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Aalesund University College

# Master's degree thesis

IP501909 MSc thesis, discipline oriented master

Generic Marine Diesel Engine Modelling and Simulation Based on 20-sim

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Aalesund, 30/05/2014

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## Preface

This thesis is written as a master thesis at the Faculty of Marine Technology and Operations, Aalesund University College. The subject of the present report is to create a generic marine diesel engine model and simulate in 20-sim. According to the Master degree education subjects and courses, using bond graph to analyze different systems and simulate in 20-sim is one of the main disciplines.

The efforts are spent on researching the previous literature information and find out how the engine model is defined, creating a generic engine model which can be used in future projects, analyzing the simulation results and give explanations. Due to my bachelor background, I have the basic knowledge of marine diesel engine working principle. Part of what I learned is applied in the model. It is helpful to go back to the text book and review the knowledge. Engine model simplification helps to transfer the real engine to the model. If so many details are taken into consideration, it is hard to make the model work at the beginning. There is also some work focusing on data collection. To find realistic data helps apply the correct parameters to the model so that reasonable results can be received. The engine manual book includes a lot of information and data. Before the simulation, some expected guesses come up and compare them to the results. If it is similar to the assumptions, then the results are reasonable. If not, try to find the theoretical reason behind is necessary.

I am really appreciated that my supervisor Vilmar Æsøy's help. He not only offers helps with the academic knowledge, but also gives me a lot of inspiration and encouragement when I encounter problems. I am also grateful for the help from other staffs in the university college. Their enthusiastic assistance and patience help me solve some questions in some field which I am not familiar with. Also my classmates in Product and System Design class, they offer a nice environment and help each other.

Aalesund 30th  May, 2014

# Summary

This Master thesis is about creating a generic marine diesel engine model and simulating in 20-sim. The methodology is applying marine diesel engine working principle into bond graph. The first step is to build a single cylinder engine model consists of governor, cylinder, reciprocating components, crank shaft and load model. The basic bond graph elements are applied. When the single cylinder model is completed, due to the same working theory, connect 6 cylinders in series to build a 6-cylinder engine model. There are small adjustments such as crank angle differences are set up by submodels. The models are built separately both for two-stoke diesel engine and four-stroke diesel engine. The differences between these two types of engines are also explained.

To achieve creating a generic model, the model should fit other types of engines. By inputting some main data of engine such as bore diameters, stroke length and maximum cylinder pressure, the model can work as selected new engine. Scaling factors and scaled parameters are discussed in details.

There is a propeller model which was built by previous student is connected to the diesel engine model. The propeller selection is discussed as well, so that the engine model and the propeller model can match each other and work well. When the bond graph model is ready, input it and parameters into 20-sim and simulate it.

After receiving the simulation results, explanations are offered to each plot. The plots are reasonable and close to the assumptions. The simulation results prove that the engine model can be used in the future projects as a generic model.

Due to the limitation of time and ability, future work is listed so the future students or researchers can develop the model more perfect or integrate this model to other ship systems.

# Contents

# 1  Background

Ships and marine equipment for advanced marine operations form the core of the maritime cluster in the West of Norway. The industrial cluster is a world leader in the design of complex and highly customized vessels and equipment to the global market. Innovation, customization, proximity to customers and short delivery times are the keys to competitiveness. Highly oriented prototype, equipment and systems are optimized to fit unique requirements. While the complexity increases, the lead time decreases. Errors can lead to fatal incidents, delays and cost overruns. Therefore there is a need to develop better design methods and tools that allow efficient variant design and the fitting of new technology.

Designing, modeling, and simulating an integrated ship machinery system in 20-sim is part of the research project in Aalsund University College. This master thesis will focus on designing a modularized and generic marine diesel engine model in 20-sim, analyzing the simulation results and offer suggestions and methods to improve the engine performance. Because the engine model is generic, by inputting the engine parameters, we can receive the performance characteristic results easily. Also it is simple to connect this model with the other component models which are implemented, so that we can build up the integrated machinery system.

# 2  Scope of work

## 2.1  Model and simulate the marine diesel engine

Design and build a diesel engine is the basic request of this Master thesis. Combining the knowledge from multi-discipline during the education period, an engine model based on mechanical, dynamic, thermodynamics theories and principles will be created. We start with modeling one simplified cylinder of the engine and input the collected data from a specific engine, after the simulation we can receive the engine cylinder performance. By deriving more cylinders a complete engine model can be built and receive the performance of the whole engine system.

## 2.2  Build a generic model

Different engine has different construction and parameters, but create a new model for each engine will cost a lot of time and effort every time. So building a generic marine diesel engine is one of the aims of this thesis. Cauterizing the engine type, dimension, power, material of each component is one of the main tasks. If we want to achieve this aim, we have to find the common relation between different types of engines, for example, input the basic dimension data into a common formula or create a global parameter sub-model. Another method is to create a library in the 20-sim and it gives an opportunity to simulate both the exist system and the systems in the future, the models can be also reused in other projects. The point is to modularize each component of the engine to meet different requirement.

## 2.3  Interface design with other components

From the literature review we can see that some parts of the propulsion systems have been implemented. How to integrate them and the engine model in this thesis together into a complete propulsion system is also under consideration. Good interface design of the engine model can help to connect it easily with other components, so that the effort and flow from the system can be input into other systems, on the other hand the effort and flow from other systems can be input into this engine system. Modularized interface between the engine system with emission model, transmission model or between transmission model and propeller model integrate the whole system work together.

# 3 Literature review

Cristian Braia (2011), Modeling and Simulation of a Simple Propulsion Machinery [1]

This is a Mater thesis from previous graduate student. In this thesis, Cristian modeled and simulated a simple propulsion system. The main segments in this system are marine diesel engine and propeller. In the first part of the thesis, the author gave a brief introduction of ship propulsion system and some background knowledge about modeling, bond graph and simulation, which make the readers more clear. In the system modeling part, the author use bond graph built a simplified diesel engine in 20-sim, the model is shown below:
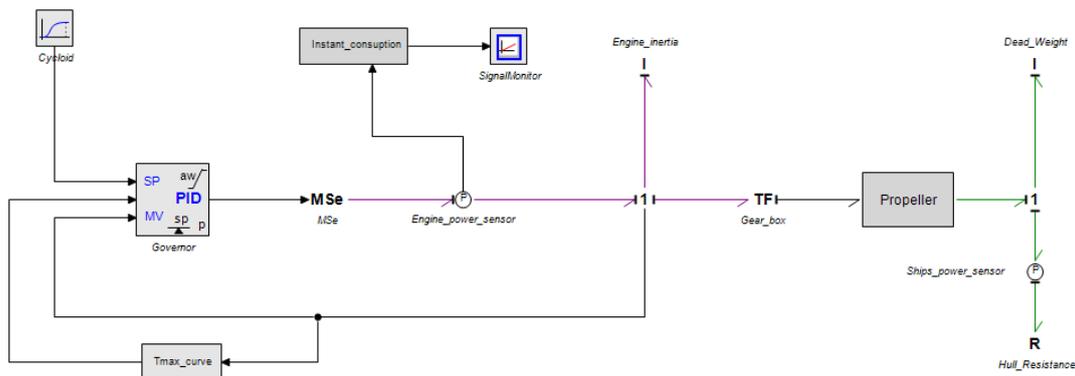


Figure 2.1 Propulsion model [1]

In this engine model, MSe is a modulated effort source which offers torque according to the speed imposed by Cycloid trough the governor. The Cycloid is the rotational speed controlled by human. The PID controller is a governor. Take the measured value from one junction which is the current speed and compares it with the set point from Cycloid. The error between them will be reduced to zero.

In the propeller part, the author introduced more detailed knowledge about the propeller, from different point of views such as propeller geometry, propeller performance characteristics. After these, a propeller bond graph model was input into 20-sim and simulated. By analyzing the plots, the propeller performance such as propeller coefficients and efficiency were explained.
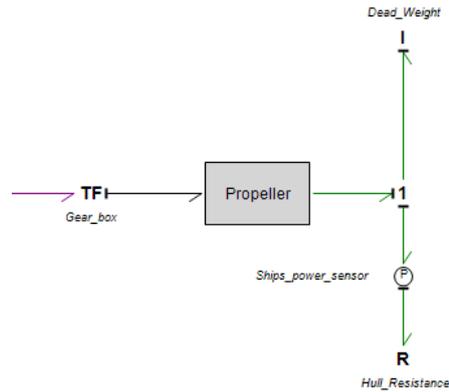
Figure 2.2 Propeller model [1]

This thesis focuses more on the propeller effects in the propulsion system. It can be introduced in the propeller part of the whole integrated machinery project. By connecting this modularized model with other parts, the propeller performance in a better organized propulsion system can be found.

Kevin Koosup Yum (2012), Modeling and Simulation of Transient Performance and Emission of Diesel Engine [2]

Nowadays the environmental foot print of industry is gaining more attention. The emission of marine engines such as nitrogen oxide ($NO_x$), sulfur oxide ($SO_x$), carbon dioxide emission ($CO_2$) and particulate matter (PM) influence the environment a lot. New regulations come up from IMO to limit the emission and also push award new green technology applied on the ship. This thesis is to develop a dynamic engine model and predict the emission from diesel engine in the generic manner. The model can capture the in-cylinder process during combustion. An overall process of the engine model building was completed.
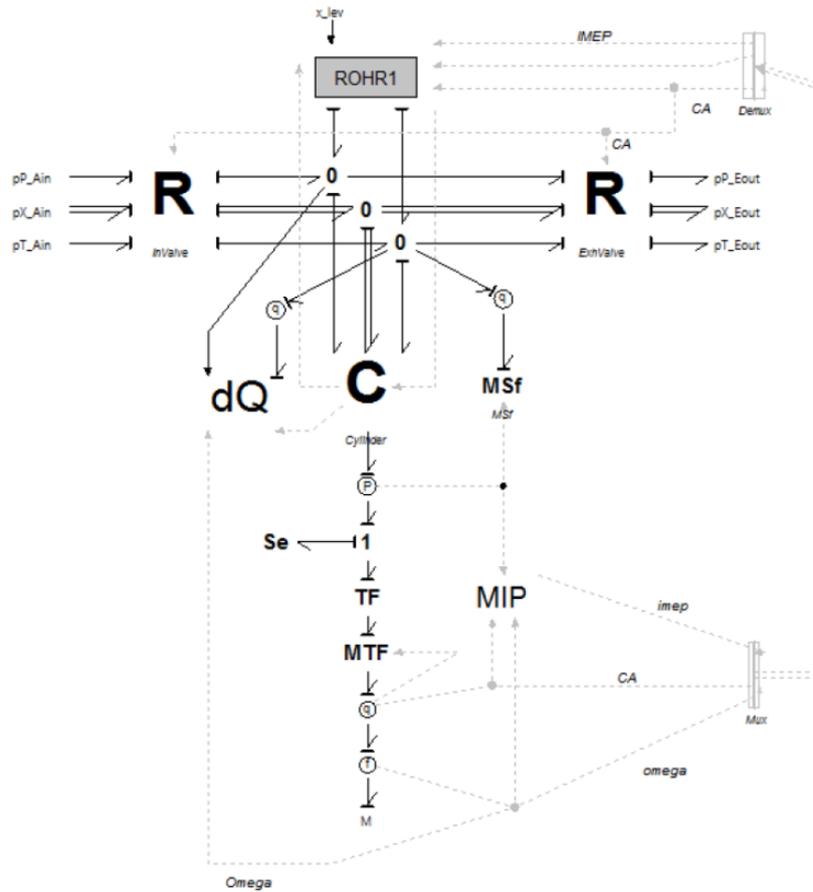
Figure 2.3 Engine emission model [2]

An implemented engine model was built. This model includes engine cylinder, intake and exhaust valve, heat loss, piping, turbocharger system and etc.

The other important section in the thesis is $NO_x$ emission model. Block Diagram of $NO_x$ Calculation is show below:
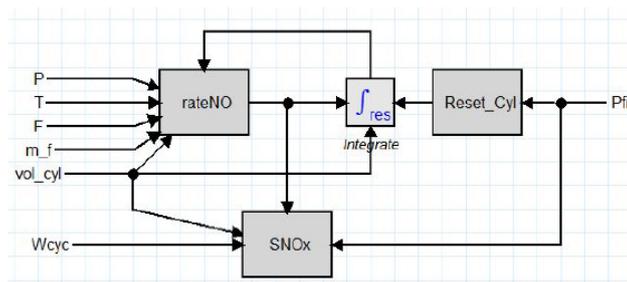


Figure 2.4 $NO_x$ emission model [2]

The cylinder control volume into the unburned zone and burned zone. Simple two zone model is considered and completed in conjunction by bond graph approach.
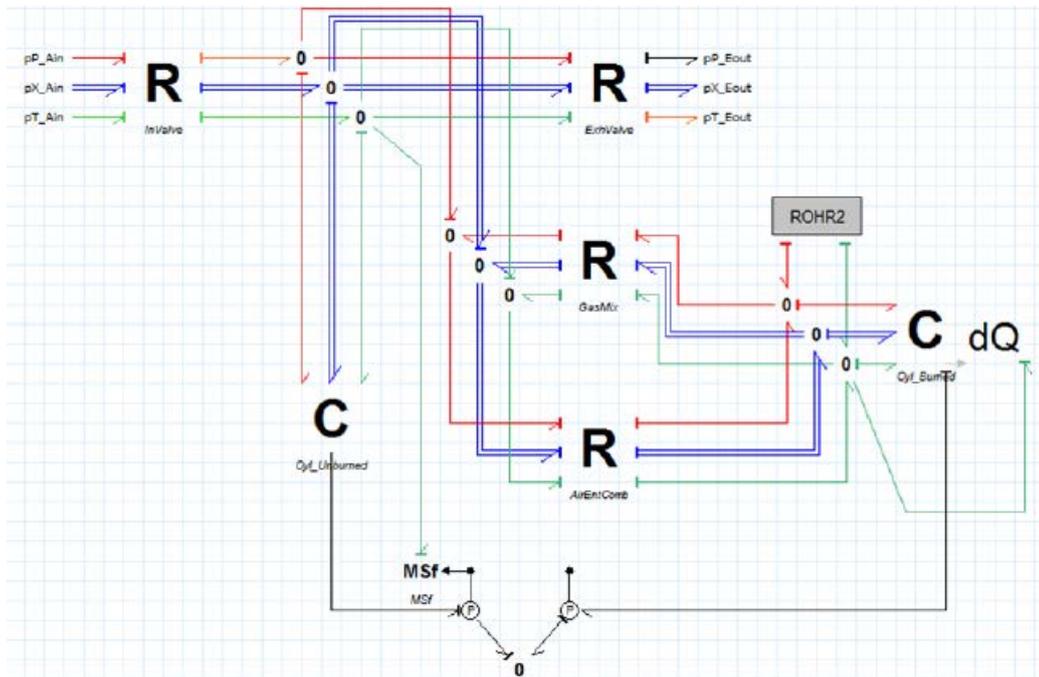
Figure 2.5 Cylinder zone model [2]

This model has some assumptions such as the fuel injection is assumed to be the same as the rate of heat release, the pressure is uniform for both zones, there is no heat exchange between zones and etc.

The engine test is done in the simulation part. By comparing and analyzing the plots, the author found the factors which result $NO_x$ formation and gave some suggestions which can help to reduce the emission.

A majority of the effort in the thesis is given to the calculation of the equilibrium composition of the combustion gas. The bond graph model of the engine is a dynamic mathematical model. The overall model was built with component libraries and could be used in other projects.

Egil Christoffer Sandbakken (2010), Marine Machinery System-Tools and Architecture [3]

The thesis presents tools and architecture regarding design of marine machinery systems in offshore supply vessels. It introduces the important aspects about the design based on a research study, and proposes a design methodology consisting of tools and architecture. In this thesis the author uses bond graph models machinery sub-systems and components, such as diesel engine, generator, electric motor, cooling system for diesel engine, governor, reduction gear, propeller and hotel loads. The model presents the components in a simplified marine machinery system configuration. The completed machinery system configuration should contain more components and it is important that it can be included at a later stage. The engine model is quite simplified. But this thesis creates a general view of the whole machinery system which we can develop each component into a more completed model in the future projects.

## MACHINERY SYSTEM
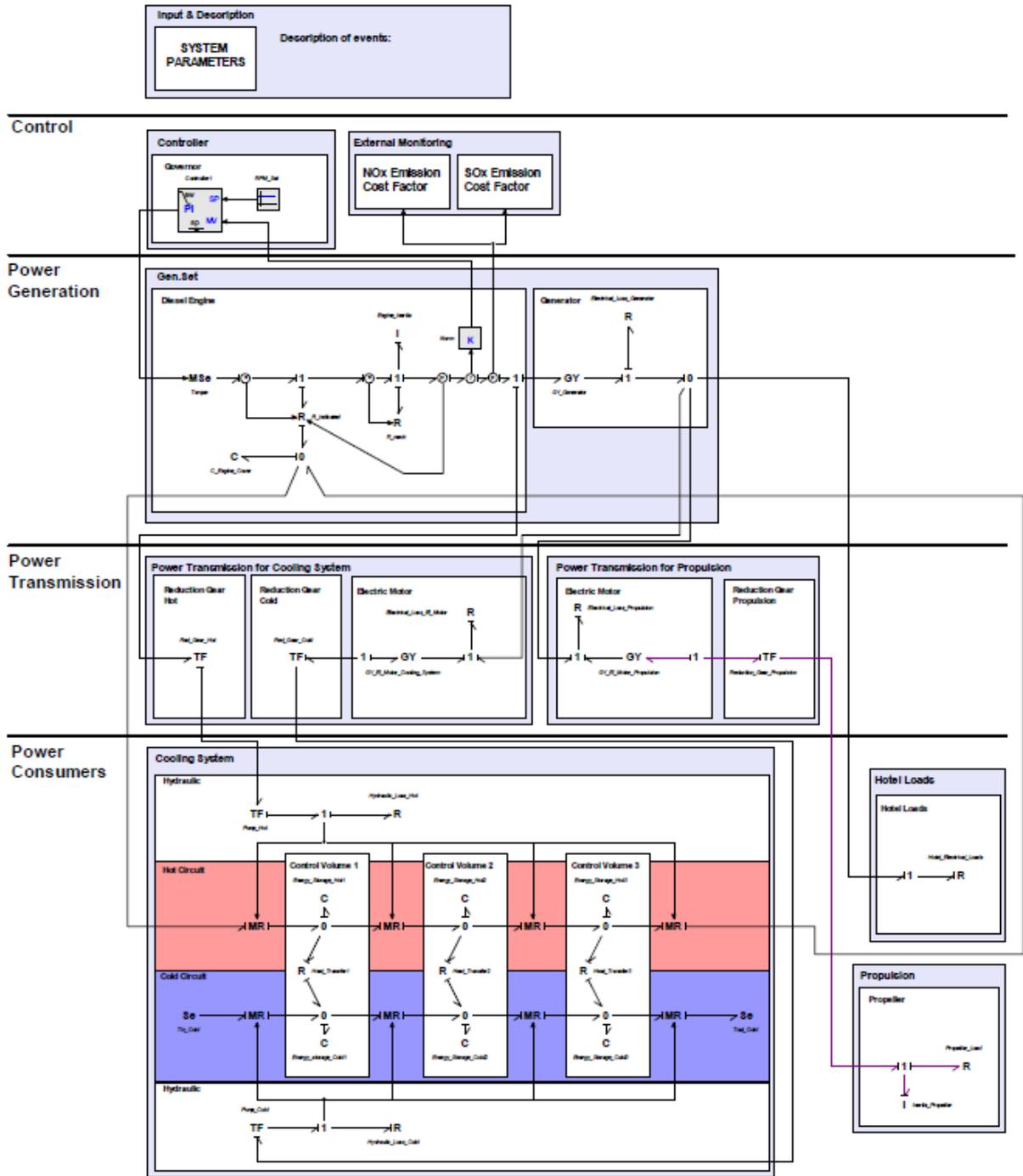


Figure 2.7 Machinery system model [3]

Jose J. Granda, Karl Sime, Computer Aided Modeling of the Four Stroke Internal Combustion Engine [4]

Computer aided modeling and simulation techniques can be used to perform dynamic analysis of a complex system. In this thesis the author uses internal combustion engine as an example to explain how to model and simulate the multi-energy system using bond graph and a computer simulation language. The internal combustion engine is a system that variety energy interrelations occur between hydraulic, thermal, and kinematic processes. This paper presents an analysis of the multi energy system using both bond graph and Pseudo bond graph model. The use of advanced modeling software such as the Computer Aided Modeling Program is illustrated. This paper is an old paper and during that time the computer aided modeling methods for bond graph was not well developed. So a lot of original methods were used for modeling and simulation. Some hand typing programs shows how the working principles are applied in the simulation software which is developed in the next few years since that time. This gives us a hint how to deal with the problems which can't be solved with present software or technology and how to use present methods to solve more difficult problems.

MAN B&W S60MC-C8-TII Project Guide, 7020-0110-00ppr Apr 2010 [5]

This paper is intended to provide the information necessary for the layout of a marine propulsion plant. It offers the general data for a typical large two-stroke marine diesel engine. The data includes the engine dimension, both the layout and load diagrams. It also offers introduction of specific systems such as lubricating oil system, cooling system, exhaust gas system and etc. The sorted out data can be input into the model and the result will be close to realistic. The input parameter in 20-sim is very expensive, if it is far from the realistic, even there is no problem with the model, it is also hard to receive the right results.

From the above literatures, the propeller, emission system are already been done by previous students. But the marine diesel engine model in their systems is quite simple or built from different theory. So building a generic engine model is necessary and it can be reused in future projects.
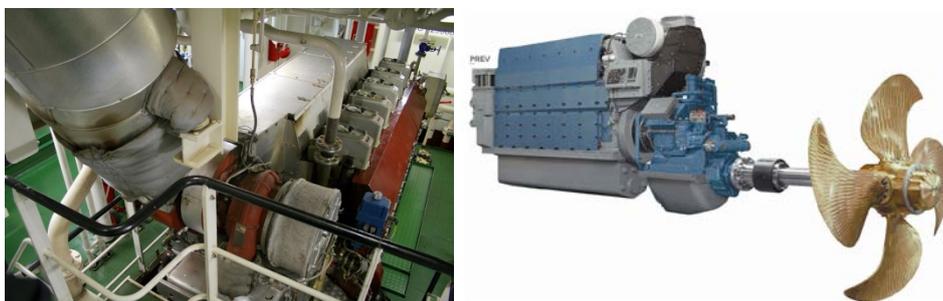


Figure 2.8 Ship propulsion system [6]

# 4　Pre-study & Methodology

## 4.1　Marine Diesel Engine

Selandia, the world's first oceangoing motor vessel, was put into service nearly a hundred years. Now the diesel engine plays an important role in ship propulsion markets. Diesel engines are categorized by many characteristics such as working principle, design, and speed of revolutions, power output, and fuel use. The entire combustion cycle is completed in one revolution of the crank shaft and consequently in two piston strokes is **two-stroke engine**. Two-stroke engines are normally very large sized engines which are without exception crosshead engines. They are usually used for ship propulsion and diesel power plants. The entire combustion cycle is completed in two revolution of the crank shaft and consequently in four piston strokes is **four-stroke engine**. They are high-speed or medium-speed diesel engines which are smaller and for driving generator sets, and also larger propulsion engines of up to 30MW shaft power all work according to the four-stroke principle.

### 4.1.1　Engine working principle

**Four-stroke**

The four stroke cycle is so called because it takes four strokes of the piston to complete the processes needed to convert the energy in the fuel into work. Because the engine is reciprocating, the piston must move up and down the cylinder twice, and therefore the crankshaft must revolve twice.

1.  Induction: The crankshaft is rotating clockwise and the piston is moving down the cylinder. The inlet valve is open and a fresh charge of air is being drawn or pushed into the cylinder by the turbocharger;
2.  Compression: The inlet valve has closed and the charge of air is being compressed by the piston while it moves up in the cylinder so the pressure and temperature increase. When the piston is approaching the top of the cylinder (known as Top Dead Centre or TDC) the pressure is over 100 bar and the temperature is over 500°C.
3.  Power: The fuel is atomized into very small droplets before the fuel is injected into the cylinder by fuel injector. These droplets heat up very quickly and start to burn when the piston passes over TDC. The expanding gas forces the piston move down in the cylinder, and turning the crankshaft. Work energy is being put into the engine in this stroke.
4.  Exhaust: The exhaust valve starts to open when the piston reaches the bottom of the cylinder (known as Bottom Dead Centre or BDC). As the piston moves up in the cylinder, the exhaust gases are expelled from the cylinder. As the Piston approaches TDC again the inlet valve starts to open and the cycle repeats itself. [7]

**Two-stroke**

The two stroke cycle is so called because it takes two strokes of the piston to complete the processes needed to convert the energy in the fuel into work. Because the engine is reciprocating, this means that the piston must move up and down the cylinder, and therefore the crankshaft must revolve once.

