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Modelling and simulation of a two-stroke engine

Master of Science Thesis

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Abstract

Engine modelling is an important task in today's engine industry since simulation is a powerful tool in order to optimize engine functionality. The two-stroke engine has not been modeled to the same extent as the four-stroke engine even though it is widely used in for example large ships, pumping stations and power plants. This report documents the first steps in the process of developing a generalized two-stroke engine model in Simulink. Focus is on defining physical and mathematical relationships which describe the engine's dynamics in order to enable the making of a potent Simulink model and also to lay the foundation for future development.

The project have resulted in relevant models for the different engine parts, but there is still work to do before being able to run one big model consisting of all parts connected together.

Preface

This report wishes to describe the work done by me during my master thesis at Hørbiger Control Systems AB, striving to achieve a master's degree in Systems, Control and Mechatronics at the institution of Signals and Systems at Chalmers University of Technology. I am grateful for the help and guidance I have received from my supervisor Ingemar Andersson throughout the project. Thank you!

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1 Introduction

Engine modelling is an important tool for engine manufacturers in their quest towards optimizing engine performance since it offers efficient testing and simulation possibilities. The existence of good testing procedures is a vital part in all engine development, particularly when designing engine control systems to cope with tightening restrictions on emissions and fuel consumption.

When testing, models have substantial advantages compared to real, full-scale prototypes, especially when it comes to availability and minimizing costs. A model has the drawback of decreased accuracy but depending on application demands this is not always a problem.

Four-stroke gasoline engines are by far the most common type of engine regarding produced quantity as well as modelling effort, the latter which has been an item in the car industry for decades. This gives the existing four-stroke engine modelling knowledge an advantage compared to that of the two-stroke engine. Not said that there is no modelling done on two stroke engines, but during project pre-studies no complete model with the characteristics needed was found.

What is needed is a model implementing only the most vital dynamics in order to be easily implemented in a bigger environment where simulation speed is more important than accuracy. This project will use the fact that a lot of the knowledge from the four-stroke engine can be applied on the two-stroke engine.

1.1 Purpose

In the process of making engine control systems, there is a great need of having good validation possibilities. It is also important that the validation process is time and money efficient in order to keep up with deadlines and customer demands. A generalized model of the engine is an asset and necessary tool to be able to prove one's theories and models in a reasonable way. At first hand this kind of models are useful when testing control strategies and general behaviour, rather than when calibrating control systems. A model that is made for calibration purposes has to be a lot more accurate and precise in order to provide useful information and take today's tight emission restrictions into consideration.

1.2 Specification

The engine type modeled in this project is a two-stroke large-bore natural gas lean burn engine with turbocharger. The model consists of a number of different sub-parts where each part has specific features being counterparts to the dynamics found in a real engine. Modeled phenomena are air-fuel ratio in combustion, turbo charge and engine torque. The sub-parts are, in order of flow direction, the compressor, the intake manifold, the combustion chamber, the exhaust manifold and the turbine. Beyond these blocks there is also a turbo shaft block, connecting the turbine and the compressor, and an external load block to simulate external load. The parts shall be possible to put together forming a bigger system that renders the most vital dynamics of a real engine and provides amount of fuel, waste-gate, turbo jet-assist and external load as control signals.

1.3 Limitations

Emphasis of the modelling will be on taking care of the most important dynamics and not so much on making a model as perfect as possible. The project will not consider development of control strategies in order to optimize the engine, but strictly consider development of a Simulink model with sufficient features and dynamics to meet the project specifications.

1.4 Method

The first step is to look at what is already done in the field and see what material that might be useful in closely related fields. Then border how deep into the engine dynamics to investigate whereupon the compilation of mathematical and physical relationships can begin. When having enough information on an engine part, those relationships makes the foundation for implementing the model in Simulink. When a sub-model is implemented, its parameters needs to be adjusted in order to better fit its real counterpart and to eventually work together with the other sub-parts as one big model.

The validation method when simulating the blocks one at a time is very much based on the common sense of the simulation operator who needs to predict the expected behaviour of the simulation in every separate case. This has to be the method until real engine simulation data can be accessible and used as validation. For more information on simulation read chapter 3.

2 Engine modeling

This chapter will focus more on the derivation of the relationships which are the foundation of Simulink implementation than covering the actual making of the Simulink blocks. Every sub-chapter from the compressor and on will more or less have the same structure, where the aim always is to present each block's output signals in terms of the input signals and parameters specified.

2.1 Theory

Large bore two-stroke engines are not suited for vehicle propulsion but are most commonly found in large boats, pumping stations or in powerplants, generating electricity. Engines used in ship environments are usually diesel fueled ([4] p.883) and natural gas fueled engines are rather found in the other two before mentioned applications. The fuel type distinction is not vital for this project since the fuel is given as an external parameter but this context information could still be interesting for the reader.

In many senses the four- and two-stroke engines are similar in their constitution. There are a multitude of different types within both families, but in main structure the most differing engine sub part is the cylinder house, the combustion chamber. Another feature that is important to consider is the high inertia in large bore engines due to larger mechanical components, for instance larger turbine wheel. One of the control signals, the jet-assist, is needed in order to get the turbo going at start-up since the exhaust gases themselves will not initially have enough energy to overcome the turbine inertia.

There are different types of combustion chamber design and also different ways of modelling the mixing of gases in the two-stroke engine. The design used in this project is called cross-scavenging ([4] p.235) which uses inlet and exhaust ports in the cylinder walls. The ports are uncovered and open when the cylinder reaches its bottom position and closes as the piston moves upwards. The mixing model resembles the isothermal three-zone model of Maekawa ([5] p.127) which separates gases in the combustion chamber into three zones; fresh gases on top, gas from previous strokes in the middle zone and short-circuiting gas that never engage in combustion running directly from intake to exhaust manifold.

For more in-depth information on the two-stroke engine, good sources are [1] and [5].

2.2 Simulink

Simulink is a Matlab application which provides a way to implement differential equations in an intuitive way, where blocks are used to represent everything from simple additions to integration of time derivative variables. Pre-defined blocks can be used just as well as self-defined functions where bigger block systems preferably are put together in bigger blocks as sub-systems.

2.3 Overview

In order to get a manageable Simulink model, different parts of the engine are separated into smaller pieces referred to as blocks and sub-blocks. This structure is adopted in the report as well, since it provides a well separated and intuitive way of describing the engine. Figure 2.1 gives a view of what the assembled engine model finally might look like. The assumed flow direction starts in the compressor and continues through the intake manifold, the combustion chamber, the exhaust manifold and finally the turbine.

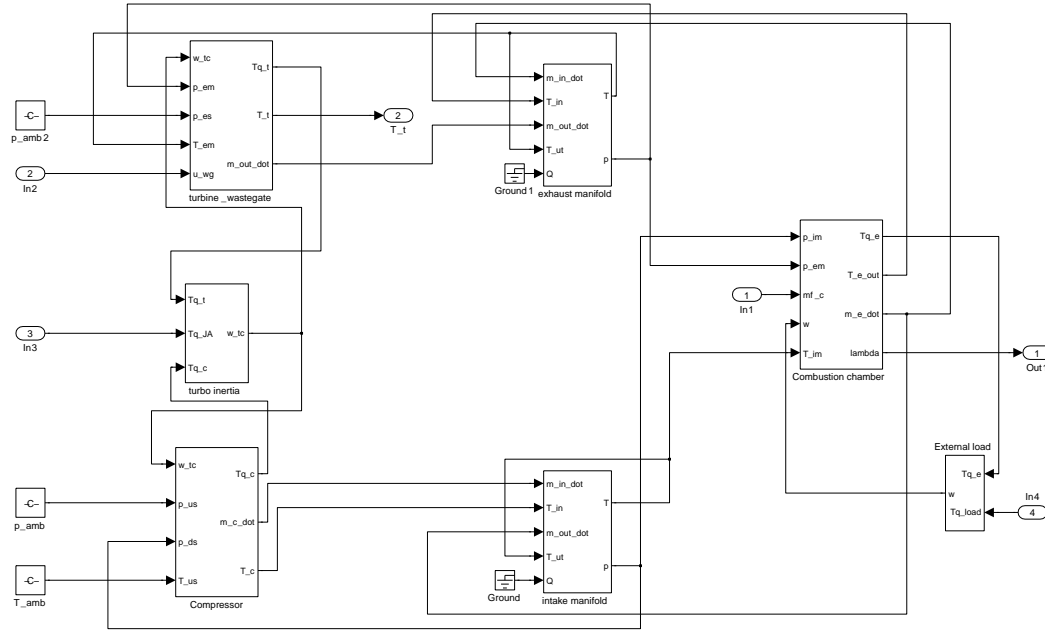


Figure 2.1 – Simulink system overview

2.4 Compressor

Following the flowchart of the engine, the first encounter is the Compressor. There exist different compressor structures but its basic element is a propeller, driven by the turbo shaft connected to the turbine. The compressor consumes the torque generated by the turbine by showing air from outside the engine into the intake manifold where pressure builds up in order to satisfy the combustion with enough air. The derivation of the compressor dynamics is to a large extent influenced by Lars Eriksson’s MVEM library ([2] p.8).

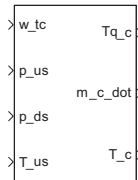


Figure 2.2 – Inputs and outputs of compressor Simulink block

In the compressor model, there are four input signals and three output signals, see figure 2.2 and table 2.1.

