

Diesel A-M (A-Modular) Model Proposal

Expanding the Diesel A model for advanced turbocharging capabilities



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Report nr.: 2018.SDPO.8252

April 2, 2017

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1. Introduction

In recent years the development of computational hardware has seen rapid increase in availability and capability. This has led to the widespread use of simulation tools and engineering research and development. The technical University of Delft is no exception, here at the department of Ship Design and Production an Organization (SDPO) many research projects are investigated with the use of a MATLAB/Simulink. This Simulink environment is very suitable to investigate large dynamic systems in the time domain, but it can also be used to investigate static systems. The graphical user interface of Simulink provides the added benefits of recreating physical components as separate blocks and to provide a clear overview of the component relations within the entire system.

Most simulation models at the department of SDPO have been dealing with ships and the components on board of ships. This has led to a wide variety of different component models such as detailed engine models, electrical machines and other commonly used auxiliary components. One component of particular interest is the Diesel engine, for which many different simulation solutions exist. Two models have been developed in-house by the SDPO department; the Diesel A and Diesel B models. These models are both mean value first principle (MVFP) models and they differ in their complexity and application.

The Diesel A model provides a lower entry level model that requires only basic project guide parameters and some other well-known parameters like gas properties. The Diesel B model requires detailed data of the turbocharger groups including the operating maps of the turbine and compressor, which unfortunately are confidential for most engines. The Diesel A model does not use operating maps for the turbocharger groups, instead it uses a model to calculate the available compressor power from which the charge pressure is derived through a first order time delay. Due to this simplification, the Diesel A model is unsuitable to simulate advanced turbocharger strategies such as: waste gate, variable turbine geometry, sequential turbocharging, two-stage turbocharging and hybrid turbocharging. This report puts forward a proposal to alter the existing Diesel A model in such a way to make it suitable to model these advanced turbocharger strategies without the need for the complex Diesel B model.

2. Proposal

The Diesel A model has been developed with the idea of looking at overall engine cycle performance in a larger system. As such, the goal has always been to provide an acceptable prediction of the engine performance without sacrificing the computational performance of the model. The simplest form of the Diesel A model has only a single integrator that is used to estimate the charge pressure based on the available heat at the end of the closed cylinder process. This method has recently been improved by adding a more realistic estimation for the charge pressure through the Bchi equation. This has proved to be an improvement but has also created a new problem: the need for a second integrator to close an algebraic loop in the model. This second integrator represents the time delay of the outlet receiver building up pressure but it has no direct physical derivation. Furthermore this method is still not suitable for advanced turbocharging strategies, and thus an improvement is proposed in this report.

2.1 Modeling Approach

The changes to the Diesel A model are aimed at solving the following issues:

- Provide a physical basis for the pressure build-up in the outlet receiver
- Provide means to implement advanced turbocharging strategies
- Clean up the model and removing redundancies.

This is achieved by implementing a volume element for the outlet receiver, this technique is taken from the Diesel B model which uses similar volume elements. A volume element is a thermodynamic model for an open control volume which can have multiple fluid-flows entering and exiting the volume. In this case the fluid consists of a mixture of air and stoichiometric gas. Three conservation balances are applied on the control volume, namely: conservation of mass, conservation of mass of gas and conservation of energy. As a result the pressure, temperature and mass fraction of the control volume can be calculated.

The inputs for this control volume are the mass flow that enters and exits the volume. The mass flow that exits is determined by the turbine mass flow equation, which is already present in the current model and this is based on flow through a nozzle. The mass flow that enters the volume is determined by the cylinder air swallow and gas disposal characteristics. These are not modeled with enough detail to supply the control volume with its necessary input and thus a model has to be made to facilitate this input.

Furthermore, the implementation of the turbocharger group in the current model is too convoluted to be split up into multiple compressors and turbines for advanced turbocharging strategies. These will have to be cleaned up and restructured into individual component systems.

The next chapter provides an overview of the model implementation on a schematic level followed by a chapter that discusses each component model in more detail with the implemented formulas.

3. Overview

This chapter provides an overview of the overall model and its components. The reason for their implementation is explained but the components themselves are discussed in more detail in the next chapter.

3.1 Diesel Engine

The Diesel engine model itself consists of two models, namely the Fuel Pump and the Diesel Engine Core. These two models have been separated to isolate the characteristics of the fuel pump and the Diesel engine itself. However, since the fuel pump is an integral part of the engine itself it is incorporated into the engine model. Furthermore on this level there is no difference between the proposed Diesel A model and the current Diesel A model. Figure 3.1 shows the Diesel engine model sub level:

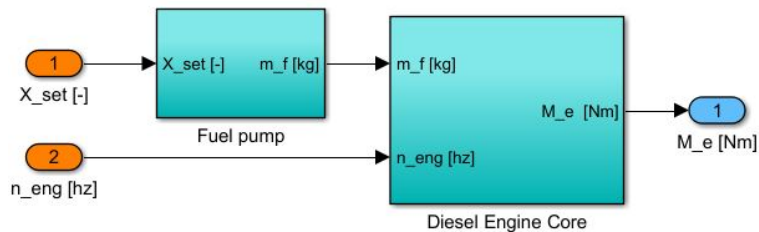


Figure 3.1: A schematic representation of the Diesel Engine sub model.

In figure 3.1, the orange ovals are the inputs and the light blue ovals are the outputs for the model that represents the engine (this color convention will be used throughout this report). The Fuel Pump takes a relative fuel rack position and converts this to the amount of fuel that is injected per cycle. The Diesel Engine Core takes this mass of fuel per cycle as an input together with its shaft speed and provides the brake effective torque as an output.