The induction and compression is the same as four-stroke engine. The useful energy from the burning fuel is expended while the piston moves down in the cylinder. At about 110° after TDC the exhaust valve opens and the hot exhaust gas starts to leave the cylinder. At about 140º after TDC the piston uncovers a set of ports known as scavenge ports. Pressurized air enters the cylinder via these ports and pushes the remaining exhaust gas from the cylinder in a process which is known as "scavenging". The piston now goes past Bottom Dead Centre (BDC) and begins to move up in the cylinder, closing off the scavenge ports. The exhaust valve then closes and compression begins

## 4.1.2 Engine parameters

**Engine power**

Power is the rate of doing work. In linear measure it is the mean force acting on a piston multiplied by the distance it moves in a given time. The force is the mean pressure acting on the piston. The mean indicated pressure can be derived by the formula:

$$p_i = \frac{W_i}{A * S} \tag{4.1}$$

$p_i$ is the indicated pressure;
$W_i$ is the indicated work;
A is the area of piston;
S is the stroke length of piston.
Then the power is:

$$\text{power} = p_i * A * S * \frac{n * m * i}{60000} \text{ (kW)} \tag{4.2}$$

n is the engine speed;
m is the number of stroke for each rev, m=0.5 for four stroke engine; m=1 for two-stroke engine.

The pressure-angle curve indicates the compression of the air fuel mixture in cylinder. Here is an example curve of four stroke engine. When the piston is moving up to TDC, the pressure in the cylinder increases, when the temperature reaches ignition point, combustion starts. The time between fuel injection and combustion is called ignition delay. The combustion causes the cylinder pressure increase and force the piston to move down to the bottom of stroke so that work is produced. The theory of two-stroke engine is similar to four-stroke engine.
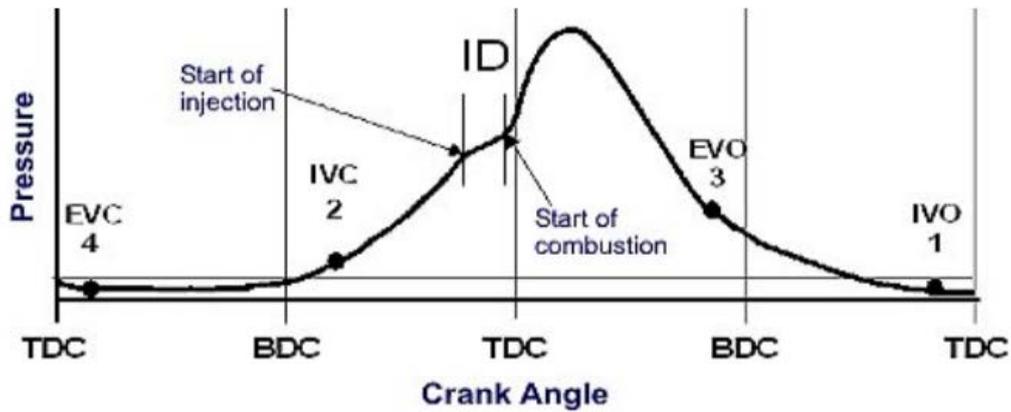
Figure 4.1 Pressure angle plot [8]

EVC-Exhaust valve closes;
IVC-Inlet valve closes;
EVO-Exhaust valve opens;
IVCO-Inlet valve opens;
ID-Ignition delay.
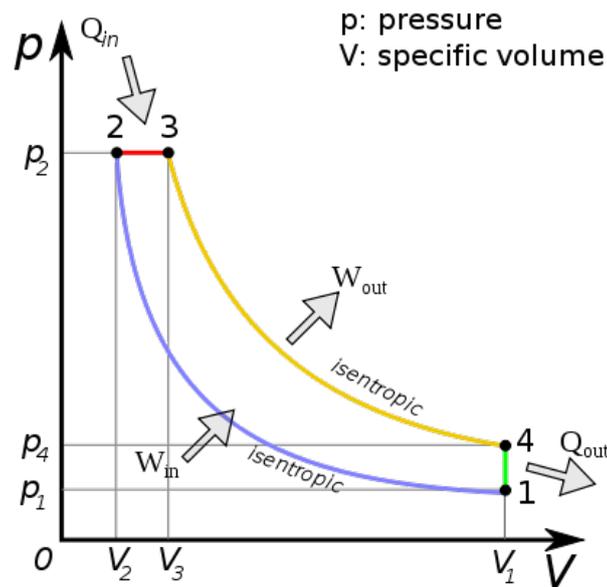


Figure 4.2 Engine P-V diagram ideal [9]

1-2: Isentropic compression of the fluid;
2-3: Reversible constant pressure heating;
3-4: Isentropic expansion;
4-1: Reversible constant volume cooling

The pressure-volume diagram describes the pressure change with volume in cylinder during a cycle. The above curve is an idealized cycle. But in real, the curve is more close to Figure 4.3
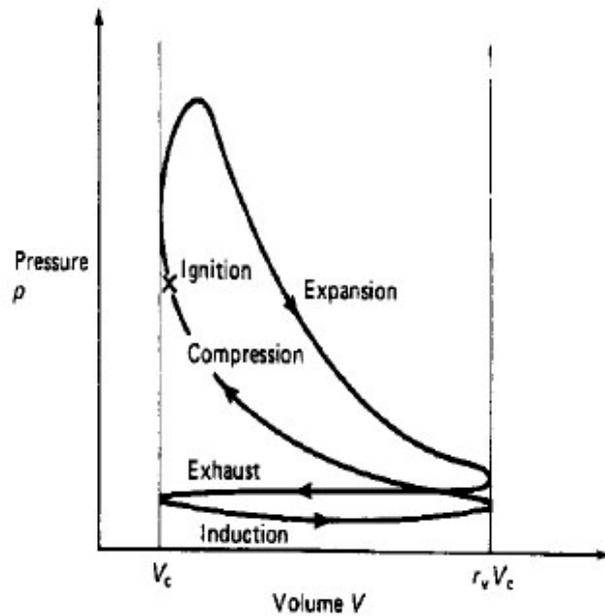
Figure 4.3 Engine P-V diagram [10]

**Mechanical Efficiency**

There are some losses when the engine is working, for example friction losses in the cylinders, bearings and together with the power consumed by engine-driven pumps and other auxiliary machines. The key issue in improving the mechanical efficiency of the engine is reducing friction on all moving parts. As shown in Figure 4.4, the main causes for frictional losses are the piston ring package and the bearing.
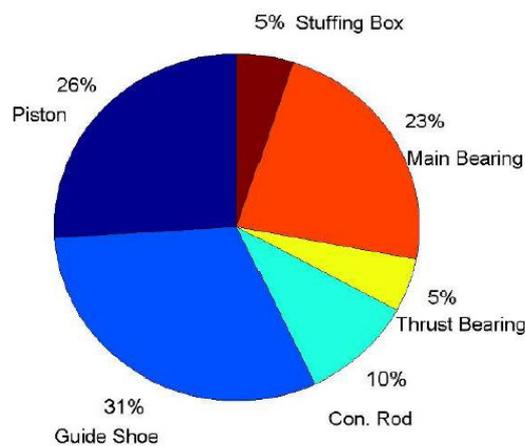


Figure 4.4 Friction loss chart [11]

The efficiency measurement can be done on the transmitted to the driven machinery by the crankshaft. In a propulsion system this can be measured by a torsion meter. Allowing for measurement error, these can be compared with figures measured on a brake in the test shop. There are two terms which are related to the mechanical efficiency:

<u>Indicated power</u>

Indicated power is the theoretical maximum output power of the engine. It is the total power available from the expanding of the fuel in the cylinders negating friction, heat loss or entropy within the system.

<u>Brake power</u>

Brake power is the power output at the drive shaft end without the power loss caused by gears, transmission, and friction.

$$Brake\ power = Torque*angular\ velocity$$

Indicated power is the sum of friction loss and brake power. The mechanical efficiency is the ratio of brake power to indicated power.

**Thermal efficiency**

The engine uses the heat to do work. But some heat exhausts so it cannot be used to do work. The law of thermodynamics constrains the operation of heat engine. The thermal efficiency is the overall measure of performance. It is equal to the ratio of heat converted into useful work to total heat supplied.
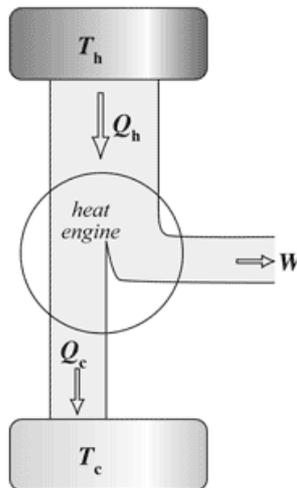


Figure 4.5 Thermal efficiency [12]

$$\varepsilon = \frac{W}{Q_h} = \frac{Q_h - Q_c}{Q_h} = 1 - \frac{Q_c}{Q_h} \qquad (4.3)$$

$$\varepsilon = \frac{T_h - T_c}{T_h} = 1 - \frac{T_c}{T_h} \qquad (4.4)$$

$\varepsilon$ is thermal efficiency
W is net work
$Q_h$ is heat flow in
$Q_c$ is heat flow out
$T_h$ is hot sink temperature
$T_c$ is cold sink temperature

But for the marine diesel engine, improving fuel efficiency is a main task to discuss. Here is an example of a ship engine energy flow.

12K98ME/MC with TES
SMCR : 68,640 kW at 94.0 r/min
ISO ambient reference conditions
TES : Single pressure (Dual pressure)

Total power output 54.2% (54.8%)

Shaft power          El. power production of
output 49.3%         TES    4.9% (5.5%)

Gain = 9.9% (11.2%)

Lubricating oil
cooler 2.9%

Jacket water
cooler 5.2%

Exhaust gas and
condenser
22.9% (22.3%)

Air cooler
14.2%

Heat radiation
0.6%

Fuel 100%
(171 g/kWh)
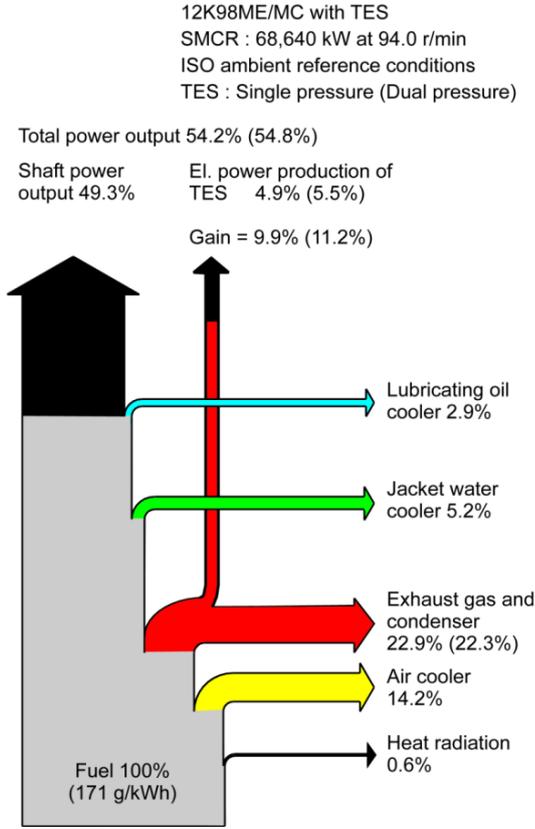
Figure 4.6 Engine thermal losses [13]

## 4.2  Bond graph

Bond graph is a graphical representation of physical dynamic system. It is a graphical approach to modeling in which component energy ports are connected by bonds that specify the transfer of energy between system components. Power is the universal currency of physical systems. Designing and analyzing the structure of a system with bond graph is easier, it is ideal for visualizing the essential characteristics of a system. The method of modeling focuses on interconnecting ideal basic elements where the interconnected elements will be able to predict the behavior of actual devices and systems within acceptable limits of accuracy. [14]

A power bond is considered to transmit power instantaneously and without loss of power from one port to another. The power direction is interpreted as power flowing in the direction of the half-arrow whenever the power is positive.

The signal bond indicates a signal flow at very low power. The notation for a signal bond is identical to that for a signal in a block diagram

In bond graph, the variables effort denoted by "e", and flow denoted by "f". There are three quantities associated with a power bond, effort, flow, and power. The power flowing into or out of a port is the effort multiplied by flow. Momentum is defined as the time integral of an effort and displacement is the time integral of a flow.

$$P(t) = e(t)f(t) = \text{power} \tag{4.5}$$
$$p(t) = \int e(t)\, dt = \text{momentum} \tag{4.6}$$
$$q(t) = \int f(t)\, dt = \text{displacement} \tag{4.7}$$

| Energy domain | Effort(e) | Flow(f) | Momentum(p) | Displacement(q) |
|---|---|---|---|---|
| Electrical | Voltage[V] | Current[A] | Flux linkage [Vs] | Charge [As] or [C] |
| Mechanical translation | Force[N] | Velocity[m/s] | Linear momentum[kg/s] | Distance [m] |
| Mechanical rotation | Torque[Nm] | Angular velocity[rad/s] | Angular momentum[N/ms] | Angle [rad] |
| Hydraulic | Pressure[Pa] | Volume flow rate[$m^3$/s] | Pressure momentum [$N/m^2$s] | Volume[$m^3$] |
| Thermal | Temperature[K] | Entropy flow [J/s] | Not defined | Entropy[J] |

Table 4.1Variables identification [14]

**Basic 1-port Elements**

<u>Source</u>

Energy sources are used to model energy supply to a system. The power is provided by an energy reservoir external to the system and ideal sources are used to describe interactions between the environment and system, or to describe an effort or a flow which is constrained to have a prescribed value.

$$\text{Se} \xrightarrow[\quad f(t) \quad]{\quad e(t) \quad} \qquad \text{Sf} \xrightarrow[\quad f(t) \quad]{\quad e(t) \quad}$$

Figure 4.7

<u>Capacitor Element</u>

Capacitor elements are used to store energy, and most systems can store energy in two forms, for example in mechanical systems they are potential and kinetic energy. Consider the 1-port device in which a single valued static constitutive relationship exists between an effort and a displacement variable. It is called C-element.

$$\xrightarrow[\quad f(t) \quad]{\quad e(t) \quad} \text{C}$$
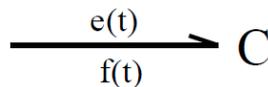
Figure 4.8

The capacitor is characterized by a constitutive relation of the form:

$$q = \Phi_c(e) \qquad (4.8)$$

In the general case,

$$q = C * e \qquad (4.9)$$

In linear case, C is a constant parameter. It is an idealization of physical components such as spring, capacitors and liquid storage tanks.

<u>Inertia Element</u>

The other energy storage element come into being when we find a single valued static constitutive relationship between a momentum p and a flow f. It is called I-element. In linear case I is a constant parameter. The inertia element is an idealization of physical objects such as a mass, inertia effects in hydraulic systems or inductance in electrical systems. The positive power direction is into the element as shown:

$$\xrightarrow[\quad f(t) \quad]{\quad e(t) \quad} \text{I}$$
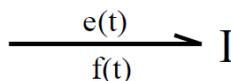
Figure 4.9

$$p = \Phi_I(f), \; p = I * f \qquad (4.10)$$

Resistor Element

Another 1-port element is a single valued static relationship between an effort e and a flow f, and is called R-element. It handles power dissipation

$$e = \Phi_R(f) \tag{4.11}$$

In general case,

$$e = R*f \tag{4.12}$$

The R-element is used to model all kinds of energy dissipation such as mechanical and hydraulic friction and electrical resistors.

**Basic 2-port Elements**

The 2-port elements exchange energy via two power bonds, we assume they are ideal in that they neither store nor dissipate but transmit power instantaneously from one bond to the other.

Transformer:

The transformer does not create, store or dissipate energy. It conserves power and transmits the factors of power with proper scaling as defined by the transformer modulus, such as a mass less ideal lever. It is assumed that the lever is rigid, which means a linear relationship can be established between power variables at both the ends of the lever.

$$\xrightarrow[f1]{e1} \overset{m}{TF} \xrightarrow[f2]{e2}$$
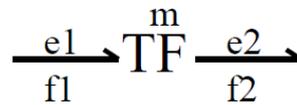
Figure 4.10

$$e_1 = m e_2 \tag{4.13}$$

$$m f_1 = f_2 \tag{4.14}$$

The m is transformer modulus.

Gyrator:

A gyrator transforms an effort on one bond into a flow on the other bond. The simplest gyrator is a mechanical gyroscope.

$$\xrightarrow[f1]{e1} \overset{r}{GY} \xrightarrow[f2]{e2}$$
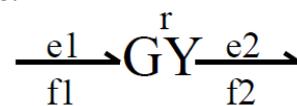
Figure 4.11

The r is the gyrator modulus.

$$e_2 = mf_1 \qquad (4.15)$$

$$r*f_2 = e_1 \qquad (4.16)$$

**Junction Elements**

In bond graph junctions can connect two or more bonds. There are two types of junctions, the 0 and the 1 junction. They conserve power and are reversible. They represent system topology and also the underlying layer of junctions. An ideal junction is considered as elements neither store nor dissipate energy in bond graphs, but only transmit it.
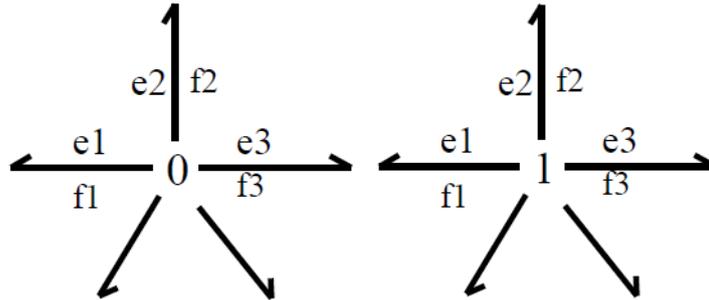


Figure 4.12

The key property of a 0-junction is that it defines a unique common effort on all bonds adjoining the junction.

$$e_1 = e_2 = e_3; \; f_1 + f_2 + f_3 = 0 \qquad (4.17)$$

It means that a 0-juntion has equal effort on all bonds adjoining and the algebraic sum of all flows equal to zero.

The 1-juntion is a multiport element where effort and flow are interchanged compare to the 0-junction.

$$f_1 = f_2 = f_3; \; e_1 + e_2 + e_3 = 0 \qquad (4.18)$$

The 1-juntion has equal flow on all bonds adjoining and the algebraic sum of all the efforts are equal to zero.

# 5   System modeling

## *5.1   Modeling & Simplification*

Modeling is an essential and inseparable part of scientific activity to make a particular part or feature easier to understand, define, quantify, visualize, or simulate. It consists of selecting and identifying relevant aspects of a situation in real world and also using different types of models for different purposes.  In this Master thesis, bond graph belongs to mathematical modeling of physical systems. The model is built from simplified single cylinder and connected in series. There are two types of model, two-stroke and four-stroke engine, but the theory is similar.
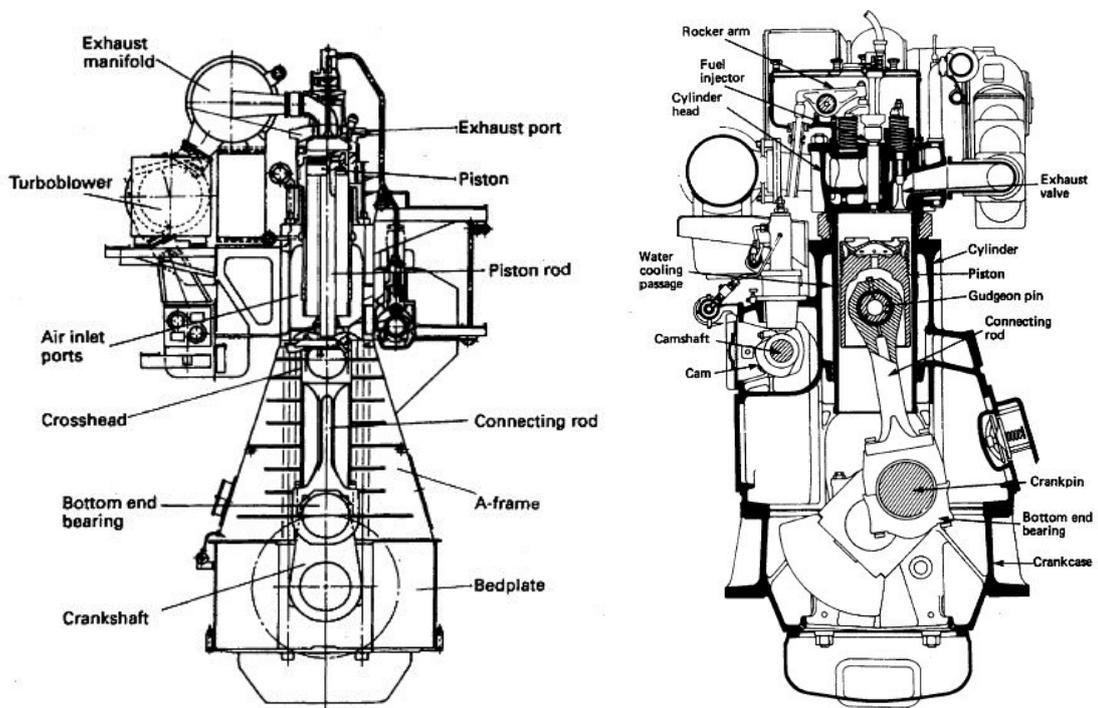
Figure 5.1 Engine structure [15][16]

## 5.2 Two-stroke engine

The engine model reference is MAN S60MC-C low speed two-stroke diesel engine. The main data of the engine is shown in the table below:

| Cylinder bore | 600mm |
|---|---|
| Piston stroke | 2400mm |
| Speed | 105rpm |
| Piston speed | 8.4m/s |
| Power output | 14280kW |
| Mean effective pressure | 20bar |
| Fuel specification | Fuel oil-ISO 8217, category ISO-F-RMK 55 |

Table 5.1 Engine data [17]
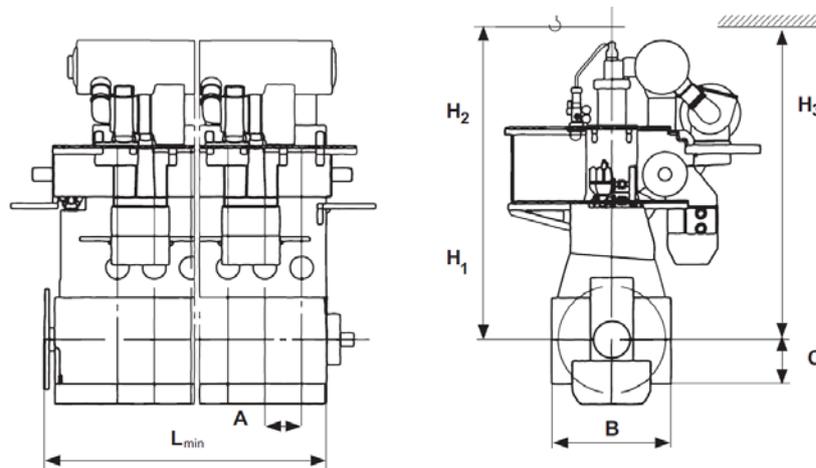


Figure 5.2 Engine dimensions [17]

Dimensions (mm) and weight (tons)

| $L_{min}$ | C | B | A | $H_1$ | $H_2$ | $H_3$ | Weight |
|---|---|---|---|---|---|---|---|
| 8271 | 1300 | 3770 | 1020 | 10700 | 10050 | 9800 | 368 |

Table 5.2 Engine dimensions [17]

### 5.2.1 One cylinder model

In order to build a complex 6-cylinder engine model, the first step is to simplify the system and start from building one cylinder model. Since the working condition and working principle is similar in each cylinder, if the one cylinder model works well, then just connect the cylinders in series and make small adjustment and the 6-cylinder can be created. So focus on the one cylinder model building is the basic but the most important process in general. If there is problem in the future complex model or optimizations want to be achieved, we always have to come back to the one cylinder model and test.