Table 2.1 – Input signals, output signals and parameters of compressor

INPUTS		
ω_{tc}	rad/s	Turbo shaft rotational speed
p_{us}	Pa	Pressure upstream: ambient pressure
p_{ds}	Pa	Pressure downstream: intake manifold pressure
T_{us}	K	Temperature upstream: ambient temperature
OUTPUTS		
Tq_c	Nm	Compressor load on turbo shaft
\dot{m}_c	kg/s	Air mass-flow out of compressor
T_c	K	Compressor out-air temperature
PARAMETERS		
c_p	J/(kg·K)	Air specific heat
η_c	—	Compressor efficiency

The compressor torque is the torque that is holding the turbine back, or in other words, the torque that the turbine has to overcome in order to run in forward direction. It can be described as a load on the turbo shaft. Using a slightly indirect way, this load can be found through the compressor power which itself is originally derived from an expression for compressor efficiency.

$$P_c = \frac{\dot{m}_c c_p T_{us}}{\eta_c} \left(\left(\frac{p_{ds}}{p_{us}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \quad (2.1)$$

The compressor power is by definition positive, meaning that this expression does not take into consideration net flows in negative direction. Having the compressor power, the torque is found by using its relationship with the turbo shaft rotational speed.

$$Tq_c = \frac{P_c}{\omega_{tc}} = \frac{\dot{m}_c c_p T_{us}}{\eta_c \omega_{tc}} \left(\left(\frac{p_{ds}}{p_{us}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \quad (2.2)$$

Just as in the case with the flow, the rotational speed is not allowed to be negative and it can furthermore not be equal to zero. The resulting temperature of the air flowing out of the compressor is a function of ambient temperature and depends on the pressure difference over the compressor.

$$T_c = T_{us} \left(1 + \frac{\left(\frac{p_{ds}}{p_{us}} \right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta_c} \right) \quad (2.3)$$

The massflow through the compressor is a function of pressure difference and turbo shaft rotational speed. The compressor will shove air into the intake manifold proportional to how fast the turbo shaft rotates depending on how much torque the turbine provides. That is the source to forward flow in the compressor. But there is also a backwards flow that appears proportional to the pressure difference over the compressor. A pressure that builds up because of the forward flow. The total flow is modelled as downstream flow subtracted by upstream flow where $k_1 > 0$ and $k_2 > 0$ are proportionality constants.

$$\dot{m}_c = \dot{m}_f - \dot{m}_b = k_1 \omega_{tc} - k_2 \left(\frac{p_{ds}}{p_{us}} - 1 \right) \quad (2.4)$$

For the flow not to become negative

$$\omega_{tc} > \frac{k_2}{k_1} \left(\frac{p_{ds}}{p_{us}} - 1 \right) \quad (2.5)$$

One thing that might need to be reconsidered about the compressor dynamics is the massflow impact on the torque calculation. If the pressure quotient is big enough making the total flow equal zero, the torque will also equal zero. An investigation of the difference when only forward massflow is considered in the torque calculation would be relevant.

2.5 Intake manifold

The intake manifold is the cavity between the compressor and the cylinder where pressure builds up due to compressor activity. This component is important from a modeling point of view since it provides both the block upstream (compressor) and the block downstream (combustion chamber) with inputs. The determining equations are derived from the laws of mass- and energy conservation and are inspired by Eriksson's MVEM library ([2] p.5). The intake and exhaust manifold share constitution and are modeled identically with the exception of a few differing parameter values. They are passive engine parts since what they receive and give away is solely controlled by other engine parts and they are thus sometimes called receivers.

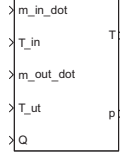


Figure 2.3 – Inputs and outputs of intake manifold Simulink block

In the intake manifold model, there are five input signals and two output signals, see figure 2.3 and table 2.2.

Table 2.2 – Input signals, output signals and parameters of intake manifold

INPUTS		
\dot{m}_{in}	kg/s	Air mass-flow from compressor
\dot{m}_{out}	kg/s	Air mass-flow towards combustion chamber
T_{in}	K	Temperature of compressor air
T_{out}	K	Temperature in intake manifold (same as output T)
Q	J/s	Heat-flow into the system (optional)
OUTPUTS		
T	K	Temperature in intake manifold
p	Pa	Pressure in intake manifold, also p_{im}
PARAMETERS		
κ_{im}	—	Ratio of specific heats
V_{im}	m ³	Volume of intake manifold
$T_{init,im}$	K	Initial temperature of intake manifold
$p_{init,im}$	Pa	Initial pressure of intake manifold
R	J/(kg·K)	Gas constant

The energy balance in the intake manifold is defined as

$$W_{im} = m_{im} c_v T_{im} \quad (2.6)$$

and its time derivative is

$$\dot{W}_{im} = \dot{m}_{im} c_v T_{im} + m_{im} c_v \dot{T}_{im} \quad \text{alt.} \quad \dot{W}_{im} = \dot{H}_{in} - \dot{H}_{out} + \dot{Q}_{in} \quad (2.7)$$

where $\dot{H} = \dot{m} c_p T$, ([2] p.5), represents enthalpy flow. The left equation in (2.7) expresses the energy flow in terms of intake manifold mass, temperature, mass flow and temperature flow. Since the objective is to find an expression for intake manifold temperature it is troublesome if there is nothing else to work with than the intake manifold temperature. It is a causality dilemma. By expressing the energy flow through the right equation in (2.7) it is possible to also use the intake manifold inputs, which are known entities. By using both equations in (2.7), equalling the right hand sides of both equalities, it is possible to solve an expression for the intake manifold temperature time derivative as a function of either already known signals or signals that can be feedbacked with an intital value to avoid non-causality.

$$\begin{aligned} \dot{T}_{im} &= \frac{1}{m_{im} c_v} \left(\dot{H}_{in} - \dot{H}_{out} - \dot{m}_{im} c_v T_{im} + \dot{Q}_{in} \right) \\ &= \frac{1}{m_{im}} \left(\kappa_{im} \dot{m}_{in} \dot{T}_{in} - \kappa_{im} \dot{m}_{out} \dot{T}_{out} - \dot{m}_{im} T_{im} + \frac{\dot{Q}_{in}}{c_v} \right) \end{aligned} \quad (2.8)$$

The external heat-flow into the system is represented by \dot{Q}_{in} and is sometimes ignored. Passing the signal through a Simulink integrator block and defining an initial condition means that T_{out} can be known from \dot{T}_{im} , because $T_{im} = T_{out}$. The only unknown parameter in equation (2.8) is the intake manifold mass m_{im} and its derivative \dot{m}_{im} . The air-mass in the intake manifold is decided upon the amount of input and output air, and the time derivative of the intake manifold mass is

$$\dot{m}_{im} = \dot{m}_{in} - \dot{m}_{out} \quad (2.9)$$

Integrating this signal will give m_{im} and the ideal gas law gives the initial condition for the integration.

$$m_{init} = \frac{p_{init} V_{im}}{R T_{init}} \quad (2.10)$$

When knowing the intake manifold temperature there is no problem finding the intake manifold pressure which is the other output signal from this block. All that is needed is an implementation of the ideal gas law.

$$p_{im} = \left(\frac{m_{im} R T_{im}}{V_{im}} \right) \quad (2.11)$$

2.6 Combustion chamber

This is the core of the model and, in the end, the purpose of all the other blocks. The combustion chamber is a complex modeling object with the main task of providing torque to the driveline shaft. Torque generation through combustion takes fuel and gives high temperature exhaust gases. An interesting output signal in an engine control perspective is the λ -value which is the quotient between the amount of air and the amount of fuel in the combustion. To describe the dynamics in the combustion process of a two-stroke engine it is also required to study the principles of scavenging which is one of the biggest differences between two-stroke and four-stroke engines.

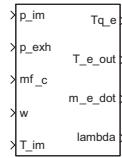


Figure 2.4 – Inputs and outputs of combustion chamber Simulink block

The combustion chamber Simulink block consists of five inputs as well as five outputs, see figure 2.4 and table 2.3.

Table 2.3 – Input signals, output signals and parameters of combustion chamber

INPUTS		
p_{im}	Pa	Pressure in intake manifold
p_{exh}	Pa	Pressure in exhaust manifold
mf_c	kg/cycle	Injected fuel per cycle
ω	rad/s	Resulting engine rotational speed
T_{im}	K	Temperature in intake manifold
OUTPUTS		
Tq_e	Nm	Engine produced torque
$T_{e,out}$	K	combustion chamber out temperature
\dot{m}_e	kg/s	Air mass-flow through combustion chamber
λ	—	Air/fuel-ratio in combustion
PARAMETERS		
η_{ig}	—	Indicated gross efficiency
q_{LHV}	J/m ³	Fuel, lower heating value
V_{cyl}	m ³	Maximum cylinder volume
V_t	m ³	Cylinder volume at top position
n	—	indicating 2- or 4-stroke engine
N	—	indicating how many cylinders in engine

Torque

When calculating the engine torque, consumed work is subtracted from produced work ([3] p.84) as

$$Tq_e = \frac{W_{ig} - W_f}{n 2\pi} \quad (2.12)$$

The input signals that have immediate impact on the engine torque are the engine rotational speed, ω , and the injected fuel per cycle, mf_c . The produced work should be seen as the energy from the delivered fuel and can be defined

$$W_{ig} = mf q_{HLV} \eta_{ig} \quad (2.13)$$

Since mf_c is delivered fuel per cycle it is multiplied by $\omega \frac{1}{2\pi} N$ to receive delivered fuel per time unit.

$$mf = mf_c \omega \frac{1}{2\pi} N \quad (2.14)$$

The gross indicated efficiency, η_{ig} is a factor combining many different kinds of losses ([3] p.85) but it is modeled as a constant, user set value $0 \leq \eta_{ig} \leq 1$. The consumed work, or the load, is due to friction in the engine and is related to the engine speed as

$$W_f = k_1 \omega + k_2 \omega^2 \quad (2.15)$$

where k_1, k_2 are proportionality constants. The indicated gross work in equation (2.13) drives the engine external load. The engine rotational speed depends on how big the inertia of the external load is.

