BACHELOR THESIS & COLLOQUIUM - ME 141502

# MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.


MUHAMMAD RAMADHAN PAMUNGKAS

NRP. 4213101037


SUPERVISORS:

Prof. Dr.-Ing. Karsten Wehner

Dipl. Ing. Hartmut Schmidt


DOUBLE DEGREE PROGRAM
MARINE ENGINEERING DEPARTMENT
FACULTY OF MARINE TECHNOLOGY
INSTITUT TEKNOLOGI SEPULUH NOPEMBER
SURABAYA 2017

**BACHELOR THESIS & COLLOQUIUM - ME 141502**

# MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.

MUHAMMAD RAMADHAN PAMUNGKAS

NRP. 4213101037

SUPERVISORS:

Prof. Dr.-Ing. Karsten Wehner

Dipl. Ing. Hartmut Schmidt

DOUBLE DEGREE PROGRAM
MARINE ENGINEERING DEPARTMENT
FACULTY OF MARINE TECHNOLOGY
INSTITUT TEKNOLOGI SEPULUH NOPEMBER
SURABAYA 2017

**APPROVAL FORM**

**MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.**

**BACHELOR THESIS**

Submitted to Comply One of The Requirements to Obtain a Bachelor Engineering Degree
in
Double Degree of Marine Engineering (DDME) Program
Department of Marine Engineering - Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember (ITS)
Department of Maritime Studies - Faculty of Engineering
Hochschule Wismar, University of Applied Sciences Technology, Business and Design

Submitted by:

**MUHAMMAD RAMADHAN PAMUNGKAS**
NRP. 4213101037
Approved by
Head of Marine Engineering Department

**Dr. Eng. M. Badrus Zaman, S.T., M.T.**
NIP. 1977 0802 2008 01 1007

SURABAYA

JULY, 2017

*"This page intentionally left blank"*

**APPROVAL FORM**

**MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.**

**BACHELOR THESIS**

Submitted to Comply One of The Requirements to Obtain a Bachelor Engineering Degree
in
Double Degree of Marine Engineering (DDME) Program
Department of Marine Engineering - Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember (ITS)
Department of Maritime Studies - Faculty of Engineering
Hochschule Wismar, University of Applied Sciences Technology, Business and Design

Submitted by:

**MUHAMMAD RAMADHAN PAMUNGKAS**
NRP. 4213101037

Approved by
Representative of Hochschule Wismar in Indonesia

**Dr.-Ing. Wolfgang Busse**

SURABAYA
JULY, 2017

*"This page intentionally left blank"*

## APPROVAL FORM

**MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.**

**BACHELOR THESIS**

Submitted to Comply One of The Requirements to Obtain a Bachelor Engineering Degree
in
Double Degree of Marine Engineering (DDME) Program
Department of Marine Engineering - Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember (ITS)
Department of Maritime Studies - Faculty of Engineering
Hochschule Wismar, University of Applied Sciences Technology, Business and Design

Submitted by:

**MUHAMMAD RAMADHAN PAMUNGKAS**
NRP. 4213101037

Approved by Bachelor Thesis Supervisor:

**Prof. Dr.-Ing. Karsten Wehner**

**Dipl. Ing. Hartmut Schmidt** for

ROSTOCK

JULY, 2017

*"This page intentionally left blank"*

**DECLARATION OF HONOR**

I, who signed below hereby confirm that:

This final project report has written without any plagiarism act, and confirm consciously that all the data, concepts, design, references, and material in this report own by Marine Power Plant Laboratory (MPP) in department of Marine Engineering ITS which are the product of research study and reserve the right to use for further research study and its development.

| | | |
|---|---|---|
| Name | : | Muhammad Ramadhan Pamungkas |
| NRP | : | 4213 101 037 |
| Bachelor Thesis Title | : | MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.<br><br>Engine |
| Department | : | Double Degree Program of Marine Engineering |

If there is plagiarism act in the future, I will fully responsible and receive the penalty given by ITS and Hochschule Wismar according to the regulation applied.

Surabaya, July 2017

Muhammad Ramadhan Pamungkas

*"This page intentionally left blank"*

**MODELLING TURBOCHARGER CUT OFF APPLICATION DUE TO SLOW STEAMING OPERATION 12RTA96C-B ENGINE.**

**Name : Muhammad Ramadhan Pamungkas**

**NRP : 4213 101 037**

**Department : Double Degree Program of Marine Engineering**

**Supervisor : Prof. Dr.-Ing. Karsten Wehner**

**Dipl. Ing. Hartmut Schmidt**

ABSTRACT

Out of the total operational costs of a ship, fuel costs account for by far the highest proportion. In view of the global economic situation and the rising oil prices, shipowners and charterers are looking for solutions to cut costs by reducing fuel consumption. Low load operation, also well-known as "slow steaming", represents the currently most effective and popular measure to cut fuel costs and, in consequence, the total operational costs for increased competitiveness in the market. Low load operation is possible and there is an increasing trend to operate in these very low engine load ranges. As the engines were not designed for this operational condition, various retrofit modifications to the engine can compensate for this.

By using low load operation, the reduction of the RPM gives problems when sailing at low speed. A turbocharger (TC) compresses inlet air to a high pressure and after cooling this compressed air it results in higher mass of air in the cylinder. But when running at a low power load this air reaches temperatures that are too low for an optimal combustion process. One of the solution comes from the company Wärtsilä. They install so called "**low steam engine kits"**. When this kit is installed it allows the engine operators to cut off one turbocharger of the engine, this result's in a higher RPM for the operating turbochargers. When the remaining TC's have a higher RPM their efficiency improves and gives the engine more air for combustion.

The goal of this Bachelor thesis is to make a calculation modelling and prove that by switching off one or more turbocharger on the system will improve the efficiency in slow steaming operation. Beside that, this thesis is aims to estimated the performance of the engine in both operation condition.

**PEMODELAN APLIKASI TURBOCHARGER CUT OFF DIKARENAKAN PENGOPERASIAN SLOW STEAMING PADA 12RTA96C-B ENGINE.**


**Nama : Muhammad Ramadhan Pamungkas**

**NRP : 4213 101 037**

**Departemen : Double Degree Program of Marine Engineering**

**Dosen Pembimbing : Prof. Dr.-Ing. Karsten Wehner**

**Dipl. Ing. Hartmut Schmidt**



ABSTRAK

Dari total biaya operasional sebuah kapal, biaya bahan bakar mencapai proporsi tertinggi. Mengingat situasi ekonomi global dan kenaikan harga minyak, pemilik kapal dan penyewa kapal mencari solusi untuk mengurangi biaya dengan mengurangi konsumsi bahan bakar. Operasi dengan beban rendah, juga dikenal sebagai "slow steaming", merupakan cara yang paling efektif dan populer saat ini untuk mengurangi biaya bahan bakar dan, akibatnya, total biaya operasional menurun untuk meningkatkan daya saing di pasar. Operasi dengan beban rendah dimungkinkan dan ada kecenderungan meningkat untuk beroperasi pada rentang beban mesin yang sangat rendah ini. Karena mesin tidak dirancang untuk kondisi operasional ini, berbagai modifikasi retrofit pada mesin dapat mengimbangi hal ini.

Dengan menggunakan operasi beban rendah, pengurangan RPM memberi masalah saat berlayar dengan kecepatan rendah. Sebuah turbocharger (TC) memampatkan udara masuk ke tekanan tinggi dan setelah mendinginkan udara bertekanan ini menghasilkan massa udara yang lebih tinggi di dalam silinder. Tapi saat beroperasi di daya rendah, udara ini mencapai suhu yang terlalu rendah untuk proses pembakaran yang optimal. Salah satu solusinya berasal dari perusahaan Wärtsilä. Mereka memasang apa yang disebut "low steam engine kits". Saat kit ini dipasang, memungkinkan operator mesin memotong satu turbocharger mesin, hasil ini ada pada RPM yang lebih tinggi untuk turbocharger operasi. Bila TC yang tersisa memiliki RPM yang lebih tinggi, efisiensinya akan meningkat dan memberi mesin lebih banyak udara untuk pembakaran.

Tujuan dari tesis ini adalah untuk membuat pemodelan perhitungan dan membuktikan bahwa dengan mematikan satu atau lebih turbocharger pada sistem akan meningkatkan efisiensi dalam slow steaming. Disamping itu, skripsi ini bertujuan untuk memperkirakan kinerja mesin pada kedua kondisi operasi

## PREFACE

This bachelor submitted to fulfil the requirement to obtain Bachelor of Engineering Degree in Department of Marine Engineering, Sepuluh Nopember Institute of Technology and Hochschule Wismar.

First of all, I would like to thank Allah SWT for blessing and helping me to complete this bachelor thesis timely. I wish to express my gratitude to my supervisor Prof. Dr.-Ing. Karsten Wehner and Dipl. Ing. Hartmut Schmidt for guidance, support, knowledge and advice given for accomplishing this thesis. My thanks to Hans Baier from MARIDIS company for the guidance to obtain some data required of engine operation from the company. I would also thanks to Mrs. Rau as Wismar University caretaker for international student, Jost, and Ben as a student buddy in Germany and all of my friends who helped me during my stay and study in Rostock, Germany.

The greatest honor and appreciation would be finally dedicated to my beloved parents and all my family for the support, love, care and motivation. I would also addressed my appreciation to my beloved family of Marine Engineering students, especially for Double Degree class of 2013, thank you so much for being such a great companion during our togetherness at campus.

Surabaya, July 2017

Writer,

Muhammad Ramadhan Pamungkas

*"This page intentionally left blank"*

**Task for Bachelor-Thesis**

Subject: **Modelling Turbocharger cut off application due to slow steaming operation 12RTA96C-B engine.**

Student: **Muhammad Ramadhan Pamungkas**

Supervising Professor: Prof. Dr.-Ing. Karsten Wehner

Assistant Supervisor: Dipl. Ing. Hartmut Schmidt
Maridis Company - Mr. Hans Baier

Date of issue: December 20$^{th}$ 2016
Filing date: July 4$^{th}$ 2017

Due to slow steaming operation trend, ships with large engine are no longer required to operated in their optimized load range. Consequently there are several optimizations that considered to be performed, one of which is turbocharger cut off operation. That change also alters the performance characteristic and that is the subject to be discussed in this thesis. Within the scope of the Bachelor thesis, the following tasks are to be implemented :

- Explanation of the slow steaming operation and technicals problem posibilities
- Explanation of turbocharger cut off method
- Mathematical calculation modelling of the performance
- Comparison model result with the reference curves

The supervising Professor reserves the rights to extend or to narrow down the scope of the task during processing. Establishing contacts with other institutions and companies must be agreed with the supervisors. The publication of the work or parts of it requires the prior permission of the supervisor. The work shall be prepared in accordance with the applicable guidelines of Hochschule Wismar for academic and scientific work. At least two consultations with the supervising Professor are required as part of the processing. The finished work is to be submitted in electronic form and in four printed copies in the organization office in Warnemünde.

Prof. Dr.-Ing. Karsten Wehn

**Table of Contents**

**List of Figure**



**List of Table**

**List of Formula**

## List of Abbraviation

| | |
|---|---|
| ME | Main Engine |
| TC | Turbocharger |
| MCR | Maximum Continous Rate |
| RPM | Rotations per Minute |
| $(SFOC)\ or\ b_e$ | Specific Fuel Oil Consumption (g/Kwh or kg/Ws) |
| $\dot{Q}$ | Heat Energy (J/s or W) |
| $H_u$ | Lower Calorific Value (kJ/kg) |
| $P_e$ | Effective Power (W) |
| $P_i$ | Indicative Power (W) |
| $\eta$ | Efficiency |
| γ/K | (Kappa Value) Specific Heat Ratio |
| $c_p$ | Specific Heat at Constant Pressure (J/kgK) |
| $\dot{m}$ | Mass flow rate (kg/s) |
| P | Pressure (bar) |
| T | Temperature (K) |
| $C$ | Heat Energy (J/kg) |
| V | Volume ($m^3$) |

# CHAPTER I
# INTRODUCTION

## 1.1 Background

Ships commonly sail at the most economic speed. This is the speed where the engines are originally designed for. The optimal load range engine lies between 70-85%. The fuel efficiency of the engine, its operational parameters, the specification of the turbochargers, coolers, auxiliary systems, exhaust gas boilers, and so on, are chosen and optimised for that normal load range. But when the engine is operated continuously in a load range below or even far below 60%, the overall system is no longer fully optimized. Recently, more ships demand to sail at lower speeds because the high speed is no longer needed due to economical section. The problem is that this is not useful to do because of the lower efficiency on these speeds. Therefore, the main question is how is it possible to sail at a low speed without losing any efficiency?

Engine's on big vessels have their own optimal load on which the efficiency of the engine is highest. When we slow down and without any adjustments to the propulsion system, the efficiency will drop down enormously. The following subjects are the most interesting to investigate when we sail at a lower speed: the cooling system, the combustion process, the turbocharger, the lubrication oil system and the engines wear and the fuel consumption. The subjects will be discussed on how the turbocharger can be adjusted to raise the efficiency on a lower speed.

The reduction of the RPM gives problems when sailing at low speed. A turbocharger (TC) compresses inlet air to a high pressure and after cooling this compressed air it results in higher mass of air in the cylinder. But when running at a low power load this air reaches temperatures that are too low for an optimal combustion process. The main issue of slow speed steaming is the low RPM of the turbochargers (TC).

On engines with two or more turbochargers, this can be overcome by applicating a turbocharger cut-off method. Thereby, the performance of the remaining turbochargers will be improved from running at higher revolutions, which again reduces the fuel oil consumption further because the better quality of the air that delivered into the combustion chamber. In this thesis, then result of modeling are displayed in the form of performance curves. Where in the curve we can see the performance of the engine in the normal operation condition with 3 TC and performance when TC cut off is performed in the low load operation.

## 1.2 PROBLEMS

Based on the description above the statement problem of this thesis are:

1. How the turbocharger cut off method will affect the performance of 12RTA96C-B engine operation?
2. How much the efficiency improvement that will be obtained by using this method on the turbocharger and compare it with the initial condition.

Scope of Problem

1. The simulation using simulation calculation and modeling
2. Simulation only used to find the performance of the turbocharger on operational method and the performance of turbocharger which had been modified by turbocharger cut-off method.
3. The experiment will be done to compare the performance of the turbocharger which had been modified by turbocharger cut-off method.

## 1.3 RESEARCH LIMITATION

1. The thesis object is limited to the performance of 12RTA96C-B engine operation on variety load.

## 1.4 RESEARCH OBJECTIVES

The objectives of this thesis are:

1. To compare the current system with the turbocharger cut off method application on 12RTA96C-B engine operation on variety loads.
2. To analyze the performance of this method in 12RTA96C-B engine operation that operated on low load condition.

## 1.5 RESEARCH BENEFITS

1. Knowing the performance of using turbocharger cut off method 12RTA96C-B engine operation.
2. Provide a reliable calculation and modelling for estimate the performance of turbocharger cut off on 12RTA96C-B engine operation.

*"This page intentionally left blank"*

# CHAPTER II
# LITERATURE STUDY

## 2.1 Overview Slow Steaming Operation ( Case Study )

Ships commonly sail in slow steaming condition which means that ship is sailing at the most economic speed. There is a lot of factor why the ships nowadays is operating in slow steaming condition, such as the global economy turndown, environmental regulation on emission, the financial crisis, the sudden fall in ship values, high fuel cost and many more. The biggest single cost factor in merchant shipping, particularly for container and other large vessels, is the fuel oil. And the easiest way to reduce this cost is to reduce the ship's speed. When ship is sailing in the slow steaming condition, the engine is only run on 60% load or less. For example, reducing the nominal ship speed from 27 to 22 knots (-19%) will reduce the engine power to 42% of its nominal output (CMCR). This results in hourly main engine fuel oil savings of approximately 58%. A further reduction down to 18 knots saves already 75% of the fuel. The reduced speed however results in a longer voyage time; therefore, the fuel savings per roundtrip (for example Asia- Europe-Asia) are reduced by 45% at 22 knots, or 59% at 18 knots.

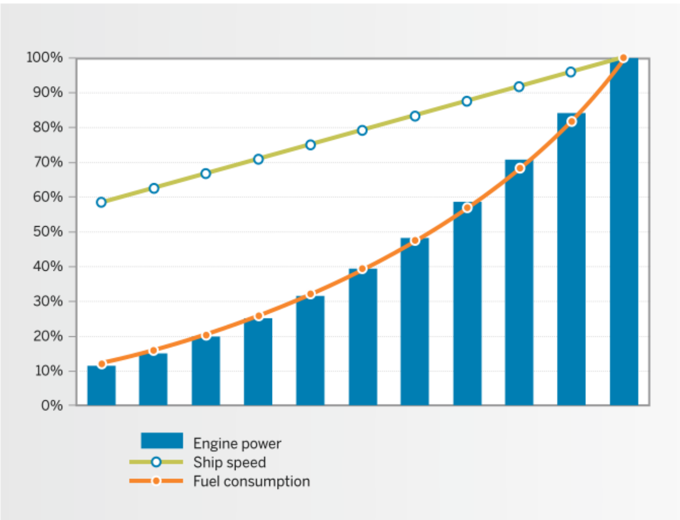



Fig. 1 – Correlation between ship speed, required engine power and fuel consumption.

*Figure 2.1.1 Correlation curves between ship speed and fuel oil consumption*
*Source : WÄRTSILÄ TECHNICAL JOURNAL 02.2010 - Slow steaming – a viable long-term option*

Slow steaming is preferred because it offers greater flexibility to increase the capacity again when the market situation changes. And there are other big advantages coming as a free side effect of slow steaming, namely that for every ton of fuel saved, the industry reduces its carbon dioxide emissions ($CO^2$) to the atmosphere by three tons, and the cylinder lubricating oil consumption of the main engine is reduced at almost the same percentages as the fuel, which also reduces solid particle emissions.

Theres other factor that led the shipping industry to consider the slow steaming operation such as The downturn in the global economy, resulting in reduced transportation capacity demand, The substantial global order book for new tonnage, the global financing crisis, high fuel costs, increasing operating costs (manning, lube oil, maintenance).The change to a long-term slow steaming scenario needs, however, a number of considerations. The optimal load range of the two-stroke engine lies between 70-85%. The fuel efficiency of the engine, its operational parameters, the specification of the turbochargers, coolers, auxiliary systems, exhaust gas boilers, and so on, are chosen and optimised for that normal load range. It is natural, therefore, that when the engine is operated continuously in a load range below or even far below 60%, the overall system is no longer fully optimised.

Wärtsilä has investigated the various concerns that have been raised across the low load range, different engine conditions can be observed. The possible consequences of continuously operating at reduced load without taking the recommended precautions are:

### 2.1.1 The cooling system

When sailing at a low speed, the cooling system doesn’t need that many adjustments. There is a negative effect: The cooling water temperature lowers quite a bit. This is because of the lower RPM of the main engine which results in lower temperatures in the cylinder. That is why the cooling water is also colder when it comes out of the cylinder. When the engine runs on lower load the mean cylinder temperature drops to a temperature where cold corrosion can occur in the cylinder.

For this problem the solution is to make sure the temperature of the cylinder is high enough to have a good combustion process and to prevent cold corrosion. This can be done by heating the high temperature water system (HT-system) and by recirculation of the water by a Thermostatic valve.

During field research it found out that container vessel Maersk Salalah also has these problems when low speed steaming. They heat the HT-system with steam to a water temperature between 86 and 96 degrees if in order to outcome these problem.

### 2.1.2 The lubricating oil system

The lubricating system has multiple functions: lubricating, cooling, cleaning, sealing and  noise reduction. Lubricating is an obvious function. A film of lubrication oil will form between the moving parts of the engine, preventing that engine parts will slide against each other. There are lubricating oils which are used for cooling some engine parts.

A negative effect that could occur is that the viscosity of the lube-oil will change. Because the lube-oil of the main system flows through the whole engine. When an engine runs at a lower load, the engines temperature will decrease. This can result in lube oil with a higher viscosity. Pumping the lube-oil through the engine becomes now more difficult. This will result in a loss of certain functions. One of the functions the lube-oil will lose, is the cooling of certain engine parts. The oil does get too thick to flow easily through the engine.