3.2 Diesel Engine Core

The Diesel Engine Core model holds all the components that make up the physical engine. There are four sub models present in the Diesel Engine Core, namely: the compressor (COM), the cylinder (CYL), the outlet receiver (OR) and the turbine (TUR). These components are connected as shown in figure 3.2:

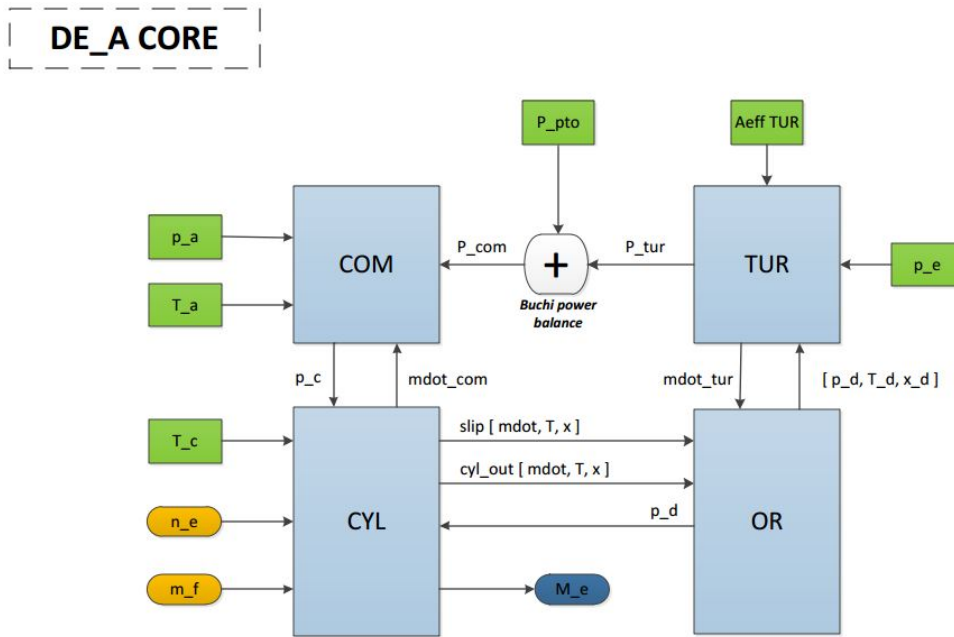


Figure 3.2: A schematic representation of the Diesel Engine Core sub model.

In figure 3.2, the green rectangles are constants that are implemented as signals. In some cases these constants can be replaced by variables to improve the model in the future, for example: the constant charge temperature T_c can be modeled with a heat exchanger to more accurately calculate its effects.

Starting with the cylinder sub model, this contains a few sub models that are explained in more detail in the next section. For now, it is enough to know what its interaction is with the compressor and outlet receiver. The cylinder contains a mass flow model that calculates the air swallow and gas disposal based on engine speed, charge conditions and scavenge conditions. The outcome of this calculation is the total amount of mass that is entering the cylinder and thus passing through the compressor. The second outcome is the mass flow out of the cylinder into the outlet receiver, this is partitioned into two separate streams (slip flow and flow out of the cylinder) each with their own mass flow, temperature and composition. The total amount of air into the engine is passed to the compressor sub model where it is used for the compressor power equation. The two streams that are exiting the engine are passed to the outlet receiver, where they are used to determine the gas properties of the gas inside the outlet receiver.

The outlet receiver is a volume element that is based on the conservation laws for the thermodynamic model of an open control volume. It takes the two streams that enter from the cylinder and the single stream that exits into the turbine and applies the conservation laws. As a result it is possible to calculate the pressure, temperature and composition of the gas inside the outlet receiver as a result of the mass and enthalpy entering and exiting the volume.

The turbine sub model contains two models: a mass flow model and a specific power model. The mass flow model is based on the equation of flow through a nozzle, known from fluid dynamics. In this case a special case has been used known as the "elliptical law". This mass flow model was also previously used in the version of the Diesel A model with the Bchi equation. This model provides the mass flow through the turbine based on the pressure ratio over the turbine and the density at the inlet (outlet receiver). This mass flow is used as an input for the outlet receiver as explained before and it is used to calculate the turbine power once the specific power is known. The specific power is calculated with the laws for polytropic expansion with heatloss. The product of the mass flow and the specific power is the available turbine power.

Before moving on to the compressor, it can be seen in figure 3.2 that the available power from the turbine

passes into a summation called the Bchi power balance. This is the power balance from which the Bchi equation is derived. The advantage of having the Bchi power balance in this form is that power can now be taken or supplied to the turbocharger shaft through an external source or load. This opens up a path for hybrid turbocharging where an electric motor/generator is connected to the turbocharger shaft.

Lastly the compressor sub-model; this is the model that makes the Diesel A able to operate without using compressor and turbine maps. It uses the same power calculations as the turbine but it does so in reverse. This can also be seen by its inputs which are the reverse of the turbine. Whereas the turbine takes in a pressure ration and determines the mass flow and available power, the compressor takes in the available power and imposed mass flow and determines what its pressure ratio should be in case the power and flow balances are in equilibrium. This estimated pressure ratio is than delayed with a first order time delay to simulate the turbocharger dynamics.

3.3 Cylinder

The cylinder sub model holds another four sub models, namely: the air swallow model, the gas disposal model, the Seiliger model and the Zinner blowdown model. These sub models and their interactions are shown in figure 3.3:

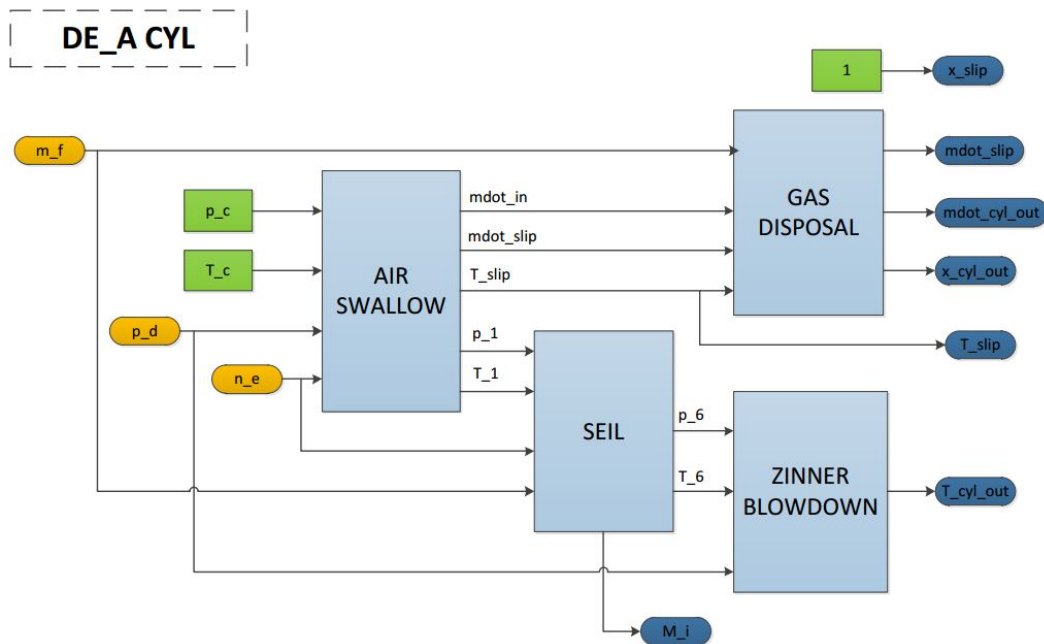


Figure 3.3: A schematic representation of the Cylinder inside the Diesel Engine Core sub model.