Scavenging

The scavenging dynamics of the two-stroke engine decides the relationship between fresh and old gas in the combustion chamber and is thereby a measurement of how well-circulated the air is in the cylinder house. In the cylinder volume there can be assumed to be three different types of gas; fresh air from the intake manifold, trapped air/fuel mixture (residual gas) from previous strokes and ‘short circuit’ air that runs directly from the intake manifold to the exhaust manifold without ever engaging in the combustion. Speaking of scavenging, the two most vital relationships to discuss are the scavenging ratio SR and the scavenging efficiency SE , defined as

$$SR = \frac{\text{massflow through engine } (= \dot{m}_e)}{\text{trapped massflow } (= \dot{m}_{cyl})}, SR > 0 \quad (2.16)$$

$$SE = \frac{\text{trapped mass/cycle}}{\text{theoretical mass/cycle}}, SE < 1 \quad (2.17)$$

The value of SE tells how much of the cylinder air that is exchanged in every cycle. A value of 1 equals the situation where all air from the previous stroke is being replaced by new air from the intake manifold. Scavenging ratio and scavenging efficiency are related to eachother as

$$SE = 1 - e^{-(SR k_{SE})} \quad (2.18)$$

where k_{SE} is a constant design parameter. The scavenging only considers the mixture in the cylinder and not the total air flow. The air flow through the combustion chamber is a result of pressure difference between the intake manifold and the exhaust manifold and must be known in order to calculate SR . Here, it is modeled as

$$\dot{m}_e = k_e \left(\frac{p_{im}}{p_{exh}} - 1 \right) \quad (2.19)$$

which is an equation describing mass flow per time unit where k_e is a proportionality constant. The theoretical mass/cycle is the air mass contained in the combustion chamber at the temperature and pressure conditions of the intake manifold.

$$m_{theoretical} = \frac{p_{im} V_{cyl}}{RT_{im}} \quad (2.20)$$

The trapped massflow in the combustion chamber depends on the theoretical mass/cycle, the engine rotational speed, ω , and the number of cylinders, N .

$$\dot{m}_{cyl} = m_{theoretical} \omega \frac{1}{2\pi} N \quad (2.21)$$

Since ω is measured in rad/s, it has to be multiplied by $\frac{1}{2\pi}$ to be converted to cycles/s. Everything needed for an implementation of equation (2.18) is thereby obtained and SR and SE are defined in known entities.

Left to describe are the three different fractions of gas in the combustion chamber. First, the fresh air into the chamber is defined as the air that is not either passing straight through (short-circuit) or is retained in the chamber between the combustion cycles. The relationship defining the quotient between fresh and retained gas is the scavenging efficiency why the airflow into the combustion chamber can be written

$$\dot{m}_{a,im} = \dot{m}_{cyl} SE \quad (2.22)$$

Since $\dot{m}_{a,im}$ is the fresh air massflow into the combustion chamber from the intake manifold at each cycle, there has to be an equally large flow of burnt gas out from the combustion chamber into the exhaust manifold,

$$\dot{m}_{ch,exh} = \dot{m}_{a,im} \quad (2.23)$$

Finally, there is a massflow not taking part in the combustion at all, short-circuiting through the combustion chamber directly into the exhaust manifold. This flow is found by subtracting the total engine massflow by the fresh gas into the cylinder.

$$\dot{m}_{shc} = \dot{m}_e - \dot{m}_{a,im} \quad (2.24)$$

A good source for finding more detailed material on scavenging modeling is ([5] Ch.4).

Lambda

The most commonly used reference term for mixtures in internal combustion engines is the air to fuel ratio (AFR) which is a measure of how the amount of air and fuel are related to each other during combustion.

$$AFR = \frac{m_a}{m_f} \quad (2.25)$$

When air and fuel appear in quantities combining all the fuel with all the free oxygen, the AFR represents chemical equilibrium and is defined as the stoichiometric AFR, AFR_{stoich} or AFR_s . The stoichiometric ratio is depending on the fuel type and for natural gas, $AFR_s = 14.5$ ([4] app.D). This condition is the reference point when defining λ since λ describes how far from stoichiometric conditions the air-fuel mixture is.

$$\lambda = \frac{AFR}{AFR_s} \quad (2.26)$$

At stoichiometric mixture in the combustion chamber the lambda value equals 1, when there is excess air (lean mixture) the lambda value is greater than 1 and when there is excess fuel (rich mixture) the lambda value is less than 1.

Assume that, for each combustion cycle, the total mass in the combustion chamber

is m_a , the mass of fresh air from the intake manifold is $m_{a,im}$ and the mass of air in the residual gas from previous strokes is $m_{a,r}$.

$$\lambda = \frac{m_{a,im} + m_{a,r}}{m_f AFR_s} \quad (2.27)$$

The scavenging efficiency defines the relationship between retained and fresh gas and can be used to rewrite the air mass of the retained gas as $m_{a,r} = (m_a - AFR_s m_f) (1 - SE)$. Giving

$$\begin{aligned} \lambda &= \frac{m_{a,im} + (m_a - AFR_s m_f) (1 - SE)}{m_f AFR_s} = \\ &= \frac{m_{a,im} + \left(\frac{m_a}{AFR_s m_f} - 1\right) (1 - SE) AFR_s m_f}{AFR_s m_f} = \\ &= \frac{m_{a,im}}{AFR_s m_f} + (\lambda - 1)(1 - SE) \implies \\ \lambda &= \frac{\frac{m_{a,im}}{SE}}{AFR_s m_f} - \frac{1}{SE} + 1 \end{aligned} \quad (2.28)$$

The quotient $\frac{m_{a,im}}{SE}$ represents the total mass m_a and can be rewritten using the ideal gas law

$$\lambda = \frac{\frac{V_{cyl} p_{im}}{RT_{im}}}{AFR_s m_f} - \frac{1}{SE} + 1 \quad (2.29)$$

For λ not to become negative, there has to be a restriction on the fuel.

$$\begin{aligned} 1 + \frac{\frac{V_{cyl} p_{im}}{RT_{im}}}{AFR_s m_f} &> \frac{1}{SE} \\ m_f &< \frac{SE}{AFR_s} \left(1 + \frac{V_{cyl} p_{im}}{RT_{im}}\right) \end{aligned} \quad (2.30)$$

Temperature

The two-stroke spark ignited combustion process can be described in terms of an Otto cycle in a p-V chart. The temperature dynamics are closely connected to the scavenging process since the retained gas after each stroke has a higher temperature than the intake manifold input. The scavenging efficiency is hence used when specifying the relationship between retained and fresh gas at the intake of each combustion cycle. First defining the residual gas fraction,

$$X_r = 1 - SE \quad (2.31)$$

makes it possible to write the combustion cycle intake temperature expression

$$T_1 = \frac{c_{V_{in}} T_{im} (1 - X_r) + c_{V_{ex}} T_{e,out} X_r}{c_{V_{in}} (1 - X_r) + c_{V_{ex}} X_r} \quad (2.32)$$

If the specific heats for intake and exhaust gases are assumed to be equal the expression can be simplified

$$T_1 = T_{im}(1 - X_r) + T_{e,out} X_r \quad (2.33)$$

The two-stroke cycle runs in four steps. Applying the ideal gas law and some basic thermodynamics on these steps will eventually yield in an expression of exhaust temperature as a function of intake manifold temperature.

- **1 → 2: Compression**

At intake, the cylinder reaches its bottom position letting the intake and exhaust valves open. Since the intake valve is open and directly connected to the intake

manifold, the pressure at step 1 in the p-V chart equals that of the intake manifold, $p_1 = p_{im}$. The volume in step 1 (and step 4) is the combustion chamber max volume $V_1 = V_{cyl}$. In step 1, using the ideal gas law, the air mass in the combustion chamber is

$$m_a = \frac{p_1 V_1}{R T_1} \quad (2.34)$$

Between step 1 and 2 the gas that is trapped in step 1 undergoes compression as the cylinder moves towards its top position. As the volume decreases the pressure increases. This is assumed to be an adiabatic process where combustion chamber external heat transfer is neglected, which gives the expression

$$p_2 V_2^\gamma = p_1 V_1^\gamma \iff p_2 = p_1 \left(\frac{V_1}{V_2} \right)^\gamma \quad (2.35)$$

Due to the ideal gas law the pressure p is a function of $\frac{T}{V}$ multiplied by two constants (the mass, since $m_1 = m_2$ and the gas constant R). Using equation (2.35) together with this knowledge results in a relationship between T_2 and T_1

$$\frac{T_2}{V_2} V_2^\gamma = \frac{T_1}{V_1} V_1^\gamma \iff T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{\gamma-1} \quad (2.36)$$

- **2 → 3: Combustion**

In step 2 the compressed gas is ignited which results in increased temperature and pressure, ending up in step 3. The volume during this process is constant, $V_2 = V_3 = V_t$. The increase in temperature between step 2 and 3 is dependant on the mass of fuel in the compressed gas, hence

$$\begin{aligned} m_f Q_{HLV} &= (m_a + m_f) c_V (T_3 - T_2) \iff \\ T_3 &= \frac{m_f Q_{HLV}}{(m_a + m_f) c_{V_{in}}} + T_2 \end{aligned} \quad (2.37)$$

- **3 → 4: Expansion**

After the moment of ignition the cylinder begin the process of returning to its bottom position, meaning that the combustion chamber volume increases while the pressure decreases. Similar to the transfer between steps 1 and 2, the temperature is assumed to remain constant.

$$p_4 V_4^\gamma = p_3 V_3^\gamma \iff p_4 = p_3 \left(\frac{V_3}{V_4} \right)^\gamma \quad (2.38)$$