The most positive effect of running at lower loads is that the cylinder lubricating system will consume less lube-oil. The engine is running at a lower RPM, thereby reducing the amount of oil needed to create an oil film on the cylinders surface. Also the film of lube oil has more time to recover. The lube-oil consumption of the main lubricating system will not change.

Normally a certain amount of lubricant is injected into the cylinder to lubricate the liner this happens at a constant amount each stroke. MAN B&W tries to prevent the built up of soot with Alpha lubricators. This lube-oil system electronically controls the amount of lube-oil pumped into the cylinder. By controlling the amount of lube-oil, a lot of lube-oil can be saved.

### 2.1.3 The combustion process

If the Maximum Continues Rate (MCR) of a vessel is lowered, which means lowering the RPM of the engine. There may be changes to the engines combustion process. This can be either positive or negative. An example of these changes are:

- Air usage of the engine
- Exhaust gas temperatures

- Mean cylinder pressures
- Fuel consumption

The efficiency of the combustion process depends on many things. The air consumption of an engine is an important factor in the combustion process. The air is used to ignite with fuel that is injected just before the piston reaches its top death center in the cylinder. This air is supplied by turbochargers which are driven by the exhaust gasses of the engine.

Insufficient air, the reduction of the RPM gives problems when sailing at low speed. A turbocharger (TC) compresses inlet air to a high pressure and after cooling this compressed air it results in higher mass of air in the cylinder. But when running at a low power load this air reaches temperatures that are too low for an optimal combustion process.

A turbocharger is a centrifugal pump that compresses air. The compressor is driven by a turbine which is driven by the exhaust gasses from the cylinder. When the revolutions of the TC drop due to less exhaust gasses when operating on low power load it is logical that the amount of air compressed is less than on a higher power load.

One of the solution solution comes from the several engine company like Wartsilla and MAN B&W. They install so called low steam engine kits, it allows the engine operators to cut off one turbocharger of the engine, this result's in a higher RPM for the operating turbochargers. The higher load of the other turbochargers will result in higher inlet air temperatures but this is no problem for the overall process. When the remaining TC's have a higher RPM their efficiency improves and gives the engine more air for combustion. (*source: Efficient low speed sailing by Christiaan Muilwijk, Wouter den Boer and Laurens van der Kooij*)

## 2.2 Turbocharger Working Principle

Turbochargers are a type of forced induction system. They compress the air flowing into the engine. The advantage of compressing the air is that it lets the engine squeeze more air into a cylinder. Therefore, you get more power from each explosion in each cylinder. A turbocharged engine produces more power overall than the same engine without the charging. A turbocharger is composed of 3 basic parts, a compressor, a turbine, and a center housing. The turbine is the section of the turbocharger where the exhaust gases of the engine are forced through to cause the turbine wheel to spin. This rotation energy is then transferred through the center housing and into the compressor by means of shaft.

The compressor is spun by the rotational force created by exhaust gases flowing through the turbine. This would feed the intake side of the motor. Air is inducted into the compressor and then compressed into the piping, feeding the air intake ports of the motor. This creates an increased flow, as well as density, of air to be fed into the combustion chambers of the engine. The more oxygen that can be forced into the motor means that more fuel can be added to maintain a stabilized combustion. This in turn causes a larger, more powerful combustion. Thus, increasing the power output of the motor.

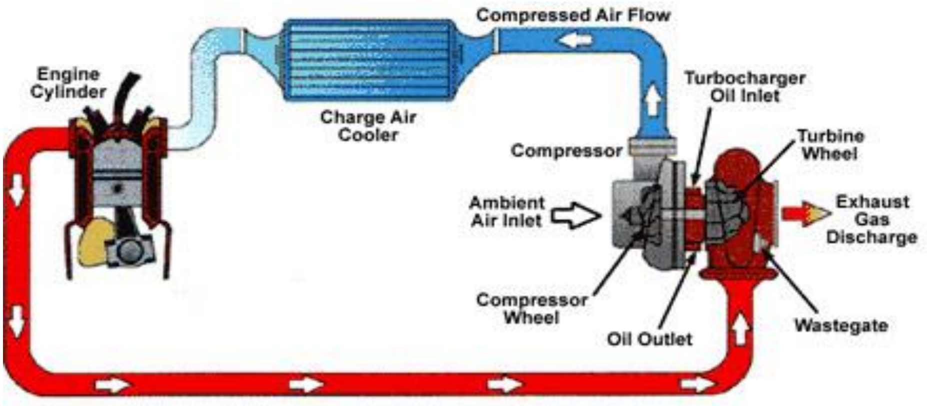



*Figure 2.2.1 Turbocharger system*
*Source : https://garrett.honeywell.com/*

The diagram above depicts the process of utilizing the engines exhaust gases to force clean air into the motor for combustion. In the diagram above, you may notice a "charge air cooler" or more commonly known as an intercooler. Most turbocharged platforms utilize an intercooler to cool the compressed air back down to the ambient air temperature.

This is due to the fact that heat is transferred from the turbine of the turbocharger to the compressor. A higher temperature air becomes less dense of oxygen molecules, which intern cause less oxygen to flow into the combustion chambers and produces a smaller, less powerful combustion. So to counter this effect, an intercooler is implemented to cool the air back down.

## 2.3 Exhaust Gas Operate Method

In general, the manner in which the energy of exhaust gases is utilized to drive the turbocharger may be ascribed to:

### 2.3.1 Constant-Pressure (Accumulation Charging)

In constant pressure type turbocharger, the exhaust gases gets collected in a single exhaust gas reservoir, where the pressure is maintained constant so as to avoid any fluctuation in the exhaust gas energy pressure. The exhaust gas is introduced to the turbine side after maintaining the pressure inside the cylinder. Exhaust gas with constant pressure type allows for the concetration of all cylinder gas output, so as the result of such things make air mass flow fluctutations, low pressure in the manifold and relatively constant. The turbochager cut-off will be perform in this type of charging. (*Turbo Magazine I: 1993*).

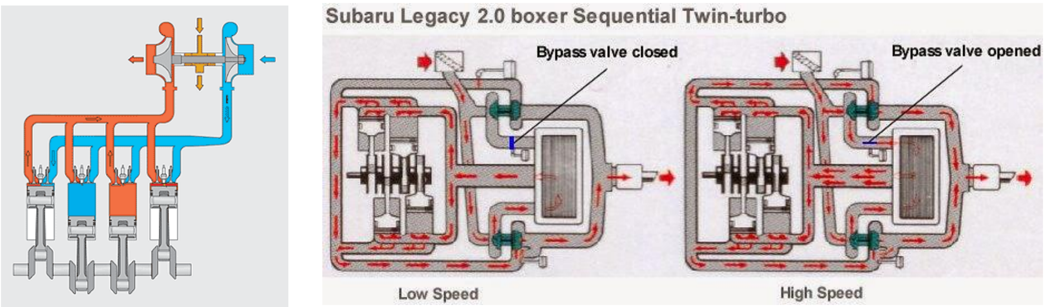



*Figure 2.3.1 Accumulation charging system*
*Source : https://www.theturboforums.com*

### 2.3.2 Pulse System

In pulse type turbocharger, the exhaust gas directly enters the turbine side and drives the turbine with the exhaust gas energy. The connection from the exhaust side of an engine is directly connected to the turbine side of a turbo charger. The pipe connections from the exhaust gas towards the turbine side are generally small in length and exhaust grouping is provided to prevent the blowback of gases from one cylinder to another. But because the exhaust gas from each cylinder goes directly to the turbine side, the turbocharger cut-off cant be applicated in this type of turbocharger.

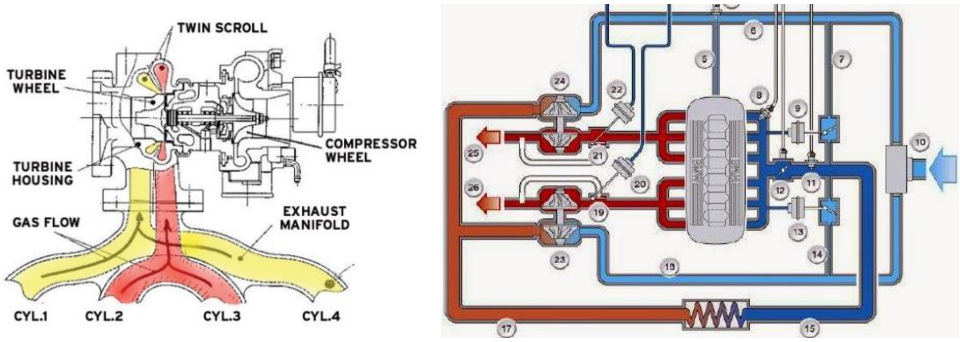



*Figure 2.3.2 Pulse charging system*

*Source : http://www.superstreetonline.com/how-to/engine/modp-0906-twin-scroll-turbo-system-design/*

## 2.4 Turbocharger Cut-off Method

### 2.4.1 Turbocharger Cut-off Overview

The reduction of the RPM gives problems when sailing at low speed. A turbocharger (TC) compresses inlet air to a high pressure and after cooling this compressed air it results in higher mass of air in the cylinder. But when running at a low power load this air reaches temperatures that are too low for an optimal combustion process.

One of the solution comes from company Wartsilla. they install so called low steam engine kits. When this kit is installed it allows the engine operators to cut off one turbocharger of the engine, this result’s in a higher RPM for the operating turbochargers. The higher load of the other turbochargers will result in higher inlet air temperatures but this is no problem for the overall process. When the remaining TC’s have a higher RPM their efficiency improves and gives the engine more air for combustion.

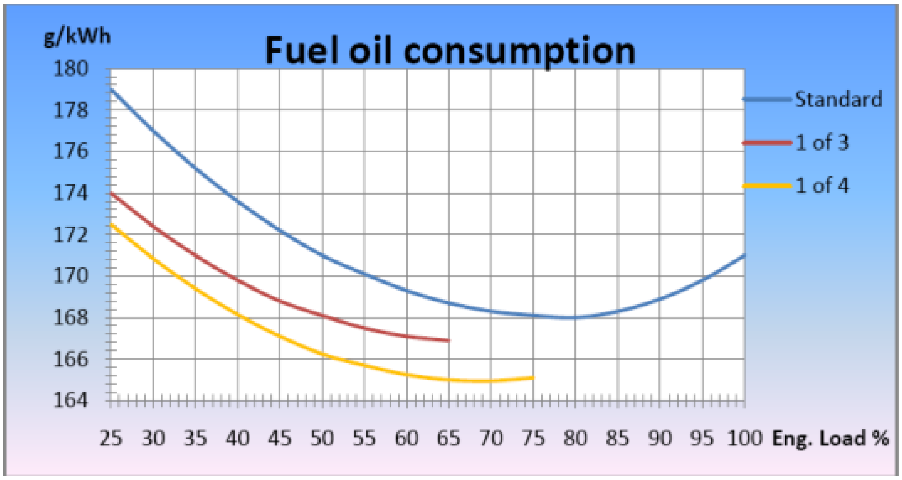



*Figure 2.4.1 SFOC vs Engine Load*
*Source : Doosan Engine TC Cut off operational Report*

Engines with four turbochargers and one turbocharger cut-out enables operation at loads from 20% to 70% MCR:

- Approx. 5g/kwh reduction of SFOC and 0.33bar increase in scavenge air pressure at 50% power. (1 T/C cut-out of 4 T/Cs)
- Approx. 3g/kwh reduction of SFOC and 0.40bar increase in scavenge air pressure at 75% power. (1 T/C cut-out of 4 T/Cs)

The solution to cut off one turbocharger is also done by MAN B&W. MAN B&W achieves to cut out a turbocharger with blind plates or valves. The following information is received from MAN during field research. The best way to cut off a turbocharger is done by valves. These valves can be operated simply by hand after the operators have slowed down the engine to low load operation.

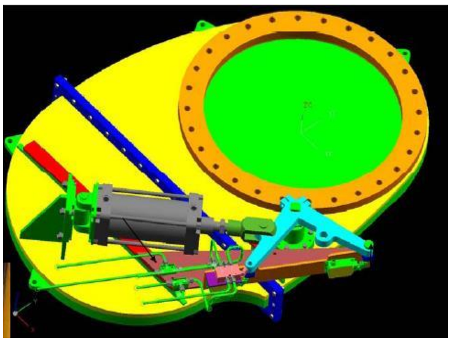


*Figure 2.4.2 Pneumatic Drive Valve*
*source: Efficient low speed sailing by Christiaan Muilwijk, Wouter den*

These are the application concept of turbocharger cut off. By installing the "blind plate" or "pneumatic drive valve" for automatic operation.

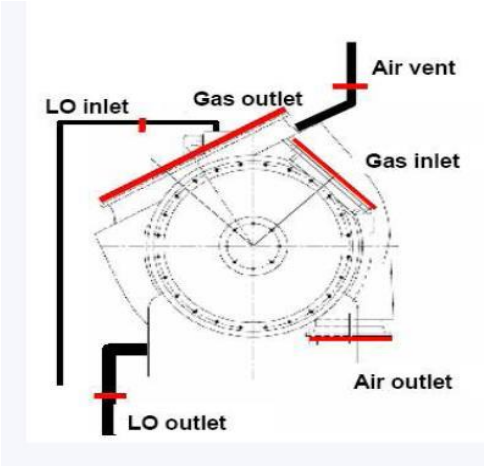



*Figure 2.4.3 Illustration of channel closure on TC cut off method.*
*Source : Doosan Engine TC Cut off operational Report*

Compare with another methods to solve the air problem in slow steaming operation:

**Table 2.4.1 Method comparison Table**

| Method | Advantage | Disadvantage |
|---|---|---|
| 1. Choose a less powerful engine | Cheaper initial investment | Limits speed for entire ship life |
| 2. Derate a new engine | Significant SFOC reduction | Typically limits speed for entire ship life |
| 3. Increase scavenge air pressure at low load (Method of TC CUT OUT) | Improve SFOC and reduce heat load on components in the 30-50% load area | Major modification with regard to IMO regulations |

# CHAPTER III
# REPORT METHODOLOGY

## 3.1 General

The methodology is a description of the steps carried out in a study. Methodology in this thesis include all activities carried out to solve a problem or process of analysis and evaluation of the problems this thesis.

## 3.2 Bachelor Thesis Methodology

In order to find the parameters of turbocharger output from the system, the data will be collected from many research related to the topic of this thesis. The methodology used in this thesis can be described as follows:

1. Indentification and Problem Formulation of the Problem

This thesis begins with indentif and formulating the problems regarding to the experiment to be carried as well limitation issue. It aims to simplify the problem.

2. Study Literature

Literature study conducted with collecting references to the turbocharger, the characteristics of turbocharger, and engine performance. The goal is to strengthen the basic theory problems as in the analysis.

3. Data Collection

The data used in this bachelor thesis is the data of 12RTA96C-B engine operation. Data for this analysis will be collected from engine specification and turbocharger specification.

4. Turbocharger Performance Simulation (Mathematical Model)

Analysis the characteristic of turbocharger based on engine performance parameter and the turbocharger cut off modification is done by making a mathematical model. The data collected will be used for making this calculation simulation. The simulation compared between the initial of turbocharger with the data of turbocharger that already modified by turbocharger cutoff method with the same engine specification.

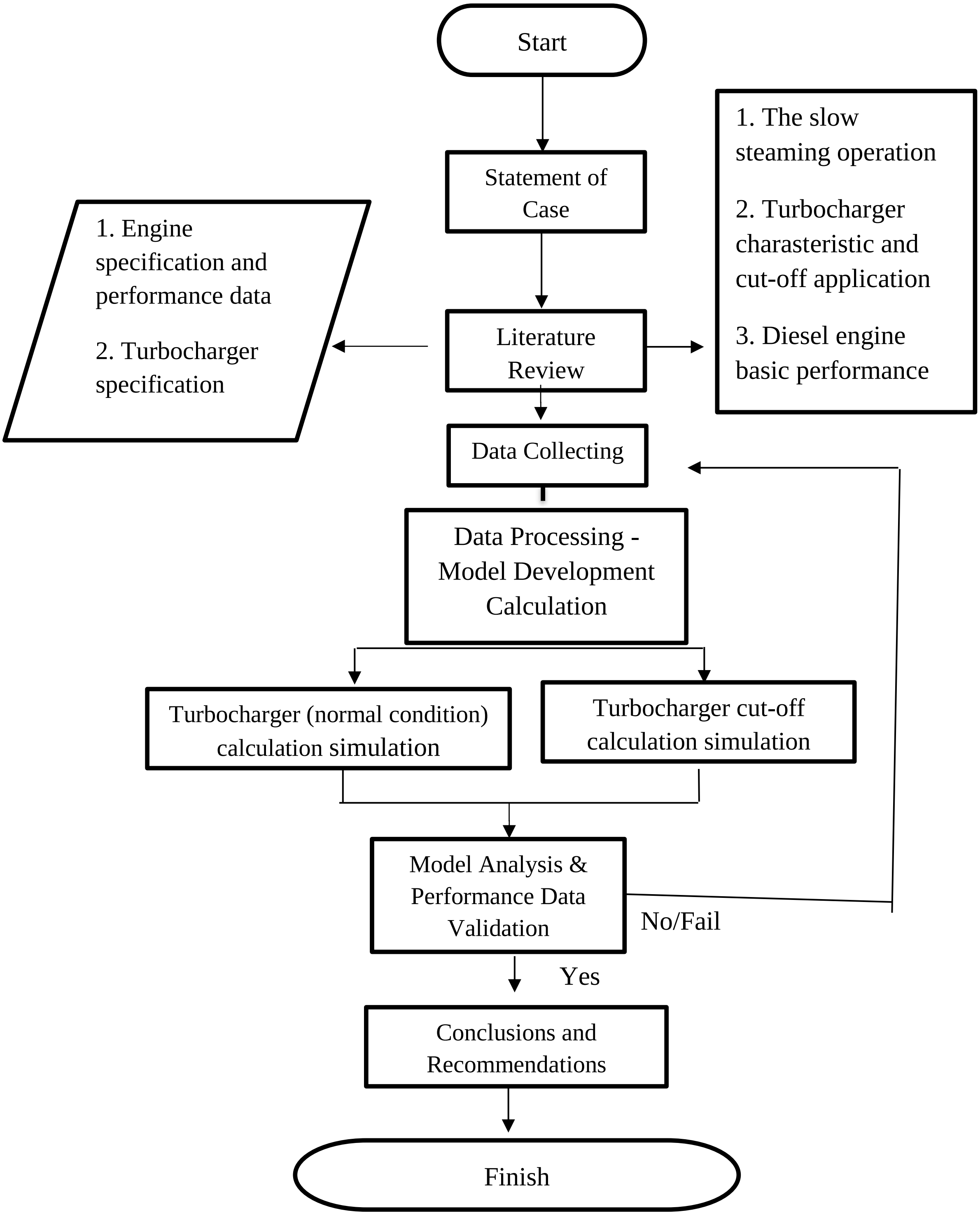
5. Turbocharger Performance Data Validation

At this stage, the output data from simulation process not exactly resulted the correct data of the performance. The output data obtained as a result from engine simulation might be had some of error. So we have to repeat the step of the simulation stage to find failures and corrected the input data.

6. Conclusions and Recommendations

Conclusions are expected in this thesis is able to answer the problem. This thesis aims to calculate and estimate the engine performance. Written advice based on data from the discussion well as the fact that there is, and given to the improvement of this bachelor thesis in order to become better.

7. Flowchart Thesis



# CHAPTER IV
# ENGINE PERFORMANCE MODELLING AND ANALYZING

## 4.1 Basis

Due to the constantly increasing demands on the company in recent years of using slow steaming operation, most of them is performing a more thoroughly monitoring and evaluation of their engine performance. Ships owners start paying more attention to their engine fuel consumption, the efficiency and also the pollutant emissions. There still quite industry that has experiencecy growth in this case in recent years. In normal operation, the evaluation is performed by testing the engine engine directly and evaluating the test bench. These record of the actual state is compared with the record from the engine maker, since the record is evaluate the same operation this will be easy to evalute.