At the start is the air swallow model that takes the following inputs: inlet conditions (p_c and T_c), exhaust conditions (p_d) and the engine speed (n_e). From these inputs, the air swallow model determines the induction mass flow (dependent on inlet conditions and engine speed) and the scavenge mass flow (dependent on inlet and exhaust conditions). These two mass fractions together make up the total mass flow into the engine ($mdot_{in}$). With two simplifications the trapped mass and slip flow can be calculated. These simplifications are already present in the current Diesel A model and are as follows:

- $T_1 = T_{ind} = \text{constant}$
- $\eta_{sc} = \text{constant}$

A further simplification is added to provide the temperature for the slip flow (T_{slip}); there is no further heat pick up of the slip flow beyond the induction and thus: $T_{\text{slip}} = T_{\text{ind}}$. This can of course be improved upon in future versions.

The gas disposal sub model is a lot simpler than the air swallow model since most of the calculations have already been done in the air swallow model. The total mass out is just a summation of the mass in plus the added mass of the burnt fuel. Furthermore the slip mass was already calculated which is the complement of the mass leaving the cylinder to make up the total mass out of the cylinder. Lastly, the composition of the mass leaving the cylinder is dependent on the air excess ratio which can be calculated with the amount of fuel.

The Seiliger sub model holds the closed cylinder process including the heat release model. This sub model has not been changed for this proposal and so it is not discussed in detail here. The start of the closed cylinder process is provided in the form of the trapped pressure and trapped temperature. The end of the closed cylinder process (p_6 and T_6) is used as an input for the Zinner blowdown model. Furthermore, the mass of fuel and engine speed are used for the heat release model and to calculate cycle values, the result is the indicated shaft torque of the engine (M_i).

Finally, the Zinner blowdown model uses the relations for the blowdown phenomena as described by Zinner to calculate the blowdown temperature. This temperature is used as the temperature for the gasses leaving the cylinder ($T_{\text{cyl_out}}$)

4. Discussion

This chapter provides some discussion on the uses of the proposed model; Future possibilities that have opened due to the new implementation and some recommendations for future work.

4.1 Future Possibilities

The individual components of the turbocharging group have now been modeled as separate blocks with clearly defined parameters. This immediately opens up a path for the following turbocharging strategies:

- Variable Turbine Geometry (VGT), by continuously varying the turbine area.
- Sequential Turbocharging (STC), by sequentially varying the turbine area.
- Hybrid Turbocharging, by supplying or consuming power from the Bchi power balance.

By adding the outlet receiver and engine mass flow model there now exists a physical basis for the pressure buildup between the engine outlet and the turbine inlet. This means that the pressure buildup can be influenced by subverting mass from the outlet receiver through a different path than the turbine. In other words, the implementation for a waste gate is now available as shown in figure 4.1

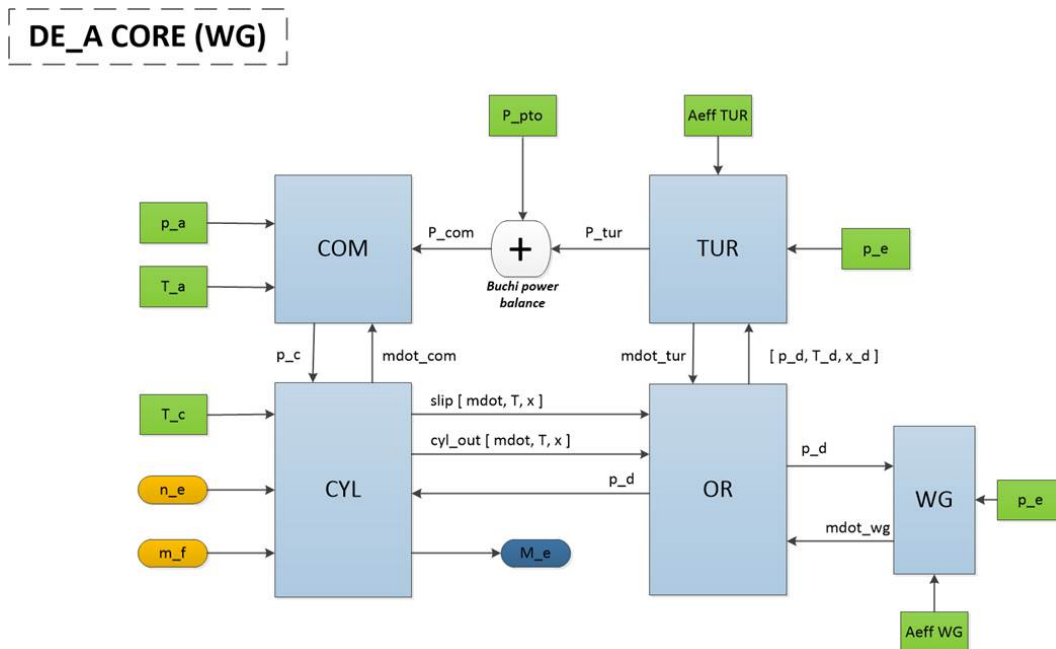


Figure 4.1: A schematic representation of the Diesel A Core with a waste gate

Previously it was not possible to simulate multi-stage turbocharging with the Diesel A model. It was

simply not possible to just multiply the pressure ratio's due to the application of inter-stage cooling. Combining the two advantages as described above (partitioning and the outlet receiver), it is now possible to create a multi-stage turbocharging engine with the Diesel A model. This is achieved by adding a second volume element between the low and high pressure turbine. The result is shown in figure 4.2:

