**BACHELOR THESIS – ME 141502**

# SELECTION OF TURBOCHARGERS ON DYNAMIC MAIN ENGINE OPERATING CONDITIONS

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SURABAYA
2017

**SKRIPSI – ME 141502**

# PEMILIHAN TURBOCHARGER PADA *DYNAMIC MAIN ENGINE OPERATING CONDITIONS*

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2017

## APPROVAL FORM

# SELECTION OF TURBOCHARGERS ON DYNAMIC MAIN ENGINE OPERATING CONDITIONS

## BACHELOR THESIS

Submitted to Comply One of the Requirements to Obtain a Bachelor of Engineering Degree
In
Double Degree Marine Engineering (DDME) Program
Department of Marine Engineering – Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Department of Maritime Studies
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SURABAYA
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**Selection of Turbochargers on Dynamic Main Engine Operating Conditions**

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## ABSTRACT

Every turbocharger has differences efficiency of its working area, it has to be considered seriously because in every volumetric flow rate and pressure ratio turbocharger has different efficiency. Moreover in dynamic main engine operating conditions the various speed of ship is influencing the process of thermodynamics in main engine combustion chamber. The value of pressure ratio and temperature of air supply is relying to its process. MV. Meratus Palembang as a container ship has a dynamic main engine operating conditions to deliver its cargo on time. Based on Grinevetsky Mazing Method, thermodynamics process can be calculated into four main steps; charging, compression, combustion, and expansion. The approach are used to obtain the value of turbocharger operating line, turbocharger efficiency, and engine power output which are the parameter for turbo-engine matching. In the engine-propeller load, at maximum pressure ratio of turbocharger have a volumetric flow rate at 3,15 $m^3$/s for VTR 304, for VTR 321 at 2,5 $m^3$/s, and for VTR 354 also at 3,15 $m^3$/s. These things happen because propeller given a high loads to the engine which affecting turbocharger to support more energy than the engine normal load. So the dynamic main engine operating conditions make the volumetric value increasing. Moreover on VTR 304 at 11,1 knots turbocharger has an efficiency at 80%, and for design speed of ship (Vs) at 10,4 knots turbocharger has an efficiency at 78%, on VTR 321 at 11,1 knots turbocharger has an efficiency at 78 % and for design speed of ship (Vs) at 10,4 knots turbocharger have an efficiency at 80%, on VTR 354 at 11,1 knots turbocharger is surging and for design speed of ship (Vs) at 10,4 knots turbocharger have a surge efficiency also. Engine power output at maximum load condition for VTR 304 the power output become 3220,595 kW, for VTR 321 become 2588,903 kW, and for VTR 354 become 3220,595 kW.

***Keywords – Diesel Engine, Engine-Propeller Turbo matching, Turbocharger Characteristic.***

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**Pemilihan Turbocharger pada Dynamic Main Engine Operating Conditions**

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## ABSTRAK

Setiap tipe turbocharger memiliki perbedaan efisiensi, karena turbocharger sangat dipengaruhi oleh nilai *volumetric flow* rate dan *compression ratio* yang berubah-ubah tergantung pada kondisi kinerja mesin. Terutama pada *dynamic main engine operating conditions*, perubahan kecepatan kapal mempengaruhi proses termodinamika pada ruang bakar mesin. Nilai *compression ratio* dan suplai temperatur udara bergantung pada prosesnya. MV. Meratus Palembang sebagai kapal kontainer bekerja pada "*dynamic main engine operating conditions*" untuk mengantarkan muatan secara tepat waktu. Berdasarkan Grinevetsky Mazing Method, proses termodinamika dapat dihitung menjadi empat bagian utama; *charging*, *compression*, *combustion*, dan *expansion*. Metode ini dipergunakan untuk mendapatkan nilai *turbocharger operating line, turbocharger efficiency,* dan *engine power output* dimana nilai-nilai tersebut merupakan parameter dari *turbo-engine matching*. Efek pembebanan mempengaruhi nilai *volumetric flow rate* 3,15 $m^3/s$ untuk VTR 304, untuk VTR 321 2,5 $m^3/s$, dan untuk VTR 354 juga pada 3,15 $m^3/s$. Hal ini terjadi karena mesin diberi beban yang tinggi oleh *propeller* yang akan mempengaruhi kinerja turbocharger untuk meberikan energi lebih. Sehingga pada *dynamic main engine operating conditions* membuat nilai *volumetric flow rate* semakin meningkat. Selain itu pada VTR 304 di kecepatan kapal 11,1 knot turbocharger memiliki efisiensi sebesar 80%, dan keceptan dinas (Vs) di 10,4 knot turbocharger memiliki efisiensi sebesar 78%, pada VTR 321 di kecepatan 11,1 knot turbocharger memiliki efisiensi sebesar 78% dan pada keceptan dinas (Vs) di 10,4 knot turbocharger memiliki efisiensi sebesar 80%, pada VTR 354 di keceptan 11,1 knot turbocharger bekerja secara *surging* dan kecepatan dinas(Vs) pada 10,4 Knot turbocharger juga bekerja secara *surging*. Daya mesin pada kondisi beban maksimum untuk VTR 304 menjadi 3220.595 kW, untuk VTR 321 menjadi 2588.903 kW, dan untuk VTR 354 menjadi 3220.595 kW.

