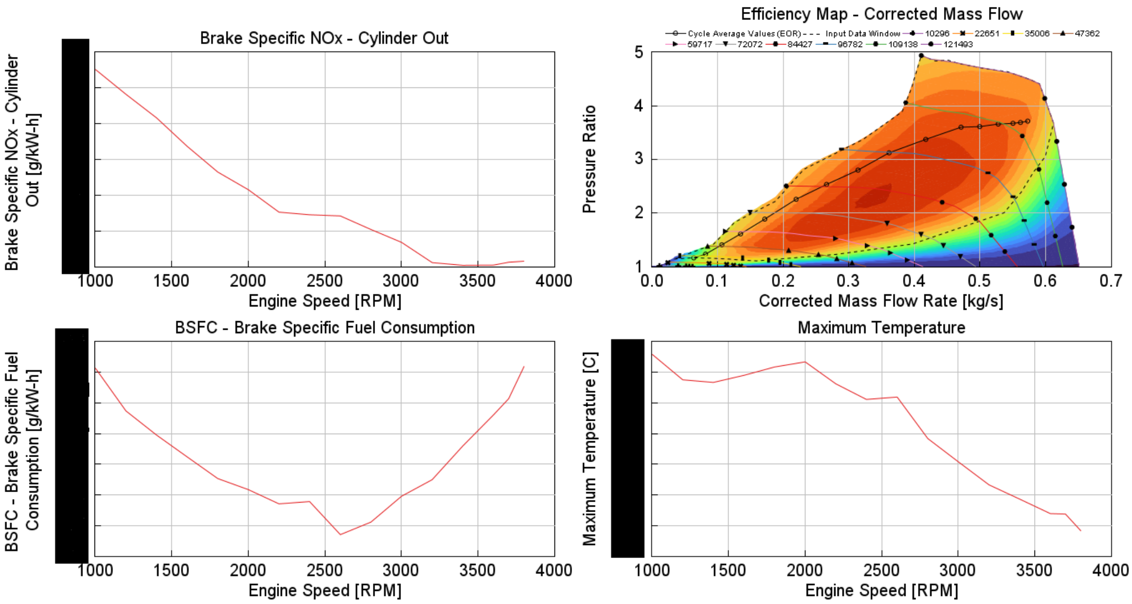


Figure 6.9

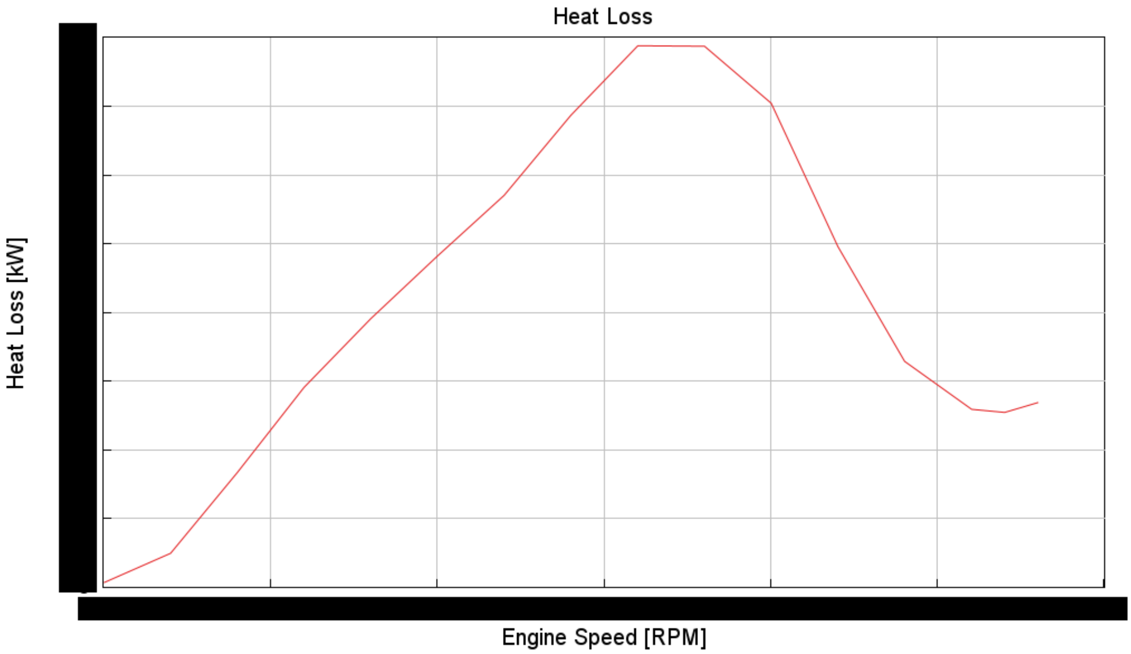
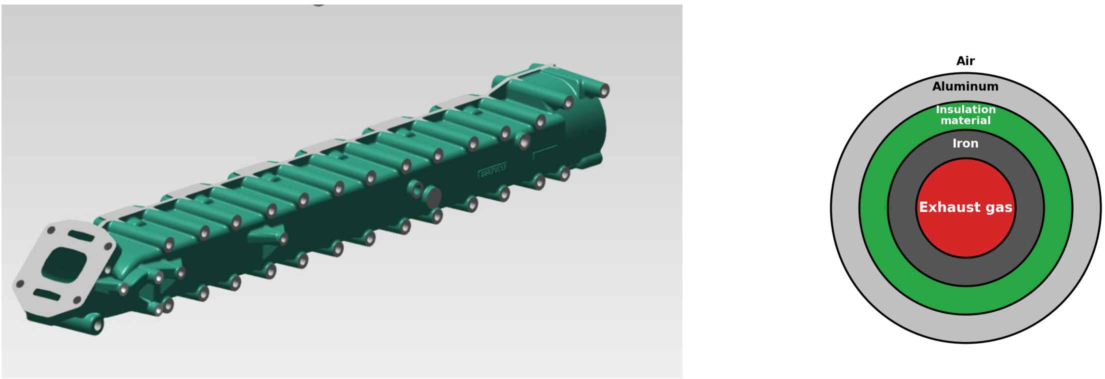


Figure 6.10

6.3.3 Original water-cooled, with less heat loss



(a) Original log type exhaust manifold   **(b)** Cut-through

Figure 6.11: Simulated system

Original manifold with reduced cooling to the exhaust, “dry” or partially insulated manifold to decrease heat loss.

Summary:

- Temperature before turbo increase significantly
- Hotter exhaust spins the turbine faster.
- Extra boost increases air into engine.This increases power and torque *until* the compressor reaches its map limit.
- At high rpm the compressor side of the turbo “runs out of map”: efficiency drops, intake air gets hotter, back-pressure rises, and power falls.
- Reasoning for next simulation: A larger turbocharger that can handle increased flow.

Reducing the cooling of the original exhaust raises the temperature of the exhaust gas. The higher enthalpy flow spins the turbine faster at all engine speeds, which shifts the compressor operating line upward and to the right on the efficiency map.

At low engine speeds, the new operating points still lie within the high-efficiency region of the map, so the increased compressor flow results in higher cylinder air charge and improved power and torque. As engine speed increases, the trajectory eventually reaches the boundary of map. At this point, GT-Power clips the operating points at the map edge, applying a last valid efficiency (around 64%). As a result there is less power and torque as a result. This loss can be explained by higher temperature of inlet air, increased backpressure and

Reducing heat loss in the exhaust manifold only improves performance while the compressor operates within its high efficiency range. Once it hits the map ceiling or the end of the map the efficiency starts to drop fast. To maintain the benefits across the full rpm range, one would need a larger compressor or some form of turbine flow management, such as a variable geometry.