The overview of one cylinder structure is shown below:
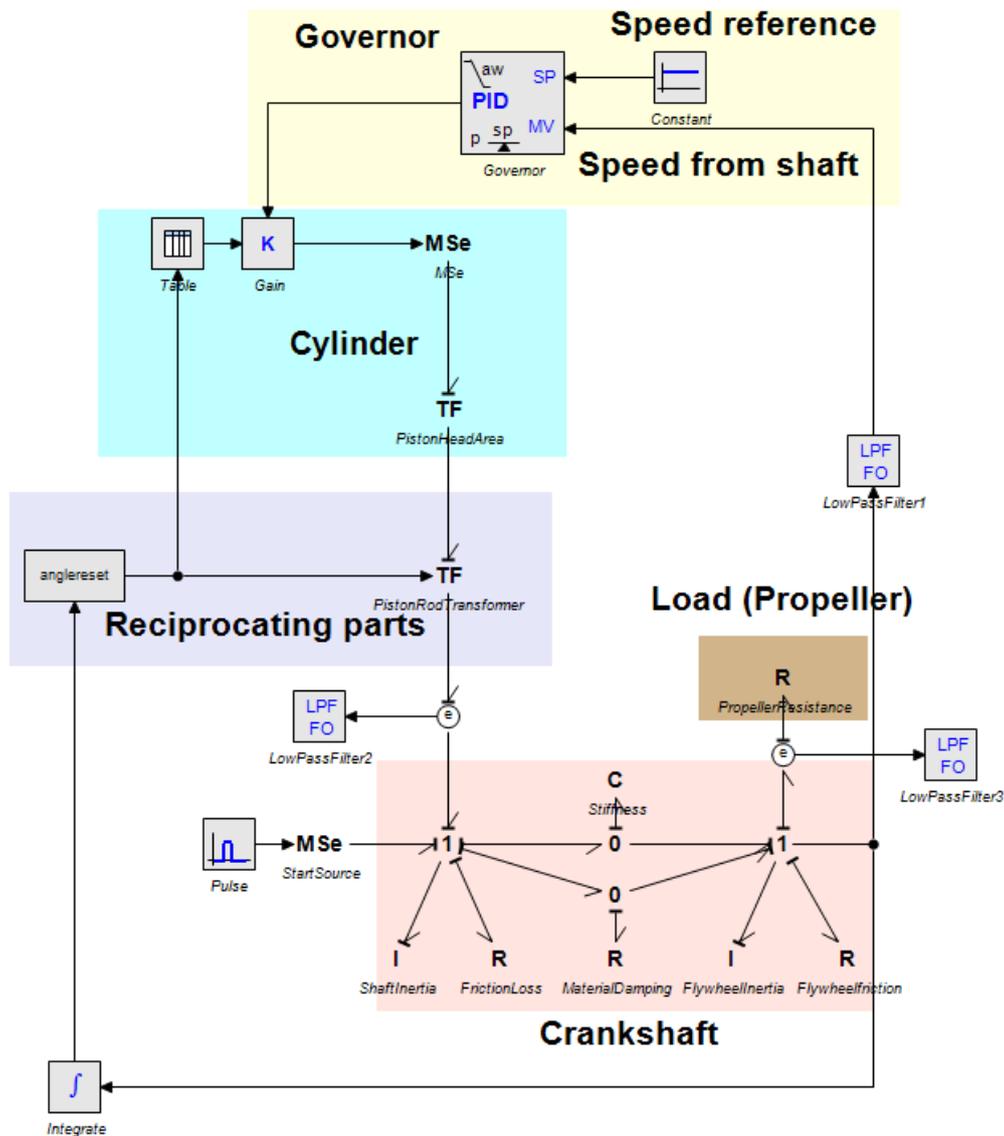


Figure 5.3 Two-stroke single cylinder engine model

The model consists of a governor, reference engine speed constant, and pressure of cylinder, piston in the cylinder, crank shaft, propeller resistance and a start source. On some bonds the Low Pass Filters are applied. The detailed information will be explained.

**Governor**

In "Modeling and Simulation of a Simple Propulsion Machinery" and "Marine Machinery System-Tools and Architecture" the governor of the engine were built by a PID controller, and the same method is used in this model.
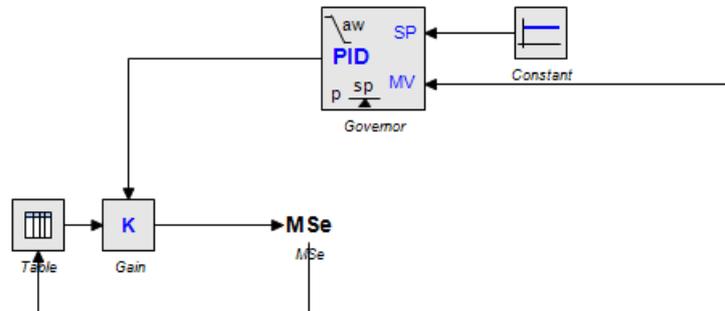


Figure 5.4 Governor model

PID control means Proportional, Integral and Derivative control. A standard control loop is shown in the figure below. A controller tries to steer a process in a way that minimizes the difference between a given set point and the output of that process [18].
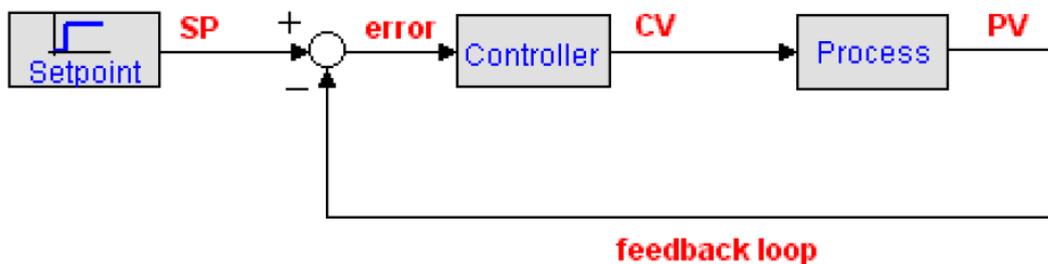


Figure 5.5 PID control loop [18]

Setpoint

The setpoint (SP) is where the process output should match. In this model the setpoint is the reference engine speed and it can be found in the manual book.

Process

The process is the system that should be controlled. The process has at least one impute to control its behavior and at least one output that is a measure of its behavior. The process input is connected to the controller variable (CV). It is mostly the input for an actuator that can drive the process to a desired setpoint. The process output is commonly known as process variable (PV), or measured variable (MV).

Error

The error is defined as the difference between setpoint and the process output. It is a measure for how much the process deviates from the desired setpoint, and thus how much

the controller should respond to get the process back to the desired setpoint. In the engine model, the error is the difference between the reference engine speed and the real engine speed measured from the shaft.

Proportional Control

The proportional action of a controller generally refers to the proportional gain. The action means that the controller moves in proportion to the error between setpoint (SP) and measured variable (MV):

$$\text{controller output} = K * \text{error} = K * (SP - MV) \tag{5.1}$$

Integral Control

With integral action, the controller output is proportional to the amount of time the error is present. Integral action eliminates offset that remains when proportional control is used.

$$\text{controller output} = \frac{1}{T_i} * \text{int(error)} \tag{5.2}$$

where the parameter Ti is called the integral time. Integral action is also known as reset and the parameter Ti as reset time.

Derivative Control

With derivative action, the controller output is proportional to the rate of change of the measurement or error. Some manufacturers use the term rate or pre-act instead of derivative. Derivative, rate and pre-act are the same thing. The controller output is calculated by the rate of change of the error with time.

$$\text{controller output} = T_d * \frac{d(\text{error})}{d(t)} = T_d * \frac{d(SP-MV)}{d(t)} \tag{5.3}$$

where the parameter Td is called derivative time.
In summary, the PID controller equation can be described as:

$$\text{controller output} = K * (SP - MV) + \frac{1}{T_i} * \text{int}(SP - MV) + T_d * \frac{d(SP-MV)}{d(t)} \tag{5.4}$$

The equations in the controller are shown in Appendix A
The reference speed of this low speed two-stroke engine is 105 rpm, which is 10.99 rad/s according to the engine manual book.

**Cylinder pressure**

In real engine, the power is from firing the fuel and the chemistry energy transfers to mechanical energy. A Table is created according to the engine data from experiment and test. The signal of engine speed output comes out from shaft end, one branch is input into the controller and another is converted to angle by integrating the angular velocity and input into the angle reset. The purpose of angle reset is to keep the angle value between 0 and $2\pi$. The input of Table is the angle and the output is the cylinder pressure. The two-

stroke engine pressure-angle curve is shown below:



## CYLINDER PRESSURE VS ANGLE
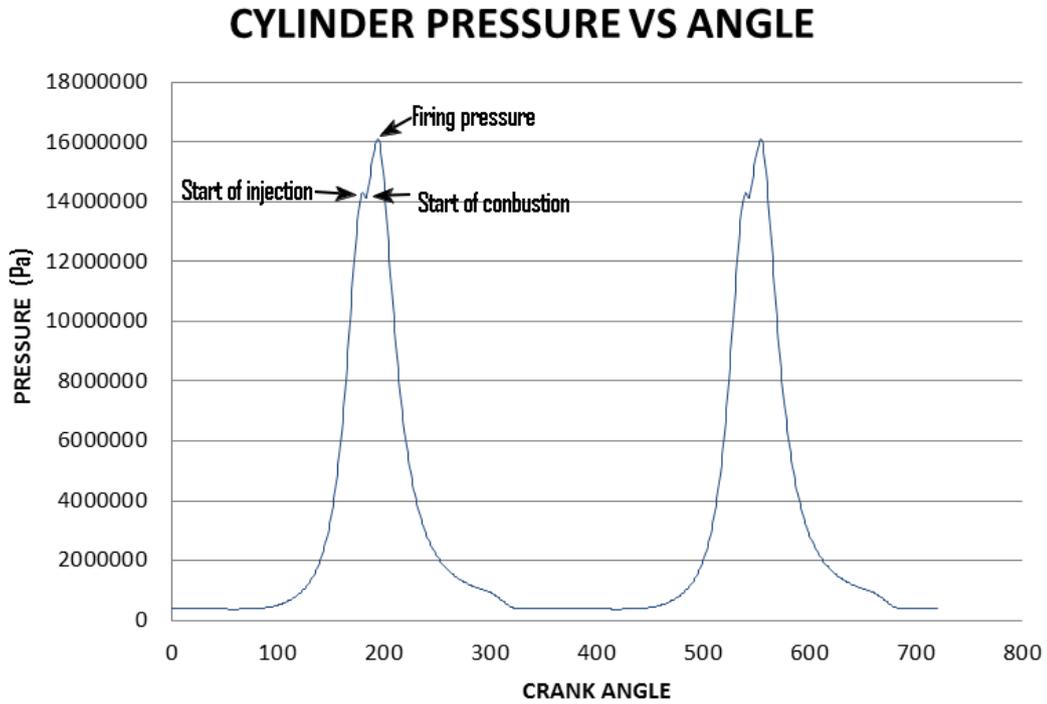
Figure 5.6 Pressure-crank angle curve

The curve shows the injection starts at the Top Dead Center (TDC) and the combustion is a little delayed. The maximum pressure happens at firing pressure which is 16.1MPa and it is after the TDC. This data is input into the Table so the engine can start working if an initial start pulse is applied.

**Piston & Connecting rod**

In most previous work these parts were simplified so it is very difficult to modularize the model and fit different types of engines. If a generic model wants to be built, it should be easy to input the parameters such as the bore diameter and stroke. MSe is the effort source which represents the pressure in the cylinder related to rotation angle. Before that there is a Gain element which is used to scale the pressure curve and also controlled by the signal from controller.
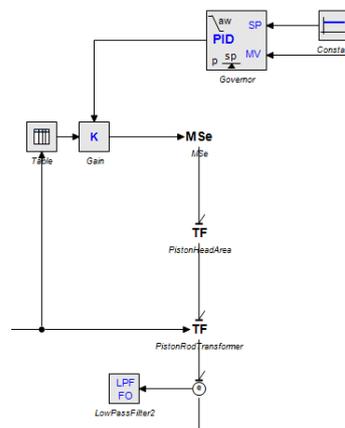
Figure 5.7 Piston and connecting rod model

The function of transformers can be obviously explained in the following structure:
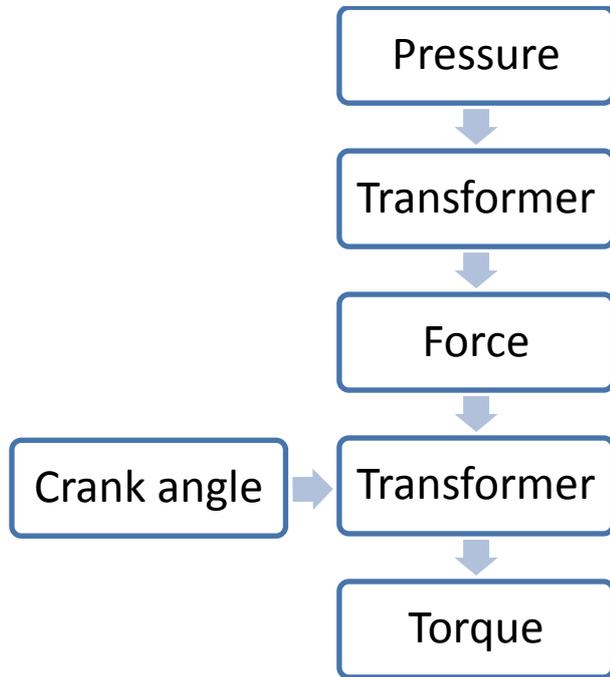


Figure 5.8 Transformer function

The first TF is a transformer which changes the pressure into force. The transformer modulus is the area of piston so it all depends on the bore diameter.

$$A = \frac{\pi D^2}{4} \tag{5.5}$$

A is the piston area, D is the bore diameter. This value can be scaled in the generic model and it will be discussed later. The s60mc-c engine bore diameter is 0.6m.
The second TF is the connecting rod transfers the force into torque. Reciprocating translational motion of a piston is converted to crankshaft rotation by a crank and slider mechanism.
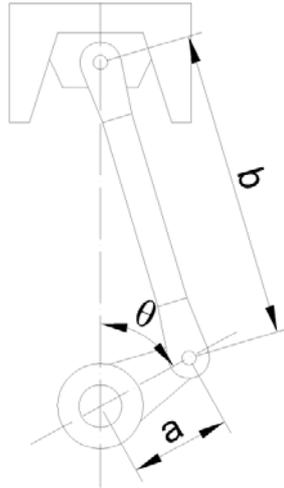
Figure 5.9 Piston-crank mechanism

The relation between the force on the piston and the twisting torque T on the crank is:

$$\left(-\mathrm{a} * \sin\theta - \frac{l}{2}\frac{a^2\sin\theta\cos\theta}{\sqrt{b^2-a^2\sin^2\theta}}\right) * F = T \tag{5.6}$$

Where F=Pressure*Piston area
So the piston rod length and connecting rod length are the input parameters. In the s60mc-c engine a=1.2m and b=2.5m.

**Crankshaft & Flywheel & Propeller**

In "Modeling and Simulation of a Simple Propulsion Machinery" crankshaft is not important compares to the propeller model, so the shaft model is inadequate. In "Marine Machinery System-Tools and Architecture", the author uses only a GY to model an electric motor on the shaft. And other papers the crankshaft is simplified by a bond. Since there is some torsion and energy loss happen on the crankshaft, we can build a model which includes the friction loss on the bearing and energy capacitor of the shaft. The model is shown below:
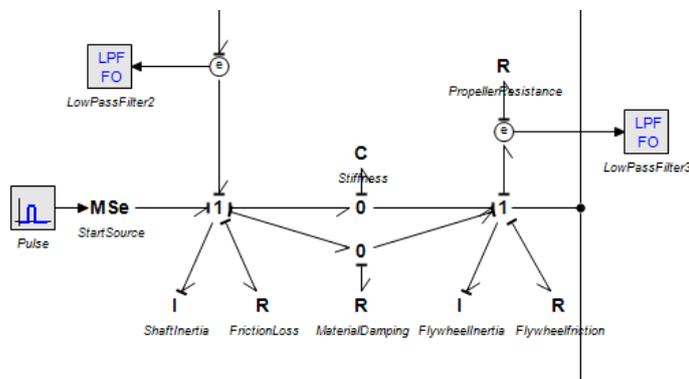


Figure 5.10 Crankshaft model

<u>Inertia:</u>

The I-element represents the moment of inertia. Moment of inertia is the mass property of a rigid body that defines the torque needed for a desired angular acceleration about an axis of rotation. It depends on the shape of the body axes of rotation. The amount and distribution of its mass also affect the inertia.

The crank shaft can be simplified as a disc with thickness rotating along the shaft line. The formula is $I = \frac{mR^2}{2}$ where m includes the mass of connecting rod, mass of crank webs and crank throw. R is the crank radius and the value is half of the stroke. Due to the complex shape of crankshaft, more detailed discussion of crankshaft moment of inertia is in chapter of scaling factor.

<u>Friction loss:</u>

R-element on the 1-junction is the friction loss. The parameter of R is received by comparing the cylinder output torque and the shaft end output torque. From the experience the friction loss is about 5%. Adjusting the R until the shaft end output torque is 95% of the total cylinder output torque. In this case R is 550.

<u>Material damping:</u>

The R-element on the shaft represents the material damping. The material damping [19]:

$$R = t * 2\sqrt{km} \qquad (5.7)$$

Where k is the young's modulus and k=210*109 (N/m2) [20], m is the mass and it can be calculated according to the dimension of the shaft, t is the material damping ratio of steel and t=0.015[20].

<u>Stiffness</u>

The C-element on the shaft is known as the compliance or flexibility, while the inverse of C is the stiffness, which is often denoted by k=1/C.
The compliance for a solid circular shaft can be expressed as [14]:

$$C = \frac{32L}{\pi D^4 G} = \frac{2L}{\pi(\frac{D}{2})^4 G} \qquad (5.8)$$

Where G is the modulus of elasticity in shear and G=77*109 (N/m2) [20], L is the length of shaft and L=0.17m, D is the diameter of the shaft and d=0.7m. The same as inertia, stiffness calculation of crankshaft is more complex than above but more details will be introduced in scaling factor part

**Flywheel**

The flywheel usually is at the end of crankshaft. It has large inertia so that it can help to reduce vibration by smoothing out the power stroke as each cylinder fires. It also can control the speed constant or stay in smooth rate. The flywheel inertia is usually lager than

the crank shaft, by comparing different data from different type of engines we assume that it is about 5 times larger than the crankshaft.

**Load model (Propeller)**

The propeller is modelled as a resistor. The torque created by the shaft is transferred to the propeller and hull to push the ship forward. There is a propeller model which is created by the previous student, and it will be connected to the engine model in the next chapter. So here the propeller is only simplified as R-element to make the model work.

### 5.2.2  Series connections (6-cylinder)

Since the one-cylinder engine model is completed and the working principle is similar in each cylinder, the 6-cylinder engine can be built by connecting the model in series. The firing order in 6-cylinder engine should be taken into consideration.
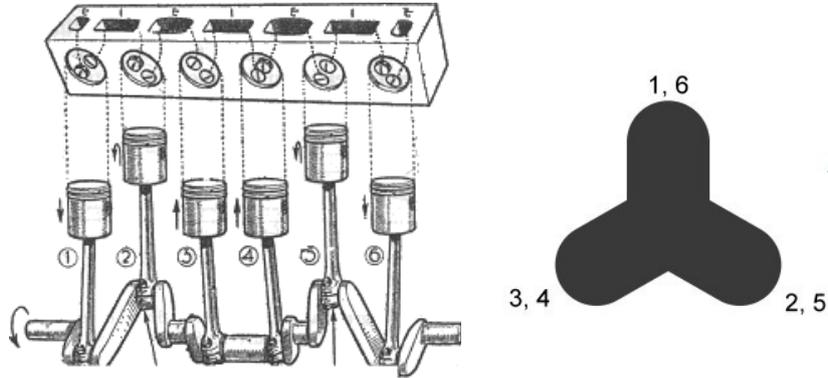


Figure 5.11 Firing order [21]

In the 6-cylinder engine, the pistons number 1 and 6, 2 and 5, 3 and 4 move as pairs. Theoretically there are several possible firing orders, and 1-5-3-6-2-4 is applied in this case.

In two-stroke diesel engine all cylinders complete the work with one flywheel revolution and the four-stroke engine all cylinders complete the work with two. Therefore, 6-cylinder **two-stroke** ignition intervals angle = 360°/ 6 (cylinders) = **60°**; while **four-stroke** engine ignition intervals angle = 720°/ 6 (cylinders) = **120°**.

So the submodel of angle reset is created to make sure the value is between 0 and 2π and distinguish the angle difference between each cylinder.
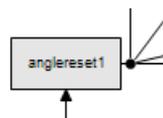


Figure 5.12

The input of the submodel angle reset is the angle from shaft end, and the output of the submodel is:

$$\text{output} = \text{input} - k * 2 * \pi \qquad (5.9)$$

where $k = \text{floor}\left(\frac{\text{input}}{2\pi}\right)$;

It means when the value of angle from the shaft end is over 2π then it becomes 0 again. For the other submodels the angle difference between each other is expressed in the equations. The details are in the Appendix A.
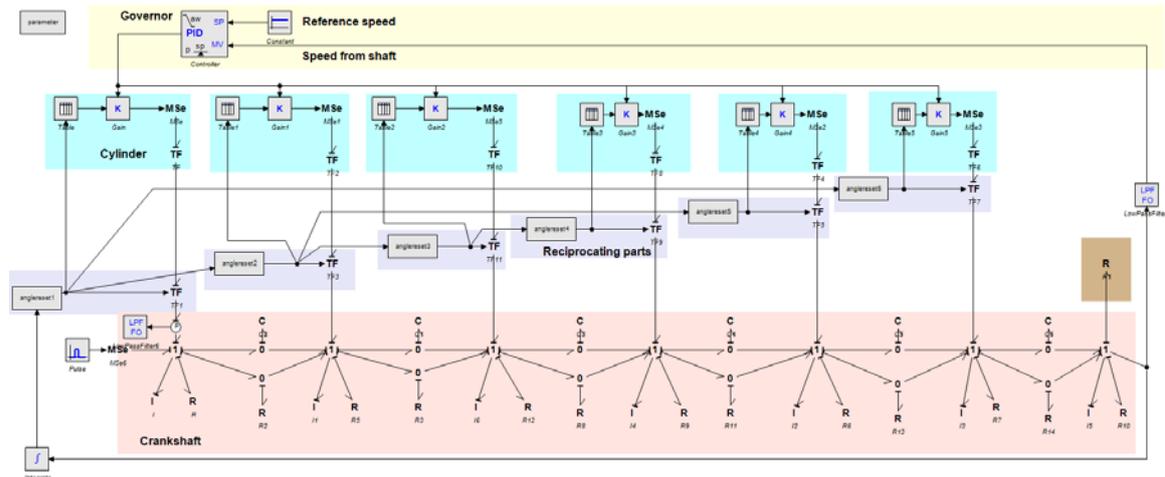
Figure 5.13 Two-stroke 6-cylinder engine model

| Constant | C=10.99 rad/s |
|---|---|
| PID controller | K = 1.0 {}; Proportional gain |
| | Td = 1 {s}; Derivative time constant: Td > 0 |
| | N = 10 {}; Derivative gain limitation. |
| | Ti = 1 {s}; Integral time constant: Ti > 0 |
| | b = 1.0 {}; Proportional setpoint weighting |
| | Ta = 1.0 {s};   Tracking time constant.t: Ta > 0 |
| | minimum = 0 {}; Minimum controller output |
| | maximum = 1 {}; Maximum controller output |
| Gain K | K=1 (scaled parameter) |
| TF (pressure to force) | D=0.6 (generic parameter) |
| TF (force to transformer) | a=1.2, b=2.5. (scaled parameter) |
| Pulse | start_time = 1.0 {s}; |
| | stop_time = 3 {s}; |
| | amplitude = 9600 |
| I (crank inertia) | i=4209 (scaled parameter) |
| R (friction loss) | R=550 |
| C (1/stiffness) | c=9.13 |
| R (material damping) | r=1490187.9 |
| R (propeller) | r=71100 (details in Table 6.2) |
| Low pass filters | Filter1 BW=0.01 Hz |
| | Filter6 BW=0.005 Hz |

Table 5.3 Two-stroke input parameters

## 5.3 Four-stroke engine

The differences between two-stroke engine and four-stroke engine have been explained in the pre-study part. Comparing to the two-stroke engine model, if some adjustment is made in the structure then the model can switch to a four-stroke engine model. The engine model reference of four-stroke engine is SCANIA DI13 074M and the data are shown below:



Figure 5.14Engine appearance [22]
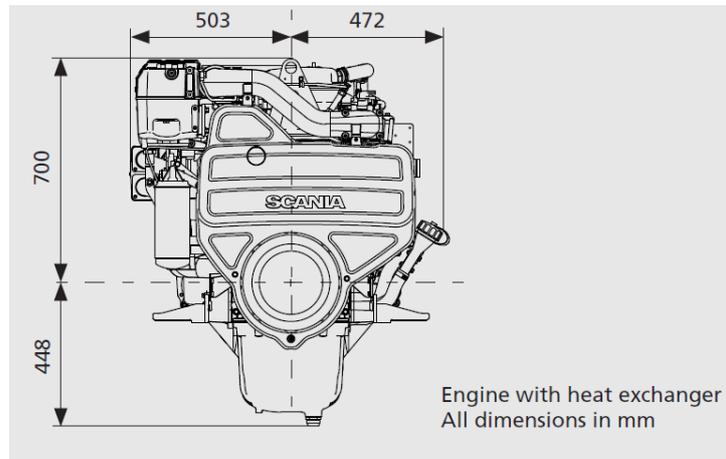


Figure 5.15 Engine dimensions [22]

| Cylinder diameter | 130 mm |
| Stroke | 140 mm |
| Displacement | 9.3 liters |
| Weight | 1150 kg |
| Engine speed | 1800 rpm |
| Piston speed | 8.4 m/s |
| Output power | 323 kW |

Table 5.4 Engine data [22]

## 5.3.1 One cylinder model
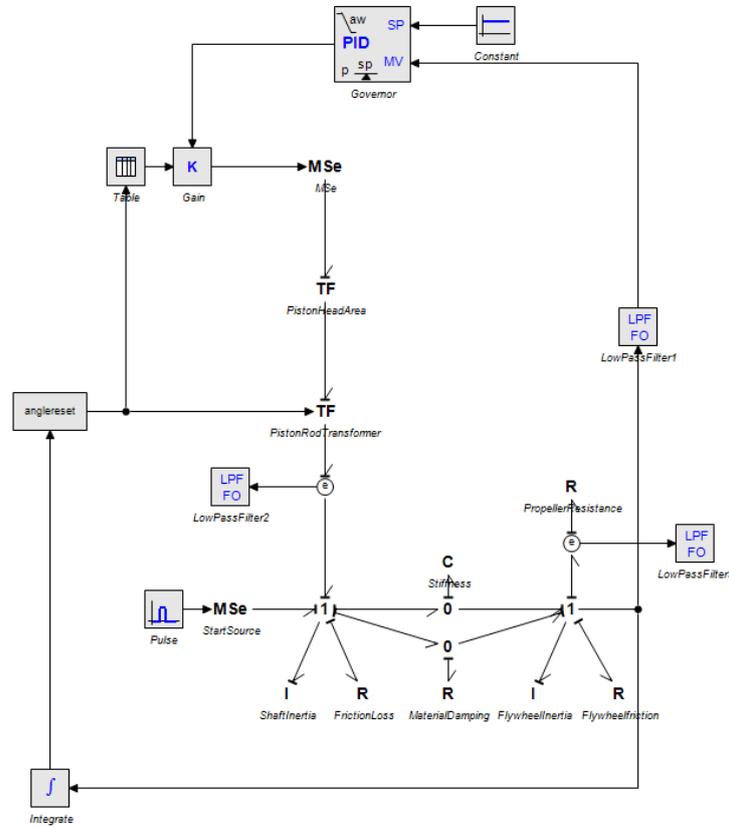


Figure 5.16 Four-stroke single cylinder engine model

**Governor**
The governor follows the same theory as two-stroke engine. Since the four-stroke is a high speed engine, the reference speed is 1800rpm according to the Scania manual book, 188 rad/s.