***Keywords – Diesel Engine, Engine Propeller-Turbo matching, Turbocharger Operating Line Characteristic.***

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## PREFACE

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# CHAPTER I
# INTRODUCTION

## 1.1 Background

Shipping is a cheap and reliable transport mode which provides significant contribution to national economics. As an archipelagic state, every island in Indonesia has its own main commodity for trading and therefore it needs effective and efficient transport mode to achieve competitive economy compared to other countries.

Container ships are important parts of a national supply chain management which needs efficient and effective goods transportation. Therefore, a continuous study on container ships capability is needed. Such technical assessment includes aspects of design, economy and performance of container ships. A more detailed study with the objective of achieving the best performance of container ships can focus only on one of those aspects.

In relation to ship performance, there are many possible topics which are prospective to be studied such as to measure performance of main engine of a ship which is an important strategy to optimize a ship performance as main mode of ocean transportation. One of the important aspects is the efficiency of fuel consumption as it relates to reduction of operational cost and air emission. Therefore, this issue has gained a strong enthusiasm of the ship owners to improve the efficiency of fuel consumption of ships.

Ravaglioli in 2015 argues that "Due to the increasing request for pollutant emissions and fuel consumption reduction, the optimization of turbocharger control has become a critical issue in modern engine management systems. Prior research demonstrates that pollutant emissions reduction and higher engine efficiency can be achieved through a proper combination of turbocharging technique and engine downsizing". Therefore, following Ravaglioli (2015), a study on improvement of engine performance is urgently needed. According to him, a more reduction of emission and better engine efficiency can be strengthened through a suitable mix between "turbocharging technique and engine downsizing". So that the use of a certain type of turbocharger will optimize the engine capability through maximising air pressure resulted by turbocharger, in parallel with air temperature decrease which enters into combustion chamber as well as increasing air gravity. This higher compressed air will maximise combustion process.

The working mechanism of turbocharger is basically to utilize efficiency of exhaust gas of engine, the kinetics and thermal energies of exhaust gas of engine are used to rotate the turbine of turbocharger. The construction of turbocharger consists of turbines and compressor which connect through shaft connector, it makes the compressor also rotates. In further development, research of turbocharger is focused on improving the resulted power optimization, so that the constant volume of combustion chamber could produce higher power. According to Kech (2014, p..): "a turbocharger compresses the air so that more oxygen flows into the combustion chamber. In this way, more fuel is burned and the power output of the engine increases accordingly".

Based on Kech in 2014, the function of turbocharging has to be suited with the engine power characteristics, however the challenge is that a turbocharger can be set up either for a wide speed range or a high boost pressure.  The operating condition of the dynamic of main engine shows that the load engine works at different characteristic to produce power output. It can be concluded that a turbocharger needs an optimum of working area, exhaust gas in varied of load engine will produce different supply of pressure and supply of temperature to be processed by turbocharger.

Further research is needed for knowing the matched turbocharger that work at certain operating conditions of the dynamic main engine. The surge line and the efficiency are shown in the compressor map graphic, indicating turbocharger working area. Surging occurs at the maximum pressure point while the flow rate capacity reached at the minimum point. The efficiency of turbocharger can be determined also from the compressor map.