Using, like in step 1 – 2, the fact that the mass and gas constant are constants gives a relationship between T_4 and T_3

$$T_4 = T_3 \left(\frac{V_2}{V_1} \right)^{\gamma-1} \quad (2.39)$$

- **4 → 1: Exhaust**

In the final step the burnt gases are released out into the exhaust manifold. To calculate the temperature of the burnt gases, the temperature of previous steps are combined

$$\begin{aligned} T_4 &= T_3 \left(\frac{V_2}{V_1} \right)^{\gamma-1} = [use\ equation\ 2.37\ and\ 2.36] = \\ &= \left(\frac{m_f Q_{HLV}}{(m_a + m_f) c_V} + T_1 \left(\frac{V_1}{V_2} \right)^{\gamma-1} \right) \left(\frac{V_2}{V_1} \right)^{\gamma-1} = [simplify] = \\ &= \frac{m_f Q_{HLV}}{(m_a + m_f) c_V} \left(\frac{V_2}{V_1} \right)^{\gamma-1} + T_1 = [use\ \lambda\text{-relationship}] = \\ &= \frac{Q_{HLV}}{(\lambda AFR_s + 1) c_V} \left(\frac{V_2}{V_1} \right)^{\gamma-1} + T_1 \end{aligned} \quad (2.40)$$

Now, this is T_4 as a function of T_1 which is close to a useful expression. There is one issue left though; T_1 is a function of T_{im} and T_4 , so the last step is to combine equations (2.33) and (2.40) to get T_4 as a function of T_{im}

$$T_4 = \frac{1}{SE} \frac{Q_{HLV}}{(\lambda AFR_s + 1)c_V} \left(\frac{V_2}{V_1} \right)^{\gamma-1} + T_{im} \quad (2.41)$$

Now, T_4 is only the temperature of the burnt gases in the combustion chamber and can not be used as the combustion chamber output temperature without some slight modification. The issue here is the short-circuit air through the combustion chamber, which has a temperature considerably lower than the burnt gas temperature T_4 . The combustion chamber output temperature has to be a weighted average between these two massflow's (equation 2.24) temperatures respectively.

$$T_{e,out} = \frac{\dot{m}_{ch,exh} T_4 + \dot{m}_{shc} T_{im}}{\dot{m}_e} \quad (2.42)$$

2.7 Exhaust manifold

The exhaust manifold is, just as the intake manifold, a receiver and there are no major differences in their modeling. It is more common to consider external energy flow, Q , in the exhaust manifold and some of the parameters are naturally different.

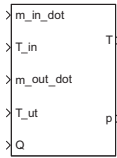


Figure 2.5 – Inputs and outputs of exhaust manifold Simulink block

Table 2.4 – Input signals, output signals and parameters of exhaust manifold

INPUTS		
\dot{m}_{in}	kg/s	Air mass-flow from combustion chamber
\dot{m}_{out}	kg/s	Air mass-flow towards turbine
T_{in}	K	Temperature of combustion chamber air
T_{out}	K	Temperature in exhaust manifold (same as output T)
Q	J/s	Heat-flow into the system (optional)
OUTPUTS		
T	K	Temperature in exhaust manifold
p	Pa	Pressure in exhaust manifold
PARAMETERS		
κ_{em}	—	Ratio of specific heats
V_{em}	m ³	Volume of exhaust manifold
$T_{init,em}$	K	Initial temperature of exhaust manifold
$p_{init,em}$	Pa	Initial pressure of exhaust manifold
R	J/(kg·K)	Gas constant

2.8 Turbine

The turbine is connected to the compressor through the turbo shaft as mentioned before. Just like the compressor the turbine consists of a rotating propeller device which is here given momentum by the exhaust manifold air. Parallel to the turbine is a device called the waste-gate which can be opened to release pressure in order to decrease the rotational speed of the turbo shaft and thereby lower the pressure in the intake manifold.

It usually takes a while for the turbine to start rotating at engine start up. Which is not hard to believe. Actually the turbo is a non-causal process, meaning that the

compressor needs the turbine to rotate in order to provide the combustion with air so that the exhausts can maintain the turbine speed. So, for the turbine to start rotating at all there is an external device added, called the jet-assist. This will make the turbine rotate during the start up process until there is a big enough exhaust air flow for it to rotate without external help.

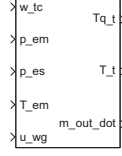


Figure 2.6 – Inputs and outputs of turbine Simulink block

The turbine Simulink block consists of five inputs and three outputs, see figure 2.6 and table 2.5. The turbine output massflow, \dot{m}_t , is the same signal connected as an input to the exhaust manifold block, also representing the exhaust manifold output massflow.

Table 2.5 – Input signals, output signals and parameters of turbine

INPUTS		
ω_{tc}	rad/s	Rotational speed for turbine and compressor
p_{em}	Pa	Pressure upstream: from exhaust manifold
p_{es}	Pa	Pressure downstream: ambient pressure
T_{em}	K	Temperature upstream: from exhaust manifold
u_{wg}	—	Control signal, wastegate
OUTPUTS		
Tq_t	Nm	Turbine generated torque on turbo shaft
\dot{m}_t	kg/s	Air mass-flow out of turbine
T_t	K	Turbine out-air temperature
PARAMETERS		
c_p	J/(kg K)	Air specific heat
d_{wg}	m	waste-gate diameter

Massflows

There are two ways for exhaust gases to pass the turbine, either through the turbine wheel to feed the turbo shaft and compressor with torque or through the waste-gate in order to release pressure built up in the exhaust manifold. When the waste-gate is fully open, a reduced amount of exhaust gases will flow through the turbine while the remainder will bypass the turbine through the waste-gate directly out in the exhaust system. The waste-gate is controlled externally and can be opened and closed independently of other signals. When it is open the flow is restricted by the intersection area and to some extent by sonic restrictions. The actual area of the waste-gate opening is

$$A_{wg} = \pi \left(\frac{d_{wg}}{2} \right)^2 \quad (2.43)$$

which is multiplied by the discharge coefficient, C_d , and by the effective waste-gate lift, $l_{wg,eff}$, to constitute the effective area. The discharge coefficient has a value between 0 and 1 and could be described as the relationship of actual versus ideal mass flow through the waste-gate opening. The waste-gate lift is a function of the waste-gate control signal and waste-gate diameter ([2] p.13) as

$$l_{wg,eff} = u_{wg} \min \left(l_{wg,max}, \frac{d_{wg}}{4} \right) \quad (2.44)$$

constituting the effective area.

$$A_{wg,eff} = C_d l_{wg,eff} A_{wg} \quad (2.45)$$

The turbine and waste-gate are both modeled as flow restrictions which gives a situation for the flow somewhat similar to that of the current in an electrical circuit. The massflow corresponds to electrical current, the pressure difference corresponds to voltage and the flow restrictions correspond to resistances. The turbine and waste-gate flows are, respectively ([2] p.7,13),

$$\dot{m}_t = \frac{p_{em}}{\sqrt{T_{em}}} C \sqrt{1 - \left(\frac{p_{es}}{p_{em}}\right)^K} \quad (2.46)$$

$$\dot{m}_{wg} = \frac{A_{wg,eff}}{\sqrt{T_{em} R}} p_{em} \Psi\left(\frac{p_{es}}{p_{em}}\right) u_{wg} \quad (2.47)$$

where C and K are adjustable constants and the sonic restriction is

$$\Psi\left(\frac{p_{es}}{p_{em}}\right) = \begin{cases} \sqrt{\gamma} \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & , \frac{p_{es}}{p_{em}} \leq \left(\frac{p_{es}}{p_{em}}\right)_{crit} \\ \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{p_{es}}{p_{em}}\right)^{\frac{2}{\gamma}} - \left(\frac{p_{es}}{p_{em}}\right)^{\frac{\gamma+1}{\gamma}} \right]} & , \frac{p_{es}}{p_{em}} > \left(\frac{p_{es}}{p_{em}}\right)_{crit} \end{cases} \quad (2.48)$$

The critical pressure ratio determines when choking of the flow occurs.

$$\left(\frac{p_{es}}{p_{em}}\right)_{crit} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (2.49)$$

In the mass flow equations, everything that is not related to pressure is lumped together forming the "conductance" G which is inverted to form the "resistance" Q

$$Q_t = \frac{\sqrt{T_{em}}}{C} \quad (2.50)$$

$$Q_{wg} = \frac{\sqrt{T_{em} R}}{A_{wg,eff}} \quad (2.51)$$

Current division then gives the weighed mass flows

$$\dot{m}_t = \frac{Q_{wg}}{Q_t + Q_{wg}} p_{em} \sqrt{1 - \left(\frac{p_{es}}{p_{em}}\right)^K} \quad (2.52)$$

$$\dot{m}_{wg} = \frac{Q_t}{Q_t + Q_{wg}} p_{em} \Psi\left(\frac{p_{es}}{p_{em}}\right) u_{wg} \quad (2.53)$$

It should be said that there might be other, more efficient and/or easier, solutions to the turbine and waste-gate massflows. At least for a model that is as generalized as the one developed in this project.

Temperature

There is a temperature drop in the air that passes the turbine that has to be taken into consideration (no temperature reduction over waste-gate). The expression for the turbine efficiency ([2] p.13)

$$\eta_t = \frac{1 - \frac{T_t}{T_{em}}}{1 - \left(\frac{p_{es}}{p_{em}}\right)^{\frac{\gamma-1}{\gamma}}} \quad (2.54)$$

leads to the expression for the difference in temperature

$$\Delta T = T_{em} - T_t = T_{em} \left(1 - \left(\frac{p_{es}}{p_{em}}\right)^{\frac{\gamma-1}{\gamma}}\right) \eta_t \quad (2.55)$$

There are different ways to estimate the efficiency η_t , but since this model is quite generalized the efficiency is set to a constant value < 1 . For other ways of modeling

the turbine efficiency see for example ([2] p.13) where use of the so called Blade Speed Ratio is covered. The temperature difference is later used to form the turbine torque expression, but it is also needed to form the output turbine temperature

$$T_t = T_{em} - \Delta T \quad (2.56)$$

Observe that this is only the temperature for the turbine, the waste-gate not included. For a correct post-turbine temperature a weighed sum of the turbine air temperature and the waste-gate (no temperature loss assumed) air temperature should be made.