The analysis of the differences between the record from the test and from the engine maker, it allows to predict the unintentional of events that may affect potential improvement or malfunctions in the machine operation. By using turbo-off technology, it will make some operating parameters value of the engine change significantly. Because this is a case of a result of operation at low load, so there is no comparative data. To monitor and evaluate this motor, it is necessary to a quality prediction or modelling of the changed operating parameters. The following model was created from this purpose.

The goal is to develop a workable model on a wide variety of Turbocharger engine configurations. So the future of this model is expected to be the basis for calculation when evaluating engine performance. So, from the beginning, a lot of simplification and idealization should be done so that calculations can be done more precisely. The calculations presented in the following chapter are thus Only the first step for a more comprehensive calculation of the circle. For modeling the count in this chapter, it is deliberately focused on modeling a closed circuit.

## 4.2 Record Data Collection

The basic calculation is an 12RTA96C-B engine performance report. 12RTA96C-B engine is built by Wartsilla company. The data for this thesis is obtained from MARIDIS Company. There There are 2 data, the data from the engine test and also reference data from TEKOMAR prediction result as a result comparasion with this model. It contains the following data.

**Table 4.2.1 Engine Performance Data Test of 12RTA96C-B**

| | Unit | 0.25 | 0.5 | 0.75 | 0.9 | 1 | 1.1 |
|---|---|---|---|---|---|---|---|
| BHP | *HP* | 23340 | 46680 | 70020 | 84024 | 933660 | 102696 |
| Rated Power | *kW* | 17404.6 | 34809.3 | 52213.9 | 62656.7 | 696230.3 | 76580.4 |
| Engine Speed (rpm) | *1/min* | 64.26 | 90.96 | 92.67 | 102 | 102 | 105.29 |
| Fuel Oil Constumption | *kg/h* | 3205.17 | 6127.48 | 8976.89 | 10908.28 | 12274 | 13944 |
| SFOC | *g/kwh* | 184.16 | 176.03 | 171.93 | 174.10 | 17.63 | 182.08 |
| Cylinder Pressure (max) | *bar* | 65.6 | 99.4 | 131.2 | 140.1 | 141.9 | 145.1 |
| Cylinder Pressure (comp) | *bar* | 44.8 | 72 | 107.2 | 125.3 | 139.3 | 151 |
| P Scavenge | *bar* | 1.24 | 2.02 | 3.02 | 3.53 | 3.85 | 4.16 |
| P Exhaust | *bar* | 1.19 | 1.76 | 2.55 | 2.97 | 3.28 | 3.36 |
| P environment | *bar* | 1.032 | 1.032 | 1.032 | 1.032 | 1.032 | 1.032 |
| T after Cylinder (T4) | *K* | 608.3 | 608.4 | 611.5 | 637.6 | 671.3 | 713.8 |
| T before Turbine (T5) | *K* | 653 | 678 | 683 | 715 | 753 | 808 |
| T after Turbine (T6) | *K* | 613 | 602 | 555 | 556 | 580 | 618 |
| T surroundings | *K* | 298 | 298 | 298 | 298 | 298 | 298 |
| Turbocharger Speed (rpm) | *1/min* | 3625 | 6730 | 8710 | 9480 | 10070 | 10620 |

For the complete data performace from the engine test and also the data from TEKOMAR company prediction result are listed in attachment A.

## 4.3 Reference Modelling (Normal Operation Model)

### 4.3.1 Engine Scheme Model

The first step was to create a simplified engine model scheme. It forms of the cylinder, the turbocharger consisting of turbine and compressor, and the charge air cooler. The charge air cooler was efficiency is regarded as ideal for this view. The inlet air cooler temperature is constant on 17 °C. For the environmental condition are state to be a constant value, where the temperature is 20 °C/ 293 K and the pressure value is 1.032 bar.

The figure below shows the engine cycle scheme used, dnd the pressure and temperature are included. Table 2 shows the the picture of it.

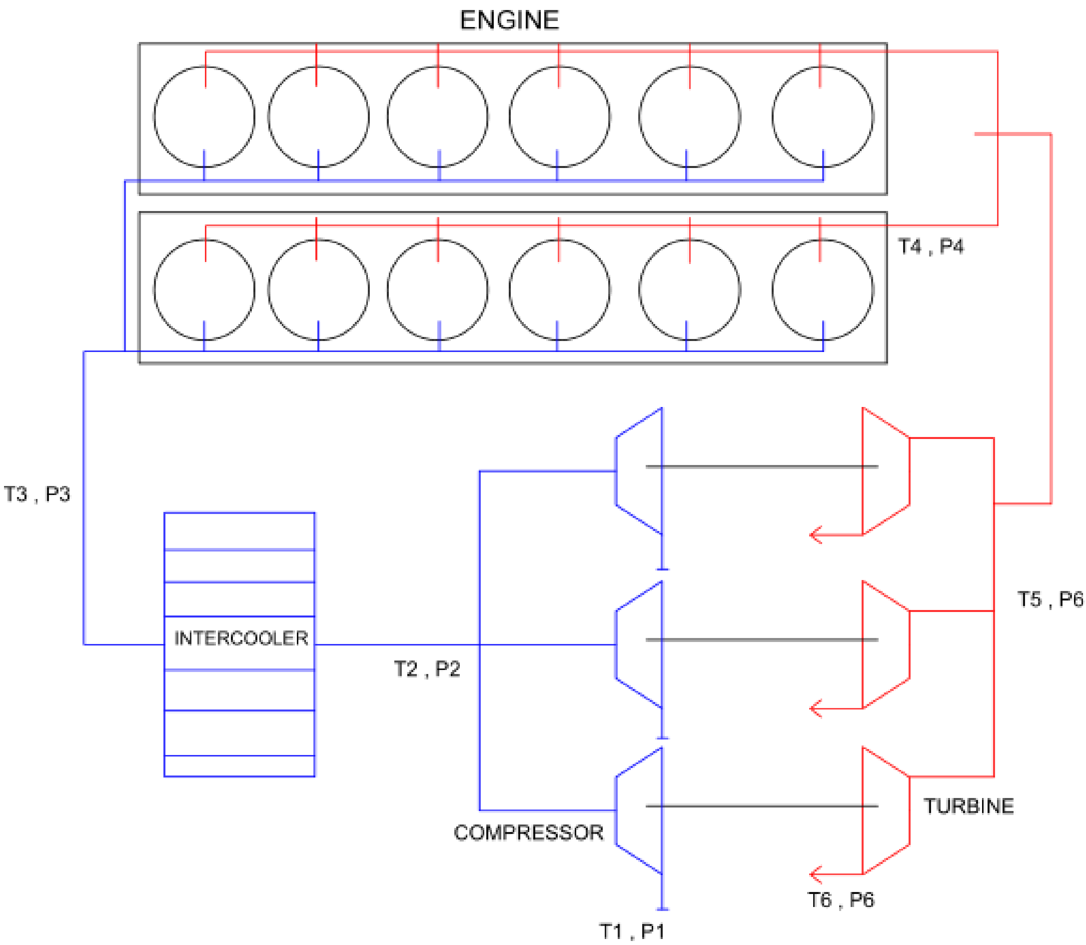



*Figure 4.3.1 Model cycle scheme*

**Table 4.3.1 Scheme variable explanation**

| Parameter | Explanation |
|---|---|
| T1, P1 | Surrounding temperature and pressure (Intake) constant at 293K and 1032 mbar |
| T2, P2 | Temperature and pressure after compressor (Scavange air) |
| T3, P3 | Temperature and pressure after scavange air cooler (before cylinder) |
| T4, P4 | Temperature and pressure after cylinder |
| T5, P5 | Temperature and pressure before turbine |
| T6, P6 | Temperature and pressure after turbine |

### 4.3.2 First Cycle to Completing the Value

The first cycle of the calculation is aimed to determine some required value of the thermodynamic model that will be used in the further calculation. For this thesis report, the example calculation is performed in 30% load power.

1. Exhaust gas energy

The first step of the calculation is to calculate the exhaust energy using energy balance method. Exhaust gas energy is the residual energy from the combustion process that goes out through the exhaust valve. Exhaust gas energy is one form the of energy distributed in the cylinder.

$$Q_{in}/\dot{Q}_{fuel} = \dot{Q}_{Exhaust} + P_i + \dot{Q}_{Wall\ Heat\ Losses}$$

*Formula 1. Cylinder Input Power (Fuel Energy) Distribution*

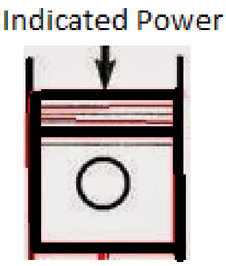
As the formula 1 above mentioned, fuel energy is the total energy that is formed from the combustion process. The fuel energy is the amount of heat that is released during the combustion of specified amount of it. For this thesis model can be assumed that fuel energy is a result of the sum of exhaust energy, indicated power, and wall heat losses energy. The input power can be calculated by cultiplying the specific fuel consumption at certain load points , the effective power and the lower calorific value.

$$\dot{Q}_{fuel} = (SFOC)b_e * P_e * H_u$$

*Formula 2. Fuel Energy (Thermal Power)*

$$\dot{Q}_{fuel} = 0.0000000508\frac{kg}{Ws} * (68640000 * 0.3)W * 42686\frac{KJ}{kg}$$
$$= 44681.997\ KW$$

As mentioned before the example calculation of this thesis is perform on 30% load points. The lower heating value of the fuel is always constant and it is assumed that for diesel heavy fuel oil the lower heat value is 42686 KJ/kg. The data of the effective power and the specific fuel consumption can be refers to engine test data sheet.




Indicated power is the theoretical maximum output power of the engine. The indicated power is the total power produced from the expanding of the gases in the cylinders negating any heat loss or entropy within the system.

*Figure 4.3.2 Indicated Power Illustration*
*Source : http://www.marinesite.info*

It is assumed in this model that indicated power is the sum of the energy that generated from burning fuel and turned into mechanical energy and friction losses that caused by mechanical movement.

$$P_i = Q_{in/fuel} * \eta_i \quad / \quad P_i = \frac{P_e}{\eta_m}$$

*Formula 3. Indicated Power*

$$P_i = 44681.997\ KW * 0.55 = 24575.10\ KW$$

By comparing the actual fuel burning power with the indicate power that goes into the piston we can acquire indicate efficiency. The theoritical assumption of indicate effieciency value is 55% of the total power, and it is assumed the value constant in the overall load range for this thesis modelling. The engine mechanical efficiency is designed with a value of about 80%-90% and with this theoretical indicate efficiency of 55%, that range of value can be achieved.

The wall heat losses is the losses of the heat energy that transfered from the combustion chamber to the engine block and system. Theoretically the losses were accepted for the full-load by 21-23% losses of the input energy. By lowering the engine power, then the value of losses will be also gradually reduced. The map of the losses and the load range is shown in the graph below.

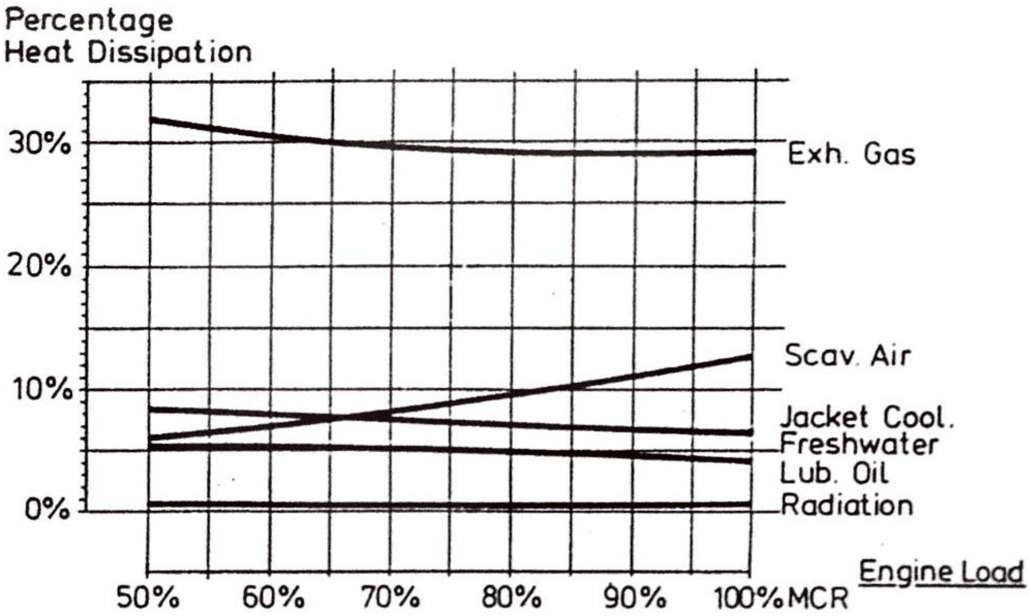



*Figure 4.3.3 Heat Losses Map Distribution*
*Source : Prof. Dr-Ing. K. Wehner & DI H. Schmidt teaching materials*

The curves are represent the percentage of heat loss distribution based on the fuel power consumption as function of the engine load. The curves are valid after up to 4 months operation using heavy fuel oil. From the wall heat losses graph figured, theses heat losses coefficient function were obtained.

$$\gamma \text{ Lube oil cooling (Pe)} = -0.03 * Pe + 0.07$$

$$\gamma \text{ Jacket cooling (Pe)} = -0.03 * Pe + 0.1$$

$$\gamma \text{ Scavange air cooling (Pe)} = 0.1 * Pe + 0.01$$

$$\gamma \text{ Radiation (Pe)} = 0.01 * Pe$$

*Formula 4. Wall heat losses coefficient function*

$$\dot{Q}_{wall\ heat\ losses} = Q_{fuel} * \gamma$$

*Formula 5. Wall heat losses*

$$\dot{Q}_{Lube\ oil\ cooling\ losses} = 44681.997\ KW * 0.061 = 2725.6\ KW$$

$$\dot{Q}_{Jacket\ cooling\ losses} = 44681.997\ KW * 0.091 = 2236.33\ KW$$

$$\dot{Q}_{Scav\ air\ cooling\ losses} = 44681.997\ KW * 0.04 = 1178.27\ KW$$

$$\dot{Q}_{Radiation\ losses} = 44681.997\ KW * 0.01 = 446.8\ KW$$

$$\dot{Q}\ Heat\ Losses\ Total = 7196.03\ KW$$

After several energy value calculated, the residual energy from the cylinder or the exhaust gas can be calculated by using formula 5. This formula is another form of formula 1.

$$Q_{exhaust} = \dot{Q}_{fuel} - P_i - \dot{Q}_{Wall\ Heat\ Losses} = \dot{m}_{exhaust} * c_{p,exhaust} * (T_4 - T_3)$$

*Formula 6. Residual energy / Exhaust gas formula*

$$Q_{exhaust} = 44681.997\ KW - 24575.10\ KW - 7196.03\ KW = 12910.86\ KW$$

2. Exhaust gas mass flow

In the next step mass flow of the exhaust gas can be determined, by using the temperature inlet (T3) and outlet (T4) value of the cylinder from the engine data test. The formula of the exhaust gas value mentioned below.

$$\dot{m}_{exhaust} = \frac{\dot{Q}_{exhaust}}{c_{p,exhaust} * (T_4 - T_3)}$$

*Formula 7. Exhaust gas flow rate*

$$\dot{m}_{exhaust} = \frac{12910.86 \text{ KW}}{1020.8\, J/Kg.K \; * (597.4\, K - 294.78\, K)} = 41.80\, Kg/s$$

The value of the Cp for exhaust gas used is the value obtained from the temperature function. By using the mean value between temperature before cylinder (T3) and after cylinder (T4) and subtitute it into the function, then the value of the Cp for exhaust gas obtained. In the next step of calculation is to calculated the fresh air mass flow rate. The value can be obtained by subtracting the fuel mass flow from exhaust gas mass flow.

3. Fresh Air Mass flow

$$\dot{m}_{Fresh\ air} = \dot{m}_{Exhaust} - \dot{m}_{Fuel}$$

*Formula 8. Freash air mass flow rate*

$$\dot{m}_{Fresh\ air} = 41.80 \frac{Kg}{s} - (68640000\, W * 0.3) * 0.0000000508 \frac{kg}{Ws} = 40.75\, kg/s$$

After getting the value of mass flow, now it is possible to make the operating line compressor. Since the fresh air mass flow is generated in normal operation by three compressors, the first step is to divide the fresh air mass flow value for 3 compressors. From the total fresh mass flow of 40.75kg/s now we divide it into 13.58 kg/s. In order to make the data is sustainable the characteristic field in the intake condition, this calculation is performed.

4. Mass flow standard coefficient

$$\dot{m}_{standard\ coefficient} = \dot{m}_{fresh\ air} \times \frac{\sqrt{T1}}{P1}$$

*Formula 9. Normal coefficient of mass flow*

$$\dot{m}_{standard\ coefficient} = 13.58\ \text{kg/s}\times \frac{\sqrt{293\ K}}{1.032\ bar} = 0.002253$$

Performance of a compressor is usually specified by curves of delivery pressure against mass flow rate for various fixed values of rotational speed and inlet temperature. These performance characteristic curves are usually plotted with dimensionless variables.

The compressor characteristics line below will show the relation between compressor pressure and the mass flow standard. The compressor pressure or scavenge air pressure data is taken from engine test performance data.

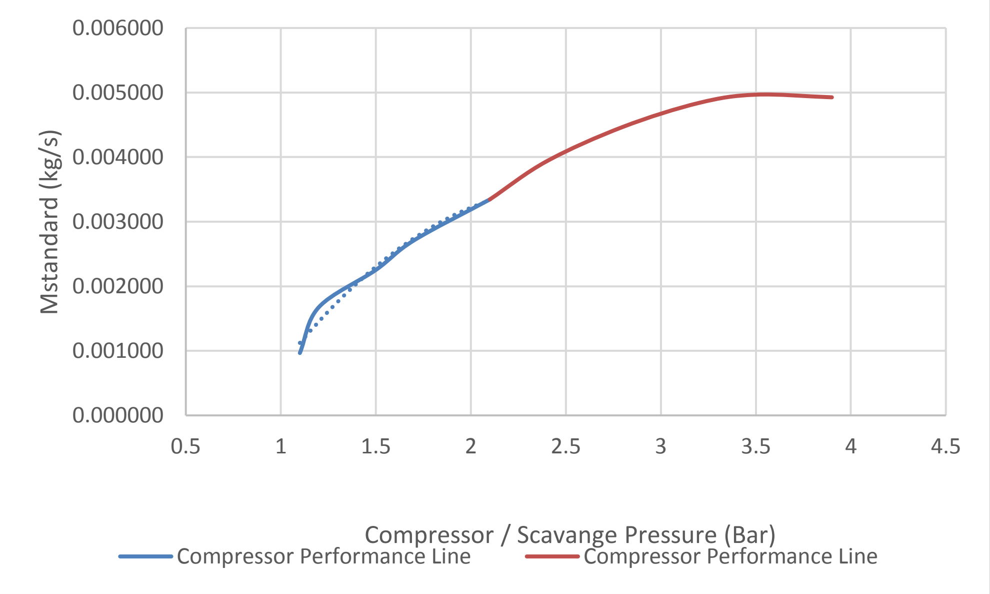



*Figure 4.3.4 Compressor characteristics curve*

Figure 4.3.4 shows the course of the operating line of compressor based on mass flow rates of all the load of engine operation. The blue line represent the compressor charasteristics when engine is operated below 50%, and the red line represent the charasteristics when the engine is operated above 50%. These charasteristics value is not affected or distorted by axiliary blower for this thesis calculation. It is expected when the value of pressure increases, then the value of mass flow will also increase gradually. However the result shows that there is a different increasing value when the pressure is higher than 2.1 bar. Although the air pressure of charging increases constantly, the increase of mass flow from load point to load point becomes smaller. From the curve, 2 mass flow standard function based on scavenge air pressure (p2) were set.