Therefore, there is an urgent need of a project research to determine which turbocharger is the most suitable based on the load engine conditions and power output of the main engine that can be increased by turbocharger. The research on MV. Meratus Palembang examines the existing turbocharger characteristic and another type of turbocharger, to know how optimal every selected turbocharger compared to the operating conditions of dynamic main engine. The main goal of such research is to achieve an optimum of engine performance on MV. Meratus Palembang.

## 1.2 Problem Formulation and Scope

The problem of operating main engine at dynamic conditions is that air pressure supply and air temperature supply produced by turbocharger will sufficiently support main engine at some point only. The existing turbocharger and the selected turbocharger are compared to know each characteristic to support main engine for achieving optimum combustion. Based on the description above the statement problems for this thesis are presented as follows:

1. How are the matched points of the turbocharger with engine in every type of turbocharger?
2. How is the influence of pressure ratio to the volumetric flow rate that supply the main engine?

## 1.3 Problem limitation

1. The project research is solved by calculation based on Grinevtsky Mazing Method in Marine Internal Combustion Engine Book written by N. Petrovsky.
2. The method is to determine the value of volumetric flow rate and the power output produced by turbocharged.
3. Performance of turbocharger is analysed based on exhaust gas constant value.
4. The mechanical efficiency of bearing friction of turbocharger is ignored.
5. Economical factor is ignored in due to the research is devoted to the value of turbocharger efficiency.

## 1.4 Objectives

1. To obtain and to compare the efficiency of each type turbocharger against the main egine from the compressor map graphic.
2. To obtain the air mass density value against the variation of turbocharger pressure.

## 1.5 Benefits

The benefits of the research are the following: (1) to provide a comparisson of exisitng turbocharger and selected type of turbocharger, (2) to provide recommendations in the form of efficiency values in each turbocharger and (3) to show the influence of dynamic of operating conditions of the main engine to the power output and turbocharger in MV. Meratus Palembang.

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## CHAPTER II
## LITERATURE STUDY

Compressed engine cycle or well known as diesel engine is device that generates power, the basic concept using a high compression to adding a pressure in combustion chamber. With high compression, air fuel mixture burnt automatically without igniter. Internal combustion engine generate power from combustion process, air and fuel mixture in chamber are compressed until its ignite than forcing the piston to rotating. The process happens repeatedly to generate energy to make a mechanical movement of working fly wheel. Diesel is one of internal combustion engine example, found by Dr. Rudolf Diesel, the high efficiency of diesel engine is usually use in large vehicles and industrial systems where the improvements in cycle efficiency make it advantageous over the more compact and lighter weight spark ignition engine. Introducing turbocharger as a tool to help internal combustion engine to achieved higher efficiency, the tool is a simple centrifugal compressor for supplying more pressured air to make the compression ratio in combustion chamber increase. As many researches approve the turbocharged engine have a high efficiency.

### 2.1 State of The Art

The idea of supplying air under pressure to a diesel engine was voiced by Dr. Rudolf Diesel, found an engine with a higher efficiency than a usual engine, it was campaigned at early 1896. Turbocharger itself was introduced by a Swiss engineer, Alfred Buchi, who patented his research "pulse system" in 1925.

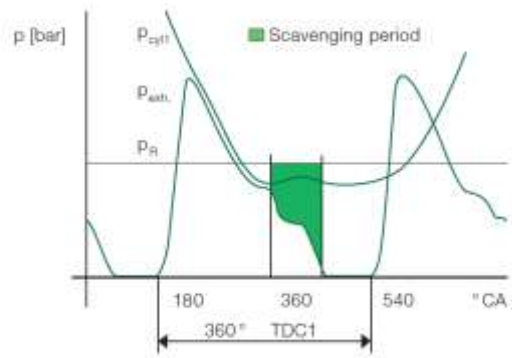


Figure.2. 1 $P_{cyl}$, $P_{exh.}$, $P_r$ and scavenging period
(Turbo Magazine, 1992-1996)

The pulse system is feeds from exhaust gases of the engine through narrow pipes to the turbocharger turbine, thus driving the compressor. The pressure variation in the small volume pipes allows overlapping of the inlet and exhaust, permitting scavenging of the compression space of the engine cylinders with clean air. Cylinders that do not disturb each other's scavenging process can be connected to one pipe (turbine gas inlet) in accordance with the firing order of the engine. The pressure $P_{cyl}$ in the cylinder, $P_{exh.}$ In the exhaust pipe, $P_r$ in the air receiver and the scavenging period (where the inlet and exhaust are simultaneously open is not disturbed by the exhaust pulses of other cylinders. This pulse system was the foundation for the future success of turbocharging.