Torque

This section covers only the torque produced as a result of the air flow from the exhaust manifold. The sum of turbo shaft impacts are treated in section 2.9.

The massflow from the exhaust manifold is splitted between the waste-gate and the turbine but only the flow through the turbine will result in torque to the turbo shaft. The expression for the difference in temperature before and after the turbine was derived in the previous section. Those variables together with the air specific heat are what is needed to form the turbine power

$$P_t = \dot{m}_t c_p \Delta T \quad (2.57)$$

The resulting turbine torque is calculated by dividing the power by the turbo shaft rotational speed

$$Tq_t = \frac{P_t}{w_{tc}} \quad (2.58)$$

2.9 Turbo shaft

The turbo shaft is the shaft connecting the compressor and the turbine. The shaft is assumed to be stiff with no dynamics of its own.

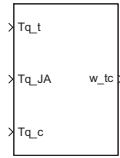


Figure 2.7 – Inputs and outputs of turbo shaft Simulink block

Three input parameters are effecting the resulting turbo torque; the massflow from the exhaust manifold over the turbine, the massflow originating from the jet-assist and the braking torque from the compressor. The jet-assist is modelled as a user maneuverable variable with an appropriate gain applied.

Table 2.6 – Input signals, output signals and parameters of turbo shaft

INPUTS		
Tq_t	Nm	Torque generated by turbine
Tq_{JA}	Nm	Torque from Jet-Assist
Tq_c	Nm	Torque consumed by compressor
OUTPUTS		
w_{tc}	rad/s	Rotational speed for turbine and compressor
PARAMETERS		
J_{tc}	kg m ²	Inertia of turbo system

The angular velocity of the shaft is calculated using Newton's second law for rotational systems

$$\dot{w}_{tc} = \frac{1}{J_{tc}} (Tq_t + Tq_{JA} - Tq_c) \quad (2.59)$$

2.10 External load

The external load originates from whatever the engine is supposed to drive. In the model it is an external parameter which can be adjusted independently of the system as a whole.

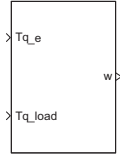


Figure 2.8 – Inputs and outputs of external load Simulink block

Table 2.7 – Input signals, output signals and parameters of external load

INPUTS		
Tq_e	Nm	Torque generated by engine
Tq_{load}	Nm	Torque from external load
OUTPUTS		
w	rad/s	Rotational speed for engine shaft
PARAMETERS		
J	kg m ²	Inertia of engine shaft

The angular velocity of the shaft is calculated using Newton's second law for rotational systems

$$\dot{w} = \frac{1}{J} (Tq_e - Tq_{load}) \quad (2.60)$$

3 Simulation results

This chapter includes simulation results for the biggest three Simulink blocks of the two-stroke engine; the Compressor, the Turbine and the Combustion chamber. The simulations are designed to resemble real conditions as well as possible. Parameters are chosen either from literature or through qualified assumptions in cases when no good reference is to be found. No real engine has been used in the process. Similarity to a real engine is restricted by the fact that each of these three sub-models are simulated separately, which disable possibility to look at interaction between blocks. Though, it is still possible to investigate if they behave in correlation with reasonable assumptions. A brief overview of the largest assembled model unit that does not produce Simulink errors will be presented as well.

The simulation cases are designed to highlight basic dynamics of the simulated component which is done by altering one or two input signals per simulation and looking at the output signals. The test input signals are fitted to be a somewhat good representation of what could appear in reality. Analyzing the input-output relationships in this way will give graphical data and thus provide more intuitive explanations and interpretations of the equations in Chapter 2.

There are some parameter values that stay unchanged throughout all the simulations and that are not component specific; ambient temperature and pressure, gas constant and ratio of specific heats:

- $T_{amb} = 300$ [K]
- $p_{amb} = 101 \text{ e}3$ [Pa]
- $R = 287$ [J/(kg K)]
- $\gamma = 1.3$ [-]

When looking at the signal values, bear in mind that many constants and parameters are quite roughly estimated which inevitably leads to slight inaccuracies. Look instead for overall trends and typical behaviour.

3.1 Compressor simulation

The compressor specific parameter values are k_1 and k_2 from equation (2.4), estimated to.

- $k_1 = 0.1$ [kg/rad]
- $k_2 = 1$ [kg/s]

This section provides two simulations with different inputs of the compressor, simulation 1 and 2, respectively. The governing equations and relationships deciding the outcome of the compressor simulations are to be found in chapter 2.4.

Simulation 1

One of the most fundamental parts of the compressor is the propeller which is connected to the turbine and described by rotational speed. Therefore, varying the rotational speed of the turbo shaft will result in valuable information on compressor performance. Typically, a high rotational speed of the turbo shaft will result in higher pressure in the intake manifold, downstream of the compressor. This effect will not be visible in this simulation, since both rotational speed and downstream pressure are input signals and therefore not feedbacked to each other. The downstream pressure will be changed separately, uncorrelated to the rotational speed.

Expected behavior following an increase in rotational speed would be increased torque (turbo shaft load) and massflow. The temperature is not directly a function of rotational speed but is a function of the pressure quotient which on the other hand is affected by

the rotational speed. An increase in the pressure downstream should give increased torque and temperature but decreased massflow.

INPUTS

The turbo shaft rotational speed varies from 10 to 1000 rad/s. The pressure before the compressor is equivalent to ambient pressure, p_{amb} while the pressure after the compressor is 1.5 times the ambient pressure at time zero. The steps have amplitude of 5 and 10 times the amplitude at time zero. Downstream pressure is altered uncorrelated to the rotational speed which provides a good feeling for how these two signals interact. The temperature before the compressor is equivalent to ambient temperature T_{amb} .

To get a good look at trends, the downstream pressure step magnitudes are set to be higher than what could be expected from reality.

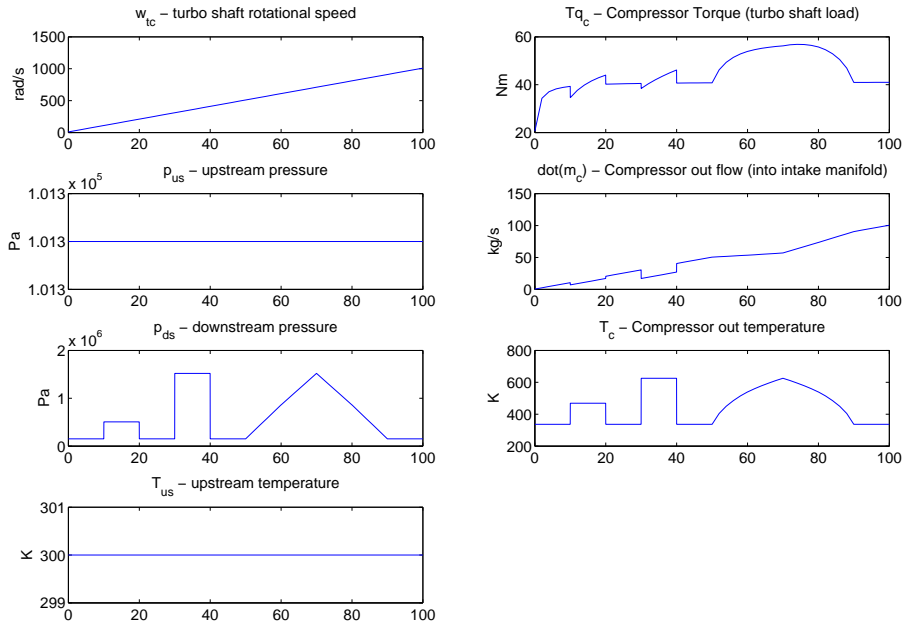


Figure 3.1 – Inputs and outputs of compressor when increasing turbo shaft rotational speed at constant rate while changing the pressure in intake manifold in three different steps

OUTPUTS

The compressor torque is the load on the turbo shaft, having the opposite direction of the turbine torque and is affected by both rotational speed and downstream pressure. For small ω_{tc} the torque is increasing rapidly but for large ω_{tc} it saturates. It seems that at high enough rotational speed with unchanged pressure the torque will not increase further meaning that an increase in rotational speed does not necessarily imply increased turbo shaft load. Now, this is slightly hypothetical since increased compressor speed will result in increased downstream pressure. At the downstream pressure steps, the saturation level of the torque increases for large ω_{tc} and the slope towards negative torque is steeper for small ω_{tc} .

Negative torque would imply a flow in the backwards direction which is reasonable if the pressure is much higher downstream than upstream. An increased downstream pressure gives an increased resistance for the flow.

The compressor temperature is not affected by rotational speed but highly correlated to downstream pressure. The higher the quotient between downstream and upstream pressure, the higher the temperature.

The result corresponds well to expected behaviour in most aspects. The torque satura-

tion phenomenon might need to be reviewed and investigated further.

Simulation 2

Changing the pressure before the compressor is basically the same thing as changing it after the compressor since upstream and downstream pressure usually appear in quotients. The effects of this were partly covered in simulation 1. When changing upstream temperature, the downstream temperature is naturally expected to change proportionally.

INPUTS

The turbo shaft rotational speed is constant 500 rad/s.

The pressure before the compressor is raised at constant rate from ambient pressure to 6 times ambient pressure.