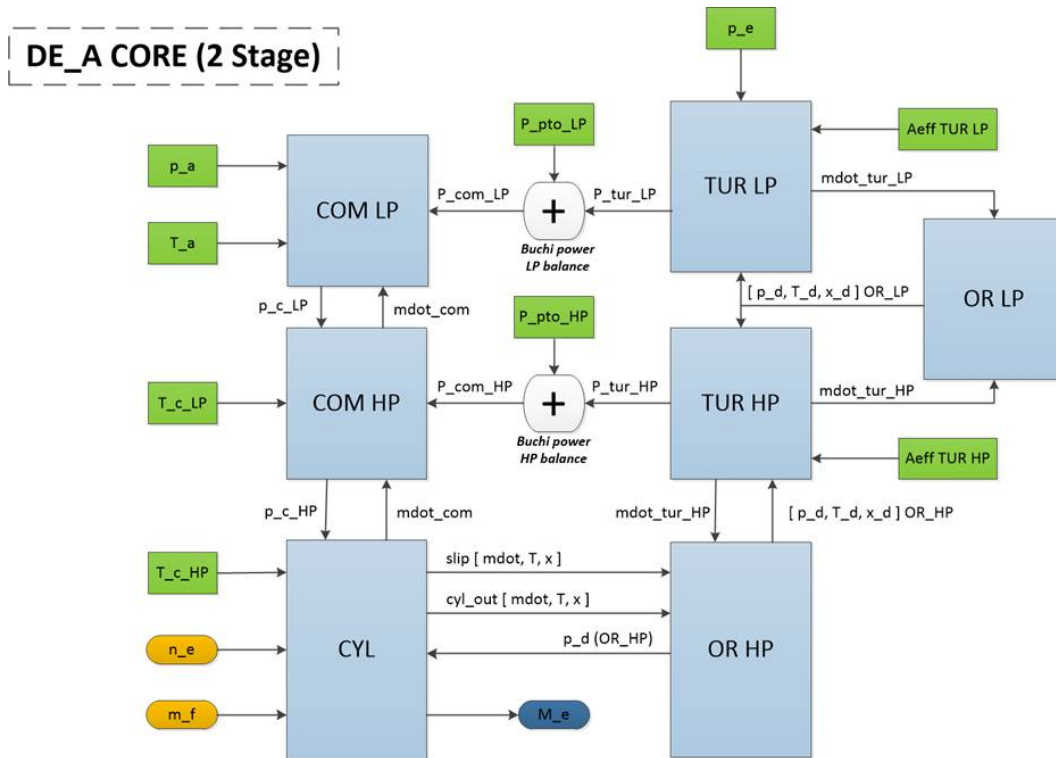


Figure 4.2: A schematic representation of the Diesel A Core with 2 stage turbocharging

The strategy shown in figure 4.2 can easily be altered to create a turbo compounding engine. By removing the low pressure compressor, the available power from the low pressure turbine can be used as if it was a free spinning power turbine.

4.2 Advantages

The advantages have already been mentioned above but are listed here once more for convenience:

- Complete re-write, Clean up and Component partitioning
 - Removes ambiguous definitions and redundancies.
 - Provides clear insight due to modeling of physical parameters.
 - Drastically improves knowledge transfer and usability.
- Outlet receiver and Engine mass flow models
 - Provides a physical basis for pressure ration over the turbine.
 - More detailed engine mass flow model and insight into flow partitions.
 - Modular approach makes it useful for different needs, see waste gate and multi-stage above.

4.3 Recommendations

The proposed model still uses some simplifications from the current Diesel A model to keep the over computational load low. Some of these are necessary while others are just used to keep things simple. Following are some recommendations and topics for future work:

- The engine mass flow model uses 2 simplifications: a constant imposed scavenge efficiency and a constant imposed trapped temperature. these can both be improved by improving the scavenge model. An alpha version of this scavenge model is already being worked on by the author.
- The turbocharger group holds three different efficiencies: compressor isentropic efficiency, turbine isentropic efficiency and the mechanical shaft efficiency. All three of these efficiencies are set to constants in the proposed model. These can be tuned to match bench data for a particular engine or a more generic approach can be investigated. For now, the implementation is good enough for the uses of the Diesel A model.
- The gas properties that are used in the outlet receiver are only dependent on species but not on temperature. These can be improved by using that gas property blocks from the Diesel B model. These blocks already exist but require the extra scripts that hold all the coefficients for the polynomial equations.
- Validation and sensitivity analysis of the model to investigate the behavior in comparison to the current Diesel A model and the current Diesel B model to compare the accuracy of the results in relation to the modeling effort (both parameter matching and computational load during simulation)
- Expand engine mass flow and scavenge model to include 2-stroke engines. Currently only 4-stroke engines are implemented. This will be included for the final version by the author.

5. Details

This chapter will be discussing all the individual blocks in detail and provide the formula as well as the sources for said formula.

5.1 Cylinder

Input

- T_c : Charge temperature [K]
- p_c : Charge pressure [pa]
- p_d : Outlet receiver pressure [pa]
- n_e : Engine speed [hz]
- m_f : Fuel mass per cycle [kg]

Output

- \dot{m}_{com} : Mass flow through the compressor [kg/s]
- \dot{m}_{slip} : Slip mass flow [kg/s]
- T_{slip} : Slip mass flow temperature [K]
- x_{slip} : Slip mass flow air mass fraction [kg/kg]
- \dot{m}_{cyl_out} : Mass flow out of the cylinder [kg/s]
- T_{cyl_out} : Temperature of the mass flow out of the cylinder [K]
- x_{cyl_out} : Air mass fraction of the mass flow out of the cylinder [kg/kg]
- M_e : Effective break torque [Nm]

As shown before, the cylinder holds several sub-models. However the cylinder model does not hold any equations itself with the exception of the declaration for the mass fraction of the slip flow ($x_{slip} = 1$), see figure 3.3 for the interconnections within the cylinder model.