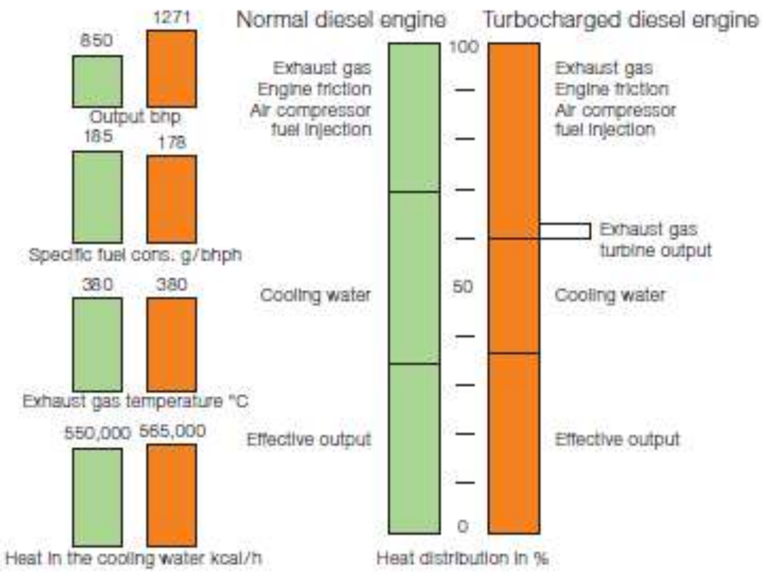


Figure.2. 2Engine test data from the lecture given by Alfred Buchi in 1928
(Turbo Magazine,1992-1996)

In December 1928, Alfred Buchi gave a lecture at the Royal Institute of Engineers at The Hague in the Netherlands. The lecture was told about thermal load of a diesel engine does not essentially increase when turbocharged. An improvement in fuel consumption due to the better mechanical efficiency could also be shown. And it was seen from a comparison of four turbocharged 10 cylinder single acting 4 stroke engines and four turbocharged 6 cylinder double acting 4 stroke engines used for a ship's propulsion installation rated at 36000 bhp that the double acting engines offered several advantages. Neither single acting nor double acting 2 stroke engines could reasonably compete. Thought

was given very early on turbocharging the engine driven scavenging pumps of the 2 stroke engine. However, since the turbocharger efficiency necessary for this was not available, it took many more years for this goal to be reached. (Turbo Magazine, 1992-1996)
The performance of turbochargers can be defined by the pressure ratio, mass flow rate and efficiency characteristics of the compressor and turbine, plus the mechanical efficiency of the bearing unit. (Diesel Engine Reference Book, 2011)

One of the first studies of this phenomenon is stated by Rautenberg's et al. in 1983. These authors emphasis the heat transfer influence on the turbine power and on the compressor outlet temperature. The increase of this temperature leads to a density decrease, which isn't favorable to the engine volumetric efficiency. The usual isentropic efficiency is wrongly used. It doesn't define the aero dynamical quality of the compression, because it considers the heat transfers between the turbine, the compressor and the surrounding area. Thanks to experimental tests, the authors note a strong dependence between turbine inlet temperature and compressor outlet temperature. The geometrical turbocharger characteristics, mainly the distance between compressor and turbine, appear to influence greatly the heat fluxes.

Concluding from the research of Kech et al at 2014, the performance of an internal combustion engine can be increased by adding turbocharging. A turbocharger compresses the air so that more oxygen flows into the combustion chamber and a vehicle engine is driven dynamically it has to deliver high performance from idling speed right through to maximum revs and the turbocharger characteristics have to be matched to the broad power band. The challenge is that a turbocharger can be set up either for a wide speed range or a high boost pressure.