**Table**
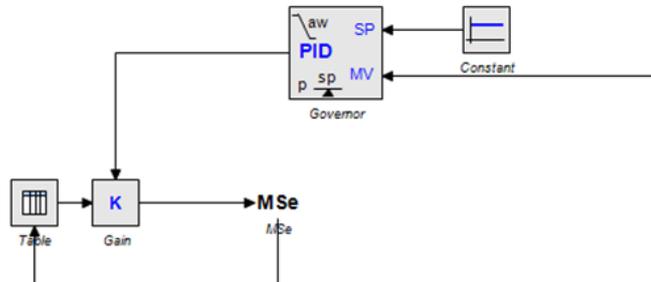


Figure 5.17 Governor model

In four-stroke engine the entire combustion cycle is completed in two revolution of the fly wheel which means the crank shaft rotates 720° each cycle. Here the data of a small four-stroke engine Scania is applied.
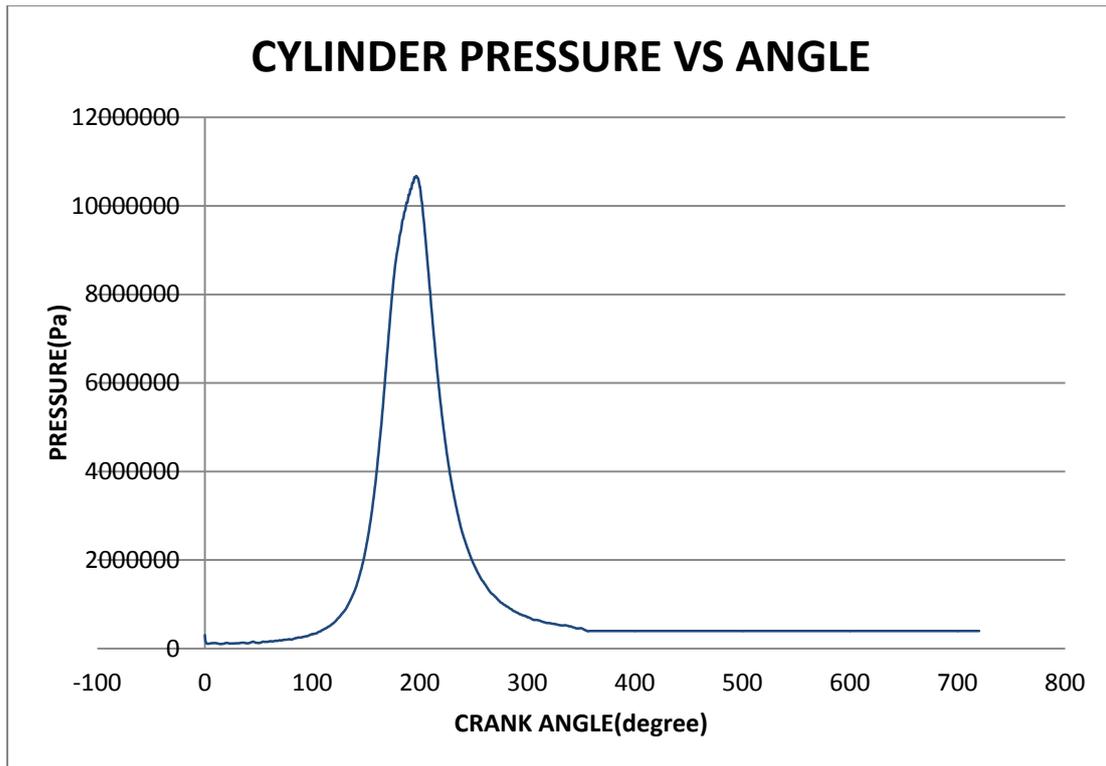
Figure 5.18 Pressure-crank angle curve

This data is also received from engine experiment test. The maximum firing pressure 10.7MPa happens at 196.5° where is after the piston reaches Top Dead Center (TDC).

**Cylinder & Piston & Connecting rod**
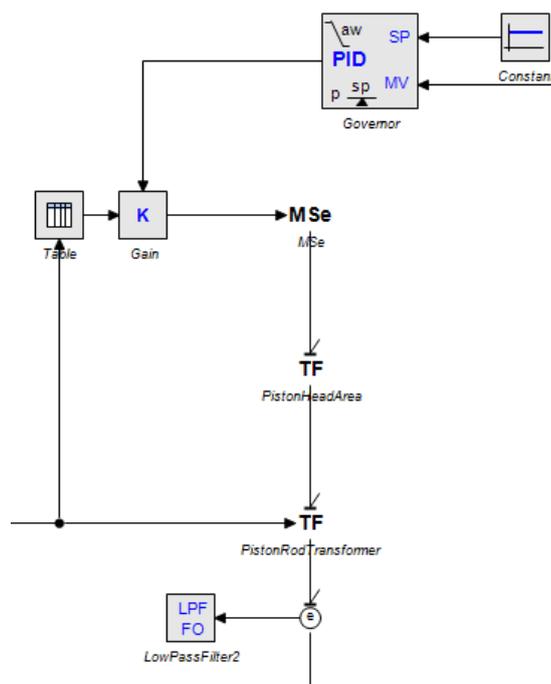


Figure 5.19 Piston and connecting rod model

33

The structure is the same as two-stroke engine. The table is replaced by the new pressure-angle data above. The bore diameter of this small Scania engine is 0.13m according to the manual [22]. So the input parameter D=0.13m. In the second TF, the relation between the force on the piston and the twisting torque T on the crank is the same as equation (5.6). The crank radius a=0.07m and piston rod length b=0.225m.

**Crankshaft & Flywheel & Propeller**
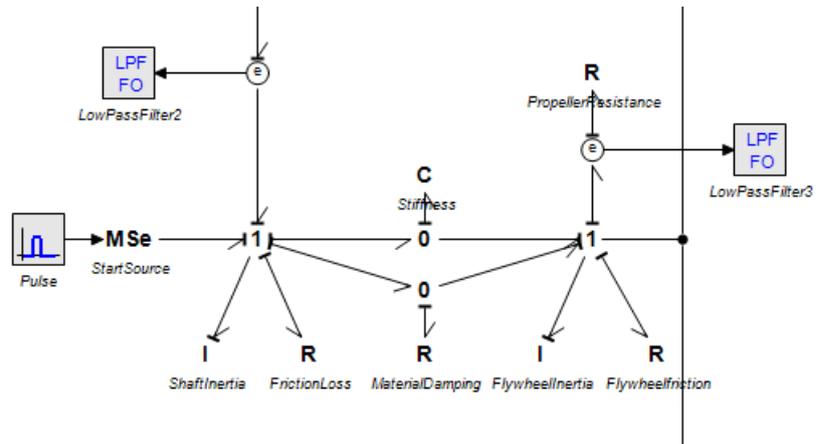


Figure 5.20 Crankshaft model

The value of inertia of crankshaft, the stiffness and the inertia of flywheel are given in the manual book:

| Inertia | 0.09 kgm$^2$ |
|---|---|
| Stiffness | 2.5 MNm/rad |
| Flywheel inertia | 2 kgm$^2$ |

Table 5.5 Engine mass elastic data [23]

The propeller calculation will be done in the next chapter and the propeller model will also be connected with the engine model.

### 5.3.2 Series connections (6-cylinder)

Same as two-stroke engine, after the one-cylinder engine model is completed the 6-cylinder engine is built by connecting it in series. The firing order still 1-5-3-6-2-4 which is the same as two-stroke engine.

In four-stroke engine all cylinders complete the work with two. Therefore, 6-cylinder two-stroke ignition intervals angle =720°/ 6 (cylinders) = **120°**.

The submodel of angle reset is different from two-stroke model, which is created to make sure the value is between 0 and $4\pi$ and distinguish the angle difference between each cylinder.
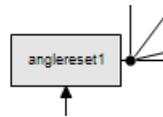


Figure 5.22

The input of the submodel angle reset is the angle from shaft end, and the output of the submodel is:

$$\text{output} = \text{input} - k * 4 * \pi \qquad (5.10)$$

where $k = \text{floor}\left(\frac{\text{input}}{4\pi}\right);$

which means when the value of angle from the shaft end is over $4\pi$ then it become 0 again. For the other submodels the angle difference between each other is expressed in the equations. The details are in the Appendix A.
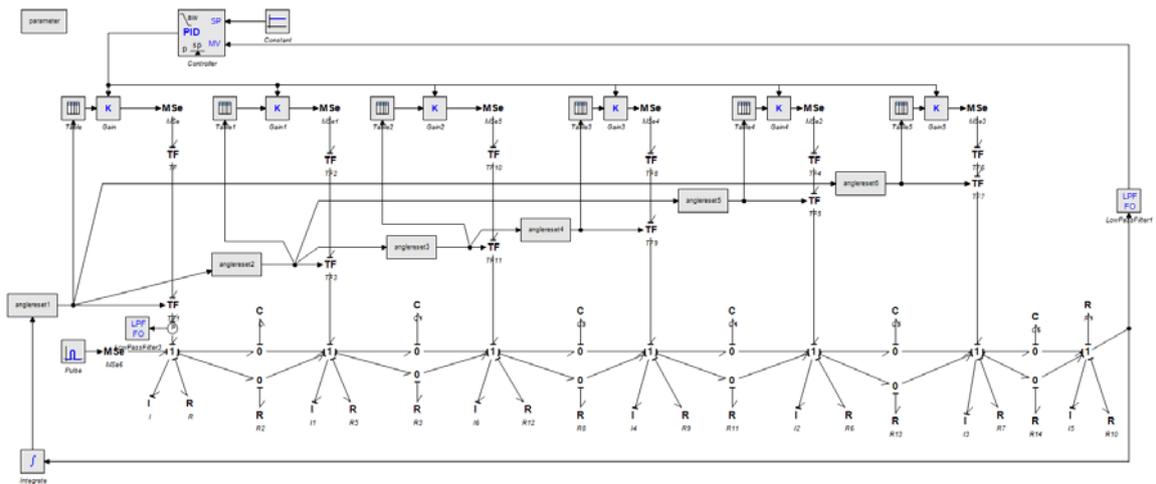
Figure 5.21 Four-stroke 6-cylinder engine model

| Constant | C=188rad/s |
|---|---|
| PID controller | K = 1.0 {}; Proportional gain<br>Td = 0.5 {s}; Derivative time constant: Td > 0<br>N = 10 {}; Derivative gain limitation.<br>Ti = 1 {s}; Integral time constant: Ti > 0<br>b = 1.0 {}; Proportional setpoint weighting<br>Ta = 1.0 {s};   Tracking time constant.t: Ta > 0<br>minimum = 0 {}; Minimum controller output<br>maximum = 1 {}; Maximum controller output |
| Gain K | K=1 (scaled parameter) |
| TF (pressure to force) | D=0.13 (generic parameter) |
| TF (force to transformer) | a=0.07, b=0.225. (scaled parameter) |
| Pulse | start_time = 1.0 {s};<br>stop_time = 3 {s};<br>amplitude = 400 |
| I (crank inertia) | i=0.09 (scaled parameter) |
| R (friction loss) | 0.02 |
| C (1/stiffness) | c=1.26 |
| R (material damping) | r=60982 |
| R (propeller) | r=7 (details in Table 6.3) |
| Low pass filters | Filter1 BW=0.1 Hz<br>Filter3 BW=0.08 Hz |

Table 5.6 Four-stroke input parameters

# 6 Connection with propeller model

As we discussed in the literature survey, there is one thesis from previous student "Modeling and Simulation of a Simple Propulsion Machinery" [1] by Cristian Braia. The author created a propeller model in 20-sim. In this model the propeller is specified while the engine is simplified. The propeller model can fit different types of engines and ships.
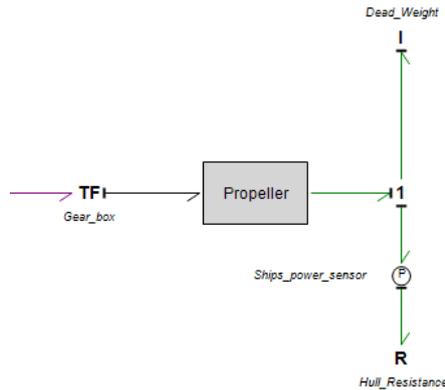


Figure 6.1 Propeller model [1]

In the thesis, the author referred to the Wageningen B-series data [24] to find the propeller properties. The open water propeller characteristics are presented in the form of the thrust and torque coefficients $K_T$ and $K_Q$ in terms of the advance coefficient J:

$$K_T = \frac{T}{\rho * n^2 * D^4} \tag{6.1}$$

$$K_Q = \frac{T}{\rho * n^2 * D^5} \tag{6.2}$$

$$J = \frac{V_A}{n * D} \tag{6.3}$$

Where T is the propeller thrust,
Q is the propeller torque.
$\rho$ is the fluid density,
n is the number of propeller revolutions per second,
D is the propeller diameter,
$V_A$ is the speed of advance.
So the formula of thrust is

$$T = K_T * \rho * n^2 * D^4 \tag{6.4}$$

And the formula of torque is

$$Q = K_T * \rho * n^2 * D^5 \tag{6.5}$$

The propeller submodel works as a gyrator and the input effort is propeller torque and input flow is angular velocity, the output effort is thrust and the output flow is the ship velocity.

If some parameters such as propeller diameter, pitch/diameter ratio, blade area ratio $A_E/A_O$, number of blades, dead weight of the ship, hull resistance factor are adjusted according to specific ship condition, then the propeller will fit.

According to the Taylors Bp-Delta diagrams [25], the non-dimensional parameters are defined as:

$$B_P = \sqrt[4]{\left(\frac{1}{2\pi\rho}\right)\frac{P_D * n_P{}^2}{V_A{}^5}} \tag{6.6}$$

Where $P_D$ is propeller shaft power in W,
$n_P$ is propeller rotational speed in revs/sec,
$V_A$ is propeller advance speed
$\rho$ is water density
The propeller selection is discussed in Appendix B.

|  | Two-stroke | Four-stroke |
|---|---|---|
| Blades | 3 | 3 |
| $A_E/A_O$ | 0.65 | 0.65 |
| Diameter | 7.4m | 1.3m |
| P/D | 0.73 | 1.08 |
| Gear ratio | 1:1 | 1:4 |

Table 6.1 Propeller selection

Input those parameters in the propeller model, the simulation result is explained in Chapter 8. The overview of the model is shown:
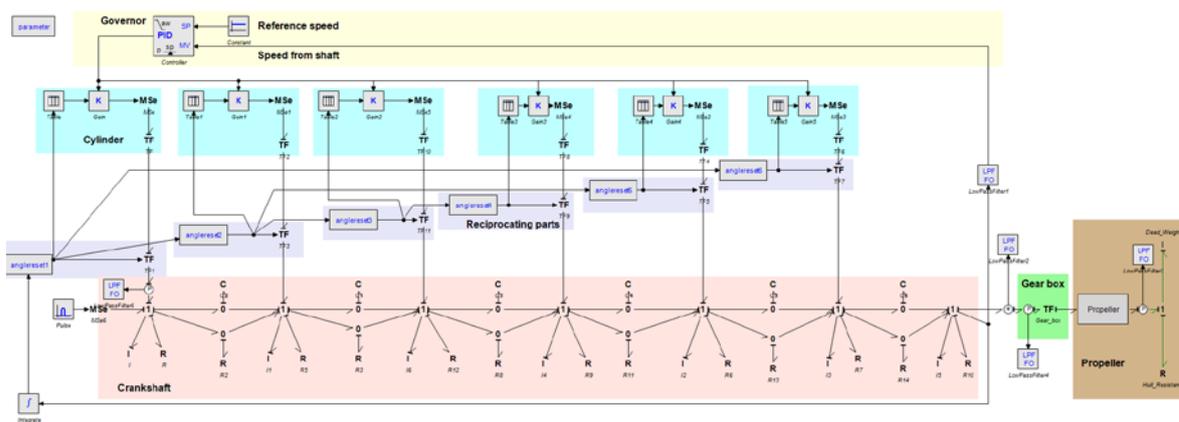


Figure 6.2

| TF (gear box) | r=1 |
|---|---|
| Propeller | AE/AO=0.65 <br> D=7.1 <br> P/D=0.72 <br> Z=3 |
| Dead weight | i=15609000 |
| Hull resistance | hull_resistance_coef = 16181 |
| Low pass filters | Filter2 BW=0.1 Hz <br> Filter4 BW=0.5 Hz |

Table 6.2 Two-stroke propeller input parameters

| TF (gear box) | r=0.25 |
|---|---|
| Propeller | AE/AO=0.65 <br> D=1.35 <br> P/D=1.08 <br> Z=3 |
| Dead weight | i=2000 |
| Hull resistance | hull_resistance_coef = 215 |
| Low pass filters | Filter2 BW=0.1 Hz <br> Filter4 BW=0.1 Hz |

Table 6.3 Four-stroke propeller input parameters

# 7 Generic model

Now both two-stroke and four-stroke engine models are built. The main purpose of this thesis is to create a generic engine model which can be reused in other future projects. A generic model constitutes a standard structure with some elements, which can facilitate its adaptation to a particular engine system. It does not correspond to a specific engine. With slight adaptation, a generic model can be used for testing other engines. However, careful restructuring and necessary adaptation is needed if one wishes to reuse it. How to input the engine data easily without many changes becomes the main task. If small adjustment and replacement can change the model to another new engine model, it will save a lot of efforts and improve the efficiency.

## 7.1.1 Scaling parameters selection

**Scaling factor**

The engine pressure in each cylinder is the power source and it decides the main character of the engine. The shape of pressure curve is similar between same types of engine. So by scaling the peak pressure in single cylinder can achieve changing the model to another. The scale factor is the ratio of the maximum pressure of engine to the standard value. This fits both two-stroke and four-stroke engine.

$$k_p = \frac{to\ be\ tested\ engine\ peak\ pressure}{standard\ peak\ pressure}$$

Another scaling factor is the ratio of the bore diameter to the standard bore diameter. By researching many engine data we found out that the other part dimension of engine is related to the bore diameter. For example the piston rod length, connecting rod length dimensions will expand with the same ratio of bore diameter in all directions.

$$k_d = \frac{to\ be\ tested\ engine\ bore\ diameter}{standard\ bore\ diameter}$$

The crank shaft shape is complex. The dimension of crank web decides the crank radius, and the value is usually half of stroke length. While the crank journal and crankpin dimensions are related to the bore diameter. So, the parameters such as crankshaft inertia, stiffness and material damping are scaled with both bore diameter and stroke length.

$$k_p = \frac{to\ be\ tested\ engine\ stroke}{standard\ stroke\ length}$$

| Generic parameters | Scaling factors | Scaled parameters |
|---|---|---|
| Cylinder pressure | $k_p$ | Peak cylinder pressure |
| Bore diameter | $k_d$ | Piston area<br>Piston rod length(four-stroke)<br>Connecting rod length(two-stroke)<br>Crankshaft inertia<br>Crankshaft stiffness<br>Material damping |
| Stroke length | $k_s$ | Crank radius<br>Crankshaft inertia<br>Crankshaft stiffness<br>Material damping |

Table 7.1 Scaling factors

As discussed above, crankshaft inertia, stiffness and material damping are scaled by both bore diameter and stroke length. So the scaling factors of these parameters should be defined.

**Crankshaft inertia**

In the modeling chapter, the crank shaft is simplified as a disc with thickness rotating along the shaft line. The formula of inertia is $I = \frac{mR^2}{2}$. If the bore diameter is scaled with a factor of $k_d$, then the mass of disc will be scaled by $k_d^3$. The diameter R is also scaled by $k_d$. Then the inertia $I' = k_d^5 \frac{mR^2}{2} = k_d^5 I$.

While referring to the engine data collection, the inertia scaling factor is not the same value as $k_d^5$. This is due to the complex shape of crankshaft.

Four types of Wartsila engine are compared here. The dimensions of each engine can be found in the manual book:

| Engine type | Diameter (mm) | Stroke (mm) | Inertia (kg/m$^2$) | Crank pin diameter(mm) | Piston weight(kg) | Connecting rod weight(kg) | Connecting rod length(mm) |
|---|---|---|---|---|---|---|---|
| W 6L20 | 200 | 280 | 90 | 190 | 21 | 39 | 777 |
| W 6L26 | 260 | 320 | 121 | 228 | 70(+pin) | 83 | 960 |
| W 6L32 | 320 | 400 | 500 | 328 | 82 | 153.5 | 1245 |
| W 6L38 | 380 | 475 | 850 | 370 | 190(+pin) | 305 | 1463 |

Table 7.2 Wartsila engine data [26]

Calculate the scaling factor:

| Scaling factor of | Diameter | Stroke | Inertia | Crank pin diameter | Piston weight | Connecting rod weight | Connecting rod length |
|---|---|---|---|---|---|---|---|
| W 6L20 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| W 6L26 | 1.3 | 1.14 | 1.34 | 1.2 | 3.33 | 2.13 | 1.24 |
| W 6L32 | 1.6 | 1.43 | 5.56 | 1.73 | 3.9 | 3.94 | 1.6 |
| W 6L38 | 1.9 | 1.7 | 9.44 | 1.95 | 9.05 | 7.82 | 1.88 |

Table 7.3 Scaling factor

| Scaling factor of | Diameter | (Diameter factor)$^3$ | (Diameter factor)$^{3.5}$ |
|---|---|---|---|
| W 6L20 | 1 | 1 | 1 |
| W 6L26 | 1.3 | 2.2 | 2.5 |
| W 6L32 | 1.6 | 4.1 | 5.18 |
| W 6L38 | 1.9 | 6.86 | 9.45 |

Table 7.4

The values in the table show that:
1. Crank pin diameter, connecting rod length are scaled by the diameter scaling factor $k_d$;
2. The piston weight and connecting rod weight are scaled by the diameter scaling factor $k_d{}^3$ so it means the dimensions are scaled by the diameter scaling factor $k_d$ in all directions;
3. The inertia scaling factor is $k_d{}^{3.5}$ instead of $k_d{}^5$ as assumed.