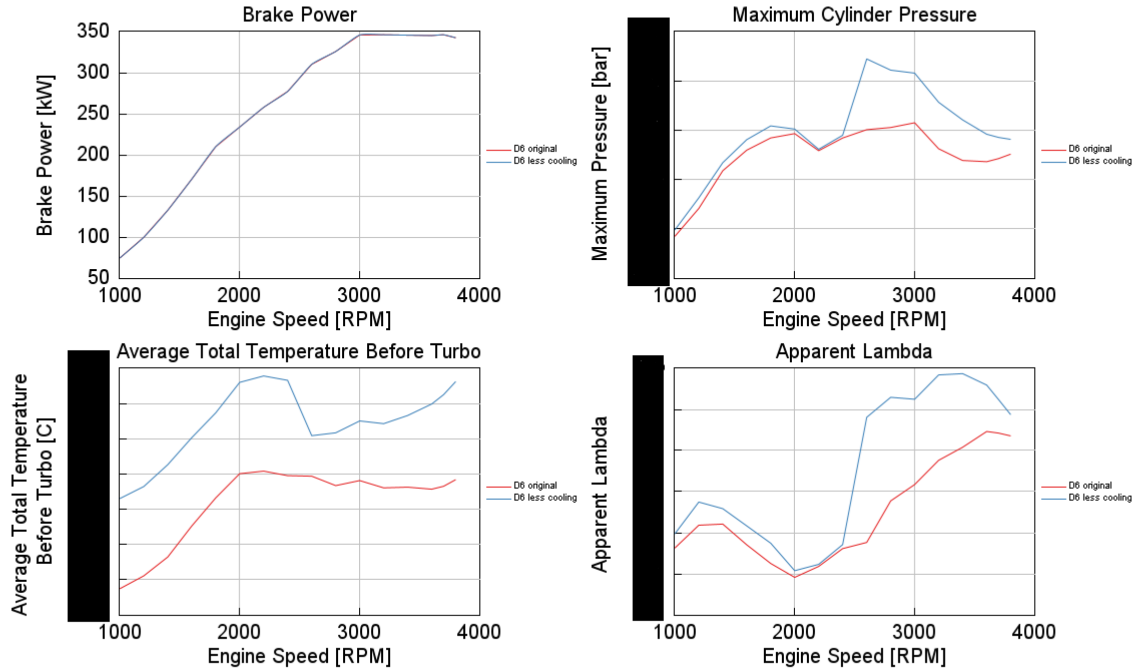


Figure 6.12

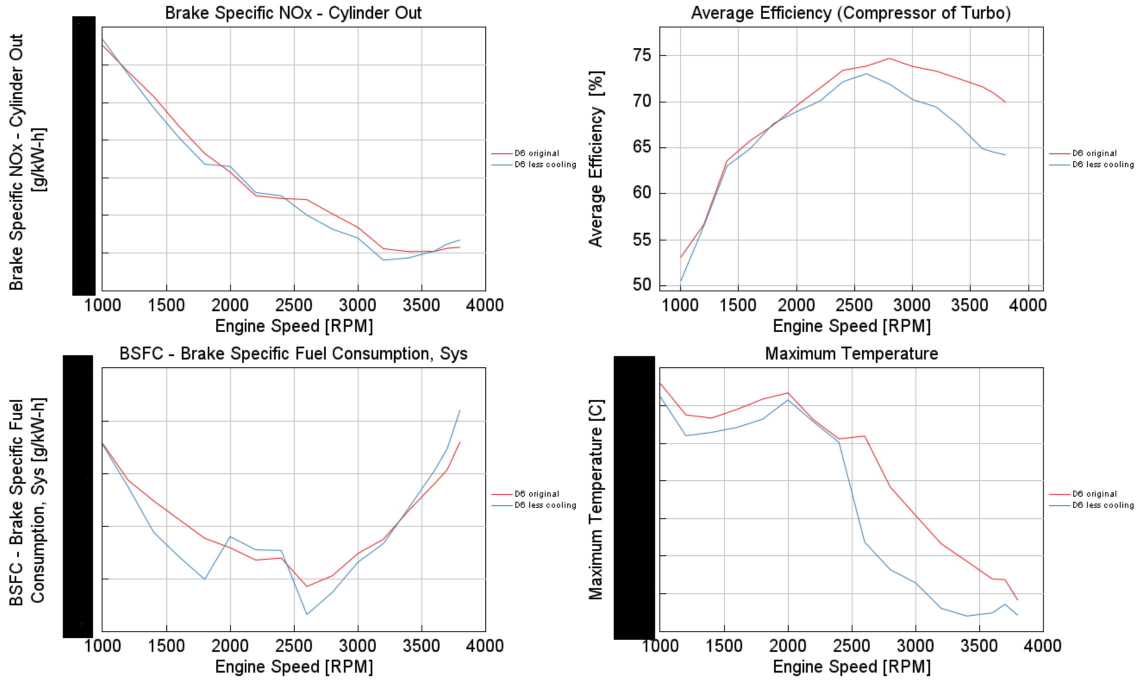
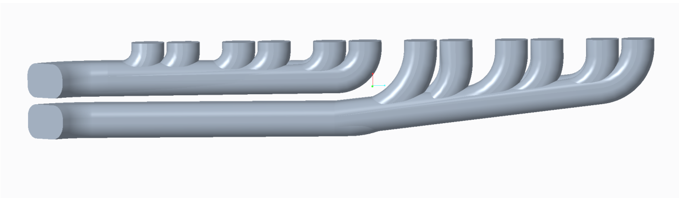
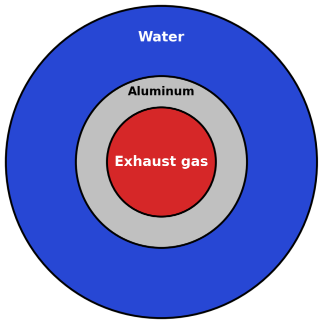


Figure 6.13

6.3.4 2 Bank system with water-cooling



(a) Dual-pipe exhaust manifold



(b) Cut-through

Figure 6.14: Simulated system

Exhaust manifold divided into two groups of three cylinders, each flowing into the stock turbo and has 100% cooling still being applied.

The 2 bank system has an increased surface area of which the exhaust gas has contact with the manifold of about 17% Summery:

- Banking of manifold is beneficial even when the boundary for cooling conditions are the same as original. The performance gain is slight.
- Heat loss was increased but the increased performance was greater than the additional loss.

In theory having the exhaust pulses interfere less with each other should result in improved efficiency. Bench-tests of a 4-2-1 pulse manifold by Galindo et al. demonstrated up to 6 % torque gain and faster transient response by reducing pulse interference in a a high-speed turbocharged diesel engine[33].

To be able to test this, a two bank system was designed that kept cylinder 1,2,3 in one bank and the reaming in another bank. From the graphs below we can see some slight gains in power mostly in the top end and a slight increase in BSFC. This show case that just changing the routing of the exhaust manifold to have the exhaust pulses interfere less as previously mentioned, will result in small improvements.