Mstandard (*p2*) = (0.005649859 + (-1555.386 - 0.005649859)/(1 + (P/0.000004907295)^1.033987)-0.0001)

*Formula 10. Mass flow standard function for p2 < 2.1 Bar*

Mstandard (*p2*) = (0.004956387 + (0.003809266 - 0.004956387)/(1 + (P/2.747092)^15.28802)-0.0003

*Formula 11. Mass flow standard function for p2 > 2.1 bar*

By this assumption, charasteristics modelling the performance of the compressor can be adequately described. For the further calculation, the fresh air mass flow value can be determined by calculating the compressor pressure and assigned it to the function above at any load points.

5. Cylinder outlet pressure function

With the given data from the engine performance report, the relation curve of cylinder inlet pressure (*P3)* and cylinder outlet pressure (*P4)* is made.

**Table 4.3.2 Cylindet inlet and outlet pressure table.**

| Rated Power % | **0.25** | **0.5** | **0.75** | **0.9** | **1** | **1.1** |
|---|---|---|---|---|---|---|
| P4 (outlet) | 1.19 | 1.76 | 2.55 | 2.97 | 3.28 | 3.36 |
| P3 (inlet) | 1.24 | 2.02 | 3.02 | 3.53 | 3.85 | 4.16 |

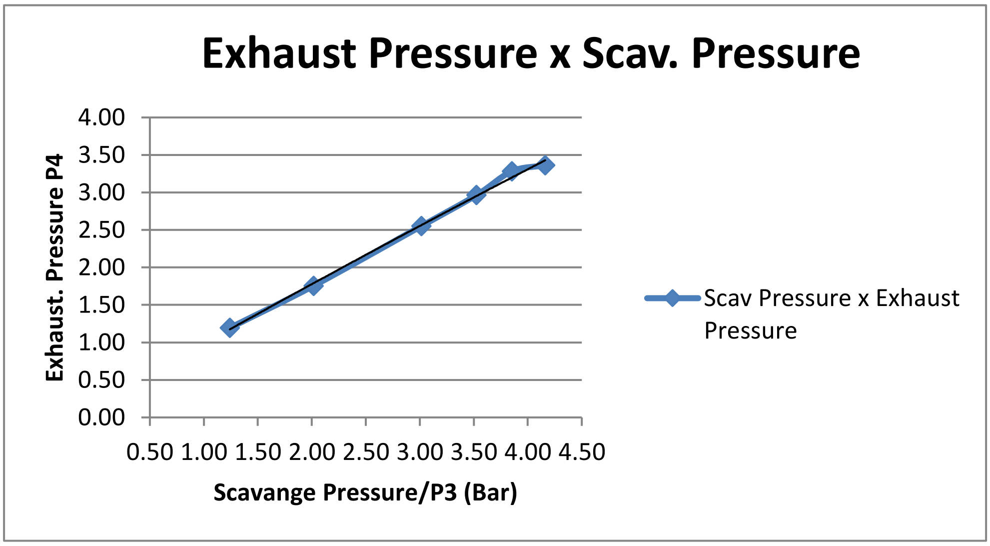



*Figure 4.3.5 Cylinder outlet and inlet relation curves*

From the curve above, This function is created to get an cylinder exhaust pressure value (*P4)* in the next cycle as an approach value.

$$P4, f(P3) = \frac{5.304972\ +\ (0.8759069\ -\ 5.304972)}{(1\ +\ (P3/3.684166)\char`^2.348152)}$$

*Formula 12. After cylinder/before turbine pressure function*

6. Kinetic energy transfer in exhaust manifold

In the turbocompressor supply system, successive pulses of hot exhaust gas which leave the engine cylinders compress adiabatically the gas column situated in front of them. The increasing hydraulic resistance slows down the flow and provokes dissipation of kinetic energy of the compressed fluid. The observable symptoms of which are the decrease of the exhaust gas flow velocity and the increase of its static temperature and pressure in the turbocompressor turbine cross-section.

$$Kinetic\ Energy\ Transfered = \frac{\dot{m}\ xC^2}{2} = \dot{m}\ Cp\ (T5 - T4)$$

*Formula 13. Kinetic energy transfer in exhaust manifold*

$$Kinetic\ Energy\ Transfered = \frac{C^2}{2} = \frac{1059.19J}{KgK}(677.56\ K - 597.4\ K)$$

$$Kinetic\ Energy\ Transfered = 412.1\ J/Kg$$

The temperature increase in front of the turbine (T5) can be very high, and exceed the exhaust gas temperature measured directly after the cylinder (T4) by 100K or more. In this kinetic energy calculation the temperature after cylinder (T4) and before turbine (T5) were obtained from the engine test data, and the Cp value of exhaust can be determined based on the mean temperature between T4 and T5. The kinetic energy transfered in exhaust manifold can be assumed to be constant for each load points engine operation. Hence in the further step, the kinetic energy value can be used to calculate the temperature before turbine (T5).

7. Turbine and compressor efficiency

In the analysis, isentropic efficiency is a parameter to measure the degree of degradation of energy in steady-flow devices. It involves a comparison between the actual performance of a device and the performance that would be achieved under idealized circumstances for the same inlet and exit states.

Physically speaking, the turbine delivers less output energy due to friction and losses in the polytropic expansion process than the possibly maximum energy given in the isentropic process. The maximum total-static isentropic efficiency of the turbine ηT is normally between 65 and 80 % at the design point of the turbine performance map.

Meanwhile the compression process in the compressor is a polytropic process with increasing entropy due to friction and losses in the compressor. The maximum total–total isentropic efficiency of the compressor $\eta_C$ is normally between 70 and 80 % at the design point in the compressor performance map. *(Japikse, and Whitfield: 1990-2003).*

For the turbine and compressor efficiency value for this thesis is adapted to the turbocharger curves from MAN B&W company. This curves are valid for all type of diesel engine with bore more than 50 cm. Its show the relation of pressure over the effciency.

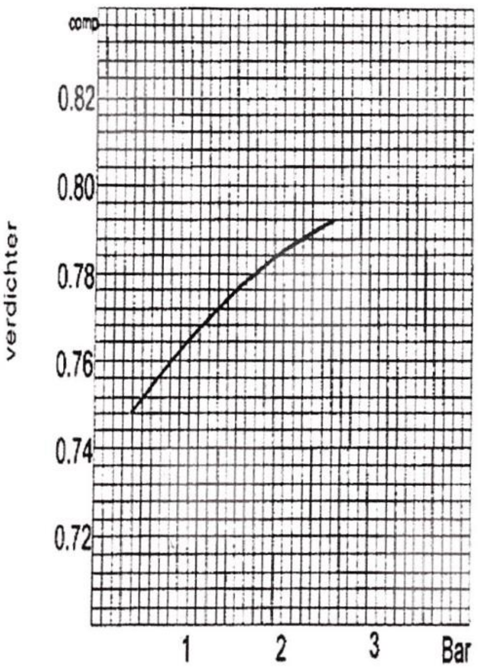



*Figure 4.3.6 Compressor characteristics curves*
*Source : MAN B&W Report*

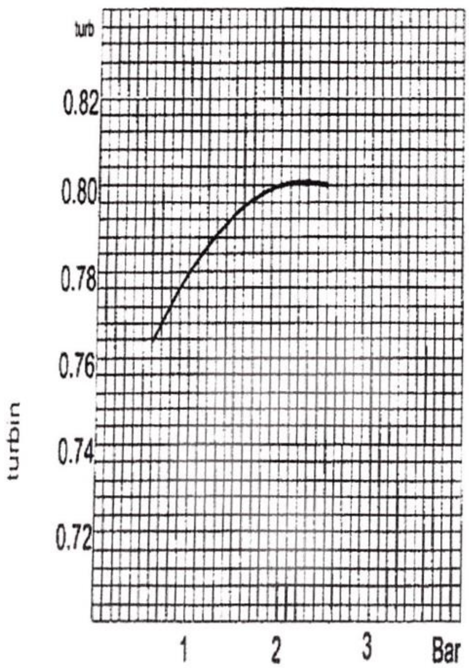



*Figure 4.3.7 Turbine characteristics curves*
*Source : MAN B&W Report*

From the characteristics above, these function were obtained.

ɳ compressor ($p$) = 0.8081947 + (0.7300196 - 0.8081947)/(1 + ( $P$ /2.16917)^2.782963)

ɳ turbine ($p$) = 0.8092781 + (0.7355279 - 0.8092781)/(1 + ( $P$ /1.805262)^3.85765)

*Formula 14. Turbine and compressor efficiency function*

By using this function, the value of the efficiency compressor and turbine can be obtained. For this example calculation on 30% load power the pressure value is on 1.5 bar, so the compressor efficiency obtained is 0.751 and for the turbine efficiency is 0.76.

8. Turbine and compressor power

The heat transfer in and out of a turbine can typically be considered negligible as well as the potential energy unless otherwise stated. In this step, the power of each turbine is determined by this formula.

$$P_T = \frac{\dot{m}_{exhaust}}{i} * c_{p\ exhaust} * (T_5 - T_6)$$

*Formula 15. Turbine Power*

The i represent numbers of turbine. Depended on how many turbines are operated, exhaust gas mass flow is evenly distributed.

$$P_T = \frac{41.8\frac{kg}{s}}{3} * 1069.1\frac{J}{kgK} * (677.56\ \mathrm{k} - 634\ \mathrm{k}) = 522191.02\ W$$

The Cp value of the exhaust gas is based of the mean temperature between temperature before turbine (T5) and after turbine (T6). For the T5 and T6 value can be obtained from the engine performance report or TEKOMAR line since the engine report does not shown the data in all load operation between 10%-100%. Then the next step is to calculate the compressor power. Both turbine and compressor wheels are fixed in the rotor shaft that is supported on the bearing system of the radial and thrust bearings.

Turbine power will be supplied to the compressor by the shaft, and certainly there will be less mechanical losses occurring in this process. Hence for this thesis assumed that the mechanical efficiency of the turbocharger is 98%.

$$P_{compressor} = P_T * \eta_{m,Turbocharger}$$

*Formula 16. Compressor power*

$$P_{compressor} = 522191.02\, W * 0.98 = 511747.20\, W$$

9. Compressor temperature (T2) and pressure (P2)

The next step is to calculate the air condition on compressor outlet. Increasing air compression, that is the amount of gas included in the same volume, at the same time, its temperature is increased. Higher temperature is connected with lower density, which means that cylinders receive the amount of oxygen smaller than if the air temperature were lower. There are 2 indications that show the process that is temperature and pressure after the compressor.

$$T_2 = \frac{P_{comp}}{\left(\dot{m}_{fresh\ air} * c_p\right)} + T_1$$

*Formula 17. Temperature after compressor*

$$T_2 = \frac{511747.20\, W}{\left(13.58 kg/s * 1013 \frac{J}{kg}\right)} + 293\ \mathrm{K} = 330.19\ \mathrm{K}$$

The next step is to determine the pressure after compressor by using law of polytropic state. The formula mentioned below.

$$p_2 = \left(\frac{T_{2,is}}{T_1}\right)^{\frac{\kappa}{\kappa-1}} * p_1$$

*Formula 18. Pressure after compressor*

Since the required data has been obtained except the after compressor isentropic temperature ($T_{2,is}$). Then the step is to calculate the isentropic temperature first and then calculate the pressure after compressor.

$$\eta_{is,C} = \frac{T_{2is} - T_1}{T_2 - T_1}$$

*Formula 19. Compressor isentropic efficiency*

$$0.751 = \frac{T_{2is} - 293\ K}{330.19\ K - 293\ K} \rightarrow T_{2is} = 320.92\ K$$

$$p_2 = \left(\frac{320.92\ \mathrm{K}}{293\ \mathrm{K}}\right)^{\frac{1.37}{1.37-1}} * 1.032\ bar = 1.42\ bar$$

10. Determine new mass flow value of the compressor

In every cycle of calculation, will eventually generate new mass flow value until the result of the calculation shows the results are constant and close to the reference value, in this thesis the benchmark reference is the performance curve from TEKOMAR company. The mass flow value can be calculated by determined the mass flow standard using formula 10 function.

$$\mathrm{Mstandard\ (1.42\ bar\ )} = \frac{(0.005649859\ +\ (-1555.386\ -\ 0.005649859)}{(1\ +\ (1.42/0.000004907295)\text{^}1.033987) - 0.0001)}$$

$$\mathrm{Mstandard\ (1.42\ bar\ )} = 0.002342$$

Since the mass flow standard value has been calculated, the new mass flow value can be determina by using formula 9. This new value will be used as the input compressor mass flow value in the next cycle of mathematical model calculation.

$$\dot{m}_{fresh\ air} = \dot{m}_{standard\ coefficient} \times \frac{P1}{\sqrt{T1}}$$

$$\dot{m}_{fresh\ air} = 0.002342 \times \frac{1.032\ bar}{\sqrt{293\ K}} = 14.12\ kg/s$$

11. Intercooler phase calculation

An intercooler's primary function is to cool the charge air after it has been heated due to boosting and the heat that is produced by the turbo before sending the air into the engine. As the air is cooled, it becomes denser, and denser air makes for better combustion (more power). There will be 2 values that change in this phase , specificaly the value of temperature and pressure. The first step is to calculate the intercooler efficiency.

$$\eta_{Cooler} = \frac{T_2 - T_3}{T_2 - T_{water\ in}}$$

*Formula 20. Intercooler efficiency*

$$\eta_{Cooler} = \frac{330.19\ K - 294\ K}{330.19\ K - 290\ K} = 0.857$$

For this thesis model, the cooling water inlet temperature value used is 290 K. For the gas after cooler temperature value was taken from the engine operation report. This cooler efficiency is assumed to be constant for each engine load operation and it will be used to calculate the gas after cooler temperature in the next cycle of the calculation model. In addition to temperature changes, the pressure value also decreased. The relations between the pressure and the pressure drop is shown in graph below.

**Table 4.3.3 Pressure drop table.**

| Pressure Drop on intercooler | | | | | | | |
|---|---|---|---|---|---|---|---|
| **Rated Power** | **% load** | **0.25** | **0.5** | **0.75** | **0.9** | **1** | **1.1** |

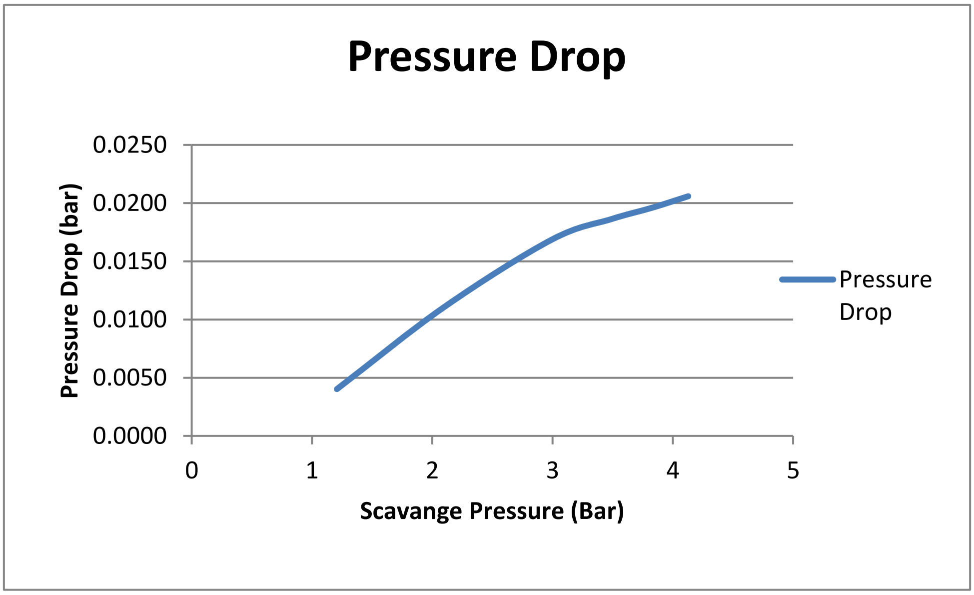



*Figure 4.3.8 Intercooler pressure drop*

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| **Scavange Pressure** | ***Bar*** | 1.207 | 2.1 | 2.994 | 3.49 | 3.83 | 4.128 |
| **Pressure Drop** | ***Bar*** | 0.0040 | 0.0111 | 0.0169 | 0.0186 | 0.0196 | 0.0206 |

Pressure loss, or pressure drop, refers to the change in pressure when comparing the air entering the intercooler with the exiting air. This change is mostly affected by the internal flow area of the intercooler. Flow loss, however, is measured not just with pressure loss but with how much restriction to airflow exists.

$$\text{Pressure drop } (P2) = \frac{0.02421102 + (0.0004137441 - 0.02421102)}{(1 + (Scav.Pressure/2.256177)\text{^}2.766972)}$$

*Formula 21. Pressure drop function*

$$\begin{aligned}\text{Pressure drop } (1.42\ bar) \\ &= \frac{0.02421102 + (0.0004137441 - 0.02421102)}{\left(1 + \left(\frac{1.42}{2.256177}\right)^{2.766972}\right)} \\ &= 0.0056\ bar\end{aligned}$$

12. Recalculate the scavange compressor (*P2*) with *the first turbocharger equation*

This recalculation is performed to get a more precise value of pressure after compressor (P2) with *the first turbocharger equation.*

$$\pi = \frac{P2}{P1} = 1 + \frac{m_{turbine} \times Cp_{turbine} \times T_{turbine\ inlet}}{m_{compressor} \times Cp_{compressor} \times T_{compressor\ inlet}} \times \eta_{TC} \times \left(1 - \left(\frac{P6}{P5}\right)^{\frac{KT-1}{KT}}\right)^{\frac{KC}{KC-1}}$$

*Formula 22. The first turbocharger equation*

For the specific heat ratio or kappa of the exhaust gas (kT or γ) , the value is based on the temperature function.

$$\gamma = \gamma_0 - \frac{K_1(T - T_{reference})}{1000}$$

*Formula 23. Kappa value function (specific heat ratio)*

This is the kappa value function of Gatowski. Where $\gamma_{0}$ is a reference value (1.38), $K_1$ is a constant (0.08) and T ref is a reference temperature (300 K).

$$\mathrm{kT}\ /\ \gamma = 1.38 - \frac{0.08 \times (677.56\ k - 300\ k)}{1000} = 1.3497$$

For the estimation of surface specific heat ratio of air (kC) as a function of temperature and pressure. For this thesis estimation of specific heat ratio or kappa for the air value is about 1.36-1.4 it depends on the compressor pressure, The higher the pressure the lower the kappa value. The value for kC in this thesis in adaptable due to get the result value as close as possible to the reference value.

Within the overall efficiency of the turbocharger ηTC is calculate as

$$\eta_{TC} = \eta_T \times \eta_C \times \eta_m$$

$$\eta_{TC} = 0.76 \times 0.751 \times 0.98 = 0.57$$

After obtaining the required value, then the value of pressure can be calculated as follows.

$$\frac{P2}{1.032bar} = 1 + \frac{\frac{41.8kg}{s} \times \frac{1069.1J}{kgK} \times 677.56K}{43.40kg/s \times \frac{1002J}{kgK} \times 293K} \times 0.57 \times \left(1 - \left(\frac{1.032bar}{1.35bar}\right)^{\frac{1.3497-1}{1.3497}}\right)^{\frac{1.4}{1.4-1}}$$

$$P2 = 1.368\ bar$$

13. Compression pressure and maximum pressure

The static compression ratio of an internal combustion engine is a value that represents the ratio of the volume of its combustion chamber from its largest capacity to its smallest capacity. In calculating the pressure ratio, assumed that an adiabatic compression is carried out. it can defined the relationship between change of volume and change of pressure as follows:

$$\frac{Compression\ Pressure}{Inlet\ air\ pressure} = \left(\frac{V1}{V2}\right)^{\gamma}$$

*Formula 24. Compression pressure ratio*

The *V1* represent the volume of the cylinder and combustion chamber when the piston is at the bottom of its stroke, meanwhile *V2* is the volume of the combustion chamber when the piston is at the top of its stroke. Since the value of V2 is unknown, the initial step is to calculate V2 value that will be used in the next step of this thesis calculation model.

$$\frac{77\ bar}{1.42\ bar} = \left(\frac{1.716\ m^2}{V2}\right)^{1.37} \rightarrow V2 = 0.1245\ m^3$$

Where the compression pressure and *V1* value can be obtained from the previous calculation and from the engine test data. For the specific heat ratio/kappa value of air approximately 1.37. After the function of the compression pressure determine, the pressure increase value to the maximum compression pressure can be estimated by using function from the curve below. Using the data of maximum pressure from engine report and calculated mass flow, this curve can be made.