Next, the reason of why the scaling factor is $k_d{}^{3.5}$ instead of $k_d{}^5$ should be found out.
The crankshaft can be considered as structural components with distributed mass and elasticity. Its motion is determined by the action of the external torque such as gas pressure and reciprocating inertia torque. To analyze these elements in diesel engines, it is important to simplify the real system with an equivalent dynamic model in a realistic way.
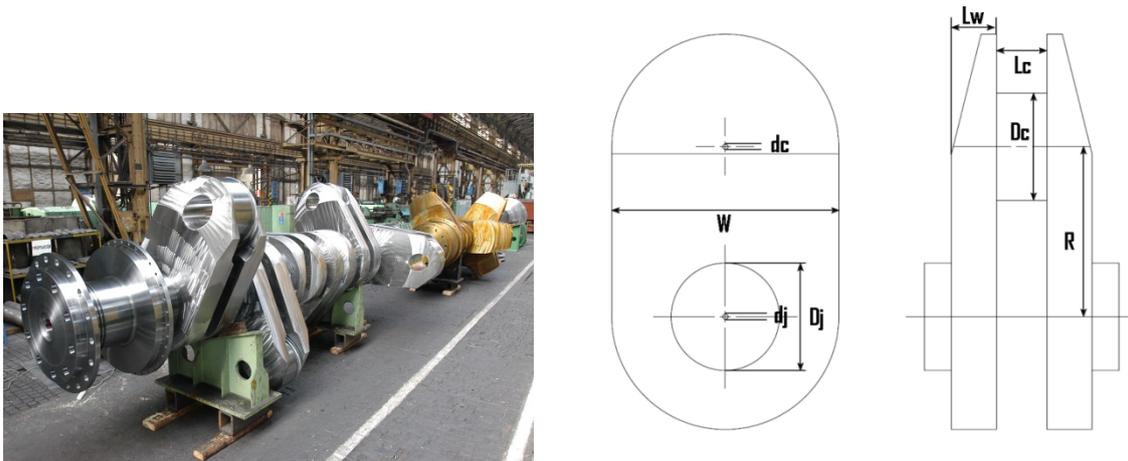


Figure 7.1 Crankshaft [27]

The moment of inertia of crankshaft can be treated as cylindrical parts, and a single crank is separated to analyze. A basic crankshaft throw includes half of the crank journal on each side, two crank webs and crankpin between the webs. The crankpin drives a connecting rod, a piston and piston rod. The moment of inertia at each throw consists of the rotating inertia and reciprocating mass. The rotating inertia is constant, but the reciprocating parts vary when the crankshaft rotates. The moment of inertia in general is expressed as an equivalent inertia and the formula is [28]:

$$I_{eqv} = I_{rot} + \frac{1}{2}M_{recip}R^2 \tag{7.1}$$

The rotational inertia includes the inertia of the journals, inertia of the crank webs, inertia of the crankpin and inertia of the connecting rod.

$$I_{rot} = 2 * I_j + 2 * I_{web} + I_{pin} + I_{rod} \qquad (7.2)$$

The crank journal can be treated as solid cylinder rotating about the cylinder axis. The crankpin follows the same theory but it rotates at the throw radius but not about its own axis, so the parallel axis theory is applied. The crank web shape is close to rectangular and parallel axis theory is also applied. The connecting rod's weight is heavier at the big end than the small end of connecting rod. In this case we assume that 2/3 of the weight is rotating and 1/3 is reciprocating.

So the formulas of inertia of each part and total inertia are shown below:
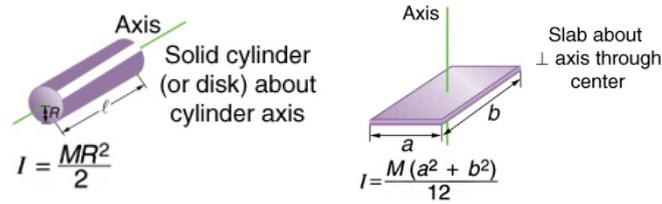


Figure 7.4 Moment of inertia calculations

Inertia of crank journal:
$$I_j = \frac{1}{2} M_j D_j^2$$

Inertia of crank web:
$$I_{web} = \frac{1}{2} M_{web}(W^2 + (R + W)^2) + M_{web}(\frac{R}{2})^2$$

Inertia of crankpin:
$$I_{pin} = \frac{1}{2} M_{pin} D_c^2 + M_{pin} R^2$$

Inertia of connecting rod:
$$I_{rod} = \frac{2}{3} M_{rod} R^2$$

Inertia of reciprocating parts:
$$M_{recip} = \frac{1}{3} M_{rod} + M_{piston}$$
$$I_{recip} = \frac{1}{2} M_{recip} R^2$$

Inertia in total:
$$I_{eqv} = 2 * \frac{1}{2} M_j D_j^2 + 2 * \left(\frac{1}{2} M_{web}(W^2 + (R + W)^2) + M_{web}\left(\frac{R}{2}\right)^2\right) + \frac{1}{2} M_{pin} D_c^2 +$$
$$M_{pin} R^2 + \frac{2}{3} M_{rod} R^2 + \frac{1}{2}(\frac{1}{3} M_{rod} + M_{piston}) R^2 \qquad (7.3)$$

If the crankshaft is scaled with bore diameter scaling factor $k_d$ and stroke scaling factor $k_s$ at the same time, then,
Inertia of crank journal with scaling factor:

$$I_j' = \frac{1}{2} k_D^3 M_j k_D^2 D_j^2 = k_D^5 \frac{1}{2} M_j D_j^2 = k_D^5 I_j$$

Inertia of crank web with scaling factor:

$$I_{web}' = \frac{1}{2}k_S{}^3 M_{web}\left(k_S{}^2 W^2 + k_S{}^2(R+W)^2\right) + k_S{}^3 M_{web}k_S{}^2 \left(\frac{R}{2}\right)^2$$

$$= k_S{}^5\left(\frac{1}{2}M_{web}(W^2 + (R+W)^2) + M_{web}\left(\frac{R}{2}\right)^2\right) = k_S{}^5 I_{web}$$

Inertia of crankpin with scaling factor:

$$I_{pin}' = \frac{1}{2}k_D{}^3 M_{pin}k_D{}^2 D_c{}^2 + k_D{}^3 M_{pin}k_S{}^2 R^2 = k_D{}^5 \frac{1}{2}M_{pin}D_c{}^2 + k_D{}^3 k_S{}^2 M_{pin}R^2$$

Inertia of connecting rod with scaling factor:

$$I_{rod}' = \frac{2}{3}k_D{}^3 M_{rod}k_S{}^2 R^2 = k_D{}^3 k_S{}^2 I_{rod}$$

$$M_{recip}' = \frac{1}{3}M_{rod}' + M_{piston}' = k_D{}^3 M_{recip}$$

$$I_{recip}' = \frac{1}{2}M_{recip}'R'^2 = k_D{}^3 k_S{}^2 \frac{1}{2}M_{recip}R^2 = k_D{}^3 k_S{}^2 I_{recip}$$

Inertia in total with scaling factor:

$$I_{eqv}' = 2k_D{}^5 I_j + 2k_s{}^5 I_{web} + k_D{}^5 \frac{1}{2}M_{pin}D_c{}^2 + k_D{}^3 k_S{}^2 M_{pin}R^2 + k_D{}^3 k_S{}^2 I_{rod}$$
$$+ k_D{}^3 k_S{}^2 I_{recip}$$

So

$$\mathbf{I_{eqv}' = k_D{}^5\left(2I_j + \frac{1}{2}M_{pin}D_c{}^2\right) + k_s{}^5 * 2I_{web} + k_D{}^3 k_S{}^2(M_{pin}R^2 + I_{rod} + I_{recip})} \quad (7.4)$$

This can be applied by both two-stroke and four-stroke engine.

**Stiffness**

The torsional stiffness of crankshaft depends on the dimensions, shape of crankshaft and also the shear modulus of shaft material. The total twist angle consists of two parts, one is the torsional twist angle of the crank journals and the crankpin, and another is the bending angle of the crank web. There are several formulas which were developed based on test data of different types of crankshafts. For example:

Carter's formula [28]

$$K_t = \frac{G * \pi}{32 * \left[\frac{L_j + 0.8 * L_w}{D_j{}^4 - d_j{}^4} + \frac{0.75 * L_c}{D_c{}^4 - d_c{}^4} + \frac{1.5 * R}{L_w * W^3}\right]} \quad (7.5)$$

Ker Wilson's formula [28]

$$K_t = \frac{G\pi}{32 * \left[\frac{L_j + 0.4 * D_j}{D_j{}^4 - d_j{}^4} + \frac{L_j + 0.4 * D_c}{D_c{}^4 - d_c{}^4} + \frac{R - 0.2 * (D_j + D_c)}{L_w * W^3}\right]} \quad (7.6)$$

For the **two-stroke** engine:

The difference between these two formulas is the first one fits more flexible crank web and stiff journal but the second one is applied to flexible journal and stiff crank web. To test and compare the formulas, the Finite Element Method (FEM) analysis is applied in this case. According to the rough sketch of the s60mc engine, a 3D model is created in UG NX.

| Lw (m) | 0.32 |
|---|---|
| Lc (m) | 0.21 |
| Lj (m) | 0.17 |
| Dj (m) | 0.6 |
| Dc (m) | 0.6 |
| dj (m) | 0.03 |
| dc (m) | 0.03 |
| G (N/m$^2$) | $7.7*10^{10}$ |
| R (m) | 0.95 |
| W (m) | 1.26 |

Table 7.5 Two-stroke crankshaft dimensions

Input the parameters into the formulas and in the Carter's formula the stiffness coefficient is $K_t = 1.12 * 10^9 \text{Nm}$ while in the Ker Wilson's formula $K_t = 0.98 * 10^9 \text{Nm}$.

Next the 3D model is meshed and boundary conditions are added. One end of the crank journal is rigidly fixed and a torque is applied on the edge of the other end. Additionally, since the bearing will support the journal, there will be no translation in X-axis and Y-axis. This constrain is added on the oil bore in the center of the journal. The calculated torsional stiffness is equal to the torque divided by the angle of twist at the free end. The material is selected as the same steel which has the same shear modulus as what is calculated in the above formulas.
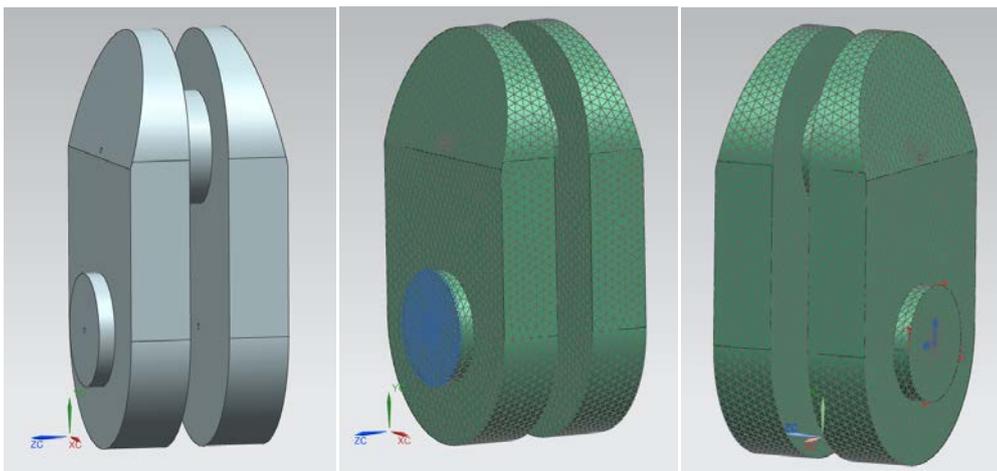


Figure 7.5 Two-stroke 3D model and boundary conditions

When these are ready to solve, run the simulation and receive the results as shown below:
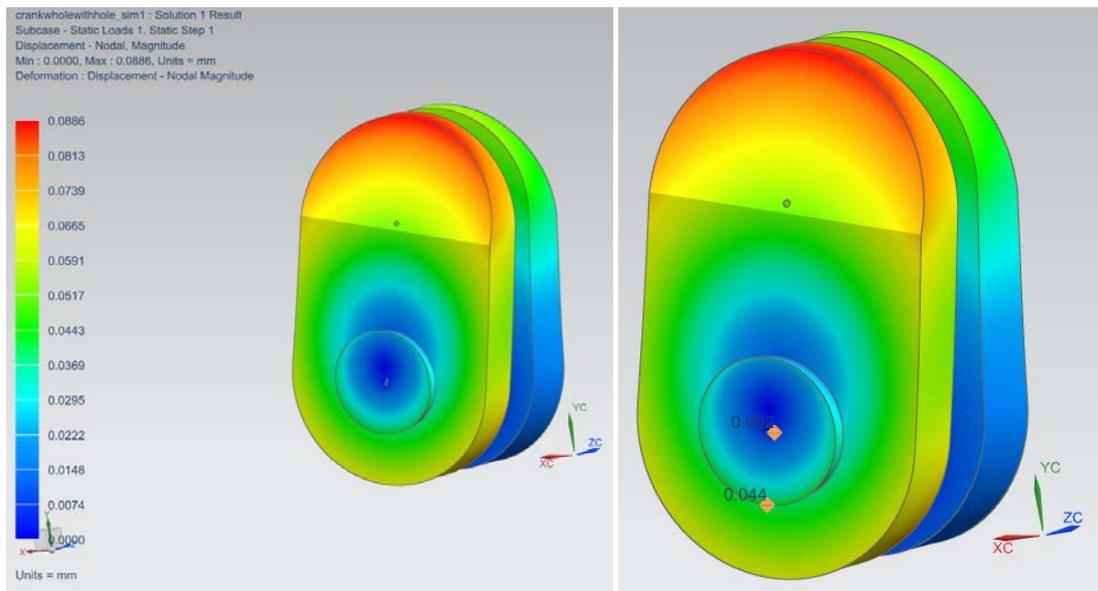


Figure 7.6 FEM analysis results

The maximum displacement happens at the top edge of the crank web and the value is 0.0886mm. To calculate the stiffness coefficient, $K = \frac{Torque}{Twist\ angle}$ the torque is 138490Nm which is received from 20-sim simulation at the cylinder torque output. The diameter of journal free end is 600mm and the displacement on the edge is 0.044mm, there is very little displacement in the center which is 0.002mm.

$$Twist\ angle = \frac{0.042mm}{300mm} = 0.00014rad$$

Then,

$$K = \frac{138490Nm}{0.00014rad} = 0.99 * 10^9 Nm.$$

This value locates between the results from the Carter's formula and Ker Wilson's formula

|  | Kt |
|---|---|
| Carter's formula | $1.12 * 10^9 Nm$ |
| Ker Wilson's formula | $0.98 * 10^9 Nm$ |
| FEM | $0.99 * 10^9 Nm$ |

Table 7.6 Kt comparison

For the **four-stroke** engine:

$$K_t = \frac{G\pi}{32 * [\dfrac{L_j + 0.4 * D_j}{D_j{}^4 - d_j{}^4} + \dfrac{L_j + 0.4 * D_c}{D_c{}^4 - d_c{}^4} + \dfrac{R - 0.2 * (D_j + D_c)}{L_w * W^3}]}$$

The same equation is applied in four-stroke engine. The dimensions are in the table below.

| Lw (m) | 0.03 |
|---|---|
| Lc (m) | 0.06 |
| Lj (m) | 0.08 |
| Dj (m) | 0.108 |
| Dc (m) | 0.087 |
| dj (m) | 0.006 |
| dc (m) | 0.004 |
| G (N/m$^2$) | 7.7*10$^{10}$ |
| R (m) | 0.07 |
| W (m) | 1.14 |

Table 7.7 Four-stroke crankshaft dimensions
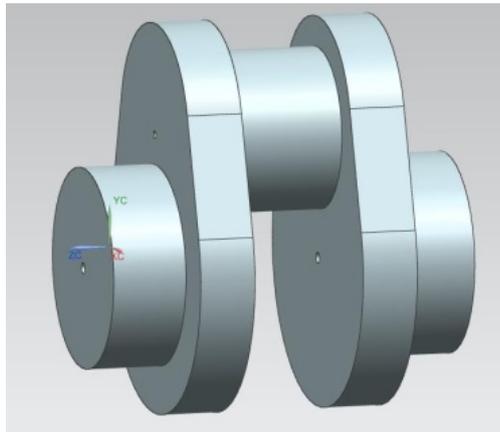


Figure 7.7 Four-stroke crankshaft
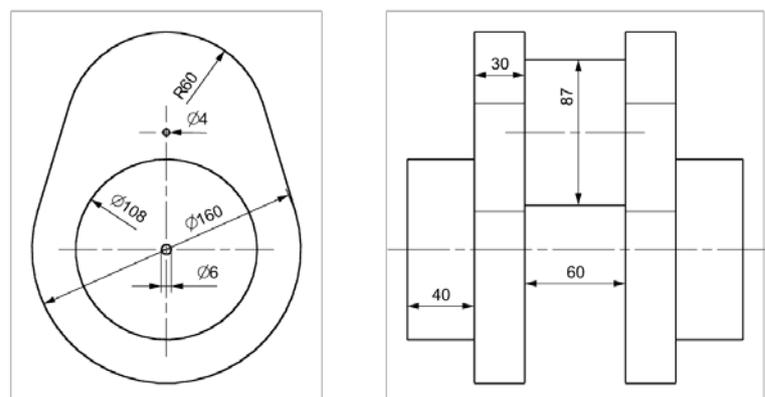


Figure 7.8 Four-stroke crankshaft dimensions

### 7.1.2 Global parameters

Parameters and variables can be shared between submodels by creating a global parameters submodel. The global parameters are accessible throughout the program. As discussed earlier, there are several parameters should be scaled by scaling factors. These parameters are related to different submodels but some have common scaling factors. So by editing the equations in the global submodel, it can be easily achieved. The detailed equations are explained in Appendix A.

Another advantage of creating a global submodel is that the interface is friendly for the users. In the future projects, just input the basic parameters of the engine, such as engine components dimensions, the other submodels will share the updated data and the whole engine can work as designed.

# 8 Simulation results and engine performance

## 8.1 Simulation & 20-sim

Simulation is the imitation of the operation of a real world process or system over time. Since we have already built a model and we need to run this model in the simulation software. The method of simulation reduces the risk of engine operation failure, cost saving in changing the product and human being effort in real world, it is more flexible to change different engine type in the propulsion system. Most importantly, we can receive the simulation results and analyze them. So it is easier for us to change the parameters in the simulation and optimize the engine performance and compare the results.

20-sim is a modeling and simulation program, it can simulate the behavior of dynamic systems, such as hydraulic systems, mechanical and electrical systems or any combination of these. 20-sim fully supports graphical modeling, and allows to design and analysis dynamic systems in an intuitive and user friendly way. With 20-sim, models can be entered as equations, block diagrams, bond graphs and physical components. The package has advanced support for bond graph modeling, making tit highly rated in bond graph communities. In the Mater education period, from the System Modeling and Simulation course we learnt the basic knowledge about how to simulate the bond graph models in 20-sim and analysis the results. Usually we have a prediction of the plot and compare it with the result, and give explanations of the result.

In this thesis, the bond graph model of marine diesel engine will be input into 20-sim and simulated. The plots such as pressure in the cylinder, the output torque of the shaft will explain the engine performance. After the simulation, some advice of how to improve the performance will be offered.

## *8.2   Simulation results*

**Load scenario analysis 1**

The engine starts working from the beginning. A start pulse is applied to start the engine. The engine runs for 400 seconds with propeller model as a load.

## 8.2.1  Two-stroke

**PID Controller output**
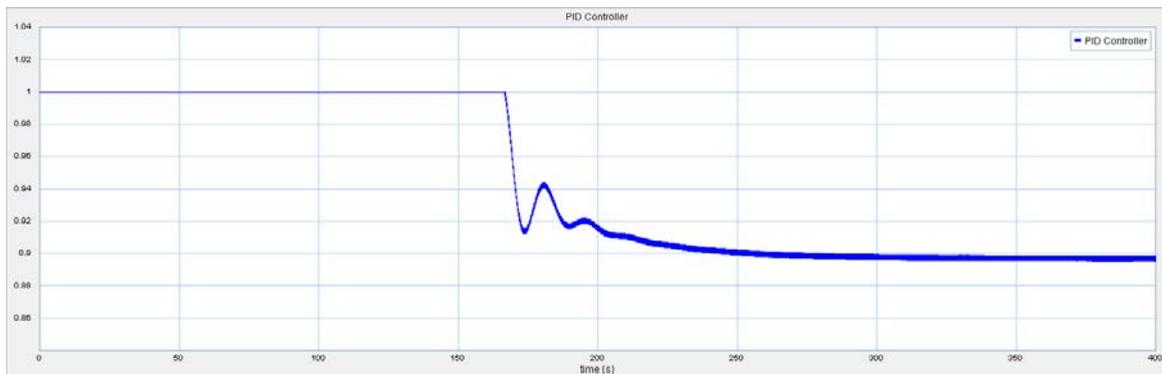


Figure 8.1 Two-stroke PID controller plot

The function of the PID controller is to maintain the engine speed stable and stay the same with the reference speed. The reference speed is 105 rpm. The plot shows that the engine starts and accelerates for about 167 seconds. Since the engine is a large two-stroke ship engine, the time is reasonable comparing to the accelerating time in realistic. When the engine speed is over the reference value, the scale factor which is the output of the PID controller decreases to less than 1. At the beginning the output signal oscillates a little but then become stable with a value of 0.9.

**Engine speed**
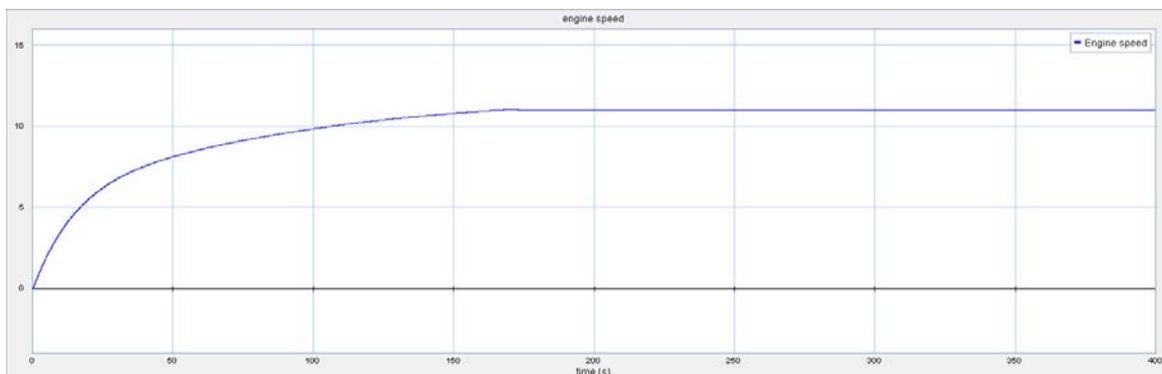


Figure 8.2 Two-stroke engine speed plot

This plot matches the PID controller plot because it shows also that the engine speed increases from the beginning and it lasts about 167 seconds before it reaches the reference speed. The reference speed of the engine is 105 rpm which is 10.99 rad/s and it can be found in the manual book of the engine. When the speed is over the reference value, it decreases back to 10.99 rad/s due to the effect of the PID controller. The following parameters can be found in the plot:

| | |
|---|---|
| Engine speed | 10.99 rad/s |
| Maximum engine speed | 11 rad/s |
| Maximum speed time | 170 s |

Table 8.1 Two-stroke engine speed plot results

**Firing order**
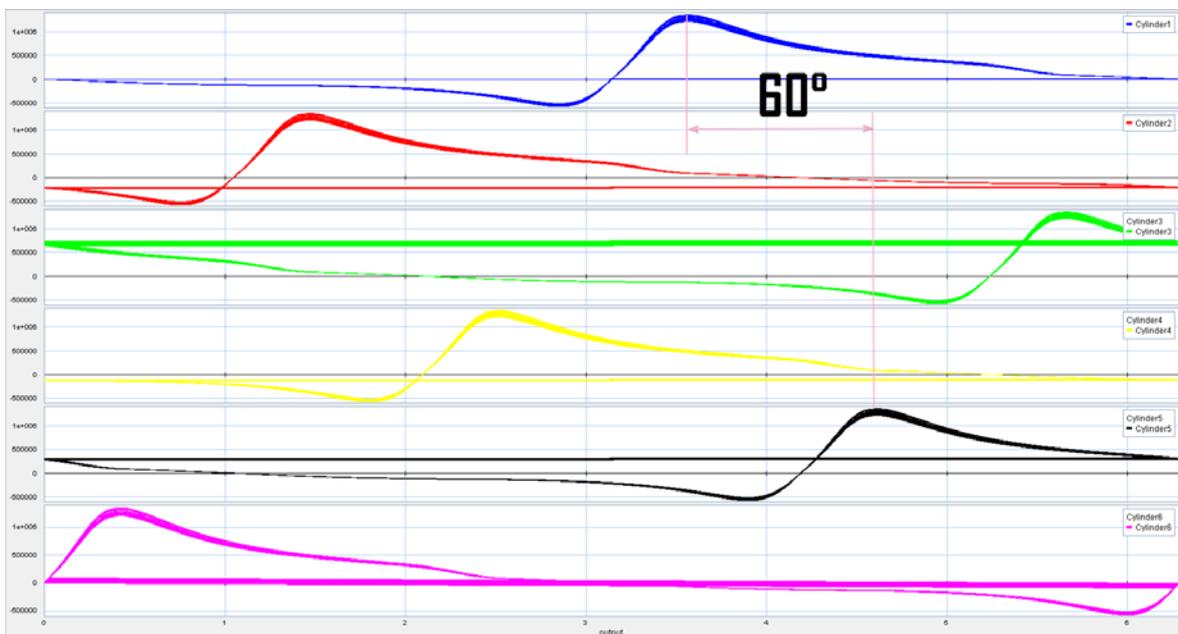


Figure 8.3 Two-stroke firing order plots

As discussed in the previous chapter, the firing order of 6 cylinders in this case is 1-5-3-6-2-4. The plot shows the same order as expected. The detailed parameters are shown below. For example, the maximum torque happens at 204.5° in cylinder 1 and 264.1° in cylinder 5. The angle difference is about 60°. The same angle difference can found between different cylinders.

| Cylinder number | Maximum position in rad | Maximum position in degree | Firing order |
|---|---|---|---|
| 1 | 3.57 rad | 204.5 | 1 |
| 2 | 1.47 rad | 84.2 | 5 |
| 3 | 5.66 rad | 324.3 | 3 |
| 4 | 2.52 rad | 144.4 | 6 |
| 5 | 4.61 rad | 264.1 | 2 |
| 6 | 0.43 rad | 24.6 | 4 |

Table 8.5 Two-stroke firing order plots results

**Pressure & Force & Torque**
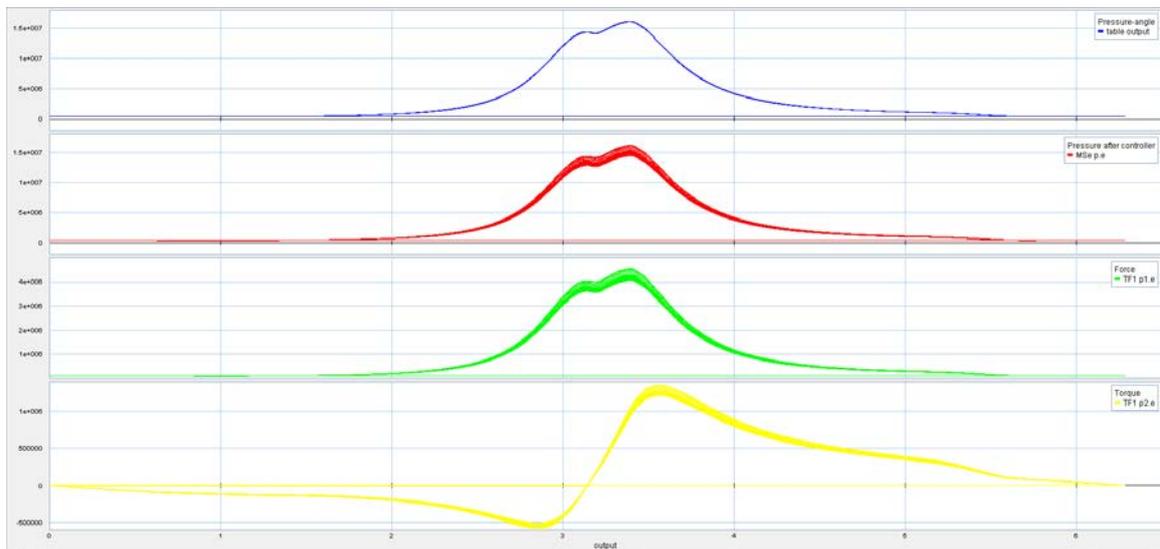


Figure 8.4 Two-stroke pressure, force, torque plots

The first plot is the pressure-angle relation input in the Table, the second plot is the output from MSe. The PID controller signal locates between these two. So the plot of MSe is affected by the PID controller scale factor. The PID controller signal is 1 at the beginning but when the speed is over reference speed, it reduces to 0.9. The maximum pressure in the cylinder is scaled by it also. The third plot is the force which is the output of the first transformer. Since the pressure is scaled by PID controller, the force will change with pressure as well because of the transform modulus is the piston area. The last plot is the torque which is after the second transformer and the force is changed into torque. It is also affected by the PID controller signal.