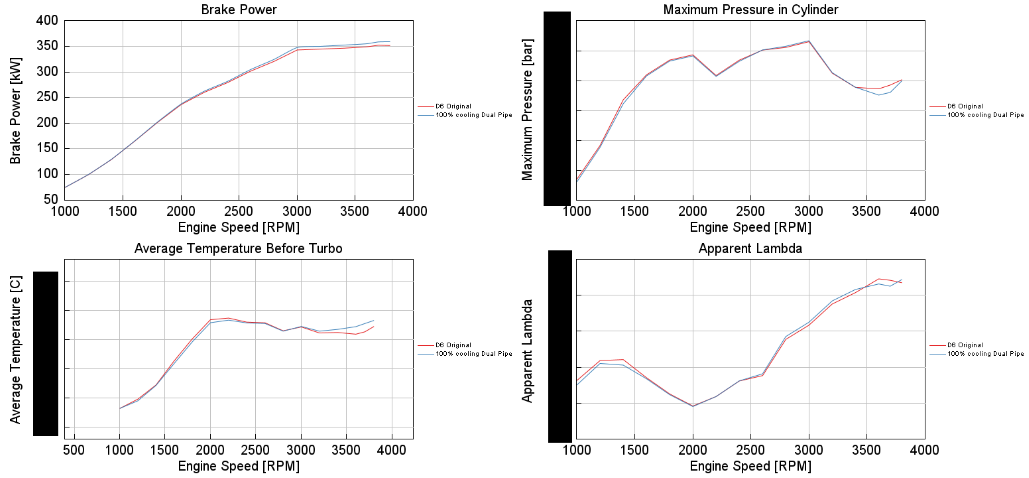


Figure 6.15

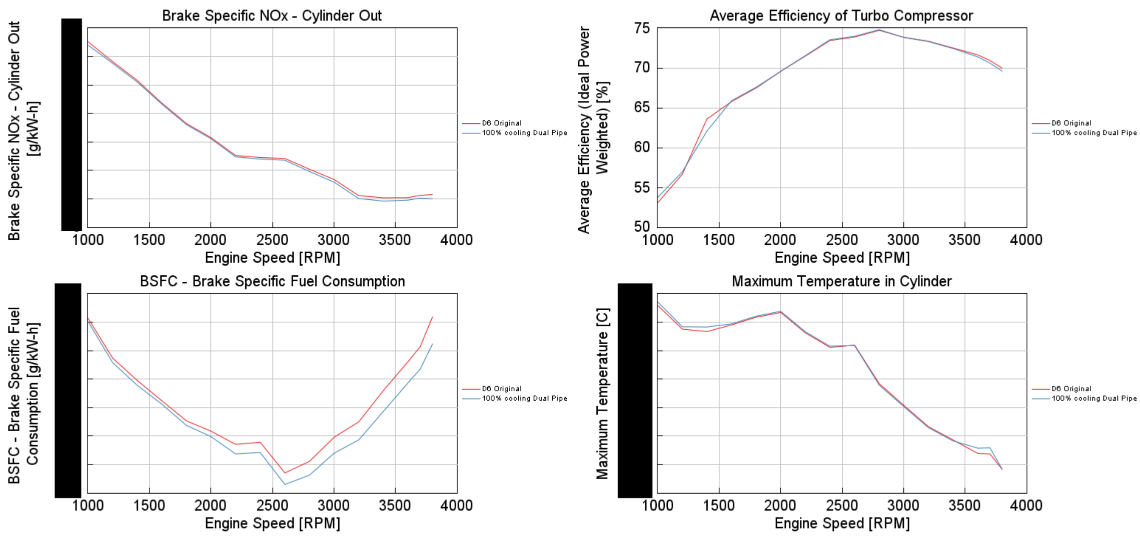


Figure 6.16

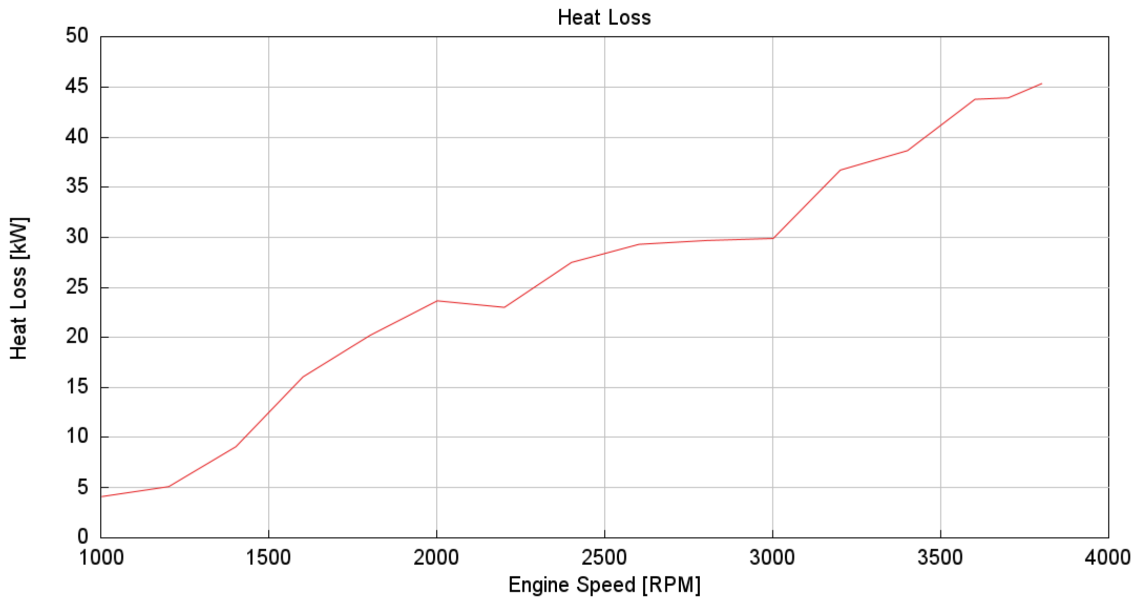
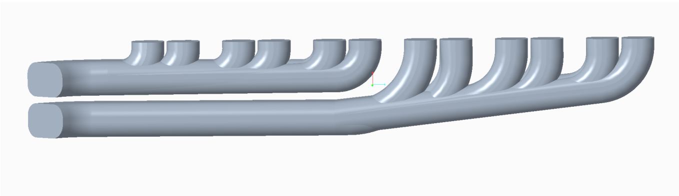


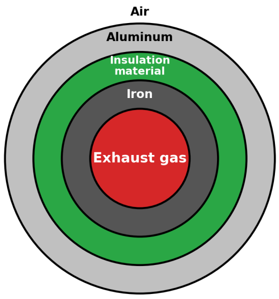
Figure 6.17

In the graph above we can see that the max heat loss of around 45 kW is around 3800 rpm. However compared to the orginal the dual piped version with water cooling is constantly increasing with rpm.

6.3.5 Dual Bank system dry pipe



(a) Bank divided exhaust system



(b) Cut-through

Figure 6.18: Simulated system

(Two-bank manifold as above, but with coolant flow removed. A “dry” manifold to maximize exhaust heat to the turbo.)

With the reduced cooling of the 2 bank system, the same type of behavior can be seen as in the original less cooled exhaust. There is initial gain at lower rpm and a loss at high rpm. Due to the turbocharger working less efficiently.

In reducing cooling for the two bank system we have increased the temperatures before turbo a lot and have some improvements in BSFC in the mid rev range. There is lower max temperatures in the cylinder and with the increased temperatures before the turbo we can see that we have gotten a lower efficiency in compressor side of the Turbo.