In 2015 Ravaglioli et al, said that turbocharger rotational speed is a very important quantity to be evaluated on-board, since it provides useful feedback information for optimizing turbocharger control strategies. As an example, many works demonstrate that the knowledge of turbo speed mean value allows optimizing compressor and turbine maps interpolation or extending turbocharger operating range. In addition, this work demonstrates that instantaneous turbo speed (i.e. the knowledge of both mean value and fluctuation) can be used to extract information about power delivered by the turbine.

From above studies turbocharger characteristic is important to consider the revs of its engine, the dynamical operation of engine will show how the turbocharger character. Also pressure air supply and temperature air supply to the combustion chamber is very effecting to the efficiency of turbocharger that support internal combustion engine to achieved more power.

## 2.2 Principal Work of Turbocharger

The development of turbocharger focusing at the improvement of power output and its efficiency. Turbocharger is a device that support internal combustion engine to generate a higher power, because of its pressure is supplying to the combustion process so the compression ratio of main engine becomes denser. Basically turbocharger is a one form of a compressor and turbine that connected each other by a shaft bearing that designed as reduce-less as it can be. Turbocharging technology is a key factor to the future engine manufacturer. This technology is very common for using in the diesel engine, because diesel engine main concept is auto-ignition, with a high compression ratio in the combustion chamber.

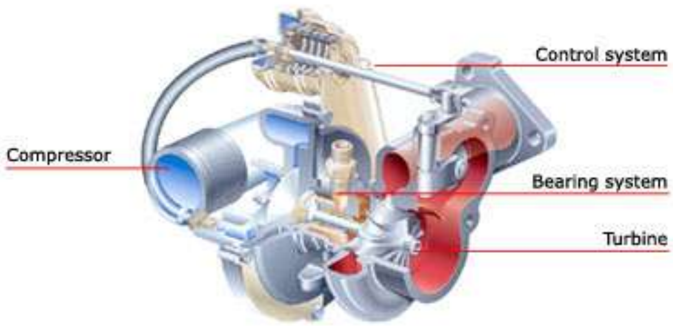


Figure.2. 3 Turbocharger
(http://foro.clubjapo.com/t/los-turbos-guia/28297/2)

Turbochargers system furthermore has many complex structures, as time goes by the form of turbocharger is develop to many type and system. Every turbo-manufacturer has their own structure as a response of the demand on industrial and automotive needs.

The principle work of turbocharger is driven the mass flow of air that entering to the combustion chamber and helping to increase the pressure ratio of combustion chamber. With using exhaust gas after combustion cycle, the exhaust gas is rotating the turbine and continues to drive the compressor. The compressor part work to suck an ambient air around the main engine, and the ambient air is going to the intercooler part, at the intercooler part the ambient air is cooled to achieve denser of air particle. The cooler ambient air temperature the density of air become higher, which mean the air ratio become more compressed, this is in line with the concept of diesel engine that higher compression ratio of a marine internal combustion can produced more efficiency to the combustion process.

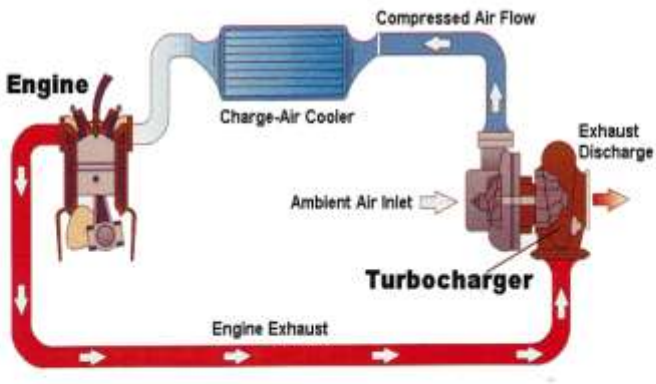


Figure.2. 4 Turbocharger Principle Work
(http://www.mech4study.com/2016/02/difference-between-supercharger-vs-turbocharger.html)

As the structure is connected to each other the turbine power must have a same power in compressor. The power required to rotate the compressor by turbine, is defined as:

$$\begin{aligned} P &= m \cdot \Delta h \\ &= m \cdot Cp.\Delta T \\ &= m \cdot u \cdot va(\tan\sigma_1 - tan\ \sigma_2) \qquad [2.1] \end{aligned}$$

m is the mass density of air (kg/m$^3$), u is the specific heat of air (J/kg-K), va is the speed of air flow mass, and tan σ is the angle of air flow (*Hamid Keshaverzi: 2005*). From the formula above shows that turbine power is correlated with the speed of air flow mass and the compressor power depends in the amount of pressure and air temperature.