**Propeller characteristics**
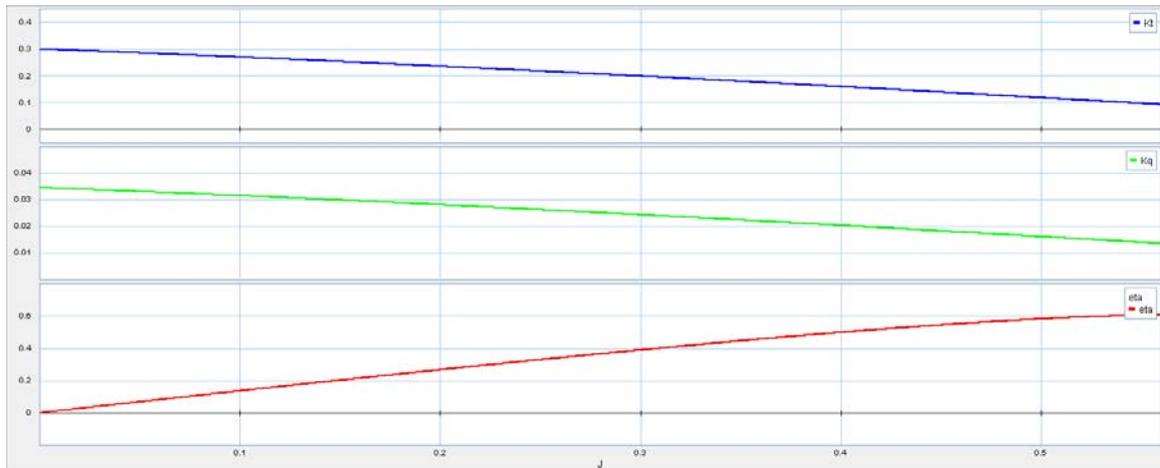


Figure 8.5 Two-stroke propeller characteristics plots

The trust and torque coefficients $K_T$ and $K_Q$ and efficiency plots are shown above. This is also discussed in the original work of the propeller simulation thesis. To prove the good combination of the engine model and the propeller model, the propeller characteristics can be explained also. According to Wageningen B-series propellers chart, 3 blades $A_E/A_O$=0.64 is selected. Since P/D=0.72 the $K_T$ and $K_Q$ and efficiency plots should close to the red line where P/D=0.7. Comparing the 20-sim plots and the Wageningen B-series propellers chart, the results prove that the propeller model fits the engine model and works well.
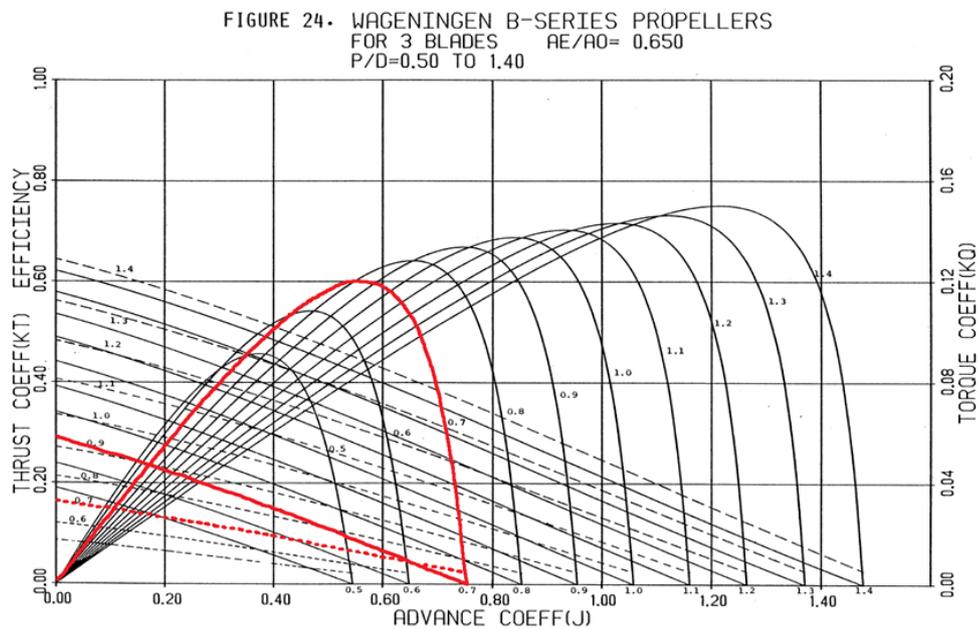


Figure 8.6 Wageningen B-series, 3 blades, AE/AO=0.65 [24]

**Torque & Trust force & Ship speed**
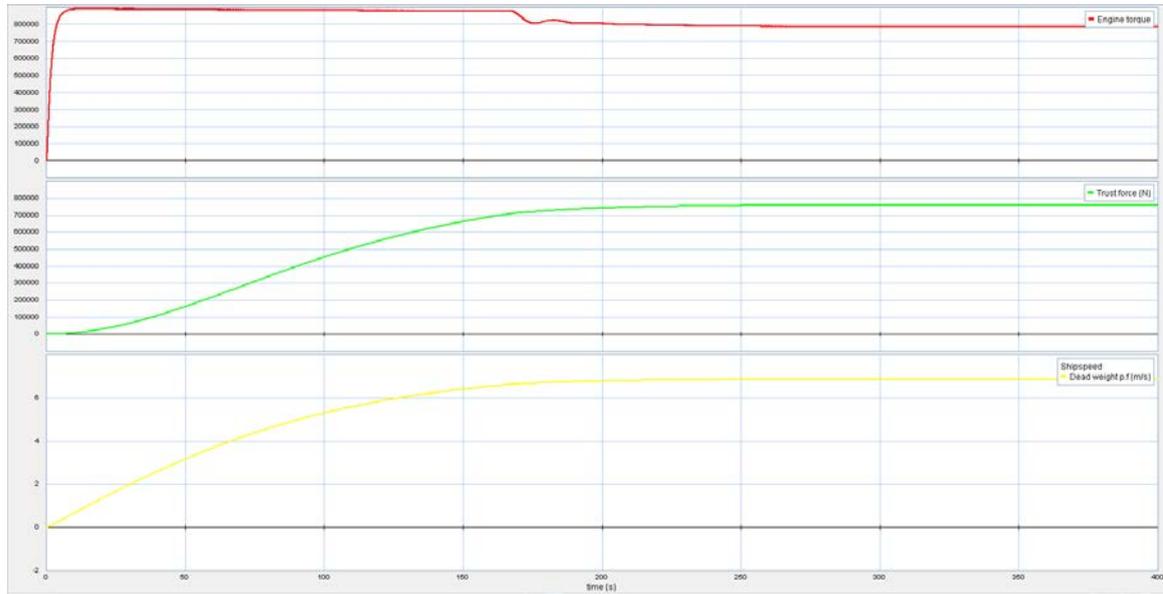


Figure 8.6 Two-stroke torque, trust force, ship speed plots

| Engine torque | 784.4 kNm |
|---------------|-----------|
| Trust force | 761.8 kN |
| Ship speed | 6.9 m/s or 13.3 knots |

Table 8.7 Two-stroke torque, trust force, ship speed plots results

The torque output from the shaft is the mean torque, and it is different from the torque in the single cylinder. The torque drops to 784.4 kNm after about 167 seconds and keep stable. The trust force is 761.8 kN and ship speed is 13.3 knots. This is according to the ship dead weight input. Referring to the ships where MAN s60mc are installed, we assume that the ship weight is 15609 tons.

**Power efficiency**
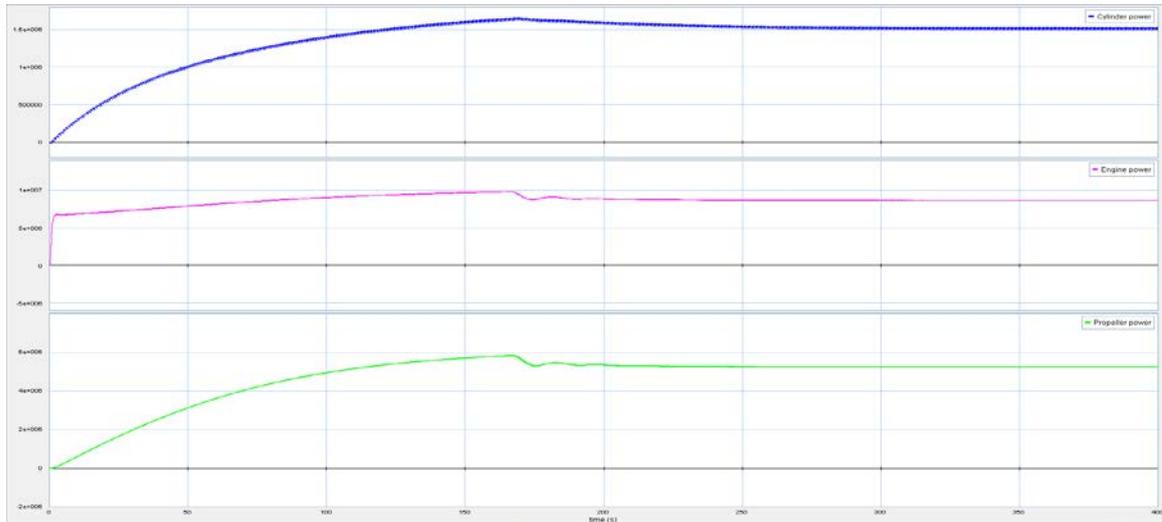


Figure 8.8 Two-stroke cylinder power, engine power, and propeller power plots

| Cylinder power | 1517.5 kW |
|---|---|
| Cylinder power*6 | 9105 kW |
| Engine shaft power | 8632.2 kW (torque*w) |
| $\eta_1$ | 94.8% |
| Propeller power | 5226.8 kW |
| $\eta_2$ | 60.6% |

Table 8.7 Two-stroke cylinder power, engine power, propeller power plots results

Power sensors are applied in the single cylinder and on the shaft. The power is each cylinder is about 1517.5 kW. Since six cylinders are connected in series so the total power is 9105 kW. But there is friction loss in the cylinder and also bearing on the shaft, the real shaft output is 8632.2 kW.  So the efficiency

$$\eta_1 = \frac{8632.2 \text{ kW}}{9105 \text{ kW}} = 94.8\%$$

The propeller efficiency

$$\eta_2 = \frac{5226.8 \text{ kW}}{8632.2 \text{ kW}} = 60.6\%$$

The propeller efficiency also matches the result from propeller characteristics plot.
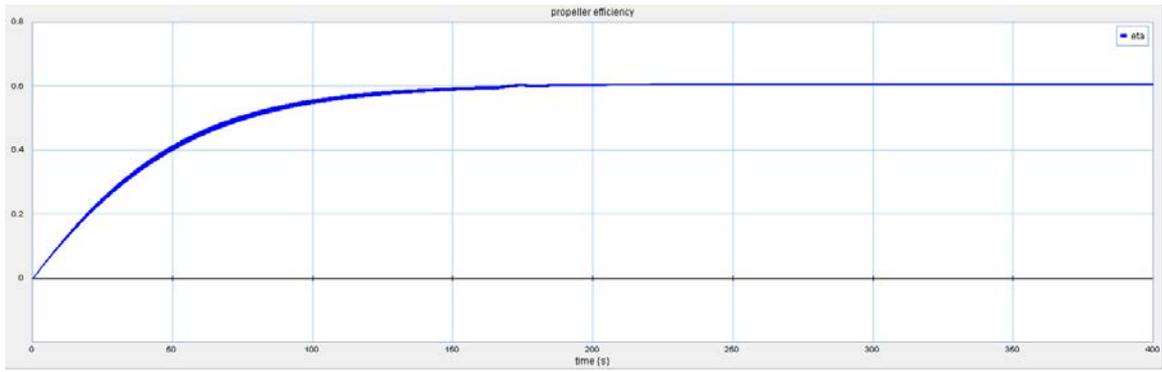
Figure 8.9 Two-stroke propeller efficiency

## 8.2.2 Four-stroke

**PID Controller**
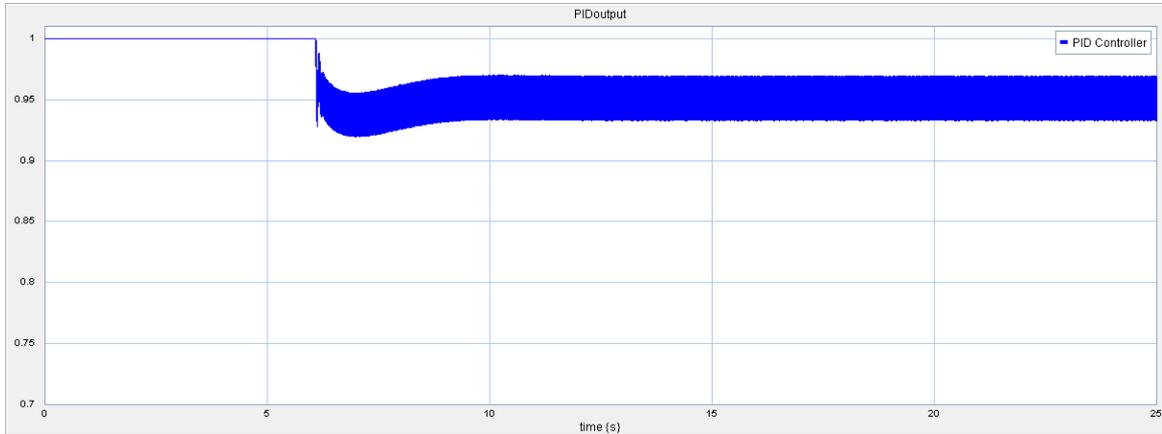


Figure 8.10 Four-stroke PID controller plot

Similar to two-stroke engine, the PID controller should maintain the engine speed stable and stay the same with the reference speed which is 1800 rpm in this case. The plot shows that the engine starts and accelerates for about 6 seconds. Because the four-stroke engine is a small high speed engine, the time of accelerating is much shorter than the two-stroke engine model. The signal stays at 0.95 when the speed is stable.

**Engine speed**
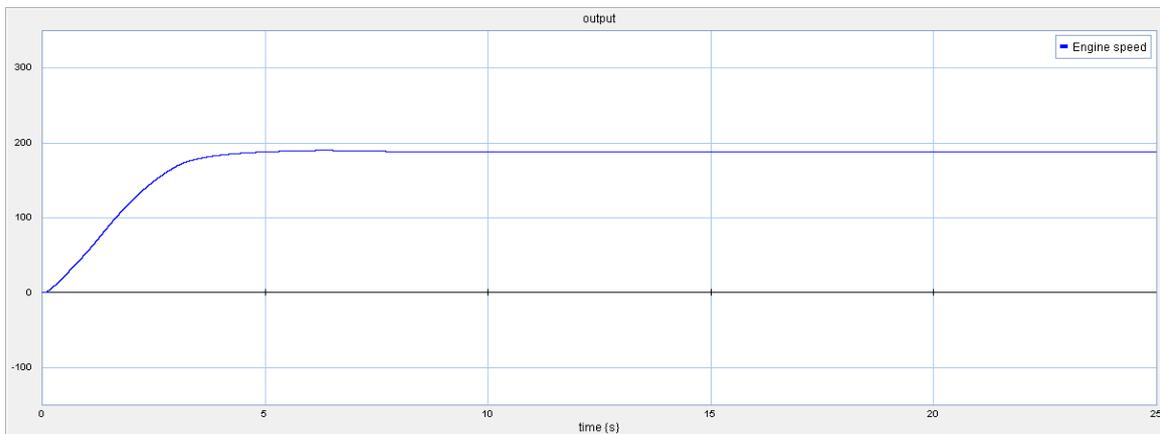


Figure 8.11 Four-stroke engine speed plot

Since the reference engine speed is 105 rpm which is 188 rad/s, the speed plot shows the engine starts acceleration and lasts for 6 seconds. Then the value keeps stable at 188 rad/s.

**Firing order**
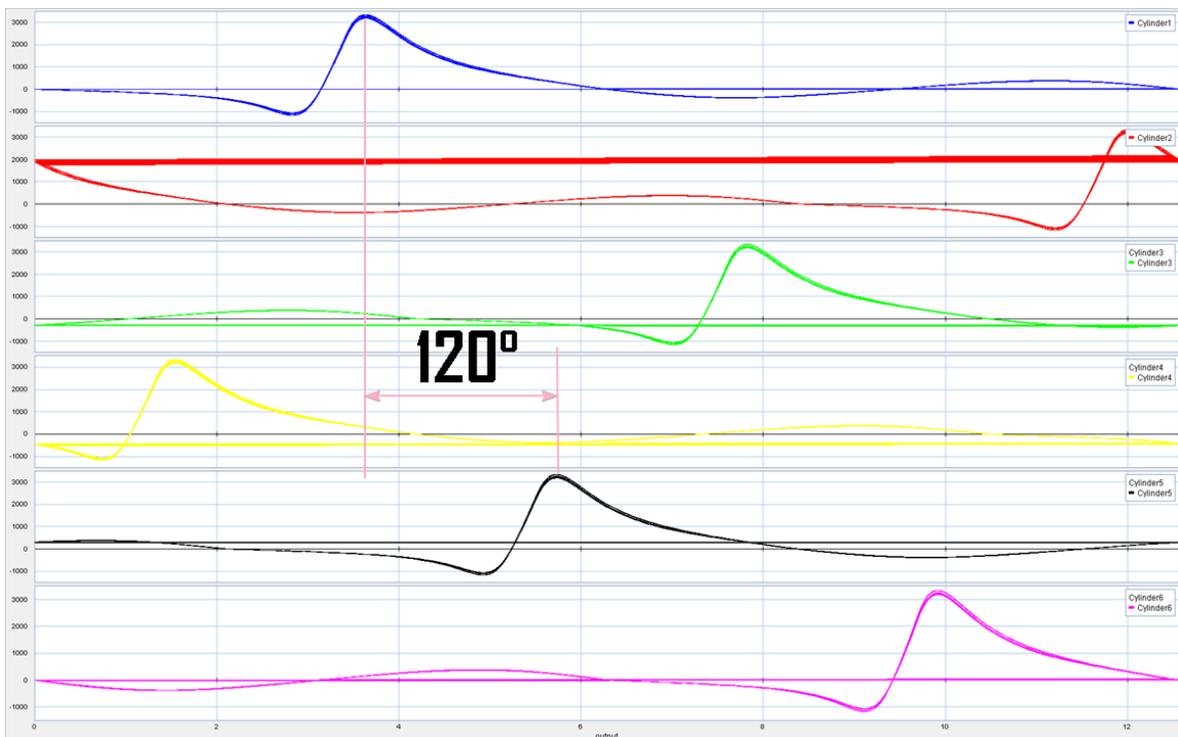


Figure 8.12 Four-stroke firing order plots

The firing order of 6 cylinders in this case is 1-5-3-6-2-4. But in four-stroke engine, the angle difference is 120°. The plot shows the same order as expected. The detailed parameters can be found in the table below. For example, the maximum torque happens at 208° in cylinder 1 and 327.7° in cylinder 5. The same angle difference can found between other cylinders.

| Cylinder number | Maximum position in rad | Maximum position in degree | Firing order |
|:---:|:---:|:---:|:---:|
| 1 | 3.63 | 208.0 | 1 |
| 2 | 12 | 687.5 | 5 |
| 3 | 7.82 | 448.0 | 3 |
| 4 | 1.53 | 87.7 | 6 |
| 5 | 5.72 | 327.7 | 2 |
| 6 | 9.91 | 567.8 | 4 |

Table 8.8 Four-stroke firing order plots results

**Pressure & Force & Torque**
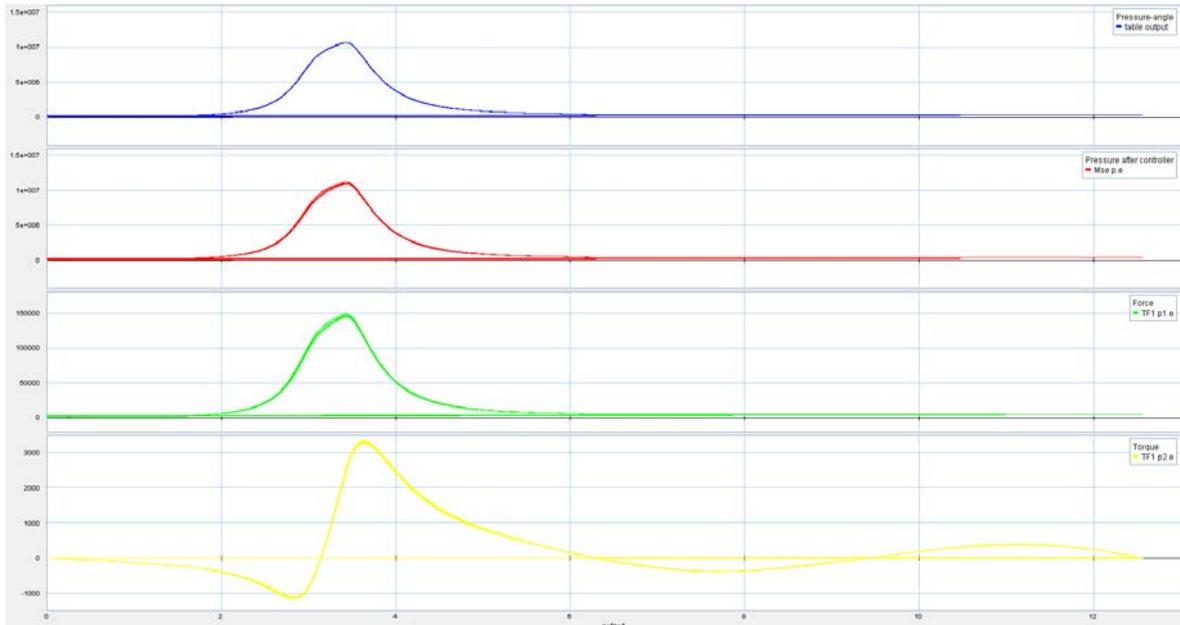


Figure 8.13 Four-stroke pressure, force, torque plots

The first plot is the pressure-angle relation input in the Table. The second plot is the output from MSe. The plot of MSe is affected by the PID controller scale factor. The PID controller signal is 1 at the beginning but when the speed is over reference speed, it reduces to 0.95. The maximum pressure in the cylinder is scaled by it also. The third plot is the force which is the output of the first transformer. Since the pressure is scaled by PID controller, the force will change with pressure as well because of the transform modulus is the piston area. The last plot is the torque which is after the second transformer and the force is changed into torque. It is also affected by the PID controller signal.