5.1.1 Air Swallow

Input

- T_c : Charge temperature [K]
- p_c : Charge pressure [pa]
- p_d : Outlet receiver pressure [pa]
- n_e : Engine speed [hz]

Output

- \dot{m}_{in} : Total mass flow into the engine [kg/s]
- \dot{m}_{slip} : Slip mass flow [kg/s]
- T_{slip} : Slip mass flow temperature [K]
- p_1 : Trapped pressure [pa]
- T_1 : Trapped temperature [K]

The air swallow of an engine can be partitioned into different fractions that each provide a different or alternative view of the air swallow characteristics. The proposed model for the air swallow is based on the terminology used in Stapersma chapter 6: Fresh air supply to the cylinder. The different partitions are shown in figure 5.1:

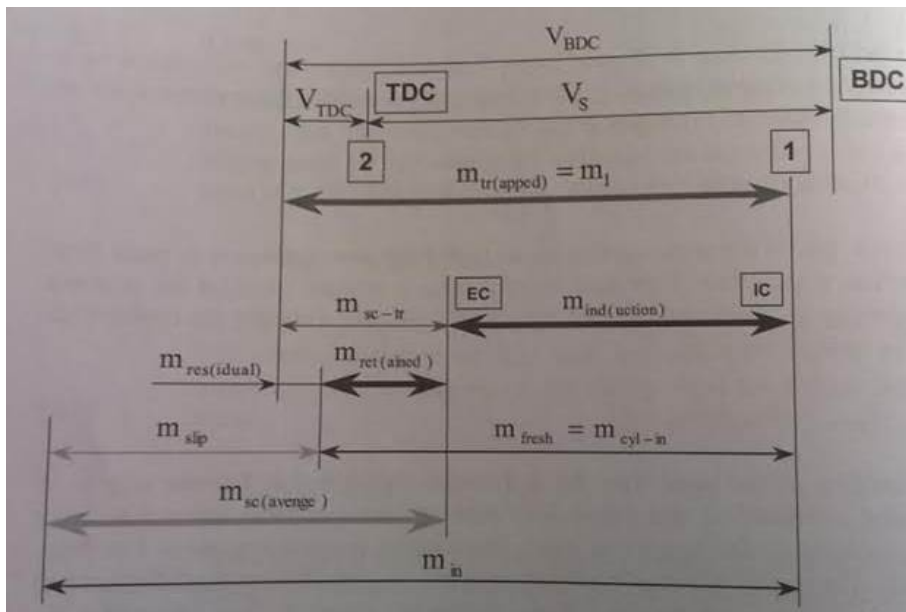


Figure 5.1: A schematic representation of partitions of the air swallow of a 4 stroke engine.

It can be seen in figure 5.1 that the total mass flow into the engine can be comprised of the induction mass flow and the scavenge mass flow:

$$\dot{m}_{in} = \dot{m}_{ind} + \dot{m}_{sc} \quad (5.1)$$

The induction mass is calculated using the ideal gas law, Stapersma [6.19]:

$$m_{ind} = \frac{p_{ind} \cdot V_{ind}}{R_{ind} \cdot T_{ind}} \quad (5.2)$$

Which for a turbocharged engine yields, Stapersma [6.52]:

$$m_{ind} = \frac{p_c \cdot (V_{IC} - V_{EC})}{R_{air} \cdot T_{ind}} \quad (5.3)$$

The induction temperature is a result of a parasitic heat pick up from the inlet port and channel, Stapersma [6.30]:

$$T_{ind} = T_c + \epsilon_{INL} \cdot (T_{INL} - T_c) \quad (5.4)$$

Finally multiplying the induction mass by the firing frequency yields the induction mass flow:

$$\dot{m}_{ind} = m_{ind} \cdot \left(\frac{i \cdot n_{eng}}{k} \right) \quad (5.5)$$

For the scavenge mass flow, the nozzle equation is used, Stapersma [6.120]:

$$\dot{m}_{sc} = i \cdot A_{sc,eff} \cdot \frac{p_c}{\sqrt{R_c \cdot T_c}} \cdot \Psi(\pi_{sc}) \quad (5.6)$$

The function for psi is modeled for non-choking and choking respectively, Stapersma [6.129] and [6.130]:

$$\Psi(\pi_{sc}) = \sqrt{1 - \frac{1}{\pi_{sc}}}; \text{ up to } \pi_{sc} < 2 \quad (5.7)$$

$$\Psi = \sqrt{1 - \frac{1}{2}}; \text{ from } \pi_{sc} \geq 2 \quad (5.8)$$

With both the induction and scavenge mass known, the two can be added to form the total mass going into the engine as well as passing through the compressor.

Next on the list are the trapped mass flow and the slip mass flow. First of it is necessary to find the trapped mass, Stapersma [6.54]:

$$m_1 = \frac{p_1 \cdot V_1}{R_1 \cdot T_1} = \frac{p_c \cdot V_{IC}}{R_{air} \cdot T_1} \quad (5.9)$$

This equation for the trapped mass requires the trapped pressure and temperature. For the pressure, all piping losses are neglected and so the trapped pressure is equal to the charge pressure. The trapped temperature is a little different, it is the result of mixing of the freshly induced cooler air and the remainder of the slightly hotter trapped scavenge air, Stapersma [6.57]:

$$T_1 = \frac{1}{\frac{1}{T_{ind}} \cdot \frac{V_{IC} - V_{EC}}{V_{IC}} + \frac{1}{T_{sc-tr}} \cdot \frac{V_{EC}}{V_{IC}}} \quad (5.10)$$

By applying the simplification that the temperature of the trapped scavenge mass is equal to the induction temperature, the resulting trapped temperature is equal to the induction temperature. This simplification is already used in the current Diesel A model:

$$T_{sc-tr} = T_{ind} \quad (5.11)$$

$$T_1 = T_{ind} \quad (5.12)$$

And so the trapped mass becomes:

$$m_1 = \frac{p_c \cdot V_{IC}}{R_{air} \cdot T_{ind}} \quad (5.13)$$

Multiplying by the firing frequency provides the trapped mass flow:

$$\dot{m}_1 = m_1 \cdot \left(\frac{i \cdot n_{eng}}{k} \right) \quad (5.14)$$

As stated before, the scavenge efficiency is imposed as a constant value; this is a simplification. The scavenge efficiency is used to find the amount of fresh trapped air, Stapersma [2.88]:

$$\eta_{sc} \equiv \frac{m_{fresh}}{m_1} \quad (5.15)$$

Finally the slip mass flow is calculated by subtracting the amount of fresh trapped air from the total amount of air entering the cylinder, figure 5.1:

$$\dot{m}_{slip} = \dot{m}_{in} - \eta_{sc} \cdot \dot{m}_1 \quad (5.16)$$

A final simplification is applied to the slip flow temperature. This is set equal to the induction temperature, in other words there is no further heat pick up to the slip flow during scavenging.