ΔP *(mass flow*) = 28.62798 + (13.23859 - 28.62798)/(1 + (*m*/44.10621)^5.357742)

*Formula 25. Pressure increase function for mass flow <65kg/s*

ΔP (*mass flow*) = -17187410 + (26.03302 - -17187410)/(1 + (*m*/183.9695)^27.47693)

*Formula 26. Pressure increase function for mass flow >65kg/s*

In this first cycle of calculation, the required value and function have been calculated and obtained such as turbine and compressor efficiency, intercooler efficiency, compressor performance function, the kinetic energy value and some other functions and values. These function and value will be used in the next cycle of the performance calculation.

### 4.3.3 Engine performance calculation in normal operation

The next step of the calculation to calculate the engine performance in normal condition with 3 turbocharger in 30% load power. This step is aimed to show that thermodynamic model is reliable and can be used in the further calculation.

1. Exhaust gas energy

- Exhaust gas energy can be calculated by using formula 1 as follows.

$$Q_{in}/\dot{Q}_{fuel} = \dot{Q}_{Exhaust} + P_i + \dot{Q}_{Wall\ Heat\ Losses}$$

- Fuel energy calculation using formula 2.

$$\dot{Q}_{fuel} = 0.0000000508 \frac{kg}{Ws} * (68640000 * 0.3)W * 42686 \frac{KJ}{kg} = 44681.997\ KW$$

- Indicated power (*Pi)* calculation using formula 3.

$$P_i = 44681.997\ KW * 0.55 = 24575.10\ KW$$

- Wall heat losses calculation using formula 5

$$\dot{Q}_{Lube\ oil\ cooling\ losses} = 44681.997\ KW * 0.061 = 2725.6\ KW$$

$$\dot{Q}_{Jacket\ cooling\ losses} = 44681.997\ KW * 0.091 = 2236.33\ KW$$

$$\dot{Q}_{Scav\ air\ cooling\ losses} = 44681.997\ KW * 0.04 = 1787.28\ KW$$

$$\dot{Q}_{Radiation\ losses} = 44681.997\ KW * 0.01 = 446.8\ KW$$

$$\dot{Q}_{WALL\ HEAT\ LOSSES\ TOTAL} = 7196.03\ KW$$

After several cylinder energy distribution value calculated, exhaust gas energy can be calculated by using formula 1.

$$Q_{exhaust} = 44681.997\ KW - 24575.10\ KW - 7196.03\ KW$$
$$= 12910.86\ KW$$

2. After cylinder temperature (T4) calculation

Cylinder outlet temperature can be calculate by using this fomula.

$$T_4 = \frac{\dot{Q}_{exhaust}}{(\dot{m}_{exhaust} * c_{p,exhaust})} + T_3$$

*Formula 27. Temperature after cylinder formula*

But before it, the exhaust gas mass flow has to be calculated first by using formula as follows.

$$\dot{m}_{Exhaust} = \dot{m}_{fresh\ air} + \dot{m}_{Fuel}$$

$$\dot{m}_{Exhaust} = \frac{43.40kg}{s} + (68640000\ W * 0.3) * 0.0000000508 \frac{kg}{Ws} = 44.45\ kg/s$$

Then the calculation of the after cylinder (*T4*) can be calculated as follows.

$$T_4 = \frac{12910.86\ KW}{\left(44.45 \frac{kg}{s} * 1020.8 \frac{J}{Kg}.K\right)} + 294.42\ K = 583.42\ K$$

For the Cp for exhaust gas value used is obtained from the previous calculation as approaching value due to the *T4* value has just been calculated. For the cylinder also obtained from previous calculation.

3. Temperature before turbine (*T5*) calculation.

The inlet turbine temperature/before turbine temperature (*T5)* can be calculated by using kinetic energy function (formula 13.)

$$\frac{(412.1)^2\ J}{2} = \frac{1059.19J}{KgK} \times (T5 - 583.42\ K) \rightarrow T5 = 667.25\ K$$

4. After cylinder pressure / before turbine pressure (*P4/P5)*

Using after cylinder pressure function (formula 12), the pressure value can be obtained.

$$P4, f(1.368\ bar) = \frac{5.304972 + (0.8759069 - 5.304972)}{\left(1 + \left(\frac{1.368\ bar}{3.684166}\right)^{2.348152}\right)} = 1.35\ bar$$

5. Temperature after turbine calculation (*T6 and T6is)*

The step is to calculate after turbine isentropic temperature (*T6is)* first and after that the calculation of after turbine temperature (*T6)* can be done.

$$T_{6,is} = T_5 * \left(\frac{p_6}{p_5}\right)^{\frac{\kappa-1}{\kappa}}$$

*Formula 28. After turbine isentropic temperature*

$$T_{6,is} = 667.25\ K * \left(\frac{1.0.32\ bar}{1.35\ bar}\right)^{\frac{1.3506-1}{1.3506}} = 621.73\ K$$

After the value of the after turbine isentropic temperature ($T_{6\ IS}$) is calculated, the after turbine tempreature ($T6$) can be calculated by using isentropic efficiency of turbine formula.

$$\eta_{is,T} = \frac{T_5 - T_6}{T_5 - T_{6IS}}$$

*Formula 29. Turbine isentropic efficiency*

$$0.748 = \frac{677.25\ K - T_6}{677.25\ K - 621.73K} \rightarrow T6 = 634.39\ K$$

6. Turbine and compressor power

The turbine power and the power distributed to the compressor can be calculated by using formula 15 and formula 16.

$$Turbine\ power, P_T = \frac{44.45\frac{kg}{s}}{3} *\ 1069.1\frac{J}{kgK} * (677.25 -\ 634.39\ \mathrm{k}\,)$$

$$Turbine\ power, P_T = 531477.03\ W$$

$$Compressor\ power, P_{compressor} =\ 531477.03\ W *\ 0.98 =\ 520847.49\ W$$

7. Temperature ($T2$) and pressure after compressor calculation ($P2$)

By using formula 17 the temperature after compressor can be calculated.

$$Temperature\ after\ comp, T_2 = \frac{520847.49\ W}{\left(\frac{44.45\ kg/s}{3} *\ 1013\frac{J}{kg}\right)} +\ 293\ \mathrm{K} =\ 329.71\ \mathrm{K}$$

To calculate the pressure after compressor, the after compressor isentropic temperature value has to be calculated first with formula 19.

$$0.751 = \frac{T_{2is} - 293\ K}{329.71\ \mathrm{K} - 293\ K} \rightarrow\ T_{2is} = 320.55\ K$$

After compressor pressure can be calculated by using formula 18.

$$p_2 = \left(\frac{320.55\ K}{293\ \mathrm{K}}\right)^{\frac{1.4}{1.4-1}} * 1.032\ bar =\ \ 1.4135\ bar$$

8. Determine new mass flow value of the compressor

To be able to determine mass flow value using formula 9, the mass flow standard coefficient should be calculated with formula 10 in advance.

$$\text{Mstandard (1.4135 bar )} = \frac{(0.005649859 + (-1555.386 - 0.005649859)}{(1 + (1.4135/0.000004907295)^\wedge 1.033987) - 0.0001)}$$

$$\text{Mstandard (1.4135 bar )} \rightarrow 0.002250$$

Since the mass flow standard value has been calculated, the new mass flow value can be determina by using formula 9. This new value will be used as the input compressor mass flow value in the next cycle of mathematical model calculation.

$$\dot{m}_{fresh\ air} = 0.001906 \times \frac{1.032\ bar}{\sqrt{293\ K}} = 13.56 \frac{kg}{s} \text{ for each turbocharger}$$

9. Intercooler phase calculation

When the air pass through the intercooler, the air temperature and pressure value change. For the after cooler temperature (*T3*) can be calculated with intercooler efficiency function (formula 20).

$$0.857 = \frac{329.71\ K - T3}{329.71\ K - 290\ K} \rightarrow T3 = 295.67\ K$$

While for the pressure drop can be calculated using formula 19.

$$\text{Pressure drop } (1.4135\ bar) = \frac{0.02421102 + (0.0004137441 - 0.02421102)}{\left(1 + \left(\frac{1.4135}{2.256177}\right)^{2.766972}\right)}$$

$$\rightarrow \text{Pressure drop } (1.4135\ bar) = 0.0055\ bar$$

10. Recalculate the scavange air compressor (*P2*) with *the first turbocharger equation*

Using formula 20 the pressure after compressor can be recalculated to get A more precies value of it .

$$\frac{P2}{1.032bar} = 1 + \frac{\frac{43.40kg}{s} \times \frac{1062.2J}{kgK} \times 677.25K}{40.94kg/s \times \frac{1002J}{kgK} \times 293K} \times 0.57 \times \left(1 - \left(\frac{1.032bar}{1.35bar}\right)^{\frac{1.3506-1}{1.3506}}\right)^{\frac{1.365}{1.365-1}}$$

$$P2 = 1.45\ bar$$

This whole calculation step is performed at all engine load points. After the calculation in the normal operation is done, the comparasion curves of temperature and pressure is created to show whether the thesis model calculation is reliable and can be used for further purpose.

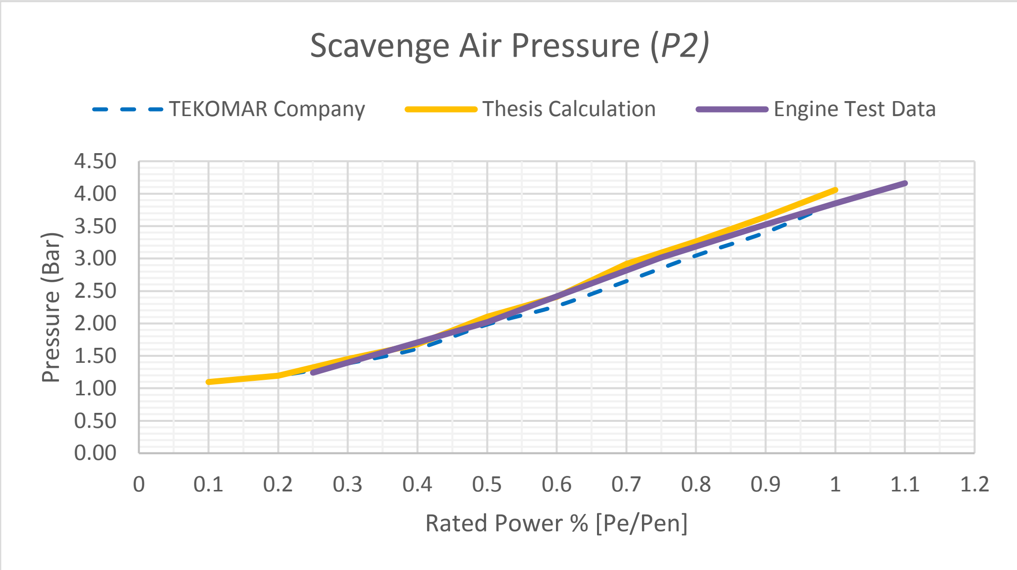



*Figure 4.3.9 Scavenge air pressure (P2) comparasion curves*

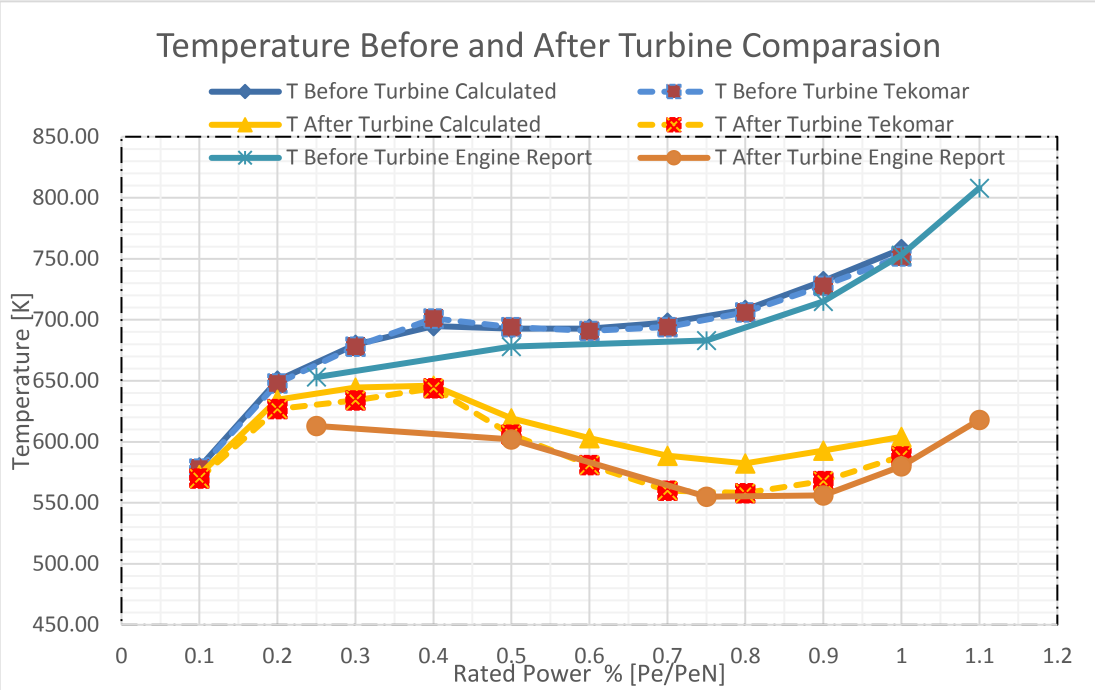



*Figure 4.3.10. Temperature before turbine (T5) and after turbine (T6) comparasion curves*

As can be seen from the comparison curves of pressure and temperature, there are indeed some differences value between engine report, tekomar, and thesis model but it is assumed that the deviation value is understandable. This difference in value is likely to be due to differences in specifications and some coefficient values in the calculations such as turbine and compressor efficiency, specific heat ratio, exhaust gas energy value and so on.

### 4.3.4 Engine performance calculation in turbocharger cutoff operation

The next step of the calculation to calculate the engine performance in 1 turbocharger cut condition in 30% load power. As pratically the slow steaming operation is operated engine in low load operation, this step is aimed to calculate the turbocharger cut off performance in only 10%-50% engine load and compare it with the normal condition operation. The calculation step practically the same as the previous calculation but with 2 turbocharger operated. The input value is obtained from the previous calculation.

1. Exhaust gas energy

- Fuel energy calculation using formula 2.

$$\dot{Q}_{fuel} = 0.0000000492\frac{kg}{Ws} * (68640000 * 0.3)W * 42686\frac{KJ}{kg} = 43217.0138\,KW$$

As the theory says when performing TC cut off operaion, the fuel consumption will decrease and the specific fuel oil cconsumption value is taken from the tekomar line as its reference.

- Indicated power (*Pi)* calculation using formula 3.

$$P_i = 43217.0138\,KW * 0.55 = 23769.36\,KW$$

- Wall heat losses calculation using formula 5

$$\dot{Q}_{Lube\ oil\ cooling\ losses} = 43217.0138\,KW * 0.061 = 2636.24\,KW$$

$$\dot{Q}_{Jacket\ cooling\ losses} = 43217.0138\,KW * 0.091 = 2163.01\,KW$$

$$\dot{Q}_{Scav\ air\ cooling\ losses} = 43217.0138\,KW * 0.04 = 1728.68\,KW$$

$$\dot{Q}_{Radiation\ losses} = 43217.0138\,KW * 0.01 = 432.17\,KW$$

$$\dot{Q}_{WALL\ HEAT\ LOSSES\ TOTAL} = 6960.10\,KW$$

After several cylinder energy distribution value calculated, exhaust gas energy can be calculated by using formula 1.

$$Q_{exhaust} = 43217.0138\,KW - 23769.36\,KW - 6960.10\,KW = 12487.56\,KW$$

2. After cylinder temperature (T4) calculation

Before calculating after cylinder temperature, the exhaust gas mass flow has to be calculated first by using formula as follows.

$$\dot{m}_{Exhaust} = \dot{m}_{fresh\ air} + \dot{m}_{Fuel}$$

$$\dot{m}_{Exhaust} = \frac{13.56kg \times 3}{s} + (68640000\,W * 0.3) * 0.0000000492\frac{kg}{Ws} = 41.70\,kg/s$$

The mass flow value obtained from the previous calculation assumed as an adaptation value for this TC cut off calculation.

Then the calculation of the after cylinder (T4) can be calculated using formula 27 as follows.

$$T_4 = \frac{12487.56\ KW}{\left(41.70\frac{kg}{s} * \ 1020.8\frac{J}{Kg}.K\right)} + \ 295.78\ K = 589.15\ K$$

3. Temperature before turbine (*T5*) calculation.

The inlet turbine temperature/before turbine temperature (*T5)* can be calculated by using kinetic energy function (formula 13.)

$$\frac{(412.1)^2\ J}{2} = \frac{1059.19J}{KgK} \times (T5 - 589.15\ K)\ \rightarrow T5\ = 672.97\ K$$

4. After cylinder pressure / before turbine pressure (*P4/P5)*

Using after cylinder pressure function (formula 12), the pressure value can be obtained.

$$P4, f(1.45\ bar) = \frac{5.304972\ +\ (0.8759069\ -\ 5.304972)}{\left(1\ +\ \left(\frac{1.45\ bar}{3.684166}\right)^{2.348152}\right)} = 1.42\ bar$$

After the cylinder outlet pressure is obtained, the cylinder inlet and outlet pressure ratio is determine. This ratio is assumed to be constant and will be used in the next cycle of calculation due to this method is provide more precise value for calculating the pressure after cylinder.