The pressure after the compressor is constantly 1.5 times the ambient pressure.

The temperature before the compressor starts at ambient temperature and increase at constant rate after half the simulation time.

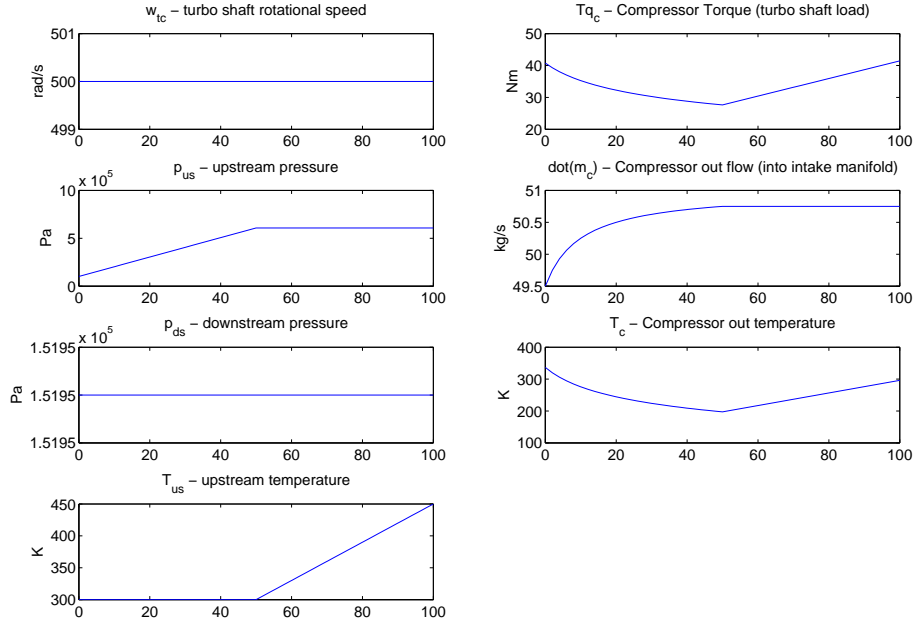


Figure 3.2 – Inputs and outputs of compressor when first increasing upstream pressure and then increasing upstream temperature at constant rates.

OUTPUTS

When upstream pressure increases, less torque is braking the turbo shaft since the pressure gradient does the job itself. The flow out from the compressor is not affected by the upstream temperature, but when upstream pressure increases, so does the flow. Increased upstream pressure implies decreased outlet temperature.

Compared to expected behaviour some things are worth looking deeper into, for example the connection between upstream temperature and compressor torque is not that intuitive. In chapter 2.4 the relationship between compressor torque and massflow is discussed.

3.2 Turbine simulation

The turbine specific parameters are K_{wg} and C_t in equation (2.46), η_t in equation (2.55), C_D in equation (2.45) and d_{wg} in equation (2.43), estimated to,

- $K_{wg} = 0.01 [-]$
- $C_t = 0.001 [-]$
- $\eta_t = 0.5 [-]$
- $C_D = 0.8 [-]$
- $d_{wg} = 0.1 [m]$

Following are three simulations with different inputs of the turbine, simulation 1, 2 and 3, respectively. The governing equations and relationships deciding the outcome of the turbine simulations are to be found in chapter 2.8.

Simulation 1

One of the major features in the turbo system is the waste-gate. The waste-gate bypasses the turbine when opened and force the flow to pass only through the turbine when closed. When closing the waste-gate while keeping everything else constant, the torque is expected to increase and the total turbine output massflow is expected to decrease since the turbine will be the only way for the air to flow through.

INPUTS

In this simulation, all input parameter values are kept constant except the waste-gate control signal. The turbo shaft rotational speed is constant at 500 rad/s. The exhaust manifold pressure is assumed to be six times the atmospherical pressure. The exhaust manifold temperature should be higher than ambient temperature and is here assumed to be 600 K. The waste-gate control start at value 1 which corresponds to a fully opened waste-gate and decreases at constant rate down to a fully closed waste-gate over 100 seconds.

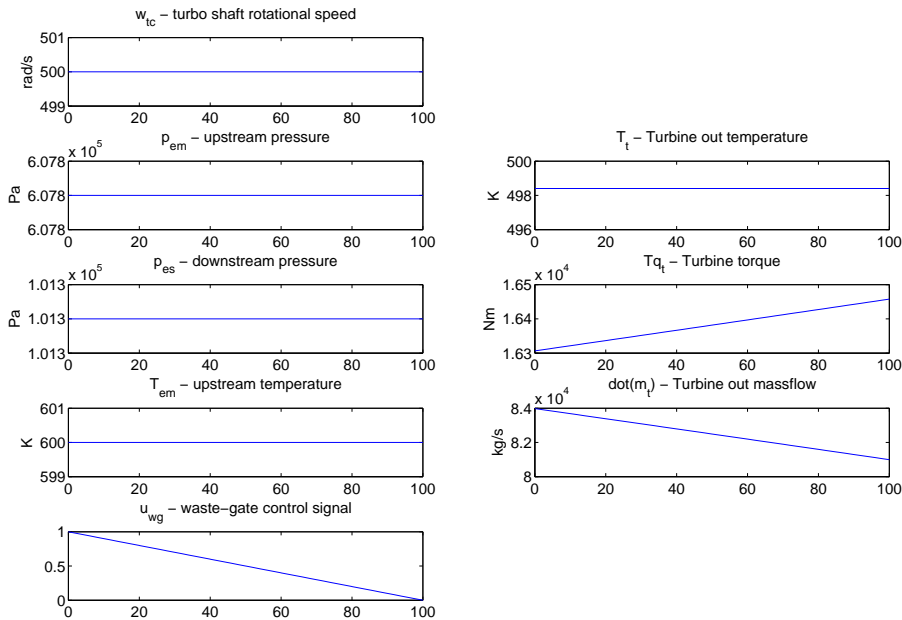


Figure 3.3 – Inputs and outputs of turbine when turning a fully opened waste-gate down to fully closed at constant rate.

OUTPUTS

A successively closing waste-gate implies less flow through the waste-gate and hence increased flow through the turbine. The total flow, as shown in the figure, will be

slightly reduced though due to there being only one valve to run through instead of two. Intuitively, a larger flow through the turbine increases the turbine output torque, driving the turbo shaft. The output temperature is not directly influenced by the waste-gate but is decreased to a lower constant level compared to the input temperature. If closing the waste-gate results in less total massflow it should be expected that the flow through the turbine would increase if not having reached saturation, and thereby also result in a change in temperature. This is probably an incompleteness in the model and a matter of future investigation. The other outputs go well with expected behaviour but the changes in torque and massflow are very small and probably not perfectly correct.

Simulation 2

The upstream pressure of the turbine is the exhaust manifold pressure, which decreases when the engine slows down. The waste-gate is shut, thus pushing all flow through the turbine. When the upstream pressure equals the downstream pressure there can not be any flow through the turbine, with the effect of zero torque and massflow. When there is no flow, there should be no temperature losses.

INPUTS

All input parameter values are constant at the same values as in Simulation 1, except for the upstream pressure which decreases from twice the ambient pressure down to ambient pressure at constant rate.

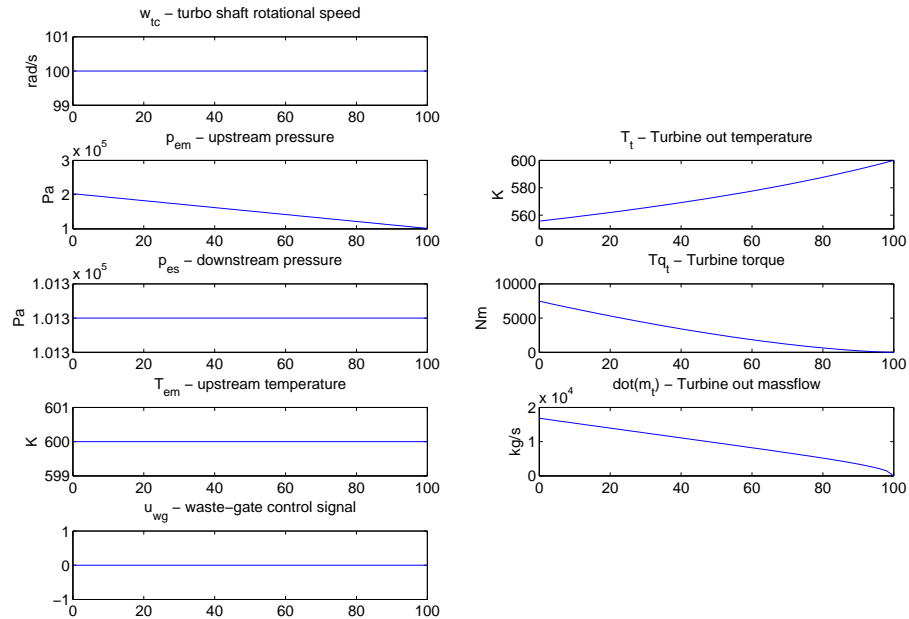


Figure 3.4 – Inputs and outputs of turbine when decreasing exhaust manifold pressure at constant rate.

OUTPUTS

It is interesting to see what happens when the pressure gradient is zero, that is, when exhaust manifold pressure equals ambient pressure, in this simulation at time 100 s. First of all, there should be no massflow which fortunately is the case. The turbine torque also follows the massflow down to zero which is also expected. When the turbine output flow is zero, there is no output temperature to speak of since the temperature is a feature of the gases that are not transported. At non-zero flow the gas releases energy when passing the turbine (but not the waste-gate) decreasing the output temperature. Maybe the temperature equations needs to be adjusted to give proper outputs at low

turbine speed.