$$T_{slip} = T_{ind} \quad (5.17)$$

Summing up the final two outputs of the air swallow model, the trapped pressure and temperature:

$$p_1 = p_c \quad (5.18)$$

$$T_1 = T_{ind} \quad (5.19)$$

5.1.2 Seiliger

Input

- T_1 : Trapped temperature [K]
- p_1 : Trapped pressure [pa]
- m_f : Mass of fuel [kg/cycle]
- n_e : Engine speed [hz]

Output

- T_6 : Stage 6 temperature [K]
- p_6 : Stage 6 pressure [pa]
- M_i : Indicated torque [Nm]

The Seiliger block contains the heat release model and the six Seiliger stages. This block has not changed in the proposed model and so it will not be discussed in this report. The details on this block can be found in Stapersma chapter 3.

5.1.3 Gas Disposal

Input

- \dot{m}_{in} : Total mass flow into the engine [kg/s]
- \dot{m}_{slip} : Slip mass flow [kg/s]
- \dot{m}_{dot_f} : Mass flow of fuel [kg/s]

Output

- \dot{m}_{dot_slip} : Slip mass flow [kg/s]
- $\dot{m}_{dot_cyl-out}$: Cylinder out mass flow [kg/s]
- $x_{cyl-out}$: Air mass fraction cylinder out [kg/kg]

As with the air swallow in figure 5.1, the gas disposal of an engine can be partitioned into different components. The proposed model for the air swallow is based on the terminology used in Stapersma chapter 7: Foul gas disposal from the cylinder. The different partitions are shown in figure 5.2:

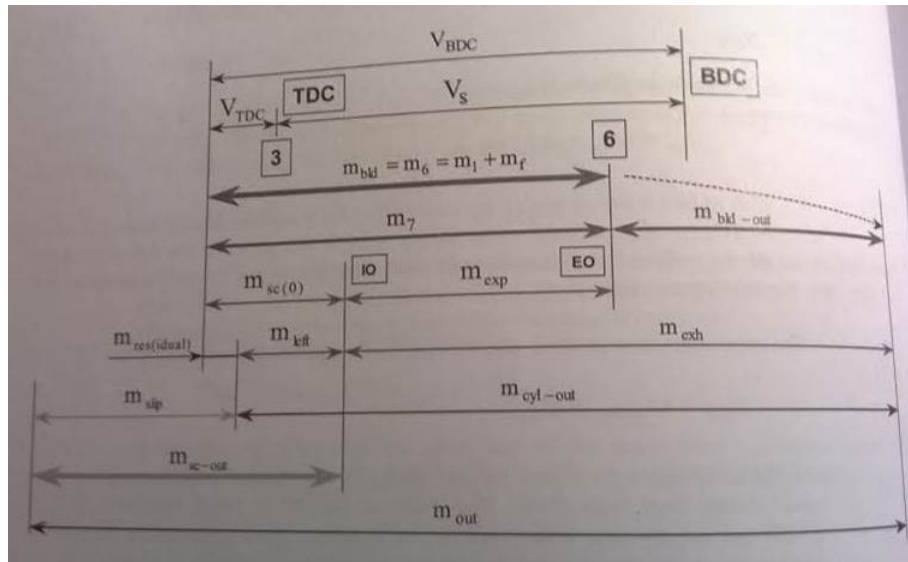


Figure 5.2: A schematic representation of partitions of the gas disposal of a 4 stroke engine.

It can be seen in figure 5.2 that the total mass flow out of the engine can be divided into the slip mass flow and the mass flow out of the cylinder [7.10]:

$$\dot{m}_{out} = \dot{m}_{slip} + \dot{m}_{cyl-out} \quad (5.20)$$

The total mass flow out of the cylinder is equal to the mass flow entering the cylinder plus the amount of burnt fuel that is injected during the closed cylinder process, Stapersma [7.12]:

$$\dot{m}_{out} = \dot{m}_{in} + \dot{m}_f \quad (5.21)$$

Combining the two equations above yields an expression for the mass flow leaving the cylinder based on the block inputs:

$$\dot{m}_{cyl-out} = \dot{m}_{in} + \dot{m}_f - \dot{m}_{slip} \quad (5.22)$$

The air mass fraction of the gas leaving the cylinder is determined by the amount of fuel that is injected/burnt and the amount of fresh air trapped in the cylinder. In other words, it is dependent on the air excess ratio:

$$x_{cyl-out} = 1 - \frac{1}{\lambda} \quad (5.23)$$

$$\lambda = (\lambda^*) \cdot \frac{1}{\eta_{sc} \cdot \eta_{pu}} = \left(\frac{\dot{m}_{in} - \dot{m}_{slip}}{\dot{m}_f} \cdot \frac{1}{\sigma_f} \right) \cdot \frac{1}{\eta_{sc} \cdot \eta_{pu}} \quad (5.24)$$

Finally the slip flow that is used as an output is the same as the slip flow that is received as an input. This value is passed through for aesthetic reasons in Simulink.