**Propeller characteristics**
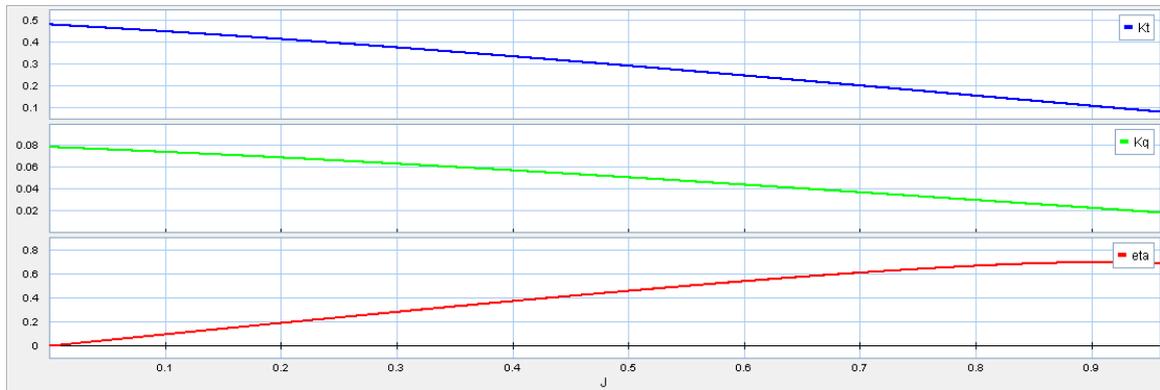


Figure 8.14 Four-stroke propeller characteristics plots

The trust and torque coefficients $K_T$ and $K_Q$ and efficiency plots are shown above. To prove the good combination of the engine model and the propeller model, the propeller characteristics can be explained also. According to Wageningen B-series propellers chart, 3 blades $A_E/A_O$=0.64 is selected. Since P/D=1.08 the KT and KQ and efficiency plots should close to the red line where P/D=1.1. Comparing the 20-sim plots and the Wageningen B-series propellers chart[24], the results show that the propeller model fits the engine model and works well.
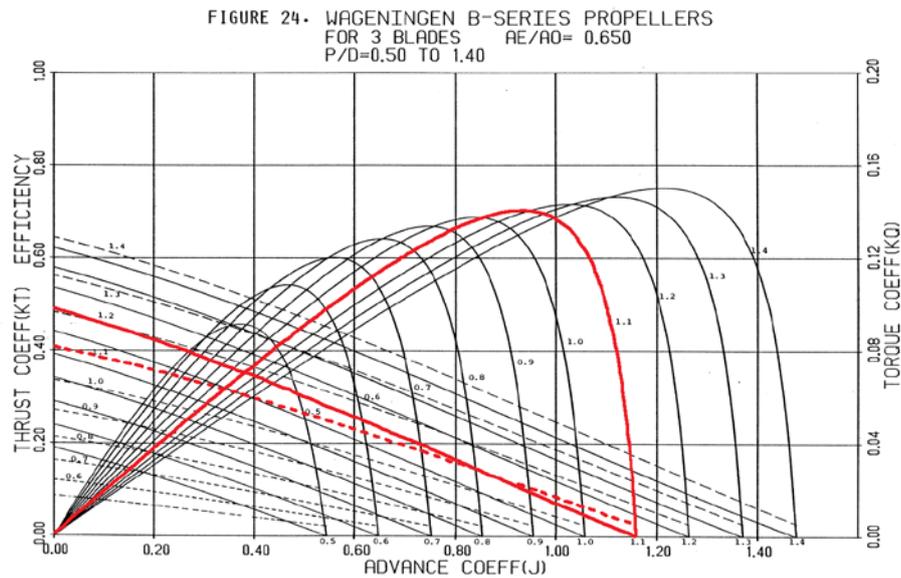


Figure 8.15 Wageningen B-series, 3 blades, AE/AO=0.65 [24]

**Torque & Trust force & Ship speed**
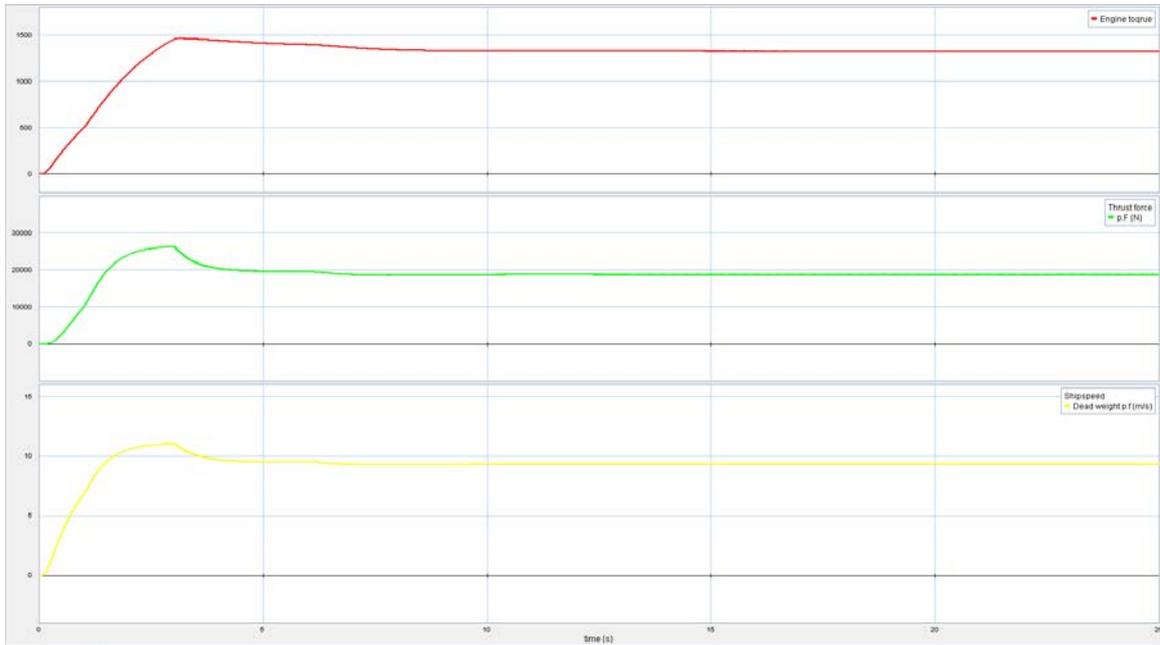


Figure 8.16 Four-stroke torque, trust force, ship speed plots

| Engine torque | 1.3 kNm |
|---|---|
| Trust force | 18.7 kN |
| Ship speed | 9.3 m/s or 18.1 knots |

Table 8.9

The torque output from the shaft is the mean torque, and it is different from the torque in the single cylinder. The torque drops to 1.3 kNm after about 6 seconds and keep stable. The trust force is 18.7 kN and ship speed is 18.1 knots. This is according to the ship dead weight input. Referring to the ships where Scania engines are installed, we assume that the ship weight is 2 tons.

**Power efficiency**
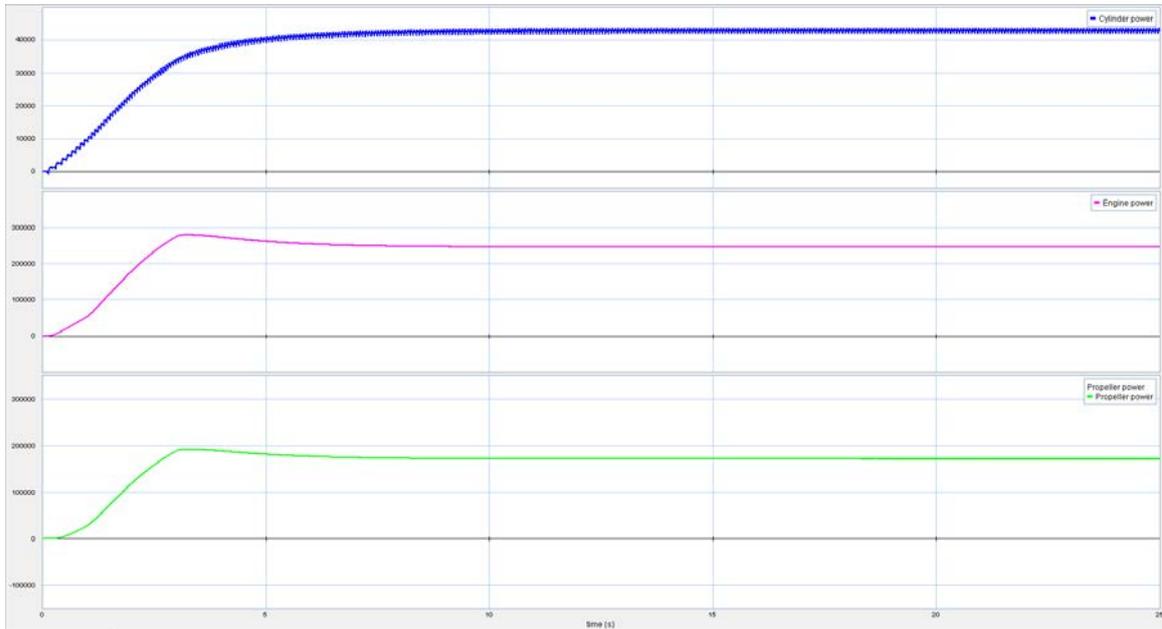


Figure 8.17 Four-stroke cylinder power, engine power, propeller power plots

| Cylinder power | 43.5 kW |
|---|---|
| Cylinder power*6 | 261 kW |
| Engine shaft power | 248.8 kW (torque*w) |
| $\eta_1$ | 95.3 % |
| Propeller power | 171.6 kW |
| $\eta_2$ | 69.0% |

Table 8.10 Wageningen B-series, 3 blades, AE/AO=0.65 [18]

Power sensors are applied in the single cylinder and on the shaft. The power is each cylinder is about 43.5 kW. Since six cylinders are connected in series so the total power is 261 kW. But there is friction loss in the cylinder and also bearing on the shaft, the real shaft output is 248.8 kW. So the efficiency

$$\eta_1 = \frac{248.8 \text{ kW}}{261 \text{ kW}} = 95.3\%$$

The propeller efficiency

$$\eta_2 = \frac{171.6 \text{ kW}}{248.8 \text{ kW}} = 69\%$$

The propeller efficiency also matches the propeller characteristics plots results.
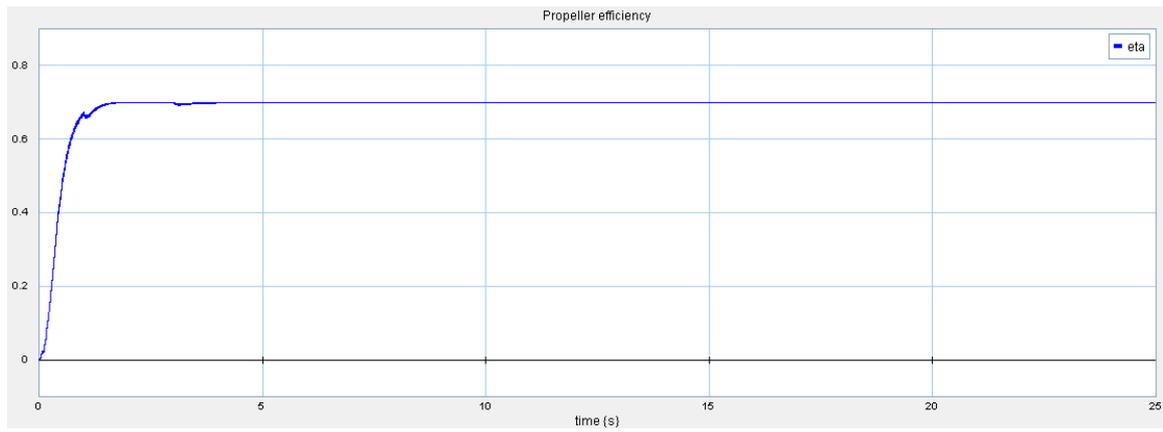
Figure 8.18 Four-stroke propeller efficiency

**Load scenario analysis 2**

To test the model works well, there should be some specific case applied to the engine. The engine load affects the working status of the engine. There is an assumption that if the load drops, how will the engine working status change.

Due to the theoretical analysis, when the engine load drops, the rotation speed of the engine will increase.
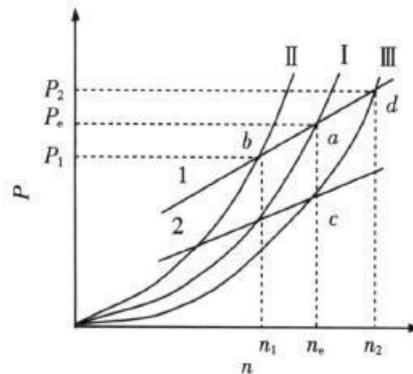


Figure 8.19 Load and speed curves

This figure describes the relation between engine load and rotation speed. Line 1 and line 2 are the speed characteristics lines and I, II, III are the propeller power curves. Assuming that the engine is working under load I with a speed Line 1 at Point a, the effective power is equal to load power. If the fuel supply is stable, when the **load suddenly drops**, curve I moves to curve III. The rotation speed increases. In this engine model, when the speed is over the reference speed, the PID controller output will drop. In real engine, the fuel supply will decrease. So Line 1 moves to Line 2, and there is a new Point c which means the engine speed back to reference after a small turbulence.

The same theory, assume the engine is working stably at Point c with Load III. When the **load increases**, curve III moves to curve I. If the fuel supply is stable, the engine speed reduces from $n_e$ to $n_1$. When the speed reduces and lower than the reference, the PID controller output will increase. In real life the engine fuel supply increases. So Line 2 moves to Line 1. The new Point a is the new engine working status. The speed comes back to $n_e$.

These guesses are based on the engine working theory. If the case is applied to the engine model, we can test if the result is similar to the assumptions.

The load model is MR element with a square signal:
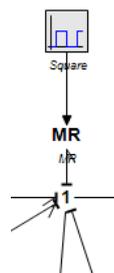


Figure 8.20

The square signal has amplitude 71100 with 0.02 rad/s frequency. So the load will be full load at the beginning and then drop to 0 at 160s and increase to full load after another 160s. The load is shown in the first plot below. Check the speed plot: the engine reach the reference speed 10.99 rad/s and keep stable in the first 160s. When the load suddenly drops, there is a small oscillation happens that the speed bump up a little and then back to 10.99 rad/s again after a few seconds. This is due to the function of PID controller. When the load is add again at 320s, the speed drops a little and back to reference speed again and keep stable.

The third figure is the PID controller output. The engine starts acceleration and reaches the reference speed. The signal keeps stable at 0.9. When the load suddenly drops, the signal reduces. In real engine, the governor reduces the fuel supply and keeps the engine running at reference speed. When the load is added, the controller signal increases. And after a few seconds the engine runs at reference speed again.
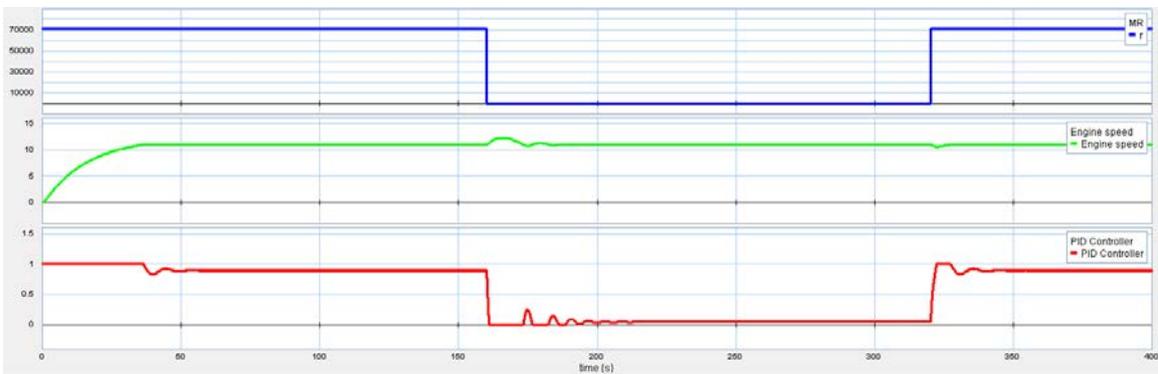


Figure 8.21 Load, speed, controller output plots

The plots are reasonable and similar to what as expected.

# 9 Discussion

**Plot of PID controller**

Both the two-stroke and four-stroke PID controller have the function of adjusting engine speed when there is a difference between the speed from shaft end and reference speed. In two-stroke engine the PID controller output keeps stable at 0.9 while four-stroke engine keeps at 0.95. This value is due to the engine load and there comparing the power the engine offers. In real life the governor of engine will adjust the speed by changing the fuel supply.

**Engine speed**

The engine speed is affected by the signal of PID controller. From the results, at the beginning, both the two-stroke and four-stroke engine speed up from zero if a start pulse is applied. When it reaches the reference speed, the controller signal keeps it stable to as the same value as the engine speed. In the two-stroke engine, the speed is 105 rpm and in four-stroke engine, the speed is 1800 rpm. The unit in the plot is rad/s. Through easy unit conversion, the stable speed in the plot is the same as reference engine speed which is defined in the constant signal.

**Firing order**

The firing order is 1-5-3-6-2-4 and the plots of both two-stroke engine and four-stroke engine show the same order as expected. This is due to the equations in angle reset. The detailed equations can be found in Appendix A. The firing order can be shown by the torque plots in each cylinder, and can be shown by other parameters as well such as pressure-angle plots in each cylinder. Just compare the maximum pressure position and the firing order and engine ignition intervals angle can be found out.

**Pressure & Force & Torque**

The pressure curve should be the same as input pressure-angle curve in the Table. The results are the same as expected in both two-stroke and four-stroke engine. The force is converted from the pressure curve. The shape is the same as pressure curve but the value is the pressure multiplied by the piston area. The results show the same value as calculated. Due to the affection of the PID controller, the curve has many layers. Because the PID controller output is stable at 0.9 in two-stroke engine and 0.95 in four-stroke engine, the maximum value is multiplied by the PID controller output. The torque is converted by the reciprocating geometry relation, which is the transformer modulus. The curve shape and value are the same as expected.

**Propeller characteristics**

The propeller model is from the previous student. In his thesis, he analyzed the propeller characteristics which including the trust and torque coefficients $K_T$ and $K_Q$ and efficiency. The engine model is connected with the propeller model and proper parameters are input into the model. The same parameters are tested in the integrated model. The results are similar to Wageningen B-series propellers chart. This means the engine and the propeller match each other and work properly.

**Torque & Trust force & Ship speed**

The torque is average torque which is different from the torque-angle in single cylinders. The trust force and ship speed are also received by the propeller model. The parameter inputs are close to real ship for example the dead weight. So the results received are also close to realistic.

**Power efficiency**

There are some losses in the propulsion system, from the engine, to the shaft, gear box and propeller. The efficiency is calculated according to the plot results and the values are reasonable to realistic. Some calculated parameters such as propeller efficiency also match the propeller characteristics plot. It proves more that the engine model and propeller model match each other.

# 10 Future work

## 10.1 More engine test

The generic diesel engine is ready to have more tests by different types of engine. Due to the limitation of data, these two models are only tested by 6-cylinder models. There are still other different types of engines. For example 4-cylinder engines, V-type engines, the working principle is the same but the models need more development to fit all kinds of engines.

## 10.2 Combination with other systems

The engine model has connected the propeller models and simulation results are acceptable. The engine model can also been applied on ship operation simulations or critical environment condition simulations. For example emission model can be connected. Integrate different systems and make them working together is the future ideal task.

## 10.3 Comparison with other software

20-sim is based on bond graph. The engine model can also been built in other software with different principle. By comparing the modelling methods and results, the engine results can be tested and at the same time, the advantages and disadvantages of different software can be found out. Choosing the better interface and more realistic simulation also benefits the future operations.

# 11 Conclusion

This thesis completed the task of creating a generic marine diesel engine model. Start from modeling a single cylinder and connect them together in series to build a 6-cylinder engine model. Input the model and realistic parameters into 20-sim. The propeller model from previous student work are also connected and tested. The results from simulation are also close to expected and explanations are offered.

In this generic model, some elements play an important role in the engine performance such as the pressure-angle data, the dimension of engine components. But some elements don't affect the results very much. This can be called sensitivity. The parameters such as crankshaft inertia and stiffness are very important in the torsional and vibration analysis of crankshaft, but these parameters don't bother the whole engine working performance as expected.

But still, more types of engines should be tested as explained in the future work. The model should be developed better and make it more convenient to use for the future projects. In a general view of whole ship systems, this model can be integrated and connected with other ship components and be tested. Wish this generic marine diesel engine model can make a contribution to the future projects and also the industry.