Turbocharger principle work is very simple to understand, but it is a complex machinery system to examine. The turbocharger which helps the combustion process of engine must be research carefully to achieve the matched point. The working area of turbocharger (can be seen at turbocharger compressor map) is much related to the engine revolutions as already describe at many references.

## 2.3 Exhaust Gas Method

Exhaust gas from internal combustion engine generally has two type forms:

1. Constant Pressure

This type of exhaust gas is supported by main manifold that connected from each of exhaust gas outlet in every cylinder and directly connected to turbine inlet port. This system enabling every exhaust gas supply same concentration to inlet port of turbine, causes fluctuate air supply to a turbine. The pressure in manifold becomes lower and constant.

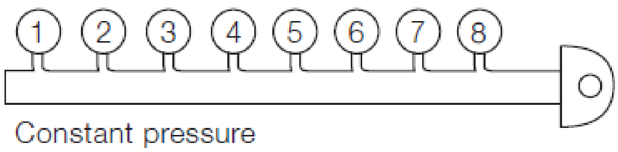


Figure.2. 5 Constant air pressure
(Turbo Magazine,1992-1996)

The constant pressure system has the same large exhaust gas receiver irrespective of how many cylinders the engine has. The same applies to single exhaust systems, where the exhaust receiver volume is reduced in comparison with constant pressure systems. (Turbo Magazine, 1992-1996)

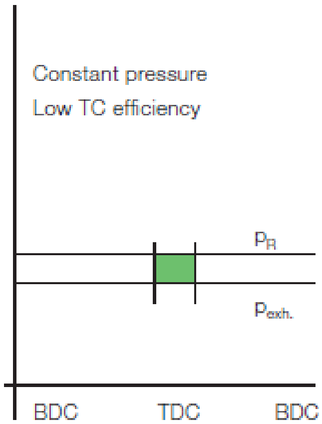


*Figure.2. 6 Turbocharger efficiency for constant pressure effects*
*(Turbo Magazine,1992-1996)*

The Figure of 2.6 shown $P_{exh}$ and $P_r$ effecting the turbocharger efficiency, due to the piston movement from BDC-TDC-BDC the constant air pressure always have a same value point.

2. 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.

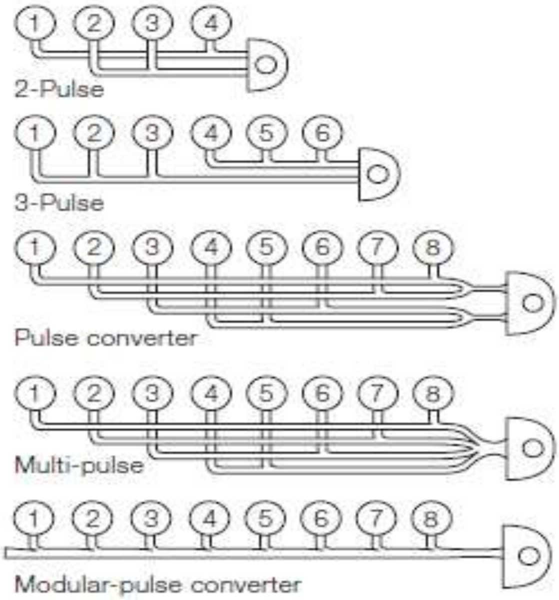


Figure.2. 7 Pulse air pressure
(Turbo Magazine,1992-1996)

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.

Pulse system allows the size of the exhaust pipe that is relatively smaller, 1 to 3 of cylinder space accommodated by exhaust pipes which directly connected to the turbine, so that what makes the pulse system type has a smaller size of exhaust gas pipe.