**Table 4.3.4 Cylinder inlet and outlet pressure ratio.**

| Power | % | **0.1** | **0.2** | **0.3** | **0.4** | **0.5** | **0.6** |
|---|---|---|---|---|---|---|---|
| Scav Pressure | *Bar* | 1.10 | 1.20 | 1.45 | 1.68 | 2.10 | 2.41 |
| Cylinder Exhaust Pressure | *Bar* | 1.09 | 1.17 | 1.42 | 1.50 | 1.81 | 2.06 |
| Ratio | | 1.007 | 1.020 | 1.022 | 1.122 | 1.161 | 1.168 |

5. Temperature after turbine calculation (*T6 and T6is)*

The step is to calculate after turbine isentropic temperature (*T6is)* using formula 28 first and after that the calculation of after turbine temperature (*T6)* can be calculated using isentropic efficiency of turbine formula (formula 29).

$$T_{6,is} = 684.993\ K * \left(\frac{1.0.32\ bar}{1.42\ bar}\right)^{\frac{1.3492-1}{1.3492}} = 638.40\ K$$

$$0.748 = \frac{684.993\ K - T_6}{684.993\ K - 638.40\ K} \rightarrow T6 = 649.82\ K$$

6. Turbine and compressor power

The turbine power and the power distributed to the compressor can be calculated by using formula 15 and formula 16.

$$Turbine\ power, P_T = \frac{41.70\frac{kg}{s}}{2} * 1066.6\frac{J}{kgK} * (684.993\ K - 649.82\ K)$$

$$Turbine\ power, P_T = 782245.75\ W$$

$$Compressor\ power, P_{compressor} = 782245.75\ W * 0.98 = 766600.83\ W$$

7. Temperature ($T2$) and pressure after compressor calculation ($P2$)

By using formula 17 the temperature after compressor can be calculated. For the turbocharger cut off operation, the mass flow only divided into half due on of the turbine is cut out.

$$Temperature\ after\ comp, T_2 = \frac{766600.83\ W}{\left(\frac{41.7\ kg/s}{2} * 1013\frac{J}{kg}\right)} + 293\ \mathrm{K} = 330.50\ K$$

To calculate the pressure after compressor, the after compressor isentropic temperature value has to be calculated first with formula 19.

$$0.751 = \frac{T_{2is} - 293\ K}{330.50\ K - 293\ K} \rightarrow T_{2is} = 321.02\ K$$

After compressor pressure can be calculated by using formula 18.

$$p_2 = \left(\frac{321.02\ K}{293\ \mathrm{K}}\right)^{\frac{1.4}{1.4-1}} * 1.032\ bar = 1.42\ bar$$

8. Determine new mass flow value of the compressor

The mass flow standard coefficient calculation with formula 10.

$$\mathrm{Mstandard\ (1.42\ bar\ )} = \frac{(0.005649859 + (-1555.386 - 0.005649859)}{(1 + (1.42/0.000004907295)\text{^}1.033987) - 0.0001)}$$

$$\mathrm{Mstandard\ (1.42\ bar\ )} \rightarrow 0.002046$$

Since the mass flow standard value has been calculated, the new mass flow value can be determined by using formula 9. For this turbocharger cut off operration, theres only 2 compressor operated.

$$\dot{m}_{fresh\ air} = 0.002046 \times \frac{1.032\ bar}{\sqrt{293\ K}} = 13.35 \frac{kg}{s} \text{ for each turbocharger}$$

9. Intercooler phase calculation

When the air pass through the intercooler, the air temperature and pressure value change. For the after cooler temperature (*T3*) can be calculated with intercooler efficiency function (formula 20).

$$0.857 = \frac{330.5\ K - T3}{330.5\ K - 290\ K} \rightarrow T3 = 295.79\ K$$

While for the pressure drop can be calculated using formula 19.

$$\text{Pressure drop } (1.42\ bar) = \frac{0.02421102\ +\ (0.0004137441\ -\ 0.02421102)}{\left(1\ +\ \left(\frac{1.42}{2.256177}\right)^{2.766972}\right)}$$

$$\rightarrow \text{Pressure drop } (1.4135\ bar) = 0.0056\ bar$$

10. Recalculate the scavange air compressor (*P2*) with *the first turbocharger equation*

Using formula 20 the pressure after compressor can be recalculated to get a more precies value of it .

$$\frac{P2}{1.032bar} = 1 + \frac{\frac{40.69kg}{s} \times \frac{1066.6J}{kgK} \times 684.99K}{26.70kg/s \times \frac{1002J}{kgK} \times 293K} \times 0.56 \times \left(1 - \left(\frac{1.032bar}{1.42bar}\right)^{\frac{1.3492-1}{1.3492}}\right)^{\frac{1.365}{1.365-1}}$$

$$P2 = 1.72\ bar$$

This calculation cycle is performed repeatedly in 10%-50% engine load power until the result value is fairly constant and and also if it is possible the value of the calculation is close to the reference value (TEKOMAR curves).

## 4.4 Engine performance calculation result

In this chapter, the performance result obtained is evaluated whether by performing turbocharger cut off will increase the turbocharger efficiency or not. Also the other indicator value will be evaluated and viewed the differences between normal operation performance and the turbocharger cut off performance. The result will also be compared with the TEKOMAR company performance line. The TEKOMAR reference curves are listed on Attachment. These are the comparisons of several aspects.

### 4.4.1 Turbocharger Efficiency Curves

As the theory and the application prove that by performing turbocharger cut off in low load range of engine, the turbochager efficiency will improve. The curves below will show it.

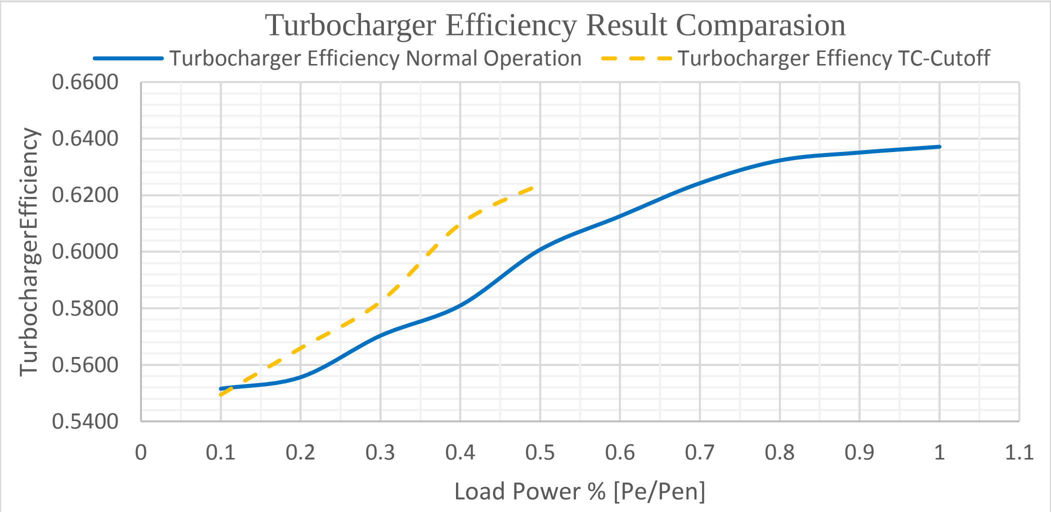



*Figure 4.4.1 Turbocharger efficiency result comparasion curves*

The result curves show that the turbocharger efficiency is increasing when the TC cut off is performed. By cutting one of the turbine, the exhaust energy will goes only into 2 remaining turbocharger which will eventually increased turbine power and also compressor capacity. Therefore it will increase the fresh air mass flow and also pressure for each compressor.

### 4.4.2 Mass Flow Comparasion

As mentioned before by performing TC cut off operation, the compressor side will be able to drag more air into the system due to more power gained from the turbine side.

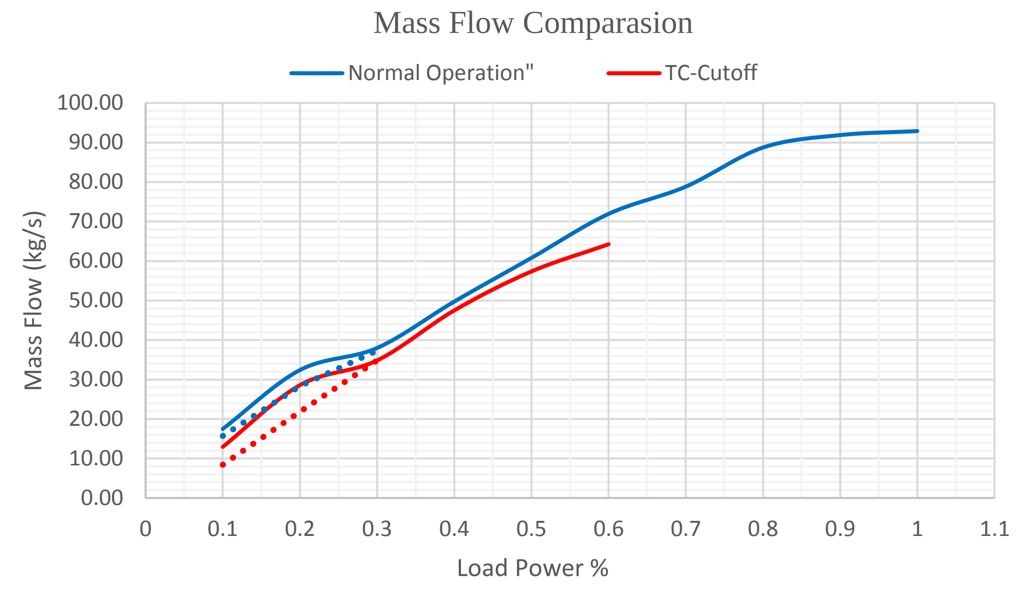



*Figure 4.4.2 Mass flow comparasion curves*

As mentioned before that the compressor power will increase in TC cut off operation, the graph above show the total amount of mass flow that the compressor can drag in to the system. Where for the normal operation there are 3 TC operated and TC cut off is only 2 TC operated.

The influence of the auxiliary blower can be evaluated. The dash line show the normal continued value of the mass flow. As can be seen in the 20% load shown that the mass flow value is increasing due to the addition of air from the blower. The blower is expected stop operating between 20%-30% engine load, for normal operation blower can bring additional air up to 4.07 kg/s meanwhile when TCcut off is performed it can bring more additional air up to 6.69 kg/s.

#### 4.4.3 Scavenge Pressure Comparasion

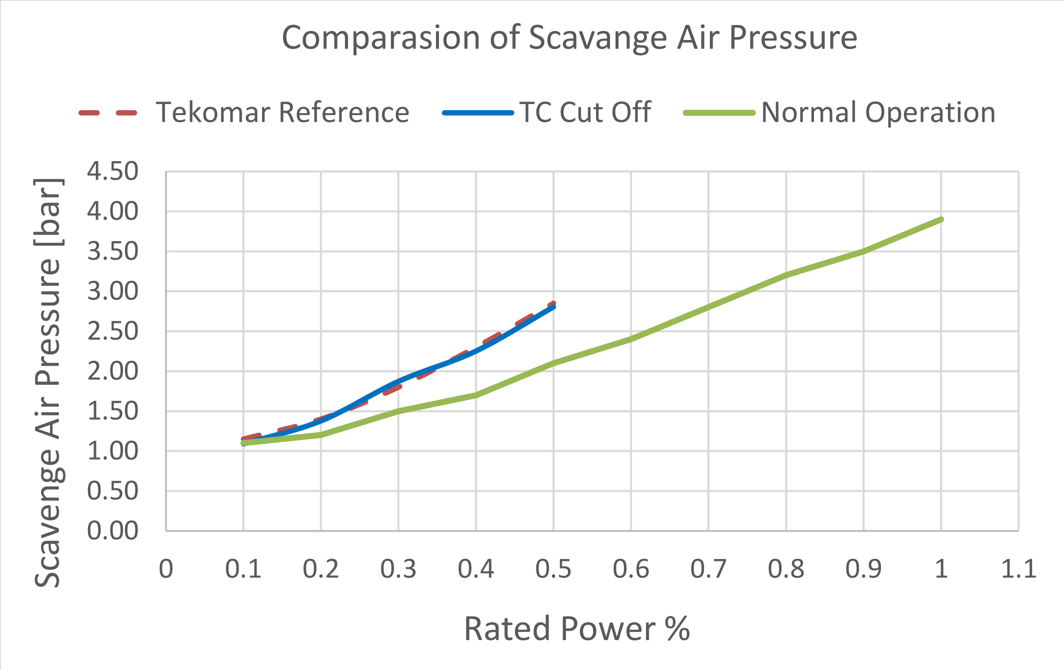



*Figure 4.4.3 Scavange air pressure comparison curves*

From curves above it is shown that the calculation result is slightly different in 40% load power from the reference curves due to some differences in calculation background value and coefficient but has the same conclusion, when TC cut off is performed the compressed air pressure is increases.

### 4.4.3 Compression Pressure and Maximum Compression Comparasion

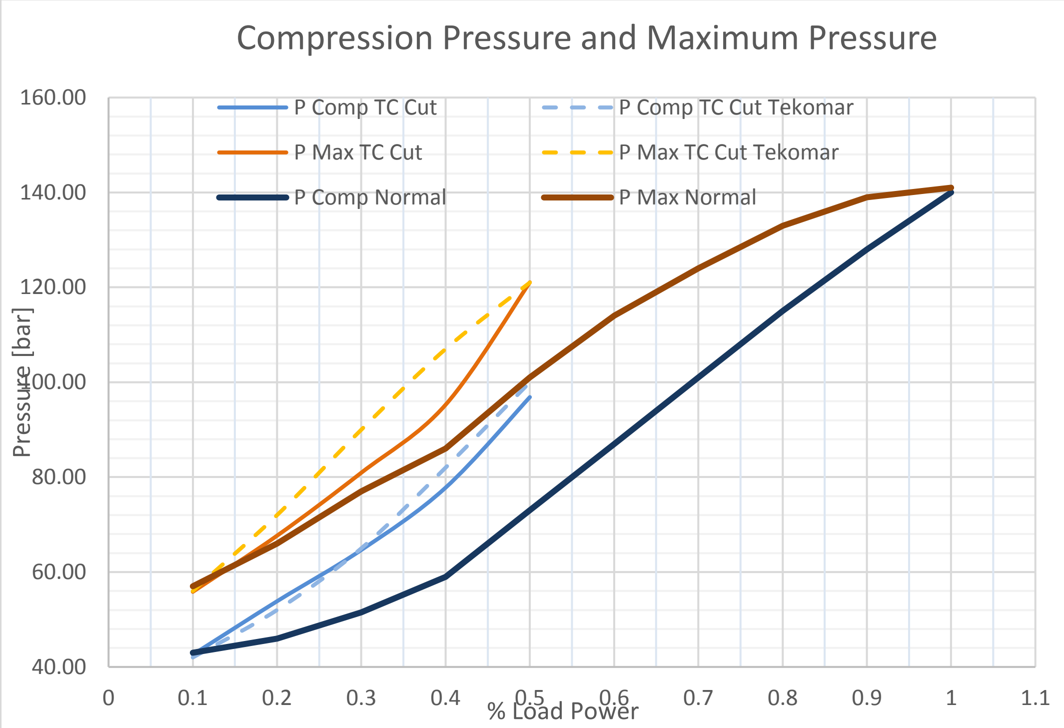



*Figure 4.4.4 Compression and maximum pressure comparison*

By using formula 24,25 and 26, the compression pressure and maximum pressure can be calculated. The result curves shown that theres a little different in value between the thesis calculation and TEKOMAR curves especially in 20% and 30% load power, but both curves shown that the compression pressure and maximum pressure value is increasing when TC cut off is operated. The biggest different between thesis and TEKOMAR curves is on 40% load points where the maximum pressure value different is 12 bar.

Due to when we operated the turbocharger cut off the scavange air pressure will increase, it is shown in figure 4.4.3. Therefore the inlet air pressure of the cylinder is higher than the air pressure in normal operation then is causes the compression and maximum pressure of the cylinder will eventually increase.

### 4.4.4 After and Before Turbine Temperature Comparasion

The next indication to be compared is the temperature before turbine and after turbine. However the biggest differences value of the thesis calculation and TEKOMAR line will be found in this section.

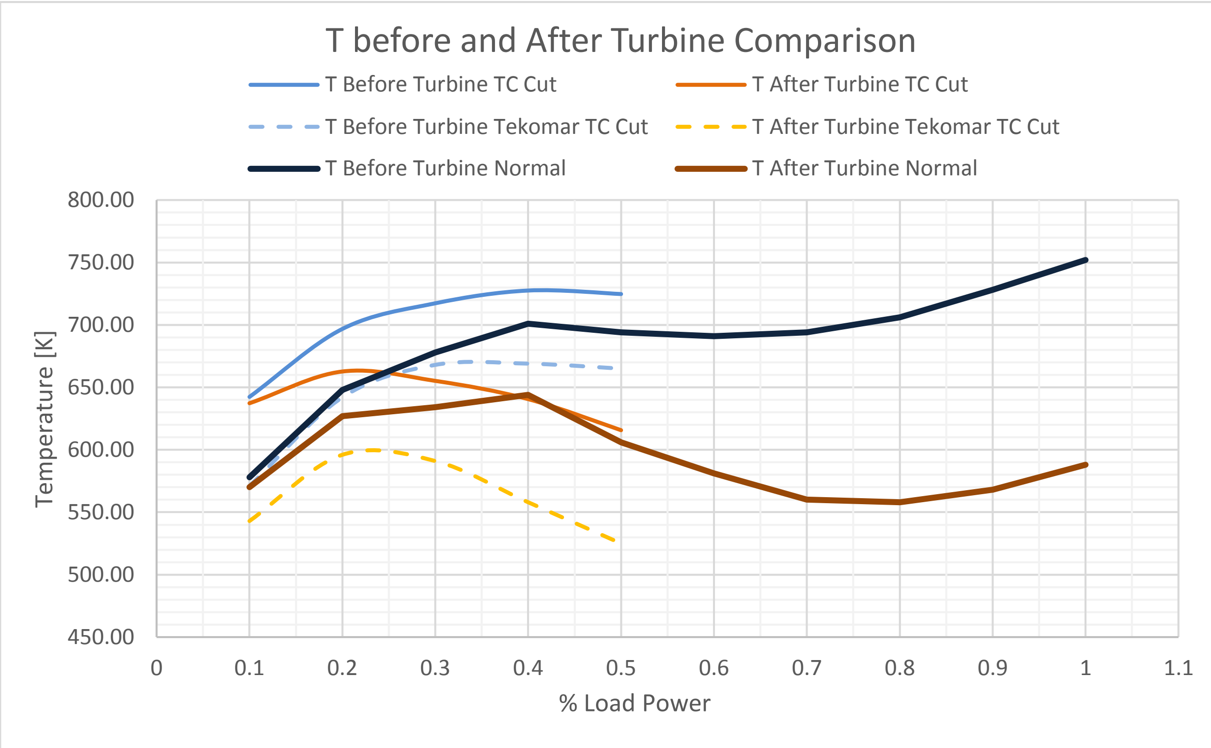



*Figure 4.4.5 Temperature before and after turbine comparison curves*

Figure 4.4.5 clearly shows that the thesis calculation result has a different conclusion with TEKOMAR curves, the thesis calculation result shown that the temperature before and after turbine in TC cut off operation is higher than the normal operation meanwhile the TEKOMAR curves shows the opposite. As an explanation it can be seen in figure 4.4.2 that TC cut operation drag less amount of air to the engine than the normal operation, but the exhaust gas energy generated between TC cut and normal operation is not much different. It can be concluded that with the quite same energy but the air mass flow is reduced, the temperature will increase and the relation can be seen in formula 27. Furthermore, it can be seen from the calculation curves the effect of the blower support, from 10 % until around 20%-30% power the temperature is slightly low due to the blower is supplying additional air the engine.

### 4.4.5 Turbocharger Speed Comparison

By using given data of turbocharger speed and scavenge pressure from the engine report data, a turbocharger characteristics speed function is created.

$$\text{TC Speed } f(scav.pressure) = \frac{16694.73 + (-1444248000 - 16694.73)}{(1 + (Scav.Pressure/1.120626e - 8)^{\wedge}0.6243389)}$$

*Formula 30. Turbocharger speed function*

By using this function on the calculation result, the comparison curves can be created.

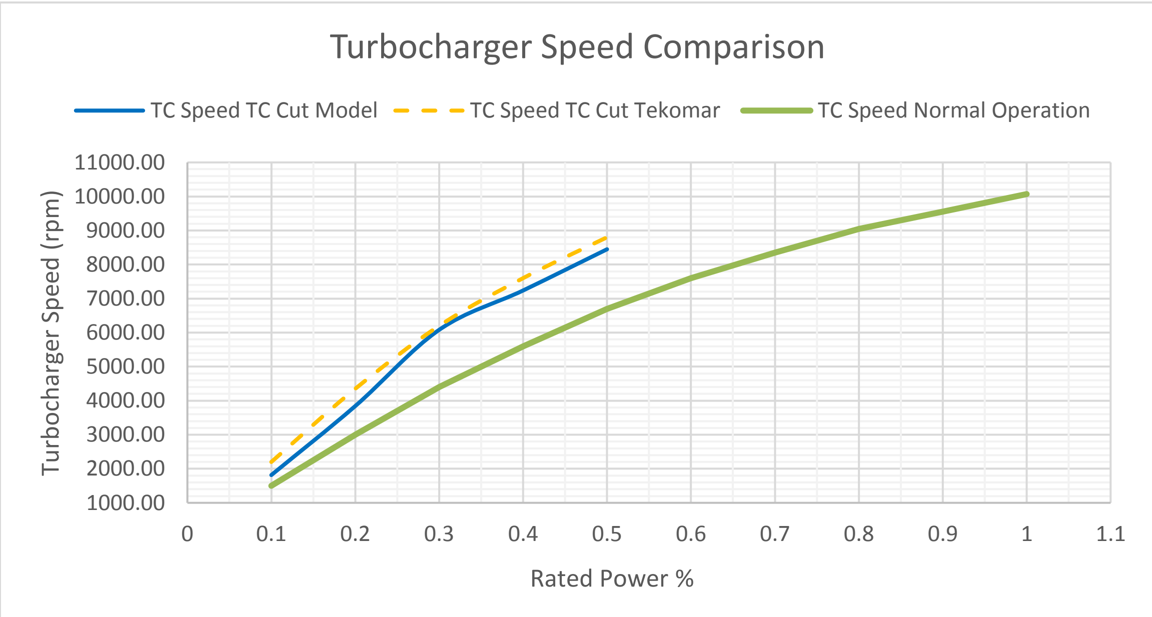



*Figure 4.4.6 Turbocharger speed comparison curves*

As mentioned before that by performing turbocharger cut off will increase the energy in the remaining turbocharger operated. Therefore it will increase the speed rotation of the turbine and also the compressor which will lead to increased of the fresh air mass flow and also pressure for each compressor. It can be seen that the thesis calculation curve result has a little difference with the TEKOMAR line due to the compressor pressure result difference. The biggest different can be seen at 20% load points where the turbocharger speed value diference is almost 500 rpm.