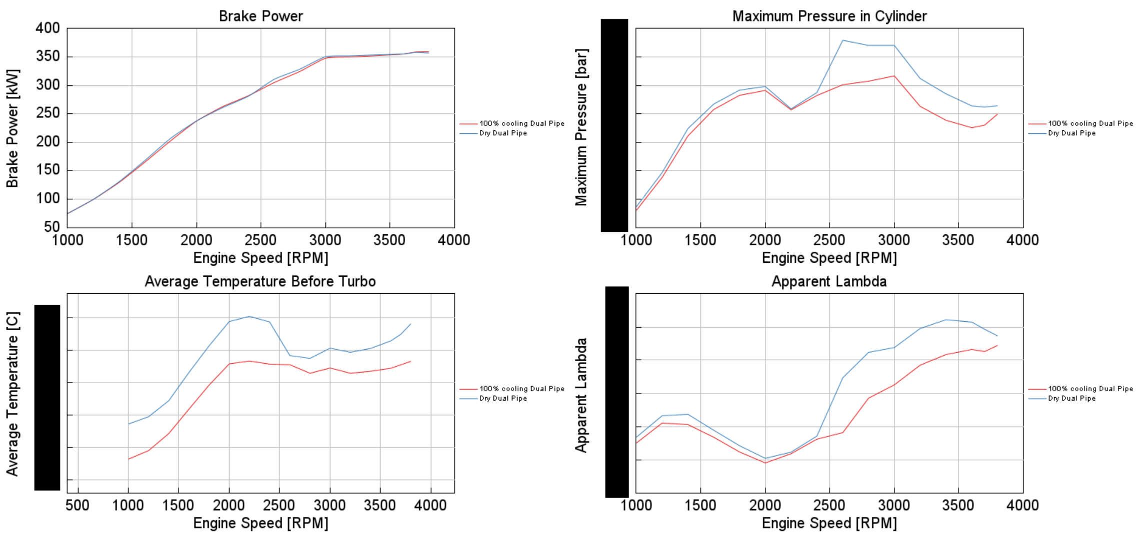


Figure 6.19

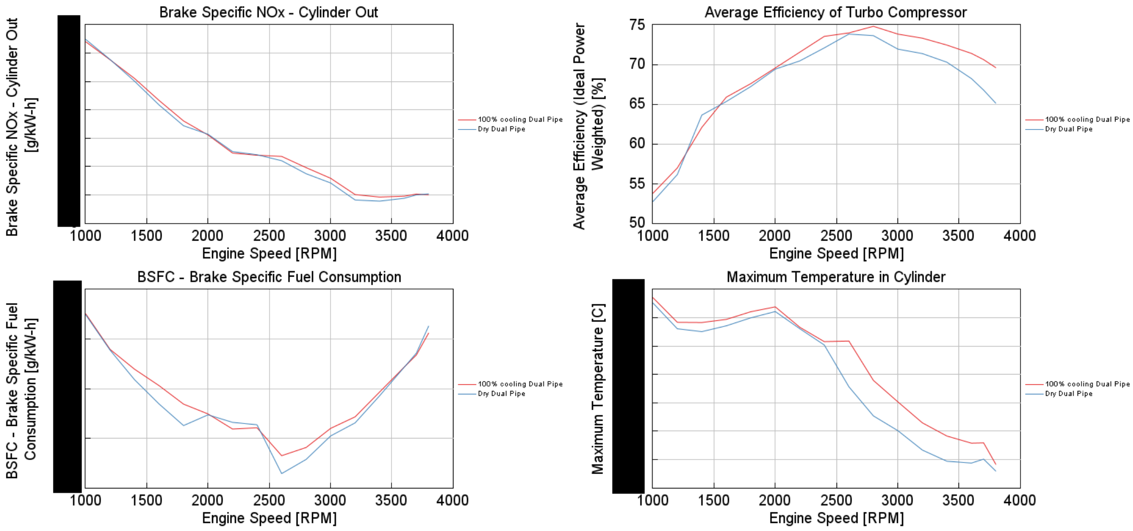
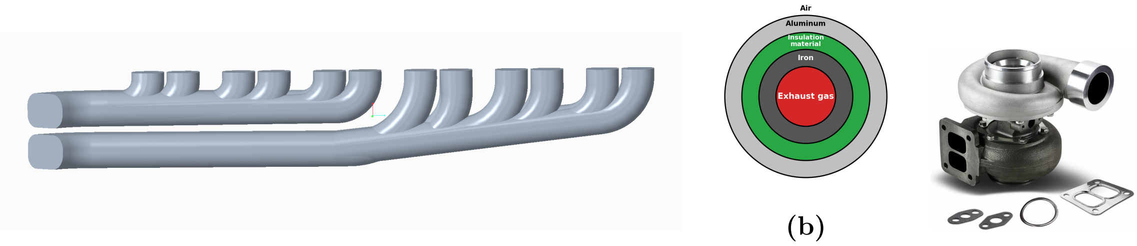

Brake Specific NOx - Cylinder Out
Brake Specific NOx - Cylinder Out [g/kW-h]
[redacted]
1000 1500 2000 2500 3000 3500 4000
Engine Speed [RPM]
100% cooling Dual Pipe
Dry Dual Pipe
Average Efficiency of Turbo Compressor
Average Efficiency (Ideal Power Weighted) [%]
50 55 60 65 70 75
1000 1500 2000 2500 3000 3500 4000
Engine Speed [RPM]
100% cooling Dual Pipe
Dry Dual Pipe
BSFC - Brake Specific Fuel Consumption
BSFC - Brake Specific Fuel Consumption [g/kW-h]
[redacted]
1000 1500 2000 2500 3000 3500 4000
Engine Speed [RPM]
100% cooling Dual Pipe
Dry Dual Pipe
Maximum Temperature in Cylinder
Maximum Temperature [C]
[redacted]
1000 1500 2000 2500 3000 3500 4000
Engine Speed [RPM]
100% cooling Dual Pipe
Dry Dual Pipe

Figure 6.20

6.3.6 Dual bank system with new turbo



(a) Bank-divided exhaust system **(b)** Cut-through **(c)** GT45

Figure 6.21: Simulated system

Summery:

- New turbo is a improvement even with same boundary conditions for cooling as original.
- The gain is more notable with insulated exhaust.
- Supplies more air after 2600 rpm.
- More power, lower BSFC.
- Abundance of air leads to lower temperatures both incylinder and in exhaust gas.

A new turbo was introduced and it was a Garret GT45. The new turbo was capable of operating at the increased mass flow and pressure that the less cooled exhaust provided. This was beneficial to engine performance and showcased the true potential of higher exhaust gas ethalpy.

With a new bigger turbo it can be seen that the engine can make more use of the mass flow and temperatures in exhaust manifold since the cylinder pressure goes up a lot for both the cooled and dry exhaust manifold. This results is a gain in power and a high lambda which indicates that there is a surplus of air during the combustion. BSFC is also a bit lower and this results in a lower NOx compared to the orginal engine even if we have a higher lambda.