Simulation 3

The turbo shaft connects the turbine and compressor and transfers the torque generated by the exhaust manifold pressure over to the compressor to increase the pressure in the intake manifold.

INPUTS

The setup is similar to Simulation 2, but exhaust manifold pressure is now constant at double ambient pressure in order to get a pressure gradient in positive flow direction. The turbo shaft rotational speed is increased at constant rate from 1000 to 2000 rad/s.

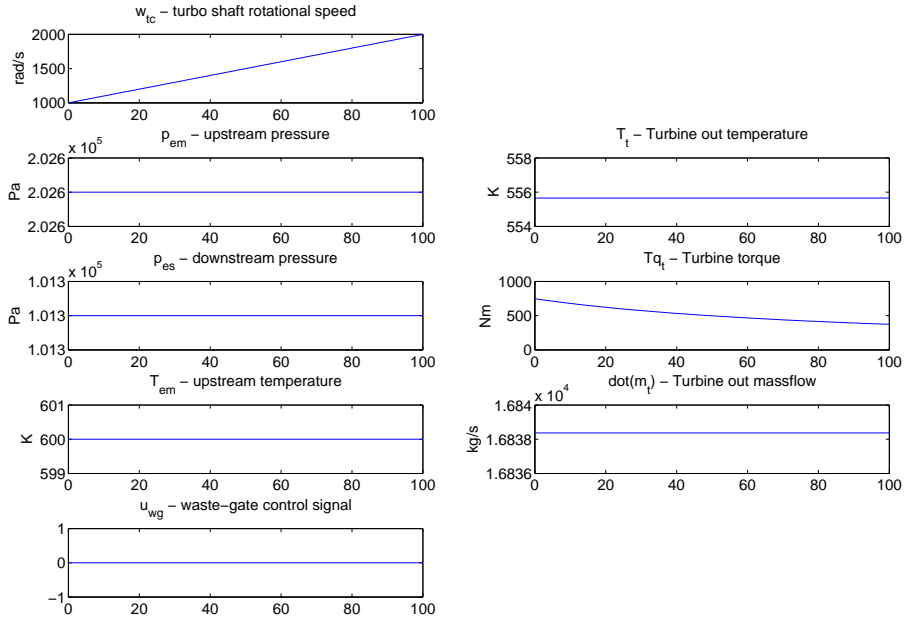


Figure 3.5 – Inputs and outputs of turbine when increasing turbo shaft rotational speed at constant rate.

OUTPUTS

In the fully assembled and simulated engine model, a change in torque would imply a change in turbo shaft rotational speed. When simulating the turbine separately this feedback is gone. What is seen in the output turbine torque is the effects of equation (2.58), an inverse proportionality relationship between rotational speed and torque when massflow, temperature and pressure are constants.

3.3 Combustion chamber simulation

The combustion chamber specific parameters are k_1 and k_2 in equation (2.15), η_{ig} and q_{HLV} in equation (2.13), n in equation (2.12), k_{SE} from equation (2.18), N from equation (2.21) and k_e from (2.19), estimated to,

- $k_1 = 1e - 6 [-]$
- $k_2 = 1e - 6 [-]$
- $\eta_{ig} = 0.75 [-]$
- $q_{HLV} = 3.4e7 [J/m^3]$ ([4] app.D)

- $n = 1$ [cycles per combustion]
- $k_{SE} = 0.5$ [-]
- $k_e = 1$ [-]
- $N = 1$ [-]

Following are three simulations with different inputs of the combustion chamber, simulation 1, 2 and 3, respectively. The governing equations and relationships deciding the outcome of the combustion chamber simulations are to be found in chapter 2.6.

Simulation 1

The intake manifold temperature tells at what temperature the massflow enters the combustion chamber.

INPUTS

All input parameter values are kept constant while increasing intake manifold temperature at constant rate, from a slightly unrealistically low temperature up to a little above room temperature.

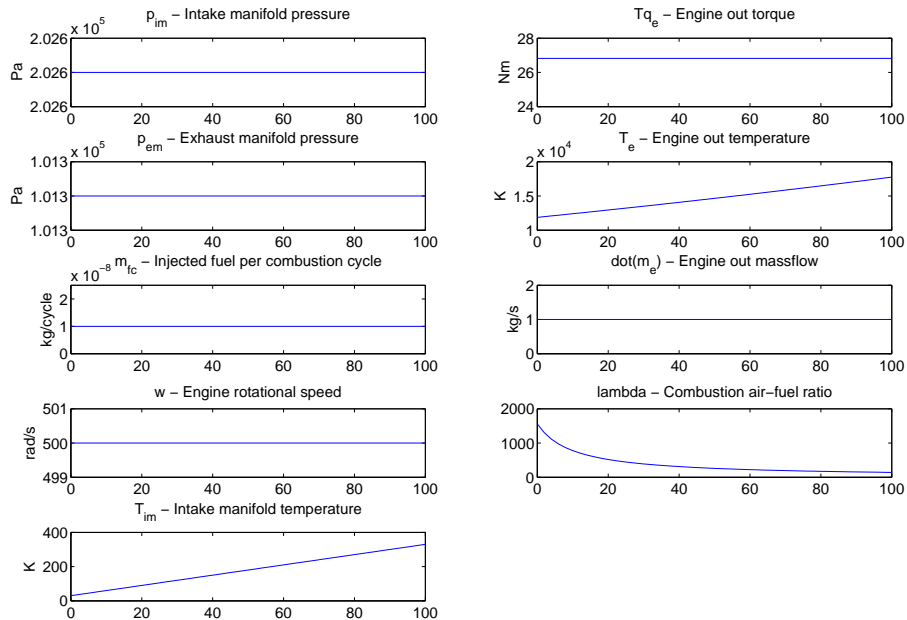


Figure 3.6 – Inputs and outputs of combustion chamber when increasing intake manifold temperature at constant rate.

OUTPUTS

A rise in input temperature results in a rise of output temperature which is expected. The lambda value increases when decreasing the input temperature. It might be a bit surprising that input temperature has such a large impact on lambda, but in reality temperatures below 300 K is not that interesting. When reaching very low temperatures the density of the gas will be considerably lower making place for a larger mass of gas per amount of fuel. Theoretically, that is. The temperature change not having any impact on the torque is a bit unexpected and should be investigated.

Simulation 2

The intake manifold pressure depends largely on the compressor activity. Changing the pressure quotient over the combustion chamber will result in altered mass flow through

the combustion chamber.

INPUTS

The input parameter value setup is much the same as in Simulation 1 but with the intake manifold temperature held constant at 300 K and the intake manifold pressure increased at constant rate originating at double the ambient pressure.

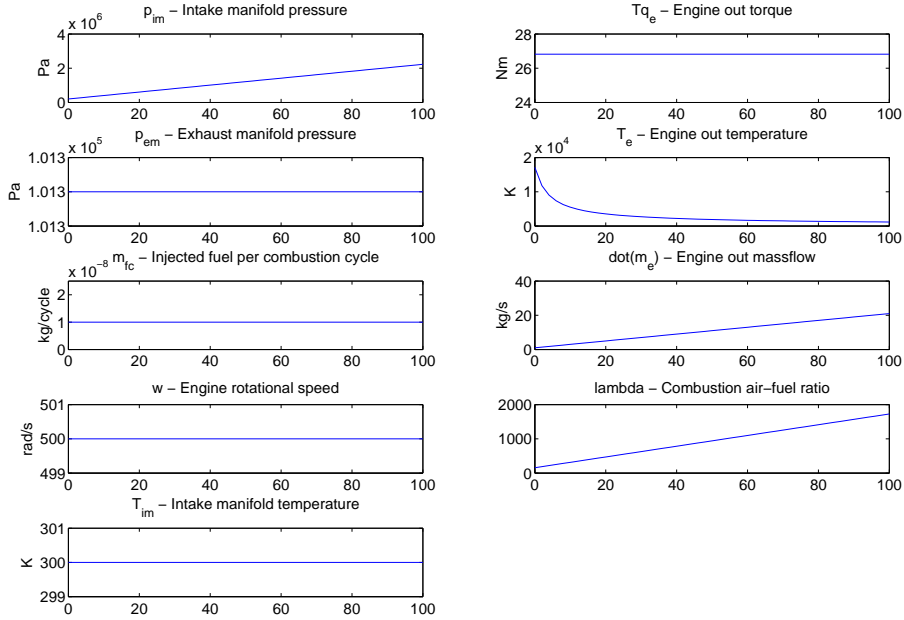


Figure 3.7 – Inputs and outputs of combustion chamber when increasing intake manifold pressure at constant rate.

OUTPUTS

A higher intake manifold pressure leads to a higher pressure quotient over the combustion chamber. At least it does in this case when the exhaust manifold pressure is not affected by pressure changes upstream. As expected the increased pressure quotient gives an increased mass flow output and more air through the combustion also implies larger λ -value. The combustion chamber output temperature is affected by increased mass flow since there will be more air to heat, hence decreased temperature.

Simulation 3

The amount of fuel per time unit depends upon the engine rotational speed since the input fuel parameter sets fuel per cycle.

INPUTS

The setup of Simulation 3 is similar to previous simulations but now the engine rotational speed is increased at constant rate and fuel per cycle is doubled as a step function after half the simulation.