# CHAPTER V
# CONCLUSION

The slow steaming trend is already part of one of the operating systems on the ship. By using this method there are some technical challenge challenges arise and one of them is the problem of air supply. The turbocharger cut off technology has been widely used among large engines user.

Turbocharger cut off method provide higher RPM and also efficiency on remaining operated turbocharger, which will lead to reducing the fuel oil consumption further because the better quality of the air that delivered into the combustion chamber. In the thesis mathematical model can be seen that turbocharger cut off method increasing the power of the remaining turbine and compressor, therefore the compressed air will have a higher pressure.

From the result curves (see figure 4.4.1 – 4.4.6) in can be conclude that by performing TC cut will significantly changing the engine performance. The scavenge air pressure, temperature before and after turbine, turbocharger speed, turbocharger efficiency, compression pressure all can be conclude that have a significant increase when TC cut is performed.

A comparison with a another model from TEKOMAR curves can be inferred to have a slight difference. But there are still some considerable value difference, especially in temperature modelling section. Therefore, the model created is still potential to be developed. With the expectation of this model can be varied in various machine operations in the future.

**Bibliography**


1) Andreas Wiesmann, General Manager Innovation & Business Development, Two-stroke, Wärtsilä Services. 2010. *Slow steaming-a viable long-term option?* Helsinki, Finlandia: Wartsilla Company.

2) Bahadori, Alireza. 2011. "Energy Conversion and Management." *Estimation of air specific heat ratio at elevated pressures using simple.*

3) Christiaan Muilwijk, Wouter den Boer, Laurens van der Kooij, Koen Stroomberg . March 31th 2011. *Good speed is low speed - Efficient low speed sailing.* Rotterdam, Netherland: Shipping and Transport College .

4) Company, Dosoan Engine Product. n.d. *TURBOCHARGER C UT O UT S YSTEM.* Seongsan-gu, Changwon-si, Gyeongsangnam-do, South Korea: Doosan.

5) Company, GT Suite Product. 2015. *GT Suite Software Overview.* Westmont, USA: Gamma Technologies.

6) Company, Tecumseh Product. 2015. "Detail compressor modeling and bottle cooler simulation." Frankfurt.

7) Coorporation, MAN B&W. 2010. "MAN B&W K90MC-C6-TII Project Guide." Copenhagen.

8) Jääskeläinen, Hannu. 2011. *Diesel Exhaust Gas.* https://www.dieselnet.com/tech/diesel_exh.php.

9) Kleimola, Matti. n.d. "Turbocharger Efficiency." *Definition and Guidelines for Measurment and Calculation.*

10) Korczewski, Prof. Zbigniew. 2015. "EXHAUST GAS TEMPERATURE MEASUREMENTS IN DIAGNOSTICS OF TURBOCHARGED MARINE INTERNAL COMBUSTION ENGINES."

11) Michael J.Moran, Howard N. Shapiro, Daisie_D.Bo. 1996. *Gas Turbine Theory 4th Edition.* London: Longman Group.

*"This page intentionally left blank"*

# AUTHOR BIODATA

The author was born on 12nd October 1995 in Bandung, as the second child of 2 siblings. His father's name is Muhammad Ramadhan Pamungkas and his mother's name is Dian Nurulqolbi. The author has completed the formal education in Salman Al-Farisi Kindergarten, Salman Al-Farisi elementary school,Salman Al-Farisi junior highschool, and Alfa Centauri senior highschool. The author continued his study for bachelor degree in Double Degree Programme of Marine Engineering (DDME) of Institut Teknologi Sepuluh Nopember (ITS) and Hochschule Wismar, with student registration number is 4213101037. In the college, the author took an area of expertise in Marine Power Plant Laboratory (MPP) . During the college, the author was active in academic and non-academic activities, namely staff of accommodation and transportation in Kejurnas Kempo in 2016, senior staff of Sponsorship for Marine Solar Boat Team for Dutch Solar Challenge, staff of publication and documentation in Marine Icon 2015, following some courses such as Project Management Course and AutoCAD Basic Training Course, SolidWork advance training and many more. The author has selected to go to Germany to do work on his thesis in the last semester. The author has done the first on the job training in PT. Dok Kodja & Bahari Shipyard, Jakarta, and the second on the job Training in PT. Garuda Maintenance Facility in Cengkareng.

## Statement of authorship

Statement of authorship

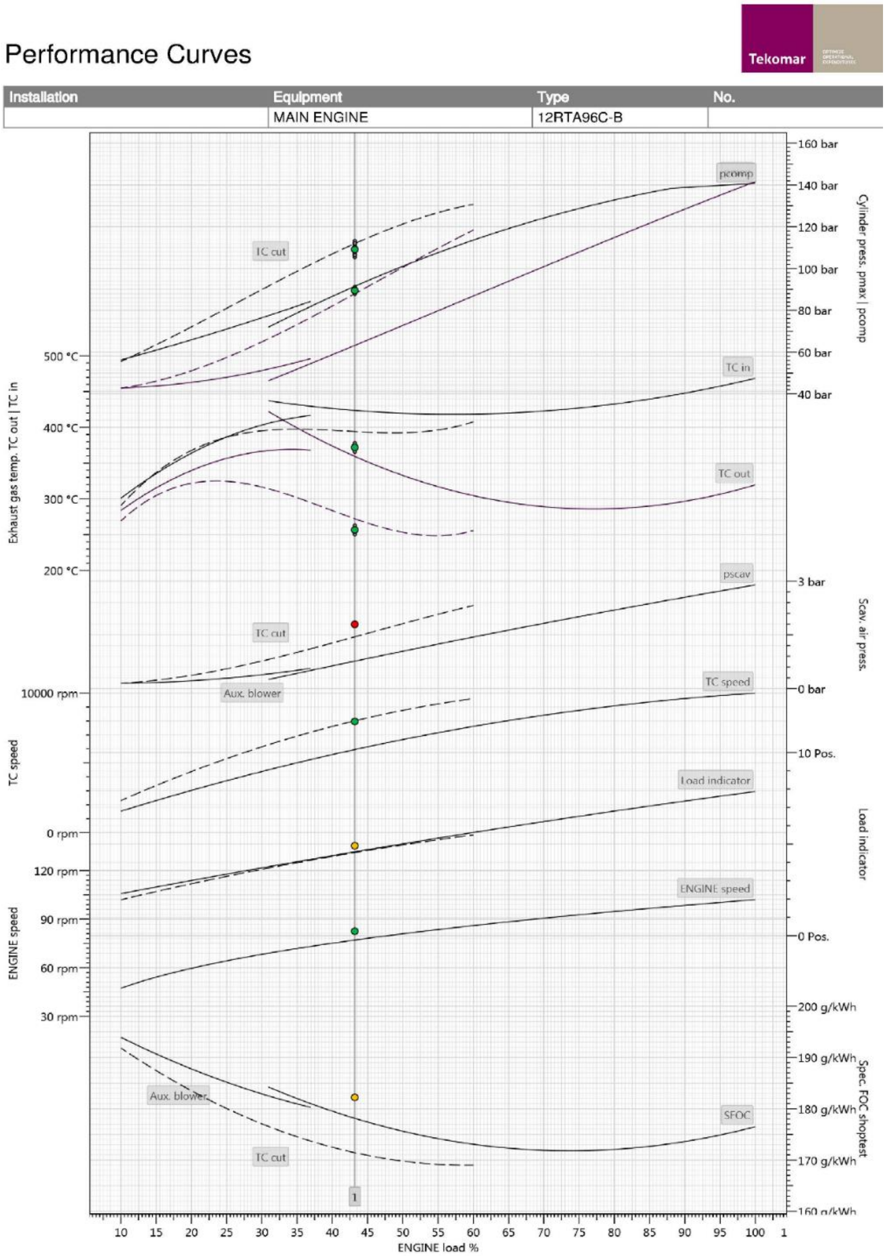
I conform that the work presented in this research proposal/research report has been performed and interpreted solely by myself except where explicitly identified to the contrary.

Rostock, 4 July 2017 Muhammad Ramadhan Pamungkas

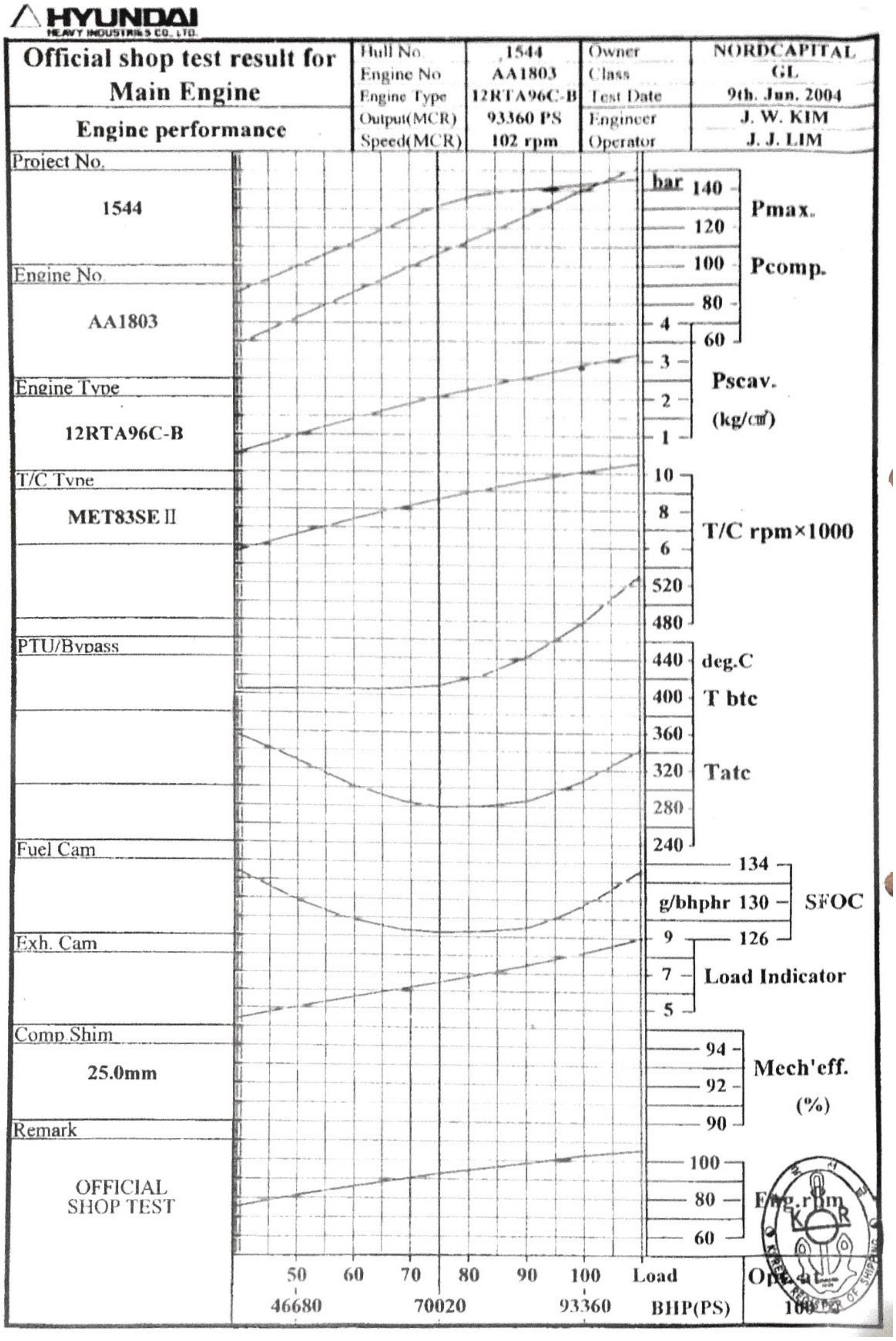
*"This page intentionally left blank"*

## Attachments

1. Engine RTA96C-B TEKOMAR performance curves



2. Engine test performance curves from MARIDIS company.



3. Engine data test performance from MARIDIS company.

**HYUNDAI** HEAVY INDUSTRIES CO., LTD. — Official Shop Test Result For Main Engine — HYUNDAI SULZER — Data Sheet 1

| | | | |
|---|---|---|---|
| Hull No. | 1544 | Owner | NORDCAPITAL |
| Engine No. | AA1803 | Class | GL |
| Engine Type | 12RTA96C-B | Test Date | 9th. Jun. 2004 |
| Output(MCR) | 93360 PS | Engineer | J. W. KIM |
| Speed(MCR) | 102 rpm | Operator | J. J. LIM |

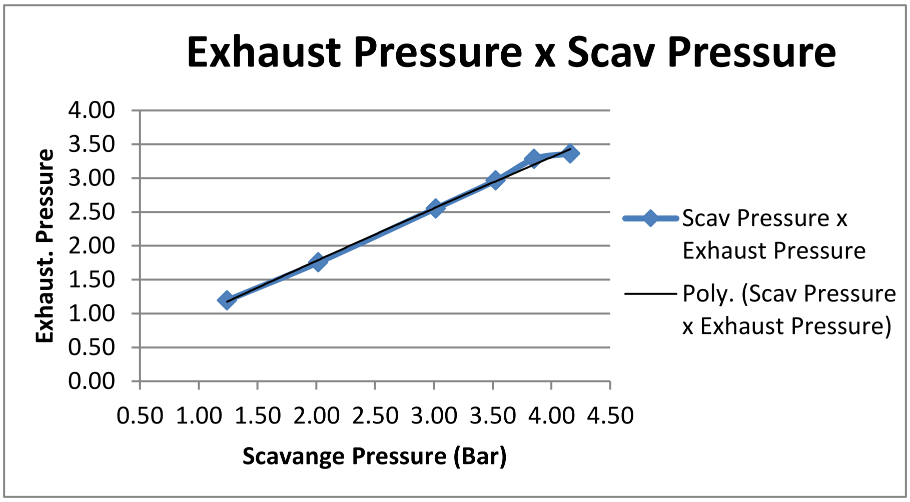
| Group | Item | | | Unit | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| General Data | Loading power(%) / Check time | | | Load | Time | 25% | 11:50 | 50% | 12:20 | 75% | 12:50 | 90%-1 | 13:20 |
| | Engine power / speed (theoretical) | | | BHP | r/min | 23340 | 64.26 | 46680 | 80.96 | 70020 | 92.67 | 84024 | 98.48 |
| | Mean eff. pressure design / actual | | | kg/cm² | | 7.38 | 7.39 | 11.72 | 11.72 | 15.35 | 15.36 | 17.34 | 17.34 |
| | Brake force (indicated) | | | KNm | | 2551 | | 4050 | | 5307 | | 5993 | |
| | Gov terminal shaft/Load indicator | Gov. | L/I | Pos. | | 3.61 | 3.60 | 5.14 | 5.10 | 6.54 | 6.50 | 7.44 | 7.40 |
| | VIT & FQS angle | VIT | FQS | Pos. | | 0.20 | 0.0 | 0.20 | 0.0 | 1.40 | 0.0 | 1.40 | 0.0 |
| | VEC length | | | Pos.(mm) | | | | | | | | | |
| | Engine speed measured | | | r/min | | 64.30 | | 81.00 | | 92.70 | | 98.50 | |
| | Engine power measured | | | BHP | | 23354 | | 46707 | | 70045 | | 84048 | |
| | Total Fuel oil consumption | | | kg/h | | 3205.167 | | 6127.480 | | 8976.885 | | 10838.836 | |
| | Fuel oil consumption MEASURED | | | g/BHP h | | 137.241 | | 131.188 | | 128.159 | | 128.960 | |
| | ISO CORRECTED | | | g/BHP h | | 136.42 | | 129.95 | | 126.42 | | 127.02 | |
| | Lubricator motor : speed | | | r/min | | 580.0 | | 984.0 | | 1401.0 | | 1682.0 | |
| | Total Cyl. lub. oil feed rate | | | kg/h | | 47.18 | | 72.40 | | 100.86 | | 126.07 | |
| | Cyl. lub. oil feed rate | | | g/BHP h | | 2.02 | | 1.55 | | 1.44 | | 1.50 | |
| Scavenge Air | Auxiliary blowers | | | on - off | | ON - 40 | | OFF | | OFF | | OFF | |
| | Turbocharger speed | 1 | 2 | r/min | | 3650 | 3620 | 6760 | 6730 | 8740 | 8710 | 9645 | 9580 |
| | | 3 | 4 | r/min | | 3625 | | 6700 | | 8680 | | 9525 | |
| | Barometric pressure / Room temp | | | mbar | ℃ | 1010 | 24.5 | 1010 | 25.0 | 1010 | 25.5 | 1010 | 25.5 |
| | T/C filter Suction pressure | 1 | 2 | mmAq | | 4 | 6 | 20 | 18 | 42 | 42 | 56 | 55 |
| | | 3 | 4 | mmAq | | 4 | | 18 | | 37 | | 47 | |
| | Pressure after blower | 1 | 2 | mmHg | | 153 | 156 | 743 | 746 | 1496 | 1496 | 1876 | 1886 |
| | | 3 | 4 | mmHg | | 163 | | 746 | | 1496 | | 1883 | |
| | Charge air pressure | | | mmHg | kg/㎠ | 157 | 0.21 | 740 | 1.01 | 1488 | 2.02 | 1871 | 2.54 |
| | Press drop across air cooler | 1 | 2 | mmAq | | 41 | 36 | 113 | 98 | 172 | 146 | 190 | 164 |
| | | 3 | 4 | mmAq | | 31 | | 86 | | 130 | | 150 | |
| | A/c water inlet temp | 1 2 | 3 4 | ℃ | | 17 | | 17 | | 17 | | 17 | |
| | A/c water outlet temp | 1 2 | 3 4 | ℃ | | 21 19 26 | | 55 50 55 | | 70 67 66 | | 73 73 68 | |
| | Temp. bef. blower | 1 | 2 | ℃ | | 25.5 | 27.5 | 26.5 | 27.5 | 27.0 | 28.5 | 26.5 | 29.0 |
| | | 3 | 4 | ℃ | | 22.0 | | 21.0 | | 21.0 | | 21.0 | |
| | Temp. bef. air cooler | 1 | 2 | ℃ | | 44.0 | 48.0 | 100.0 | 102.0 | 154.0 | 155.0 | 182.0 | 182.0 |
| | | 3 | 4 | ℃ | | 45.0 | | 99.0 | | 154.0 | | 180.0 | |
| | Temp. after air cooler | 1 | 2 | ℃ | | 21.5 | 19.0 | 27.0 | 26.5 | 34.5 | 34.0 | 37.0 | 37.0 |
| | | 3 | 4 | ℃ | | 21.0 | | 29.0 | | 35.0 | | 37.0 | |
| Exhaust Gas System | Pressure bef. turbine | 1 | 2 | mmHg | | 118 | 122 | 545 | 541 | 1150 | 1142 | 1477 | 1468 |
| | | 3 | 4 | mmHg | | 125 | | 543 | | 1147 | | 1472 | |
| | Pressure aft. turbine | 1 | 2 | mmAq | | 1 | 1 | 2 | 2 | 6 | 6 | 10 | 10 |
| | | 3 | 4 | mmAq | | 1 | | 2 | | 6 | | 10 | |
| | Temp. bef turbine | 1 | 2 | ℃ | | 380 | 375 | 418 | 400 | 422 | 407 | 458 | 442 |
| | | 3 | 4 | ℃ | | 380 | | 397 | | 397 | | 426 | |
| | Temp. after turbine | 1 | 2 | ℃ | | 340 | 335 | 338 | 325 | 290 | 278 | 298 | 285 |
| | | 3 | 4 | ℃ | | 345 | | 324 | | 270 | | 274 | |
| | Temp. after cylinder | Ave | | ℃ | | 335.3 | | 335.4 | | 338.5 | | 366.7 | |
| | | No, | 1 2 | ℃ | | 338 | 330 | 349 | 339 | 350 | 348 | 378 | 374 |
| | | No, | 3 4 | ℃ | | 322 | 345 | 330 | 339 | 338 | 342 | 369 | 378 |
| | | No, | 5 6 | ℃ | | 358 | 339 | 348 | 331 | 350 | 332 | 381 | 360 |
| | | No, | 7 8 | ℃ | | 335 | 318 | 325 | 318 | 330 | 322 | 360 | 350 |
| | | No, | 9 10 | ℃ | | 335 | 317 | 336 | 327 | 339 | 330 | 365 | 355 |
| | | No, | 11 12 | ℃ | | 341 | 345 | 339 | 344 | 341 | 340 | 366 | 364 |
| Cyl. Pressures | Maximum / Compression press | Ave | | bar | bar | 65.6 | 44.8 | 99.4 | 72.0 | 131.2 | 107.2 | 139.5 | 126.6 |
| | | No, | 1 | bar | bar | 66.0 | 45.0 | 100.0 | 72.0 | 131.0 | 107.0 | 140.0 | 128.0 |
| | | No, | 2 | bar | bar | 65.0 | 45.0 | 100.0 | 72.0 | 131.0 | 107.0 | 140.0 | 127.0 |
| | | No, | 3 | bar | bar | 66.0 | 45.0 | 100.0 | 72.0 | 131.0 | 108.0 | 140.0 | 127.0 |
| | | No, | 4 | bar | bar | 65.0 | 44.0 | 99.0 | 72.0 | 131.0 | 107.0 | 140.0 | 126.0 |
| | | No, | 5 | bar | bar | 65.0 | 44.0 | 99.0 | 71.0 | 130.0 | 106.0 | 140.0 | 126.0 |
| | | No, | 6 | bar | bar | 65.0 | 45.0 | 99.0 | 72.0 | 131.0 | 107.0 | 140.0 | 127.0 |
| | | No, | 7 | bar | bar | 67.0 | 45.0 | 99.0 | 72.0 | 131.0 | 107.0 | 139.0 | 126.0 |
| | | No, | 8 | bar | bar | 66.0 | 45.0 | 100.0 | 73.0 | 132.0 | 108.0 | 140.0 | 126.0 |
| | | No, | 9 | bar | bar | 66.0 | 45.0 | 100.0 | 72.0 | 131.0 | 107.0 | 139.0 | 126.0 |
| | | No, | 10 | bar | bar | 66.0 | 45.0 | 99.0 | 72.0 | 132.0 | 108.0 | 139.0 | 126.0 |
| | | No, | 11 | bar | bar | 65.0 | 45.0 | 99.0 | 72.0 | 131.0 | 107.0 | 139.0 | 127.0 |
| | | No, | 12 | bar | bar | 65.0 | 45.0 | 99.0 | 72.0 | 132.0 | 107.0 | 138.0 | 127.0 |
| System | Air pressure Valve air spring supply | | | kg/cm² | | 6.5 | | 7.5 | | 7.5 | | 7.3 | |
| | Coolant press Cyl Cooling F.W inlet | | | kg/cm² | | 3.70 | | 3.70 | | 3.80 | | 3.70 | |
| | Coolant temp. Cyl inlet / outlet min-max | | | ℃ | | 57 | 62-64 | 68 | 75-77 | 66 | 77-78 | 70 | 79-81 |
| | T/C oil pressure (inlet) | 1 | 2 | kg/cm² | | 1.40 | 1.20 | 1.05 | 1.04 | 1.20 | [illegible] | [illegible] | 1.30 |
| | | 3 | 4 | | | 1.40 | | 1.04 | | 1.15 | | | |
| | T/C oil temp (inlet) | 1 2 | 3 4 | ℃ | | 38 41 37 | | 40 40 36 | | 39 42 [illegible] | | [illegible] 44 46 | |
| | T/C oil temp (outlet) | 1 2 | 3 4 | ℃ | | 38 43 44 | | 44 45 45 | | 54 60 [illegible] | | 61 64 [illegible] | |
| | Oil pressure : Bearing / Cross head | | | kg/cm² | | 5.10 | 11.60 | 5.10 | 12.00 | 5.20 | [illegible] | [illegible] | 11.80 |
| | Oil temp. Main L.O / Cross head in | | | ℃ | | 38 | 40 | 38 | 40 | 40 | 42 | 40 | 44 |
| | Fuel oil press (inlet,outlet)/ Temp.bef engine | | | kg/㎠ | ℃ | 8.7 2.3 | 35.0 | 8.3 1.9 | 38.0 | 8.8 2.9 | 40.0 | [illegible] | 39.0 |
| | Axial detuner pressure | free | drive | kg/cm² | | 4.70 | 4.50 | 4.70 | 4.50 | 4.50 | 4.20 | 4.20 | 4.00 |
| | Thrust pad temp. | | | ℃ | | 38.0 | | 38.0 | | 39.0 | | 41.0 | |