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# Appendix A

**Two-stroke engine**

**Constant**


Constant

```
parameters
        real C =10.99;              // output value
equations
        output = C;
```

**PID Controller**


Controller
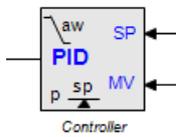
```
// Created by C. Kleijn, 22 Sep 1999
parameters
        real K = 1.0 {};                            // Proportional gain
        real Td = 1 {s};                    // Derivative time constant: Td > 0
        real N = 10 {};                             // Derivative gain limitation.
        real Ti = 1 {s};                            // Integral time constant: Ti > 0
        real b = 1.0 {};                            // Proportional setpoint weighting parameter:
0 <= b <= 1
        real c = 1.0 {};                    // Derivative setpoint weighting parameter: 0 <= c <=
1
        real Ta = 1.0 {s};                  // Tracking time constant.t: Ta > 0
        real minimum = 0 {}; // Minimum controller output
        real maximum = 1 {}; // Maximum controller output
variables
        real error,PB_high,PB_low;
        real hidden uP,uI,uD,uDstate,ideal_output;
equations
        error = SP - MV;
        uP = K * ( b * SP - MV );
        uI = int ( K * error / Ti  -  ( ideal_output - output ) / Ta );
        uDstate = int ( uD * N / Td );
        uD = K * (c * SP - MV ) * N - uDstate;
        ideal_output = uP + uI + uD;
        output = limit (ideal_output, minimum, maximum);
        PB_low = b * SP + (uI + uD - maximum)/K;
        PB_high = b * SP + (uI + uD - minimum)/K;
```

## Table

parameters
       string filename= 'E:\AAUC\20sim\Masterthesis\engine1.txt';
equations
       output = table ('E:\AAUC\20sim\Masterthesis\engine1.txt', input);

## Gain

parameters
       real global K;         // gain
variables
       real global KK;
equations
       output = KK*PID_input * input;

## MSe

variables
       real flow;
equations
       p.e = effort;
       flow = p.f;

## TF

parameters
  real global D;//piston diameter
variables
  real A;//piston head area
equations
  $A=(pi*D^2)/4$;
       p2.e = A * p1.e;
       p1.f = A * p2.f;

## TF1

parameters
  real global a,b;//piston rod length and connecting rod length
variables
  real r;

equations
      r = -a*sin(angle)-0.5*(a^2*sin(angle)*cos(angle))/sqrt(b^2-(a*sin(angle))^2);
  p2.e = r * p1.e;
      p1.f = r * p2.f;

**Angle reset**



variables
  real k;
equations
k=floor(input/(2*pi));
output=input-k*(2*pi);




variables
  real k;
  real angle;
equations
    angle = input-4*pi/3;
  k=floor(angle/(2*pi));
  output=angle-k*(2*pi);

**Inertia**



parameters
    real global i;
variables
    real global ii;
equations
  state = int(p.e);
  p.f = state / ii;

**Friction loss**



parameters
    real global r;
variables
    real global rr;
equations
    p.e = rr * p.f;

**Stiffness**

C
q3

parameters
  real global Lj;
  real global Dj;
  real global Lc;
  real global Dc;
  real global dj;
  real global dc;
  real global web;
  real global Lw;
  real global rad;
  real G=77000000000;
variables
  real global LJ;
  real global DJ;
  real global LC;
  real global DC;
  real global dJ;
  real global dC;
  real global Web;
  real global LW;
  real global Rad;
  real global c;
equations
  c=32*((LJ+0.4*DJ)/(DJ^4-dJ^4)+(LJ+0.4*DC)/(DC^4-dC^4)+(Rad-
0.2*(DJ+DC)/LW/Web^3))/(G*pi);
  state = int(p.f);
  p.e = state / c;

**Material damping**

R
R2

parameters
 real t=0.015;//material damping ratio of steel
 real k=200000000000;//stiffness youngs modulus=200GPa
 real global m1;
 real global m2;
variables
 real m, r1;
 real global m1m1;
 real global m2m2;
equations
 m=m1m1+m2m2;//mass of shaft
 r1=t*2*sqrt(k*m);
  p.e = r1 * p.f;

**Global parameters**



```
parameters
        //standard parameters
        //piston
        real global K=1;
        real global D=0.6;
   real global S=2.4;
   //propellers
   real global T= 1110;
   real global w= 10.9;
        //parameters can be changed automatically;
        real global a= 1.2;
        real global b=2.5;//piston rod length and connecting rod length
        //inetia value
        real global i= 4209;
        real global flywheel=20400;
        //to be tested engine parameter
   real DD= 0.6;
   real SS=2.4;
        //crank shaft friction loss
   real global r = 550;
   //crank shaft damping
        real global m1=11499.4;//web mass kg
        real global m2=837.6;//journal and crankpin mass kg
        //crankshaft dimesion
        real global Lj=0.17;
        real global Dj=0.6;
        real global Lc=0.21;
        real global Dc=0.6;
        real global dj=0.03;
        real global dc=0.03;
        real global web=1.26;
        real global Lw=0.32;
        real global rad=0.95;
variables
   //scaling parameter
   real global kD;
   real global kS;
   real global KK;
   //parameters to be scaled
   real global ii;
   real global rr;
   real global cc;
   real global aa;
   real global bb;
   real global m1m1;
   real global m2m2;
   real global LJ;
```

```
        real global DJ;
        real global LC;
        real global DC;
        real global dJ;
        real global dC;
        real global Web;
        real global LW;
        real global Rad;
equations
        kD=DD/D;
        kS=SS/S;
        KK=1*K;
        ii=kD^3.5*i;
        rr=kD*r;
        aa=kD*a;
        bb=kD*b;
        m1m1=kS^3*m1;
        m2m2=kD^3*m2;
        LJ=kD*Lj;
        DJ=kD*Dj;
        LC=kD*Lc;
        DC=kD*Dc;
        dJ=kD*dj;
        dC=kD*dc;
        Web=kS*web;
        LW=kS*Lw;
        Rad=kS*rad;
```

## Gear box

```
 ⇁ TF⊢⇁
 Gear_box
```

```
parameters
        real r = 1;
equations
        p1.e = r * p2.e;
        p2.f = r * p1.f;
```

## Propeller

```
⇁  Propeller  ⇁
```

```
constants
// Salt water density
        real ro=1025;
```

```
// Coefficients used for calculating Kt
        real C1[39,1] = [0.00880496; -0.204554; 0.166351; 0.158114; -0.147581; -
0.481497; 0.415437; 0.0144043; -0.0530054; 0.0143481; 0.0606826; -0.0125894;
0.0109689; -0.133698; 0.00638496; -0.00132718; 0.168496; -0.0507214; 0.0854559; -
0.0504475; 0.010465; -0.00648272; -0.00841728; 0.0168424; -0.00102296; -0.0317791;
```

0.018604; -0.00410798; -0.000606848; -0.0049819; 0.0025983; -0.000560528; -0.00163652; -0.000328787; 0.000116502; 0.000690904; 0.00421749; 0.0000565229; -0.00146564];

    real s1[39,1] = [0; 1; 0; 0; 2; 1; 0; 0; 2; 0; 1; 0; 1; 0; 0; 2; 3; 0; 2; 3; 1; 2; 0; 1; 3; 0; 1; 0; 0; 1; 2; 3; 1; 1; 2; 0; 0; 3; 0];

    real t1[39,1] = [0; 0; 1; 2; 0; 1; 2; 0; 0; 1; 1; 0; 0; 3; 6; 6; 0; 0; 0; 0; 6; 6; 3; 3; 3; 3; 0; 2; 0; 0; 0; 0; 2; 6; 6; 0; 3; 6; 3];

    real u1[39,1] = [0; 0; 0; 0; 1; 1; 1; 0; 0; 0; 0; 1; 1; 0; 0; 1; 2; 2; 2; 2; 2; 0; 0; 0; 1; 2; 2; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 2];

    real v1[39,1] = [0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2];

    real unit1[39,1] = [1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1];

// Coefficients used for calculating Kq

    real C2[47,1] = [0.00379368; 0.00886523; -0.032241; 0.00344778; -0.0408811; -0.108009; -0.0885381; 0.188561; -0.00370871; 0.00513696; 0.0209449; 0.00474319; -0.00723408; 0.00438388; -0.0269403; 0.0558082; 0.0161886; 0.00318086; 0.015896; 0.0471729; 0.0196283; -0.0502782; -0.030055; 0.0417122; -0.0397722; -0.00350024; -0.0106854; 0.00110903; -0.000313912; 0.0035985; -0.00142121; -0.00383637; 0.0126803; -0.00318278; 0.00334268; -0.00183491; 0.000112451; -2.97228E-05; 0.000269551; 0.00083265; 0.00155334; 0.000302683; -0.0001843; -0.000425399; 8.69243E-05; -0.0004659; 5.54194E-05];

    real s2[47,1] = [0; 2; 1; 0; 0; 1; 2; 0; 1; 0; 1; 2; 2; 1; 0; 3; 0; 1; 0; 1; 3; 0; 3; 2; 0; 0; 3; 3; 0; 3; 0; 1; 0; 2; 0; 1; 3; 3; 1; 2; 0; 0; 0; 0; 3; 0; 1];

    real t2[47,1] = [0; 0; 1; 2; 1; 1; 1; 2; 0; 1; 1; 1; 0; 1; 2; 0; 3; 3; 0; 0; 0; 1; 1; 2; 3; 6; 0; 3; 6; 0; 6; 0; 2; 3; 6; 1; 2; 6; 0; 0; 2; 6; 0; 3; 3; 6; 6];

    real u2[47,1] = [0; 0; 0; 0; 1; 1; 1; 1; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 0; 0; 0; 1; 1; 2; 2; 2; 2; 0; 0; 0; 1; 1; 1; 1; 2; 2; 2; 2; 2];

    real v2[47,1] = [0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 1; 0; 0; 0; 0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2];

    real unit2[47,1] = [1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1];

variables

    real Kt;
    real Kq;
    real J;
    real n;
    real Va;

    /*real c;
    real d;
    real global i;
    real global j;*/

    real J_array1[39,1];
    real P_over_D_array1[39,1];
    real Ae_over_Ao_array1[39,1];
    real z_array1[39,1];

```
        real Kt_array[39,1];

        real J_array2[47,1];
        real P_over_D_array2[47,1];
        real Ae_over_Ao_array2[47,1];
        real z_array2[47,1];
        real Kq_array[47,1];

        real eta;
parameters
        real D;
        real P_over_D;
        real Ae_over_Ao;
        real z;

equations
        n = (p1.f+10E-4) / (2*pi); // in revolutions per second.
        Va = p2.f+10E-8;
        J = Va / (n * D)+10E-10;

        J_array1 = J * unit1;
        P_over_D_array1 = P_over_D * unit1;
        Ae_over_Ao_array1 = Ae_over_Ao * unit1;
        z_array1 = z * unit1;

        Kt_array = C1 .* (J_array1 .^s1) .* (P_over_D_array1 .^ t1) .*
(Ae_over_Ao_array1 .^ u1) .* (z_array1 .^ v1);
        Kt = msum(Kt_array);

        J_array2 = J * unit2;
        P_over_D_array2 = P_over_D * unit2;
        Ae_over_Ao_array2 = Ae_over_Ao * unit2;
        z_array2 = z * unit2;

        Kq_array = C2 .* (J_array2 .^s2) .* (P_over_D_array2 .^ t2) .*
(Ae_over_Ao_array2 .^ u2) .* (z_array2 .^ v2);
        Kq = msum(Kq_array);

  p1.e = Kq * ro * D^5 * n^2;
        p2.e = Kt * ro * D^4 * n^2;
  eta= J*Kt/(2*pi*Kq);
```

Parameters | Initial Values | Constants

| Name | Value | Quantity | Unit | Description |
|---|---|---|---|---|
| ◆ Ae_over_Ao | 0.65 | | | |
| ◆ D | 7.1 | | | |
| ◆ P_over_D | 0.72 | | | |
| ◆ z | 3 | | | |

**Dead weight**
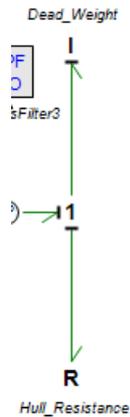


parameters
       real i = 15609000;
equations
  state = int(p.e);
  p.f = state / i;

**Hull resistance**

parameters
       real hull_resistance_coef = 16181;
equations
       p.e = hull_resistance_coef * p.f^2;

**Startup pulse**
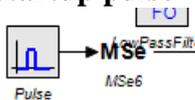


parameters
       real start_time = 1.0 {s};
       real stop_time = 3 {s};
       real amplitude = 9600 {none};
variables
       boolean hidden change;
equations
       "calculate at least at the start and stop time"
       change = timeevent (start_time) or timeevent (stop_time);

       "calculate the block signal"
       output = amplitude * (step (start_time) - step (stop_time));

**Four-stroke**

**Constant**


Constant

parameters
      real C = 188;           // output value
equations
      output = C;

**PID Controller**


Controller
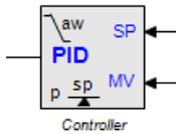
```
// Created by C. Kleijn, 22 Sep 1999
parameters
        real K = 1.0 {};                        // Proportional gain
        real Td = 0.5 {s};              // Derivative time constant: Td > 0
        real N = 10 {};                         // Derivative gain limitation.
        real Ti = 1 {s};                        // Integral time constant: Ti > 0
        real b = 1.0 {};                        // Proportional setpoint weighting parameter:
0 <= b <= 1
        real c = 1.0 {};                 // Derivative setpoint weighting parameter: 0 <= c <=
1
        real Ta = 1.0 {s};               // Tracking time constant.t: Ta > 0
        real minimum = 0 {}; // Minimum controller output
        real maximum = 1{};  // Maximum controller output
variables
        real error,PB_high,PB_low;
        real hidden uP,uI,uD,uDstate,ideal_output;
equations
        error = SP - MV;
        uP = K * ( b * SP - MV );
        uI = int ( K * error / Ti  - ( ideal_output - output ) / Ta );
        uDstate = int ( uD * N / Td );
        uD = K * (c * SP - MV ) * N - uDstate;
        ideal_output = uP + uI + uD;
        output = limit (ideal_output, minimum, maximum);
        PB_low = b * SP + (uI + uD - maximum)/K;
        PB_high = b * SP + (uI + uD - minimum)/K;
```

## Table



parameters
       string filename= 'E:\AAUC\20sim\Masterthesis\engine5.txt';
equations
       output = table ('E:\AAUC\20sim\Masterthesis\engine5.txt', input);


## Gain



parameters
       real global K;           // gain
variables
       real global KK;
equations
       output = KK*PID_input * input;


## MSe



variables
       real flow;
equations
      p.e = effort;
      flow = p.f;


## TF



parameters
  real global D;//piston diameter
variables
  real A;//piston head area
equations
  A=(pi*D^2)/4;
      p2.e = A * p1.e;
      p1.f = A * p2.f;


## TF1



parameters
  real global a,b;//piston rod length and connecting rod length
variables

real r;
equations
    r = -a*sin(angle)-0.5*(a^2*sin(angle)*cos(angle))/sqrt(b^2-(a*sin(angle))^2);
    p2.e = r * p1.e;
    p1.f = r * p2.f;

**Angle reset**
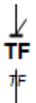


variables
    real k;
equations
k=floor(input/(4*pi));
output=input-k*(4*pi);



variables
    real k;
    real angle;
equations
        angle = input-8*pi/3;
    k=floor(angle/(4*pi));
    output=angle-k*(4*pi);

**Inertia**



parameters
        real global i;
variables
        real global ii;
equations
    state = int(p.e);
    p.f = state / ii;

**Friction loss**



parameters
        real global r;
variables
        real global rr;
equations
        p.e = rr * p.f;

**Stiffness**

C
φ3

parameters
      real global Lj;
      real global Dj;
      real global Lc;
      real global Dc;
      real global dj;
      real global dc;
      real global web;
      real global Lw;
      real global rad;
      real G=77000000000;
variables
      real global LJ;
      real global DJ;
      real global LC;
      real global DC;
      real global dJ;
      real global dC;
      real global Web;
      real global LW;
      real global Rad;
      real global c;
equations
      c=32*((LJ+0.4*DJ)/(DJ^4-dJ^4)+(LJ+0.4*DC)/(DC^4-dC^4)+(Rad-0.2*(DJ+DC)/LW/Web^3))/(G*pi);
      state = int(p.f);
      p.e = state / c;

**Material damping**

R
R2

parameters
  real t=0.015;//material damping ratio of steel
  real k=200000000000;//stiffness youngs modulus=200GPa
  real global m1;
  real global m2;
variables
  real m, r1;
  real global m1m1;
  real global m2m2;
equations
  m=m1m1+m2m2;//mass of shaft
  r1=t*2*sqrt(k*m);
  p.e = r1 * p.f;

**Global parameters**

```
parameter
```

parameters
        //standard parameters
        //piston
        real global K=1;
        real global D=0.13;
        real global S=0.14;
        //crank shaft
        real global d=0.05;//shaft diameter
    real global l=0.03;//shaft length
    //propellers
    real global T= 1110;
    real global w= 188;
    real global flywheel=2;
        //parameters can be changed automatically;
        real global a= 0.07;
        real global b=0.225;//piston rod length and connecting rod length
        real global i= 0.09;
        //changing parameter
    real DD= 0.13;
    real SS= 0.14;
        //crank shaft friction loss
    real global r = 0.02;
    //crank shaft damping
        real global m1=12.17;//web mass kg
        real global m2=8.49;//journal and crankpin mass kg
        //crankshaft dimesion
        real global Lj=0.08;
        real global Dj=0.108;
        real global Lc=0.06;
        real global Dc=0.087;
        real global dj=0.006;
        real global dc=0.004;
        real global web=0.14;
        real global Lw=0.03;
        real global rad=0.07;
variables
    //scaling parameter
    real global kD;
    real global kS;
    real global KK;
    real global dd;
    real global ll;
    real global ii;
    real global rr;
    real global cc;
    real global m1m1;
```

```
    real global m2m2;
    real global LJ;
        real global DJ;
        real global LC;
        real global DC;
        real global dJ;
        real global dC;
        real global Web;
        real global LW;
        real global Rad;
equations
        kD=DD/D;
        kS=SS/S;
        KK=1.05*K;
        dd=kD*d;
        ll=kD*l;
        ii=kD^3.5*i;
        rr=kD*r;
        m1m1=kS^3*m1;
        m2m2=kD^3*m2;
        LJ=kD*Lj;
        DJ=kD*Dj;
        LC=kD*Lc;
        DC=kD*Dc;
        dJ=kD*dj;
        dC=kD*dc;
        Web=kS*web;
        LW=kS*Lw;
        Rad=kS*rad;
```

## Gear box


Gear_box

```
parameters
        real r = 0.25;
equations
        p1.e = r * p2.e;
        p2.f = r * p1.f;
```

## Propeller



```
constants
// Salt water density
        real ro=1025;

// Coefficients used for calculating Kt
        real C1[39,1] = [0.00880496; -0.204554; 0.166351; 0.158114; -0.147581; -
0.481497; 0.415437; 0.0144043; -0.0530054; 0.0143481; 0.0606826; -0.0125894;
```

0.0109689; -0.133698; 0.00638496; -0.00132718; 0.168496; -0.0507214; 0.0854559; -0.0504475; 0.010465; -0.00648272; -0.00841728; 0.0168424; -0.00102296; -0.0317791; 0.018604; -0.00410798; -0.000606848; -0.0049819; 0.0025983; -0.000560528; -0.00163652; -0.000328787; 0.000116502; 0.000690904; 0.00421749; 0.0000565229; -0.00146564];

    real s1[39,1] = [0; 1; 0; 0; 2; 1; 0; 0; 2; 0; 1; 0; 1; 0; 2; 3; 0; 2; 3; 1; 2; 0; 1; 3; 0; 1; 0; 0; 1; 2; 3; 1; 1; 2; 0; 0; 3; 0];

    real t1[39,1] = [0; 0; 1; 2; 0; 1; 2; 0; 0; 1; 1; 0; 0; 3; 6; 6; 0; 0; 0; 0; 6; 6; 3; 3; 3; 3; 0; 2; 0; 0; 0; 0; 2; 6; 6; 0; 3; 6; 3];

    real u1[39,1] = [0; 0; 0; 0; 1; 1; 1; 0; 0; 0; 0; 1; 1; 0; 0; 0; 1; 2; 2; 2; 2; 2; 0; 0; 0; 1; 2; 2; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 2];

    real v1[39,1] = [0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2];

    real unit1[39,1] = [1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1];

// Coefficients used for calculating Kq

    real C2[47,1] = [0.00379368; 0.00886523; -0.032241; 0.00344778; -0.0408811; -0.108009; -0.0885381; 0.188561; -0.00370871; 0.00513696; 0.0209449; 0.00474319; -0.00723408; 0.00438388; -0.0269403; 0.0558082; 0.0161886; 0.00318086; 0.015896; 0.0471729; 0.0196283; -0.0502782; -0.030055; 0.0417122; -0.0397722; -0.00350024; -0.0106854; 0.00110903; -0.000313912; 0.0035985; -0.00142121; -0.00383637; 0.0126803; -0.00318278; 0.00334268; -0.00183491; 0.000112451; -2.97228E-05; 0.000269551; 0.00083265; 0.00155334; 0.000302683; -0.0001843; -0.000425399; 8.69243E-05; -0.0004659; 5.54194E-05];

    real s2[47,1] = [0; 2; 1; 0; 0; 1; 2; 0; 1; 0; 1; 2; 2; 1; 0; 3; 0; 1; 0; 1; 3; 0; 3; 2; 0; 0; 3; 3; 0; 3; 0; 1; 0; 2; 0; 1; 3; 3; 1; 2; 0; 0; 0; 0; 3; 0; 1];

    real t2[47,1] = [0; 0; 1; 2; 1; 1; 1; 2; 0; 1; 1; 1; 0; 1; 2; 0; 3; 3; 0; 0; 0; 1; 1; 2; 3; 6; 0; 3; 6; 0; 6; 0; 2; 3; 6; 1; 2; 6; 0; 0; 2; 6; 0; 3; 3; 6; 6];

    real u2[47,1] = [0; 0; 0; 0; 1; 1; 1; 1; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 0; 0; 0; 1; 1; 2; 2; 2; 2; 0; 0; 0; 1; 1; 1; 1; 2; 2; 2; 2; 2];

    real v2[47,1] = [0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 0; 0; 0; 0; 0; 0; 0; 0; 0; 0; 1; 1; 1; 1; 1; 1; 1; 1; 1; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2; 2];

    real unit2[47,1] = [1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1; 1];

variables
    real Kt;
    real Kq;
    real J;
    real n;
    real Va;

    /*real c;
    real d;
    real global i;
    real global j;*/

    real J_array1[39,1];
    real P_over_D_array1[39,1];

```
        real Ae_over_Ao_array1[39,1];
        real z_array1[39,1];
        real Kt_array[39,1];

        real J_array2[47,1];
        real P_over_D_array2[47,1];
        real Ae_over_Ao_array2[47,1];
        real z_array2[47,1];
        real Kq_array[47,1];

        real eta;

parameters
        real D;
        real P_over_D;
        real Ae_over_Ao;
        real z;

equations
        n = (p1.f+10E-4) / (2*pi); // in revolutions per second.
        Va = p2.f+10E-8;
        J = Va / (n * D)+10E-10;

        J_array1 = J * unit1;
        P_over_D_array1 = P_over_D * unit1;
        Ae_over_Ao_array1 = Ae_over_Ao * unit1;
        z_array1 = z * unit1;

        Kt_array = C1 .* (J_array1 .^s1) .* (P_over_D_array1 .^ t1) .*
(Ae_over_Ao_array1 .^ u1) .* (z_array1 .^ v1);
        Kt = msum(Kt_array);

        J_array2 = J * unit2;
        P_over_D_array2 = P_over_D * unit2;
        Ae_over_Ao_array2 = Ae_over_Ao * unit2;
        z_array2 = z * unit2;

        Kq_array = C2 .* (J_array2 .^s2) .* (P_over_D_array2 .^ t2) .*
(Ae_over_Ao_array2 .^ u2) .* (z_array2 .^ v2);
        Kq = msum(Kq_array);
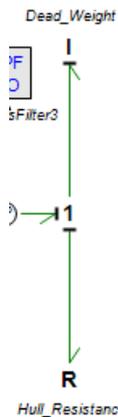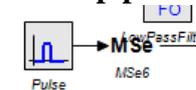
  p1.e = Kq * ro * D^5 * n^2;
        p2.e = Kt * ro * D^4 * n^2;
        eta= J*Kt/(2*pi*Kq);
```

| Parameters | Initial Values | Constants |

| Name | Value | Quantity | Unit | Description |
|---|---|---|---|---|
| ◆ Ae_over_Ao | 0.65 | | | |
| ◆ D | 1.35 | | | |
| ◆ P_over_D | 1.08 | | | |
| ◆ z | 3 | | | |

## Dead weight

Dead_Weight

I

PF
D

sFilter3

)—|1

R
Hull_Resistance

parameters
      real i = 2000;
equations
  state = int(p.e);
  p.f = state / i;

## Hull resistance
parameters
      real hull_resistance_coef = 215;
equations
      p.e = hull_resistance_coef * p.f^2;

## Startup pulse

FO

MSe  LowPassFilt
Pulse  MSe6

parameters
      real start_time = 1.0 {s};
      real stop_time = 3 {s};
      real amplitude = 400 {none};
variables
      boolean hidden change;
equations
      "calculate at least at the start and stop time"
      change = timeevent (start_time) or timeevent (stop_time);

      "calculate the block signal"
      output = amplitude * (step (start_time) - step (stop_time));

## Appendix B

**Two-stroke**
Propeller selections
For the two-stroke engine in this case, the calculation of propeller size is shown below:
Propeller shaft power: $P_D = 1428000kW$
Engine speed: $n_E = 105rpm$
Gear ratio:$i = 1:1$
Ship speed:$v_S = 15knots$
Wake coefficient:w $= 0.2$
Sea water density:$\rho = 1030kg/m^3$
Propeller speed: $n_P = \frac{n_E}{i*60} = 1.75revs/sec$
Propeller advance speed in water $v_A = (1 - w) * v_S = 6.2m/s$
Blade number: z $= 3$

$$B_P = \sqrt[4]{(\frac{1}{2\pi\rho})\frac{P_D*n_P{}^2}{V_A{}^5}} = 0.93$$

Blade area ratio is a parameter used to relate the size of a propeller blade to its diameter. It is critical to the control of cavitation and changes to blade area ratio affect its efficiency and thrust-making performance.

| Blade number (Z) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Blade area ratio$A_E/A_O$ | | | | | | | | | | | |
| 2 | 0.30 | | | | | | | | | | | |
| 3 | | 0.35 | | | 0.50 | | | 0.65 | | | 0.80 | | |
| 4 | | | 0.40 | | | 0.55 | | | 0.70 | | | 0.85 | 1.00 |
| 5 | | | | 0.45 | | | 0.60 | | | 0.75 | | | 1.05 |
| 6 | | | | | 0.50 | | | 0.65 | | | 0.80 | | |
| 7 | | | | | | 0.55 | | | 0.70 | | | 0.85 | |

Table of Extent of theWageningen B-screw series [18]

In this case, blade area ratio $\frac{A_E}{A_O} = 0.65$

**Four-stroke**

Pitch diameter ratio

$$\frac{P}{D} = 0.73$$
$$\sigma = 2.1$$
$$D = 7.4m$$
$$k = \frac{P_D}{v_s{}^3} = 4231.1$$

Since there is small error between the calculation result and realistic value, after adjusting the propeller diameter value, when D=7.1m the engine model works well with 90% of full load.

For four-stroke engine:

Propeller shaft power: $P_D = 234005kW$

Engine speed: $n_E = 1800rpm$

Gear ratio:$i = 4:1$

Ship speed:$v_s = 20knots$

Wake coefficient:$w = 0.2$

Sea water density:$\rho = 1030kg/m^3$

Propeller speed: $n_P = \frac{n_E}{i*60} = 7.5revs/sec$

Propeller advance speed in water $v_A = (1 - w) * v_s = 8.2m/s$

Blade number: $z = 3$

$$B_P = \sqrt[4]{(\frac{1}{2\pi\rho})\frac{P_D*n_P{}^2}{v_A{}^5}}=0.48$$

$$\frac{A_E}{A_O} = 0.65$$



Pitch diameter ratio $\frac{P}{D} = 1.08$

$$\sigma = 1.18$$
$$D = 1.3m$$
$$k = \frac{P_D}{v_s{}^3} = 215.4$$