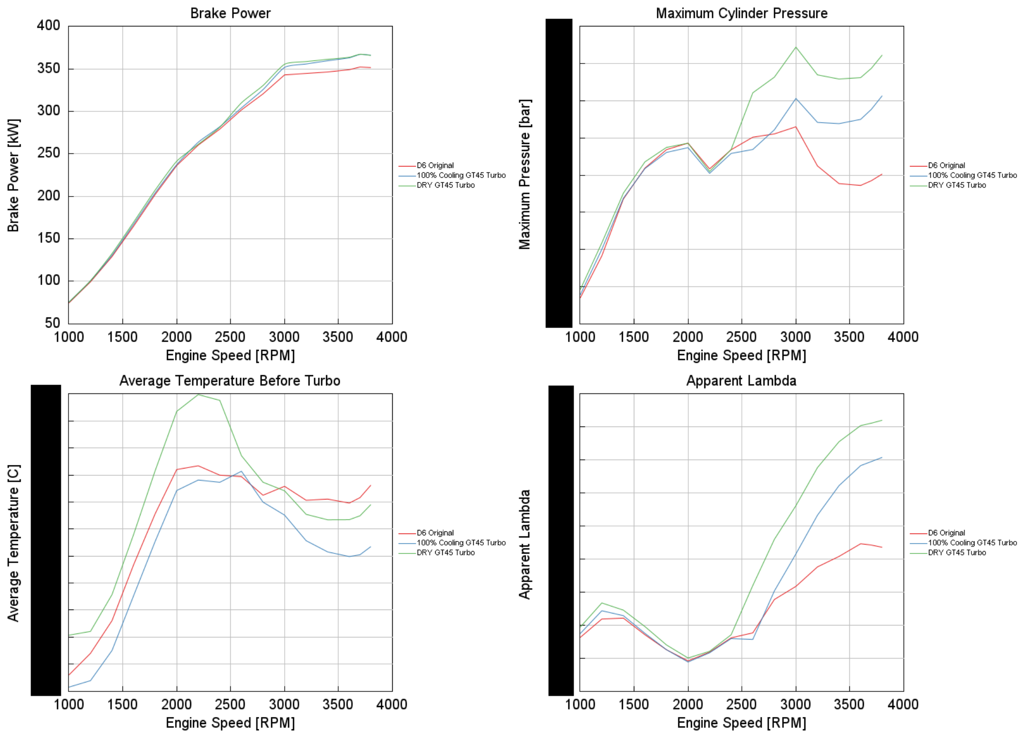


Figure 6.22

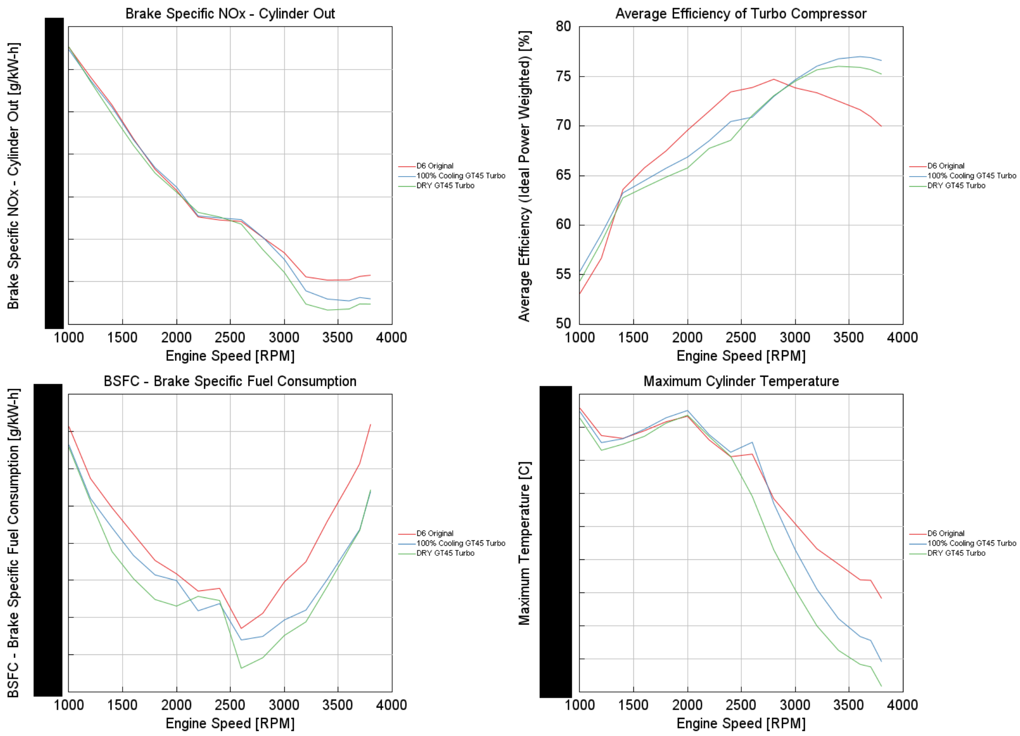
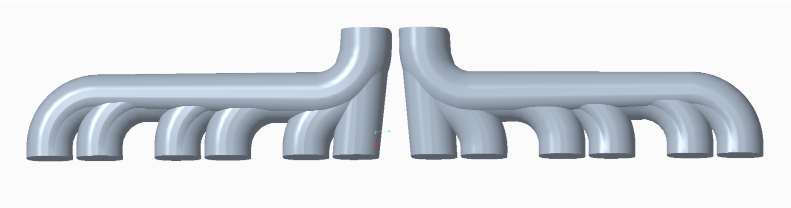


Figure 6.23

6.3.7 Mid mounted 2 bank system.



(a) CAD of mid-mounted turbo exhaust manifold

(b) GT45

Figure 6.24: Mid-mounted turbo manifold and GT45 turbocharger

In the graphs below are a comparison between the original exhaust manifold and turbo setup vs a a mid mounted with the new GT45 turbo. The mid mounted version is shown as both 100 % cooled and dry.

In theory having a shorter distance between the engine and turbo results in less losses. By having the Turbo mounted in the middle the length the exhaust pules have to travel can be limited.

A gain in peak power of around 4.5% can be seen for the cooled Mid mounted compared to the original. Turbo temperatures staying similar below 2750 rpm for the cooled variants and the dry mid mounted having a higher temperatures until around 2750 rpm. Where both cooled and dry mid mounted setups has lower temperatures before the turbo compared to the original.

Better BSFC, NOx values for both mid mounted version compared to the original and slightly lower turbo compressor efficiency in the mid range for the mid mounted setups and higher in the top range compared to the original.