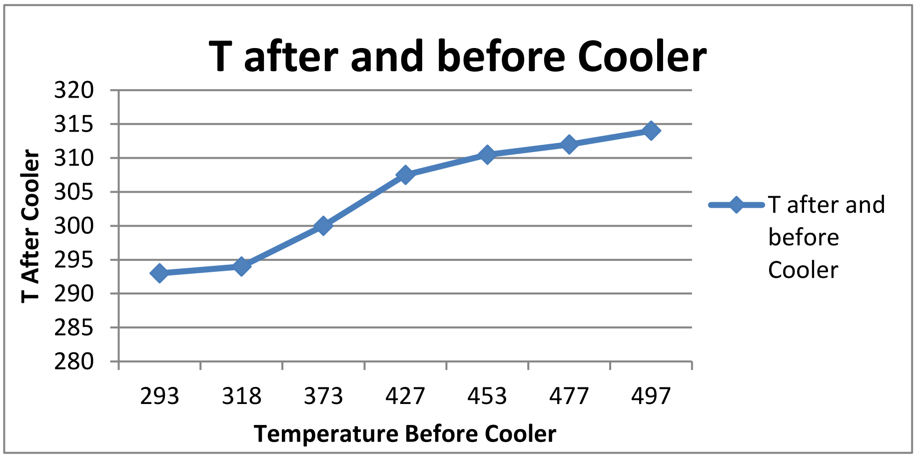
**HYUNDAI** HEAVY INDUSTRIES CO., LTD. — Official Shop Test Result For Main Engine — HYUNDAI SULZER — Data Sheet 2

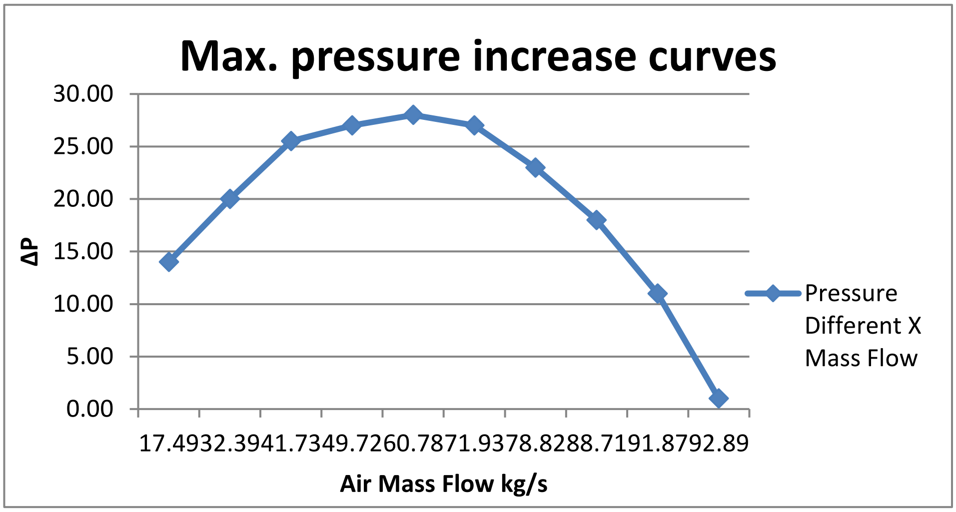
| | | | |
|---|---|---|---|
| Hull No. | 1544 | Owner | NORDCAPITAL |
| Engine No. | AA1803 | Class | GL |
| Engine Type | 12RTA96C-B | Test Date | 9th. Jun. 2004 |
| Output(MCR) | 93360 PS | Engineer | J. W. KIM |
| Speed(MCR) | 102 rpm | Operator | J. J. LIM |

| Group | Item | | | Unit | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| General Data | Loading power(%) / Check time | | | Load | Time | 90%-2 | 13:50 | 100%-1 | 14:20 | 100%-2 | 14:50 | 110% | 15:20 |
| | Engine power / speed (theoretical) | | | BHP | r/min | 84024 | 102.00 | 93360 | 102.00 | 93360 | 102.00 | 102696 | 105.29 |
| | Mean eff. pressure design / actual | | | kg/cm² | | 16.74 | 16.74 | 18.60 | 18.60 | 18.60 | 18.60 | 19.82 | 19.82 |
| | Brake force (indicated) | | | KNm | | 5787 | | 6429 | | 6429 | | 6850 | |
| | Gov terminal shaft/Load indicator | Gov. | L/I | Pos. | | 7.19 | 7.15 | 8.15 | 8.10 | 8.17 | 8.10 | 8.95 | 8.90 |
| | VIT & FQS angle | VIT | FQS | Pos. | | 1.90 | 0.0 | 0.40 | 0.0 | 0.30 | 0.0 | -0.30 | 0.0 |
| | VEC length | | | Pos.(mm) | | | | | | | | | |
| | Engine speed measured | | | r/min | | 102.00 | | 102.00 | | 102.00 | | 105.30 | |
| | Engine power measured | | | BHP | | 84043 | | 93366 | | 93366 | | 102699 | |
| | Total Fuel oil consumption | | | kg/h | | 10908.280 | | 12274.000 | | - | | 13944.000 | |
| | Fuel oil consumption MEASURED | | | g/BHP h | | 129.794 | | 131.461 | | - | | 135.776 | |
| | ISO CORRECTED | | | g/BHP h | | 127.79 | | 129.30 | | - | | 133.38 | |
| | Lubricator motor /shaft speed | | | r/min | | 1675.0 | | 1901.0 | | 1909.0 | | 2174.0 | |
| | Total Cyl. lub. oil feed rate | | | kg/h | | 121.86 | | 140.05 | | 140.05 | | 147.89 | |
| | Cyl. lub. oil feed rate | | | g/BHP h | | 1.45 | | 1.50 | | 1.50 | | 1.44 | |
| Scavenge Air | Auxiliary blowers | | | on - off | | OFF | | OFF | | OFF | | OFF | |
| | Turbocharger speed | 1 | 2 | r/min | | 9540 | 9480 | 10140 | 10070 | 10160 | 10090 | 10710 | 10620 |
| | | 3 | 4 | r/min | | 9425 | | 10010 | | 10025 | | 10550 | |
| | Barometric pressure / Room temp. | | | mbar | ℃ | 1010 | 25.5 | 1010 | 26.0 | 1010 | 26.5 | 1010 | 26.5 |
| | T/C filter Suction pressure | 1 | 2 | mmAq | | 58 | 56 | 64 | 62 | 65 | 63 | 74 | 71 |
| | | 3 | 4 | mmAq | | 47 | | 56 | | 56 | | 60 | |
| | Pressure after blower | 1 | 2 | mmHg | | 1847 | 1860 | 2128 | 2130 | 2140 | 2148 | 2352 | 2356 |
| | | 3 | 4 | mmHg | | 1860 | | 2130 | | 2152 | | 2356 | |
| | Charge air pressure | | | mmHg | kg/㎠ | 1845 | 2.51 | 2122 | 2.89 | 2133 | 2.90 | 2347 | 3.19 |
| | Press drop across air cooler | 1 | 2 | mmAq | | 190 | 165 | 200 | 175 | 200 | 176 | 210 | 185 |
| | | 3 | 4 | mmAq | | 150 | | 155 | | 156 | | 165 | |
| | A/c water inlet temp | 1 2 | 3 4 | ℃ | | 17 | | 17 | | 17 | | 17 | |
| | A/c water outlet temp | 1 2 | 3 4 | ℃ | | 73 73 67 | | 70 73 66 | | 70 72 66 | | 67 65 64 | |
| | Temp. bef. blower | 1 | 2 | ℃ | | 26.75 | 29.0 | 27.0 | 29.5 | 27.0 | 29.5 | 27.5 | 30.25 |
| | | 3 | 4 | ℃ | | 21.5 | | 21.5 | | 21.5 | | 21.0 | |
| | Temp. bef. air cooler | 1 | 2 | ℃ | | 183.0 | 183.0 | 204.0 | 204.0 | 205.0 | 205.0 | 224.0 | 226.0 |
| | | 3 | 4 | ℃ | | 180.0 | | 202.0 | | 202.0 | | 222.0 | |
| | Temp. after air cooler | 1 | 2 | ℃ | | 37.5 | 37.5 | 39.0 | 39.0 | 39.0 | 39.0 | 41.5 | 41.0 |
| | | 3 | 4 | ℃ | | 37.5 | | 39.0 | | 39.0 | | 40.5 | |
| Exhaust Gas System | Pressure bef. turbine | 1 | 2 | mmHg | | 1463 | 1450 | 1695 | 1680 | 1698 | 1680 | 1897 | 1748 |
| | | 3 | 4 | mmHg | | 1449 | | 1688 | | 1688 | | 1660 | |
| | Pressure aft. turbine | 1 | 2 | mmAq | | 10 | 10 | 20 | 20 | 20 | 20 | 30 | 30 |
| | | 3 | 4 | mmAq | | 10 | | 20 | | 20 | | 30 | |
| | Temp. bef turbine | 1 | 2 | ℃ | | 458 | 439 | 496 | 479 | 499 | 480 | 550 | 531 |
| | | 3 | 4 | ℃ | | 424 | | 463 | | 465 | | 516 | |
| | Temp. after turbine | 1 | 2 | ℃ | | 300 | 283 | 320 | 304 | 321 | 307 | 357 | 338 |
| | | 3 | 4 | ℃ | | 272 | | 296 | | 297 | | 330 | |
| | Temp. after cylinder | Ave | | ℃ | | 364.6 | | 398.3 | | 398.8 | | 440.8 | |
| | | No, | 1 2 | ℃ | | 378 | 375 | 412 | 408 | 415 | 406 | 465 | 453 |
| | | No, | 3 4 | ℃ | | 369 | 376 | 399 | 406 | 398 | 407 | 442 | 452 |
| | | No, | 5 6 | ℃ | | 374 | 359 | 408 | 396 | 410 | 396 | 452 | 435 |
| | | No, | 7 8 | ℃ | | 357 | 345 | 390 | 379 | 391 | 378 | 429 | 418 |
| | | No, | 9 10 | ℃ | | 364 | 352 | 396 | 388 | 399 | 388 | 438 | 430 |
| | | No, | 11 12 | ℃ | | 364 | 362 | 398 | 399 | 398 | 399 | 433 | 442 |
| Cyl. Pressures | Maximum / Compression press | Ave | | bar | bar | 140.1 | 125.3 | 141.9 | 139.3 | 142.0 | 140.6 | 145.1 | 151.0 |
| | | No, | 1 | bar | bar | 139.0 | 126.0 | 142.0 | 140.0 | 142.0 | 141.0 | 145.0 | 151.0 |
| | | No, | 2 | bar | bar | 140.0 | 126.0 | 142.0 | 140.0 | 142.0 | 141.0 | 146.0 | 151.0 |
| | | No, | 3 | bar | bar | 140.0 | 125.0 | 142.0 | 140.0 | 143.0 | 141.0 | 145.0 | 150.0 |
| | | No, | 4 | bar | bar | 140.0 | 125.0 | 142.0 | 138.0 | 142.0 | 141.0 | 144.0 | 150.0 |
| | | No, | 5 | bar | bar | 140.0 | 125.0 | 142.0 | 139.0 | 142.0 | 141.0 | 145.0 | 150.0 |
| | | No, | 6 | bar | bar | 140.0 | 125.0 | 141.0 | 139.0 | 143.0 | 141.0 | 145.0 | 151.0 |
| | | No, | 7 | bar | bar | 140.0 | 125.0 | 142.0 | 139.0 | 142.0 | 141.0 | 145.0 | 151.0 |
| | | No, | 8 | bar | bar | 141.0 | 125.0 | 142.0 | 138.0 | 142.0 | 140.0 | 146.0 | 151.0 |
| | | No, | 9 | bar | bar | 141.0 | 125.0 | 142.0 | 138.0 | 142.0 | 140.0 | 145.0 | 152.0 |
| | | No, | 10 | bar | bar | 140.0 | 126.0 | 142.0 | 140.0 | 142.0 | 140.0 | 145.0 | 152.0 |
| | | No, | 11 | bar | bar | 140.0 | 125.0 | 142.0 | 140.0 | 141.0 | 140.0 | 145.0 | 152.0 |
| | | No, | 12 | bar | bar | 140.0 | 126.0 | 142.0 | 140.0 | 141.0 | 140.0 | 145.0 | 151.0 |
| System | Air pressure Valve air spring supply | | | kg/cm² | | 7.0 | | 7.4 | | 7.3 | | 7.3 | |
| | Coolant press Cyl.Cooling F.W inlet | | | kg/cm² | | 3.70 | | 3.70 | | 3.70 | | 3.70 | |
| | Coolant temp Cyl.inlet / outlet min-max | | | ℃ | | 67 | 79-80 | 67 | 81-83 | 67 | 81-83 | 69 | 84-86 |
| | T/C oil pressure (inlet) | 1 | 2 | kg/cm² | | 1.20 | 1.20 | 1.30 | 1.32 | 1.30 | [illegible] | [illegible] | 1.18 |
| | | 3 | 4 | | | 1.15 | | 1.30 | | 1.30 | | | |
| | T/C oil temp (inlet) | 1 2 | 3 4 | ℃ | | 43 46 42 | | 46 48 43 | | 46 48 [illegible] | | [illegible] | |
| | T/C oil temp (outlet) | 1 2 | 3 4 | ℃ | | 63 68 67 | | 67 72 70 | | 68 72 70 | | [illegible] | |
| | Oil pressure Bearing / Cross head | | | kg/cm² | | 4.90 | 11.40 | 5.00 | 11.40 | 4.90 | [illegible] | [illegible] | [illegible] |
| | Oil temp Main L.O / Cross head in | | | ℃ | | 44 | 45 | 44 | 46 | 44 | 46 | [illegible] | 48 |
| | Fuel oil press (inlet outlet)/ Temp bef.engine | | | kg/㎠ | ℃ | 8.6 2.4 | 37.0 | 9.5 3.6 | 38.0 | 9.5 4.0 | 36.0 | [illegible] | 36.0 |
| | Axial detuner pressure | free | drive | kg/cm² | | 4.20 | 4.00 | 4.00 | 3.80 | 3.80 | 3.60 | 3.80 | 3.60 |
| | Thrust pad temp. | | | ℃ | | 43.0 | | 44.0 | | 45.0 | | 46.0 | |

4. Model calculation curves reference




Max. pressure increase curves
30.00
25.00
20.00
15.00
10.00
5.00
0.00
ΔP
17.4932.3941.7349.7260.7871.9378.8288.7191.8792.89
Air Mass Flow kg/s
Pressure Different X Mass Flow