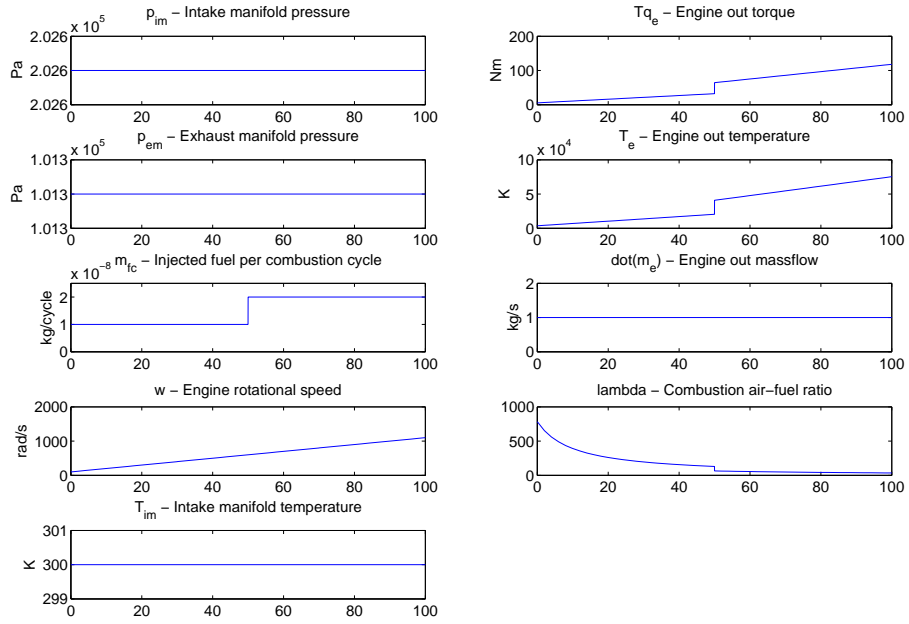


Figure 3.8 – Inputs and outputs of combustion chamber when making a step increase in fuel per cycle while increasing rotational speed of engine at constant rate.

OUTPUTS

Both engine torque and output temperature increase with increased rotational speed. The temperature increases because the amount of fuel increases with the rotational speed, and more fuel per combustion cycle means more energy released, thus a higher temperature. The same reasoning goes for the output torque. The output mass flow should depend on the rotational speed, since the parameter k_e in equation (2.19) in reality depends on rotational speed, but this is not implemented. The air-fuel ratio decrease with increased amount of fuel as expected.

3.4 Largest no-Simulink-error model unit

When making a Simulink model consisting of more than a few separate blocks, the complexity of the assembled system increase with every added block. This is particularly obvious when trying to put together a system where every block in some way have impact on all other blocks, directly or indirectly.

Figure (3.9) presents the largest unit of assembled Simulink blocks that runs in Simulink without errors. This model leaves the turbine out, forcing constant values to be connected to \dot{m}_{out} of the exhaust manifold and to Tq_t of the turbo shaft.

Actually, not only the turbine is removed, there is one more restriction implemented. The pressure from the exhaust manifold is disconnected and replaced by a constant value where it should have been input to the combustion chamber. And this seems to be the crucial matter. When the exhaust manifold output pressure is connected to the combustion chamber input (or the turbine input), Simulink will not run the model. Though, more thorough testing and validation is needed in order to verify this guess.

However, even though this model runs without any Simulink errors, it is very hard to give a relevant interpretation of the result since the model is mutilated. Following are the global input values together with input/output plot for the combustion chamber.

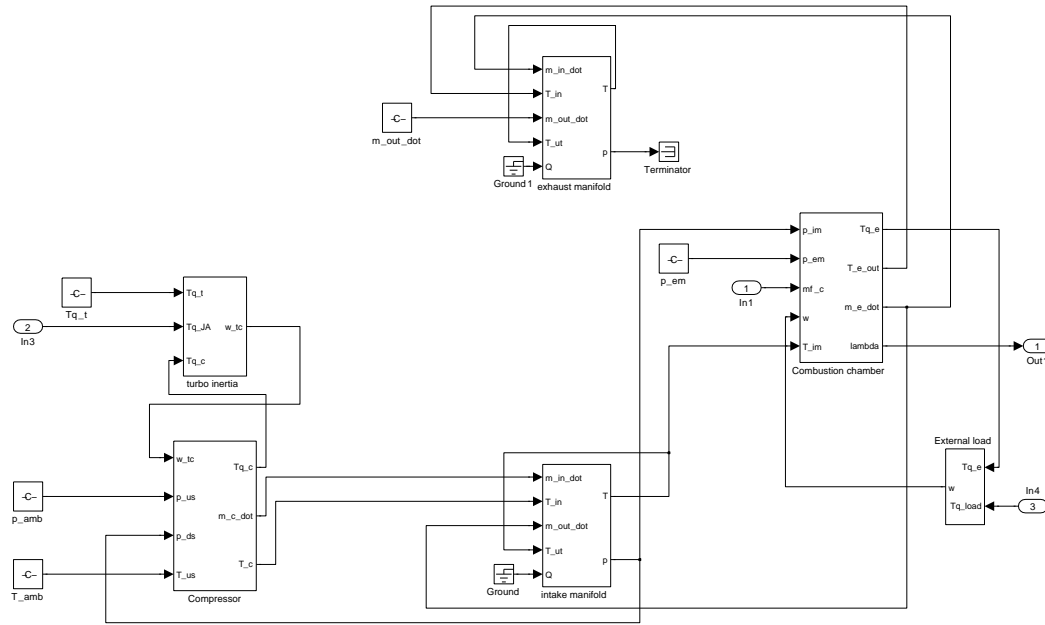


Figure 3.9 – The largest model unit that does not produce Simulink errors

INPUTS

The inputs used for running the Simulation resulting in figure 3.10 are; compressor upstream pressure, $p_{us} = p_{amb}$, compressor upstream temperature, $T_{us} = T_{amb}$, exhaust manifold pressure into combustion chamber, $p_{em} = 2p_{amb}$, amount of fuel per cycle, $m_{f_c} = 1e-8$ kg/cycle, external load torque, $T_{q_{load}} = 500$ Nm, exhaust manifold output massflow, $\dot{m}_{out} = 50$ kg/s, turbine torque, $T_{qt} = 100$ Nm, Jet-assist torque, $T_{q_{JA}} = 10$ Nm.

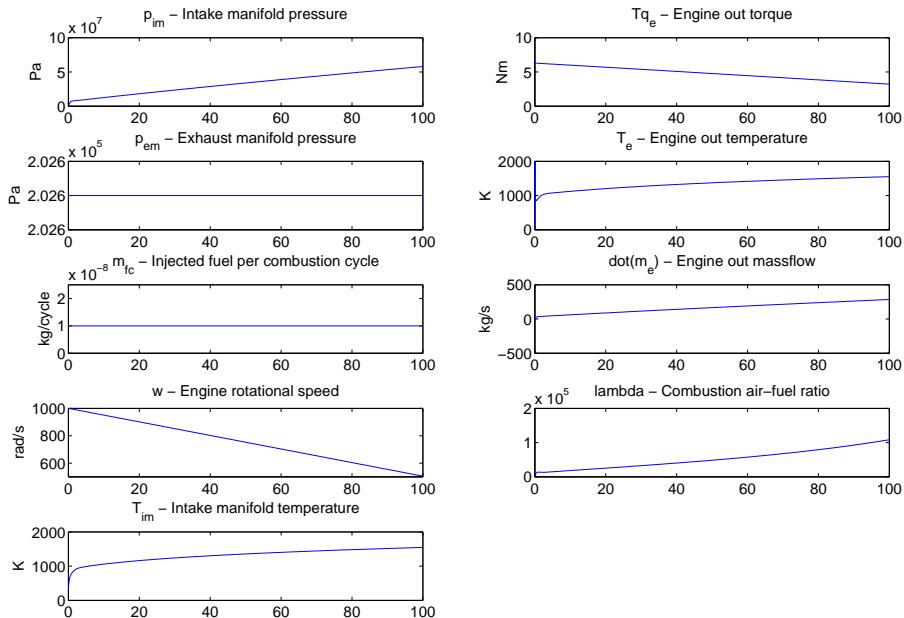


Figure 3.10 – Inputs and outputs of combustion chamber at simulation with the above stated inputs.

4 Conclusions

Looking at the project specification, most of the pre-defined goals are fulfilled. Serious attempts to describe the dynamics of each component have been made, and have resulted in a number of Simulink blocks that each are generalized models of actual engine dynamics. Individually and taken out of context, the three main blocks - the compressor, the turbine and the combustion chamber - show a behaviour that is relevant and reasonable, but the output signal magnitudes of each block are not properly scaled and is one of the greatest challenges to overcome in order to obtain a useful model of the engine.

4.1 Discussion

It has been harder than expected to assemble all separate blocks into one well-functioning unit mainly because of the complexity of such a system. The complexity combined with a slightly ineffective assembling strategy, connecting the blocks together at a too early stage, has proven to be time-consuming. Ideally each block would be completely validated and tested before connected together.

This kind of modeling work is always a question of re-considering your solutions and refining your strategies. Since the model only is put into limited function there are most likely good alternative solutions to many of the modeled dynamics.

4.2 Future work

In order to complete the construction of the two-stroke engine model there are issues on different levels that should be looked into. There are high level issues such as making all blocks work together in a simulation giving relevant output. There are low level issues such as choosing accurate and correct physical relationships to describe the dynamics in the engine. It is also a question about refining parameter values which now, in some cases, are quite roughly estimated.

References

- [1] Gordon P. Blair. *Design and Simulations of Two-Stroke Engines*. Society of Automotive Engineers, Inc., 1996.
- [2] Lars Eriksson. Modeling of Turbocharged Engines with MVEM_lib. Vehicular Systems, Linköping University, Linköping, Januari 2008.
- [3] Lars Eriksson and Lars Nielsen. Automotive Control Engineering, Modeling and Control of Internal Combustion Engines (Gröna Bilen). Vehicular Systems, ISY, Linköping Institute of Technology, Linköping, 2005.
- [4] John B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, Inc., 1988.
- [5] John B. Heywood and Eran Sher. *The Two-Stroke Cycle Engine, it's development, operation and design*. Taylor and Francis, Inc., 1999.