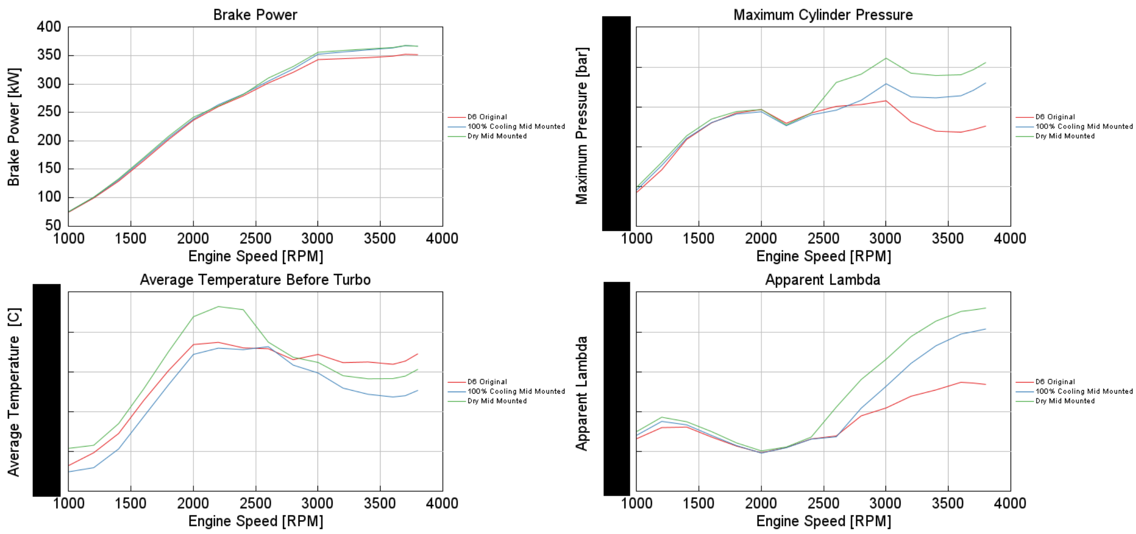


Figure 6.25

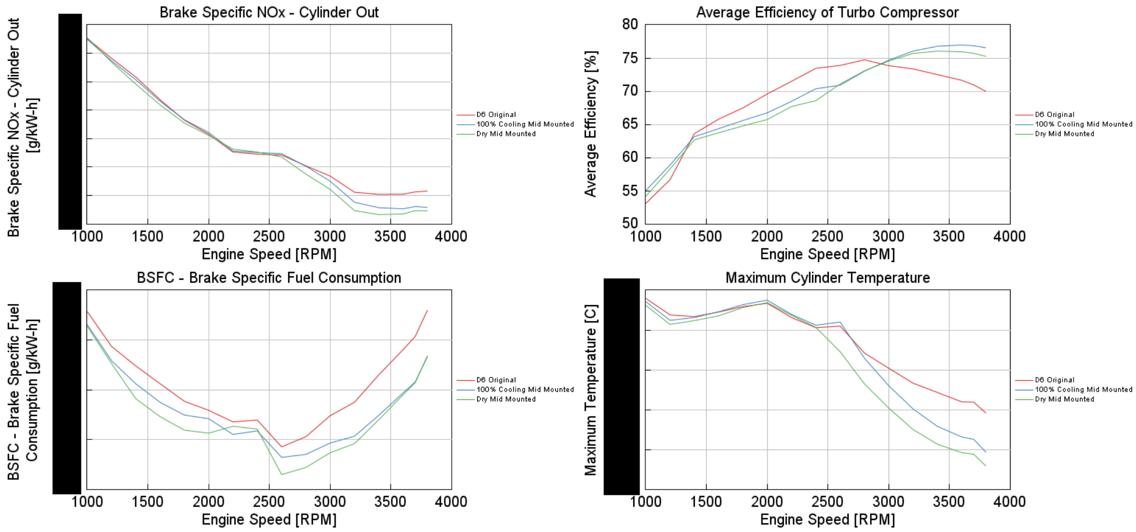


Figure 6.26

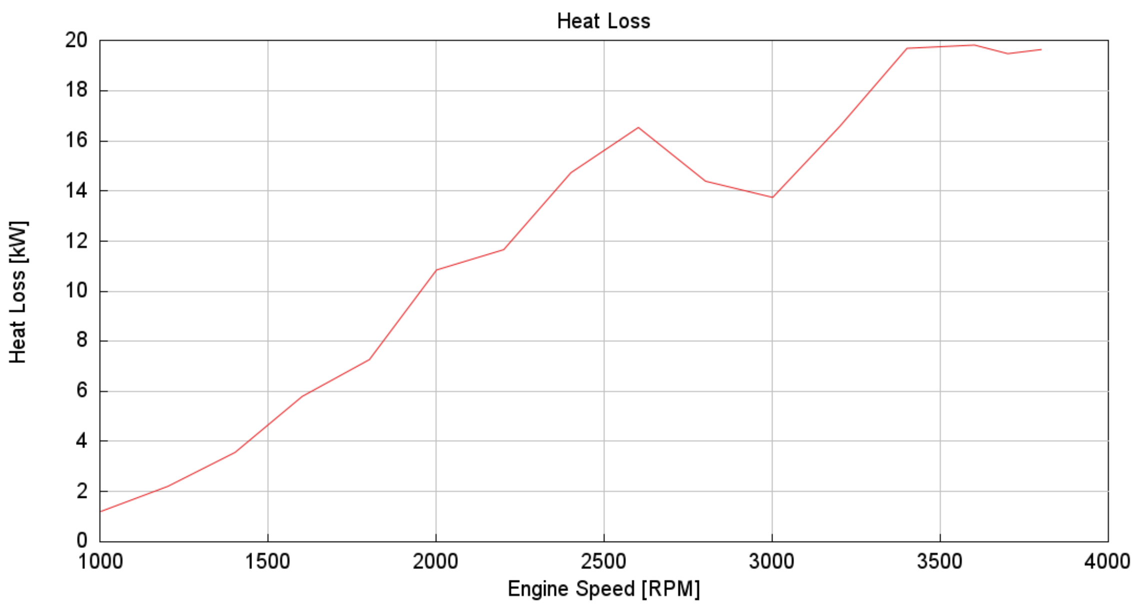
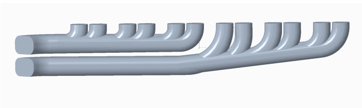


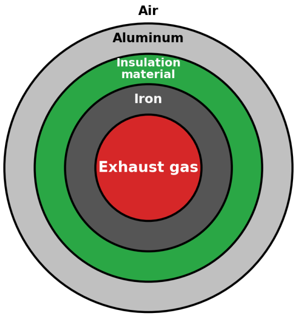
Figure 6.27

In the figure above we can see that the mid mounted setup as the maximum heat loss around 20 kW at around 3600 rpm. We can also see that it is increasing as the rpm increases.

6.4 Final model

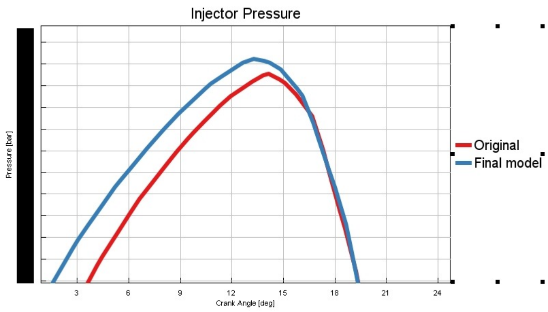


(a) Bank-divided exhaust system

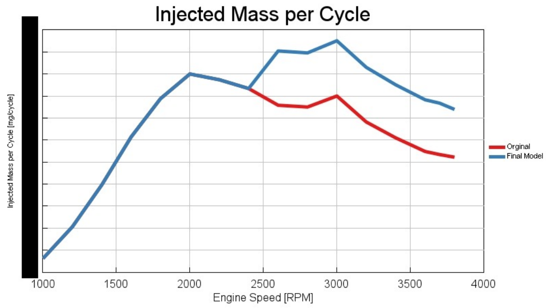


(b) Cut-through

(c) GT45



(d) Injection starts 2 CAD earlier and ends at the same CAD.



(e) 10 % more injected mass above 2600 rpm.

Figure 6.28: (a)–(e) Simulated system

(Two-bank manifold with the GT45 turbo, turbocharger repositioned to the rear (original position), plus engine tuning. Advanced injection timing and increased fuel at high rpm)

The final model is serves as a suggestion of what could be achieved with the new pulse divided, less cooled exhaust manifold, a GT45 turbo, more fuel after 2600 rpm and 2 degrees earlier injection timing after 2600 rpm.