5.1.4 Zinner blowdown

Input

- T_6 : Stage 6 temperature [K]
- p_6 : Stage 6 pressure [pa]
- p_d : Outlet receiver pressure [pa]

Output

- T_bld : Blowdown temperature [K]

The blowdown ratio during Zinner blowdown is calculated with a polytropic relation with a constant polytropic exponent. This exponent is higher than the isentropic exponent to take into account the minor heat losses to the exhaust valve, Stapersma [7.33]:

$$\frac{T_{bld}}{T_6} = \frac{1}{n_{bld}} + \frac{n_{bld} - 1}{n_{bld}} \cdot \frac{p_d}{p_6} \quad (5.25)$$

Finally the Zinner blowdown temperature is calculated by multiplying the temperature ratio with the temperature at stage 6 of the Seiliger cycle:

$$T_{bld} = T_6 \cdot \frac{T_{bld}}{T_6} \quad (5.26)$$

5.2 Outlet receiver

Input

- mdot_tur : Turbine mass flow [kg/s]
- mdot_slip : Slip mass flow [kg/s]
- T_slip : Slip mass flow temperature [K]
- x_slip : Slip mass flow air mass fraction [kg/kg]
- mdot_cyl_out : Mass flow out of the cylinder [kg/s]
- T_cyl_out : Temperature of the mass flow out of the cylinder [K]
- x_cyl_out : Air mass fraction of the mass flow out of the cylinder [kg/kg]

Output

- p_d : Outlet receiver pressure [pa]
- T_d : Outlet receiver temperature [K]
- x_d : Outlet receiver air mass fraction [kg/kg]

The outlet receiver uses three conservation laws to balance the mass and enthalpy entering and leaving the volume. In the following equations the subscript "cv" will be used to denote the control volume, for the Diesel model this is interchangeable with the subscript "d" for the outlet receiver. In this model, the gas properties used for calculation are dependent on the species (their respective mass fraction) but not on the temperature:

$$x = \frac{m_{air}}{m_{air} + m_{gas}} = \frac{m_{air}}{m_{total}} \quad (5.27)$$

$$cp_d = cp_{air} \cdot x_d + cp_{gas} \cdot (1 - x_d) \quad (5.28)$$

$$cv_d = cv_{air} \cdot x_d + cv_{gas} \cdot (1 - x_d) \quad (5.29)$$

$$\kappa_d = \kappa_{air} \cdot x_d + \kappa_{gas} \cdot (1 - x_d) \quad (5.30)$$

$$R_d = R_{air} \cdot x_d + R_{gas} \cdot (1 - x_d) \quad (5.31)$$

The first conservation law is that of mass, the change of mass inside the volume is dependent on the mass flow into and out of the volume, Stapersma [B.71]:

$$dm_{cv} = dm_{in} - dm_{out} \quad (5.32)$$

Integrating the change of mass in the volume over time gives the total mass inside the volume:

$$m_{cv} = \int dm_{cv} \cdot dt \quad (5.33)$$

When considering perfect mixing of the gasses inside the volume, the amount of stoichiometric gas inside the volume is defined by the mass flows and the air fraction of said flows. This is the second conservation law:

$$dm_{g,cv} = dm_{g,in} - dm_{g,out} \quad (5.34)$$

Integrating in the same way as with the total mass gives the mass of gas in the volume:

$$m_{g,cv} = \int dm_{g,cv} \cdot dt \quad (5.35)$$

The outlet receiver has two streams entering (slip and cyl-out) and one stream leaving. For the particular application of the proposed Diesel A model, the above mass conservation laws above are defined be as follows (back flow or negative scavenging is not considered):

$$dm_d = \dot{m}_{slip} + \dot{m}_{cyl-out} - \dot{m}_{tur} \quad (5.36)$$

$$m_d = \int dm_d \cdot dt \quad (5.37)$$

And for the mass of gas:

$$dm_{g,d} = \dot{m}_{g,slip} + \dot{m}_{g,cyl-out} - \dot{m}_{g,tur} \quad (5.38)$$

$$dm_{g,d} = \dot{m}_{slip} \cdot (1 - x_{slip}) + \dot{m}_{cyl-out} \cdot (1 - x_{cyl-out}) - \dot{m}_{tur} \cdot (1 - x_{tur}) \quad (5.39)$$

The air mass fraction of the slip flow is set to unity and this is constant, so the first term on the right hand side disappears. The mass leaving the volume (tur) has the air mass fraction of the volume itself and so the above equation becomes much simpler:

$$dm_{g,d} = \dot{m}_{cyl-out} \cdot (1 - x_{cyl-out}) - \dot{m}_{tur} \cdot (1 - x_d) \quad (5.40)$$

And integrating over time yields the mass of gas in the volume, in the same way as with the total mass of gas in the volume:

$$m_{g,d} = \int dm_{g,d} \cdot dt \quad (5.41)$$

Finally the air fraction of the gas in the volume is calculated with the two masses found, total mass and mass of gas:

$$x_d = \frac{m_{air,d}}{m_d} = \frac{(m_d - m_{g,d})}{m_d} \quad (5.42)$$

In a general form the change in temperature of the gasses inside the volume is calculated by using the 1st law of thermodynamics, conservation of energy, for an open control volume, Stapersma [B.72]:

$$m_{cv} \cdot du_{cv} + (h_{out} - u_{cv}) \cdot dm_{out} - (h_{in} - u_{cv}) \cdot dm_{in} = \delta Q - \delta W \quad (5.43)$$

Eliminating work and heat transfer to the volume and moving the in- and outflow terms to the right side yields:

$$m_{cv} \cdot du_{cv} = (h_{in} - u_{cv}) \cdot dm_{in} - (h_{out} - u_{cv}) \cdot dm_{out} \quad (5.44)$$

By replacing the specific enthalpy and specific internal energy with specific heat at constant pressure and specific heat at constant volume multiplied by their appropriate temperatures, the previous equation becomes:

$$m_{cv} \cdot cv_{cv} \cdot dT_{cv} = (cp_{in} \cdot T_{in} - cv_{cv} \cdot T_{cv}) \cdot dm_{in} - (cp_{out} \cdot T_{out} - cv_{cv} \cdot T_{cv}) \cdot dm_{out} \quad (5.45)$$

When this is applied to the outlet receiver the following equation for the change in temperature is used:

$$m_d \cdot cv_d \cdot dT_d = dH_{slip} + dH_{cyl-out} - dH_{tur} \quad (5.46)$$

$$dH_{slip} = (cp_{slip} \cdot T_{slip} - cv_d \cdot T_d) \cdot dm_{slip} \quad (5.47)$$

$$dH_{cyl-out} = (cp_{cyl-out} \cdot T_{cyl-out} - cv_d \cdot T_d) \cdot dm_{cyl-out} \quad (5.48)$$

$$dH_{tur} = (cp_d \cdot T_d - cv_d \cdot T_d) \cdot dm_{tur} \quad (5.49)$$

$$dT_d = \frac{dH_{slip} + dH_{cyl-out} - dH_{tur}}{m_d \cdot cv_d} \quad (5.50)$$

The change in temperature is integrated to find the current temperature of the gasses inside the outlet receiver volume:

$$T_d = \int dT_d \cdot dt \quad (5.51)$$

With the mass, air fraction and temperature known; the pressure can be calculated with the use of the ideal gas law:

$$p_d = \frac{m_d \cdot R_d \cdot T_d}{V_d} \quad (5.52)$$

With this, all three outputs are documented. It can be seen how applying these basic conservation laws yields a physical basis for the outcome of the model; pressure, temperature and species make-up.