Fuel quantity was raised by 10 % and start-of-injection advanced by 2°CA; rail pressure and end-of-injection remained at stock limits (late closing not permitted by Penta). The simulation delivered 550 hp while keeping brake-specific NOx at or below the unmodified level. Cylinder peak pressure increased by about 20 %, staying within the allowed +21 % margin, and injector peak pressure rose by 4.4 %. Temperatures in the exhaust are higher than the original att all times, however at high rpm, when lambda rises significantly, and the abundance of air lowers exhaust temperatures to equal of that of the original.

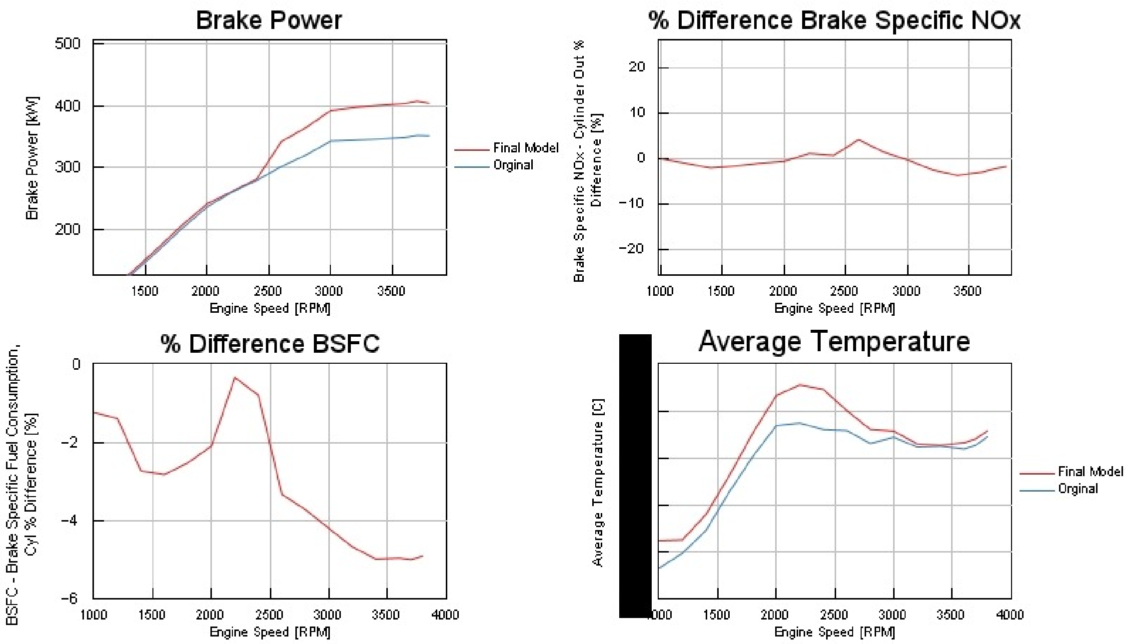

Brake Power
500
400
300
200
Brake Power [kW]
1500
2000
2500
3000
3500
Engine Speed [RPM]
Final Model
Orginal
% Difference Brake Specific NOx
Brake Specific NOx - Cylinder Out % Difference [%]
20
10
0
−10
−20
1000
1500
2000
2500
3000
3500
Engine Speed [RPM]
% Difference BSFC
BSFC - Brake Specific Fuel Consumption, Cyl % Difference [%]
0
−2
−4
−6
1000
1500
2000
2500
3000
3500
4000
Engine Speed [RPM]
Average Temperature
Average Temperature [C]
[redacted]
1500
2000
2500
3000
3500
4000
Engine Speed [RPM]
Final Model
Orginal

Figure 6.29

7

Discussion

7.1 Model credibility

Before interpreting any results, it is worth noting that the GT-Power model reproduced measured fullload points for brake power, torque, BMEP and BSFC with average absolute errors of only ≈1-2 %, giving an indication that the model is accurate for full load. NOx prediction remained less accurate (≈15 %).

The part load validation however showed that the errors increased as the operating point moved further away from the calibrated maximum torque curve. This raises concerns about the accuracy of the model, especially when values for flow, pressure, etc, differ significantly from those used during model calibration.

The exact reason for greater error at low loads is unclear . However, we theorize that it may be caused by hard coded parameters or scaling factors. To make the model more dynamic, scaling the heat transfer rates within the engine should be investigated and using pressure traces from actual in-cylinder pressure measurements should be implemented, as this approach would improve accuracy [20].

When simulating other turbocharger alternatives, the supplied maps were based on adiabatic conditions, and both sides of the scroll used the same map with a modeled leak between them. Since no leak data was available for the GT45 turbo, the D8 leak rate was used for all twinscroll simulations. In reality, twin-scroll turbochargers have different leak rates and different maps for each side, so using two separate maps and a turbo specific leak rate would improve accuracy. This could possibly increase gain of bank division.

7.2 What happens when cooling is reduced

Insulating the original manifold (no geometry change) raised turbine inlet temperature and boosted low- and mid-range power, but the original turbo quickly ran off the edge of its compressor map. Above 3000 rpm the engine lost efficiency and ultimately power because intake air became hotter and turbine back pressure climbed. The result answers the Research Question 1 (RQ1)- heat retention alone is not enough, turbo must be matched to the new energy level. This opened the door for a next question, what can be achieved in terms of performance with a better turbo and how should the engine be controlled for max benefit.

Due to the model uncertainties below max load, no investigation was done below these points. It can however be assumed that the gain of insulating the exhaust is even greater at lower loads. This due to the fact that the coolant pump output only varies with the speed of the engine, and is therefore dimensioned for max load, and therefore overcool at low load.

7.3 How should the engine be controlled

The implementation of a larger, more efficient turbo provides a performance gain on its own after 2600 rpm, but more importantly, it creates the conditions to supply more fuel as more air and therefore more oxygen molecules are available for combustion. This increase in fuel quantity slows the combustion process, making it then a suitable to advance the injection timing. Advancing timing allows more time for combustion and moves the main combustion phase closer to top dead center, which improves power output. This while keeping the end of the injection intact.

With the combination of increased air, more fuel and earlier timing, 550 horsepower can be achieved. However, each of these factors also contributes to higher cylinder pressure which will increase stains on the engine, and needs further investigation. A ten percent increase in fuel and a two crank angle degree advance in timing after 2600 rpm were found to deliver 550 horsepower while keeping peak cylinder pressure within the limits defined by Penta for a future engine, around 20% difference from the original.

Looking ahead, further optimization of this engine with the addition of a wastegate controller(which is currently not implemented) will give calibration engineers three variables to manage: air, fuel, and injection timing. These can be adjusted to balance performance while maintaining peak pressure and emissions within acceptable limits for optimal efficiency.

It should be noted that the increased airflow from a larger turbo allows more air to absorb combustion heat, which results in lower exhaust gas temperatures at high rpm. While this is not an issue when performance is the focus, it may pose a challenge if a selective catalytic reduction system is implemented, as it relies on higher exhaust temperatures. This can be addressed by using waste-gating to reduce the air supply when higher exhaust temperatures are needed for the SCR system.

The fuel delivery system needs to able to deliver 10% more fuel, which will increase peak pressure in the injector by 4.4%. This might degrade the injector faster, and should be considered.