5.3 Turbine

Input

- p_e : Exhaust receiver pressure [pa]
- p_d : Outlet receiver pressure [pa]
- T_d : Outlet receiver temperature [K]
- x_d : Outlet receiver air mass fraction [kg/kg]
- A_eff_tur_r : Relative effective turbine area [0-1]

Output

- mdot_tur : Turbine mass flow [kg/s]
- P_tur : Available turbine power [W]

The turbine model consists of two separate models; the mass flow model and the specific power model. The mass flow model is based on the simplified turbine characteristic as described by Stapersma in chapter 8.2.2, Stapersma [8.56] (note that the effective area can be made variable to accommodate STC or VGT):

$$A_{tur,eff} = A_{tur,eff,relative} \cdot A_{tur,eff,nominal} \quad (5.53)$$

$$\dot{m}_{tur} = \alpha \cdot A_{tur,eff} \cdot \frac{p_d}{\sqrt{R_d \cdot T_d}} \cdot \Psi(\pi_{tur}) \quad (5.54)$$

The psi function is modeled approximated with the following equation, Stapersma [8.60]:

$$\Psi(\pi_{tur}) = \sqrt{1 - \left(\frac{1}{\pi_{tur}}\right)^f} \quad (5.55)$$

As a default starting point, f is chosen as 2 according to the elliptical law as given by [Dixon 1998].

The specific power model is using a constant isentropic efficiency and thermodynamic relations for polytropic expansion. starting with the turbine pressure ratio:

$$\pi_{tur} = \frac{p_d}{p_e} \quad (5.56)$$

From this, the isentropic temperature ratio can be determined:

$$\tau_{tur,s} = \pi_{tur}^{\left(\frac{\kappa_d - 1}{\kappa_d}\right)} \quad (5.57)$$

Dividing the inlet temperature by this ratio yields the isentropic exhaust temperature:

$$T_{e,s} = \frac{T_d}{\tau_{tur,s}} \quad (5.58)$$

From the definition of the isentropic turbine efficiency, the real exhaust temperature can be derived:

$$\eta_{tur,s} = \frac{h_1 - h_2}{h_1 - h_{2,s}} \equiv \frac{T_d - T_e}{T_d - T_{e,s}} \quad (5.59)$$

$$T_e = T_d + \eta_{tur,s} \cdot (T_{e,s} - T_d) \quad (5.60)$$

The specific work of the turbine is given by the specific heat and the temperature difference:

$$w_{tur} = cp_d \cdot (T_d - T_e) \quad (5.61)$$

Finally combining the mass flow and the specific heat yields the turbine power, where beta is the Zinner pulse correction factor, Stapersma [8.30]:

$$P_{tur} = \beta \cdot \dot{m}_{tur} \cdot w_{tur} \quad (5.62)$$

5.4 Compressor

Input

- T_a : Compressor inlet temperature [K]
- p_a : Compressor inlet pressure [pa]
- m_dot_com : Mass flow through the compressor [kg/s]
- P_com : Available compressor power [W]

Output

- p_c : Charge pressure [pa]

As mentioned before, in the Diesel A model the compressor acts as the inverses of the turbine. Its inputs are inverted but it uses the same equations, only in reverse. Whereas the process in the turbine started at a pressure ratio and yielded the power, the compressor starts with the available power from the turbine. The mass flow is also used as an input for the compressor block; from the available power and mass flow, the specific power can be derived:

$$w_{com} = \frac{P_{com}}{\dot{m}_{com}} \quad (5.63)$$

As before this specific power is the result of the product of the specific heat and the temperature difference over the compressor. And with a known entry temperature, this yields the real exit temperature of the compressor:

$$w_{com} = cp_{air} \cdot (T_b - T_a) \quad (5.64)$$

$$T_b = T_a + \frac{w_{com}}{cp_{air}} \quad (5.65)$$

With a fixed isentropic efficiency of the compressor, the isentropic exit temperature and isentropic temperature ratio can be derived from the real exit temperature:

$$\eta_{com,s} = \frac{h_{2,s} - h_1}{h_2 - h_1} \equiv \frac{T_{b,s} - T_a}{T_b - T_a} \quad (5.66)$$

$$T_{b,s} = T_a + \eta_{com,s} \cdot (T_b - T_a) \quad (5.67)$$

The isentropic temperature ratio can be used to retrieve the (Buchi) pressure ratio and charge pressure:

$$\tau_{com,s} = \frac{T_{b,s}}{T_a} \quad (5.68)$$

$$\pi_{com,Buchi} = \tau_{com,s}^{\left(\frac{\kappa_{air}}{\kappa_{air}-1}\right)} \quad (5.69)$$

$$p_{c,Buchi} = p_a \cdot \pi_{com,Buchi} \quad (5.70)$$

However, this whole process started by looking at the available power from the turbine. The charge pressure that was found in the previous equation is only valid in case the Buchi equation holds up, in other words: the compressor and turbine are in dynamic equilibrium. To differentiate this charge pressure from the real charge pressure, it is referred to as the Buchi charge pressure. The real charge pressure is derived from a first order time delay of the Buchi charge pressure to accommodate a dynamic imbalance between the compressor and turbine, for instance during acceleration and deceleration. The first order time delay is delay with a time constant τ_{TC_mech} to get the real charge pressure that is used by the rest of the model:

$$p_c = \int \frac{1}{\tau_{TC_mech}} \cdot (p_{c,Buchi} - p_c) \cdot dt \quad (5.71)$$