7.4 Manifold Geometry

The new dual bank rear mounted exhaust manifold had a 17% bigger surface area compared to the original. This means that there will be a 17 % increase in surface that can transfer heat and needs to be isolated. Even when this was the case we saw improvements in performance with the new exhaust manifold.

This shows that separating the pulses to avoid interference from each other show improvements even if the cooling loses increased. This aspect could be further enhanced by separating each cylinder and attempting to make each pulse patch to the turbo equal. However Flow splits are one volume with an average in the middle which isn't great to fully capture the pulse behavior. To be able to study the behavior

Mid mounted turbo exhaust manifold had a 13% smaller surface area than the original. Which results in a 13% smaller area that can transfer heat. It also had the shortest path for the exhaust pulses and the same type of exhaust pulses dividing as the rear mounted dual bank. The biggest drawback of the mid mounted was the packaging which would make the engine very wide and this wasn't desired or really possible in the engines current use.

The exhaust manifold got its diameters and cross section areas by sweeping from the exhaust ports of the engine to the turbo inlets as previously mentioned the methodology chapter. The idea behind this was to allow the exhaust gases to flow freely without having too much restriction or freedom to expand. However this was not tested in depth and only quickly designed to have a CAD model for GEM3D. This would need to be studied further and explore how different diameters and cross sectional areas affects pressure and flow.

Having a larger surface area gives more area that can transfer heat and more area that needs to be insulated. Ideally you would want a small as possible surface area for performance to be able to retain as much thermal energy as possible and deliver it to the turbo. With increasing the temperatures it was seen that the maxiumum exhaust temperatures before the turbo went from around 600 °C to 680 °C. This puts a additonal requirement on the material. The old exhaust manifold was alumnium and with this increase in temperatures, the new manifold would have to be something that can handle more temperatures since alumnium would be close or over its melting point to avoid failure and would likely have to be cast iron. This answer research question 2 in terms of geometry and manifold material.

7.5 Suggestions for insulation materials

The simple calulation on a cast iron-insulation material-aluminum cover 5mm-25mm-5mm in the methods chapter indicates that the insulation properties of the material to insulate the exhaust needs to be somewhere around 0.034 W/m*K.

Following list are alternatives.Most are mats that could be stapled in layers and secured around the manifold with wires or staps.

#	Material	k @ 400 °C ($\mathrm{W \cdot m^{-1} K^{-1}}$)	Source
1	Siltherm **AluFlex 1000** — bendable panel 5–10 mm	0.023	[34]
2	Microtherm **Thin Sheet Super G** — rollable sheet 3 mm	0.027	[35]
3	Unifrax **Excelfrax 1800 Flexliner** — stitched wrap 5–10 mm	0.026	[36]
4	Siltherm **Quilt** — quilted blanket 10 mm	0.026	[37]
5	Morgan **WDS MultiFlex Plus** — flexible blanket 5–10 mm	0.026	[38]

Table 7.1: Thermal conductivities of flexible 400 °C insulations

We also calculated that the heat transfer through conduction could reach around 40–70 kW if the mating surfaces are assumed to have zero thickness. However, this is not realistic. A more accurate estimate of about 8 kW was obtained by including assumed material thicknesses and a graphite gasket.

This shows that conduction can contribute significantly to heat transfer, and strategies to reduce it should be considered in manifold design. For example, using a thicker or less thermally conductive gasket could help limit unwanted heat transfer.

8
Conclusions

		Peak power	Mean BSFC	Heat-loss	Max avg temp before turbo	Max cylinder pressure	Max Lambda
Water cooled	Original	352.3 kW	*	46.9 kW	*	*	*
Water cooled	Midmounted dual bank	+4.5%	-2.3%	-57.8%	-3.6%	+4.4%	+14.9%
Water cooled	Rear-mounted dual bank	+2.0%	-1.0%	-3.2%	-2.8%	+0.1%	+0.1%
W/o water cooling	Original	-1.7%	+0.5%	-77.4%	+12.7%	+12.3%	+8.0%
W/o water cooling	Rear-mounted + new turbo	+4.3%	-2.7%	-85.7%	-2.9%	+10.2%	+21.0%
W/o water cooling	Rear-mounted + new turbo + more fuel + earlier timing	+15.8%	-3.1%	-86.2%	+10.1%	+22.0%	+14.1%

* Confidential

Figure 8.1: The most important models and results.

Among the manifold designs evaluated, a mid-mounted turbo configuration delivered the highest performance due to shorter exhaust paths and minimal flow losses, but it proved impractical because it interfered with existing engine components and widened the engine profile, which will be problematic for customers upgrading their engine . In contrast, a dual-piped rear-mounted manifold emerged as the most viable design, substantially outperforming the original manifold by separating exhaust pulses and reducing pulse interference, thereby improving breathing. Although this dual-pipe setup has a greater surface area (potentially increasing heat losses), it fits within the current engine envelope and only needs space for a larger turbocharger.

Engine tuning adjustments provided insights. Retarding the injection timing in the original engine improved performance and lowered fuel consumption, but at the cost of higher peak cylinder pressures and increased NOx emissions, where as dumping turbo pressure by wastegating effectively raised exhaust gas temperatures (reducing risk for aftertreatment) but reduced overall power. Reducing or eliminating water-cooling in the manifold significantly increased exhaust gas enthalpy, which improved turbocharger performance at low-mid range in the stock configuration of the engine. However, under these hotter conditions the stock turbo quickly exceeded its efficient operating range at high rpm, resulting in lower top end performance. **Only reducing cooling and keeping the stock Turbocharger makes the engine preform worse**.

This limitation was resolved by upgrading to a larger turbocharger optimized for the higher exhaust energy, which delivered greater airflow above 2600 rpm and enabled additional fueling with earlier injection. This was enough to reach the targeted 550hp without increasing brake-specific NOx emissions over the baseline. Achieving this higher output did demand substantially increased peak cylinder and fuel injection pressures, associated with potential durability and reliability concerns.

With the manifold running hotter, thermal insulation becomes critical to keep external surface temperatures within safe limits and to minimize unwanted heat loss. This needs careful selection of insulation materials and thickness, and suggestions for those were presented in the discussion chapter.

Importantly, while a higher lambda from larger turbo lowers exhaust gas temperature, insulation increases it. The resulting exhaust gas temperature is slightly higher than the original, but because lambda can be reduced on demand via control of the wastegate, we can still raise temperature when needed. Offering the more precise thermal management required for integration of SCR systems.

8.1 Recommendations for Future Work

Future work should start with more detailed FEM analyses to check how the new manifolds handle higher thermal and mechanical loads. Temperatures and pressures from the 1-D model can serve as boundary conditions in a 3D CFD study of the manifold to locate hot spots and estimate heat loss more accurately. Finally, building and testing a prototype on the engine is needed to confirm the simulations.

It would also require specific calculations on the mounting bolts, washer and eventually expansion cylinders on the bolts to handle the exhaust manifolds change depending on the temperature of the material and the G-forces the exhaust manifold will experience from its own and the turbos weight during harsh sea conditions.

The insulating materials that is showed and recommended would need a lot of testing on their own to see how they handle the conditions the engine operate in and how they behave over time. It was also be required to make sure they live up to everything the manufacturer says they do.

Overall, this thesis shows that modifying exhaust manifold geometry, reducing cooling, and adjusting engine tuning strategies can significantly improve performance. However, real-world practicality, material choices, and reliability remain critical aspects that must be addressed before applying these changes in actual engine designs.

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