The pressure that occurs in the manifold tends low, but this this is the one of advantages of this system related to the scavenging process. When the first cylinder in open condition, the exhaust gas pressure in the exhaust pipe will increase, and when the pressure value is greater than the air pressure into the engine can result in improved of turbine performance. (Turbo Magazine, 1992-1996)

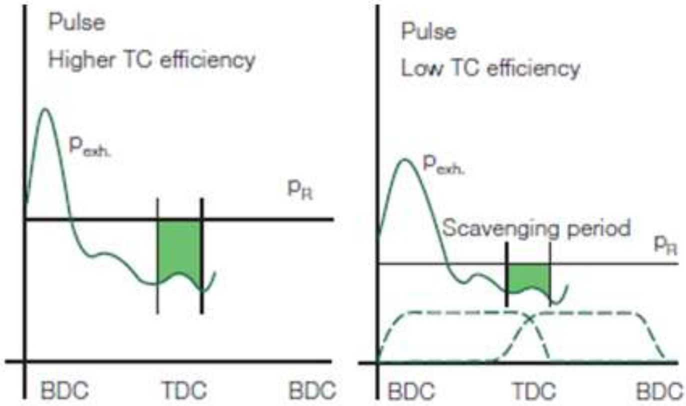


Figure.2. 8 Turbocharger efficiency for pulse pressure effect
(Turbo Magazine,1992-1996)

From the above figure pulse air pressure can support turbocharger with both characteristic, during the piston movement from BDC-TDC-BDC air pressure can supply from each manifold because every manifold its own direct supply to inlet port of turbine.

## 2.4 Compressor

The main parts of the turbocharger compressor are the compressor wheel (inducer and impeller), diffusor, air inlet and air outlet casing. This part is the part to forcing an air supply to the combustion chamber. The characteristic of a compressor is shown in the compressor map graphic; from the compressor we can get the value of turbocharger efficiency, surge line, and operating line.

Compressor surging phenomenon appear at maximum discharge of pressure with minimum volumetric flow rate and at high revolutions of crankshaft engine. Maximum pressure capacity is reached, then pressure in difusser will be greater than pressure at impeller outlet. This will prevent fluid from moving further at impeller outltet and causes the fluid in difusser to flow back. This phenomenon can be affecting to the damage of bearings and other rotating parts, and also making a high vibrations.

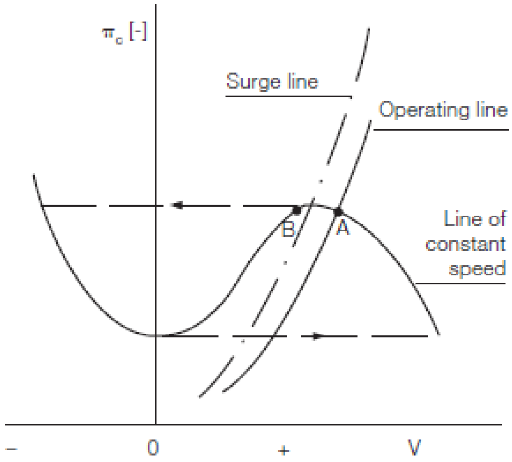


Figure.2. 9 Surging cycle
(Turbo Magazine,1992-1996)

The figure shows the line of constant speed and the operating line of the engine. The intersection of the two lines is working point A if a slight increase in air volume occurs, more pressure is required on the operating line and the pressure becomes lower on the constant speed line. The volume has to decrease again to the point of equlibrium A if, at the same charger speed, a slight reduction in airflow occurs, the pressure will increase although less pressure is required on the working line. Eqilibrium is then once more at point A. The working point A is stable on the part of constant speed line inclined downwards with increasing volume. If a slight decrease in volume occurs at point B (at the same pressure as A), then the pressure on the constant speed line decreases. The compressor cannot maintain the required pressure, the volume continues to decrease and the compressor surge. Point B is not stable on that part of the constat speed line that is inclined upwards (with increasing volume). (Turbo Magazine, 1992-1996)

Bottom lined surging phenomenon can occur when the revolutions of engine is increasing aloft and the volumetric air flow is decreasing aloft. This kind is usually happen when the scavenging air pressure that forcing the compressor to rotate is at the low value point. So far in the modern technology of turbocharging there is a valve to control the air pressure ratio to become normal. So it can occur when the impeller blade of compressor is damaged by time. This is by far every turbo-manufacturer providing a manual book of turbocharger, so the turbocharger can be scheduled for maintenance. The maintenance procedure must be considered with proper, to avoid the failure for maintenance assets.

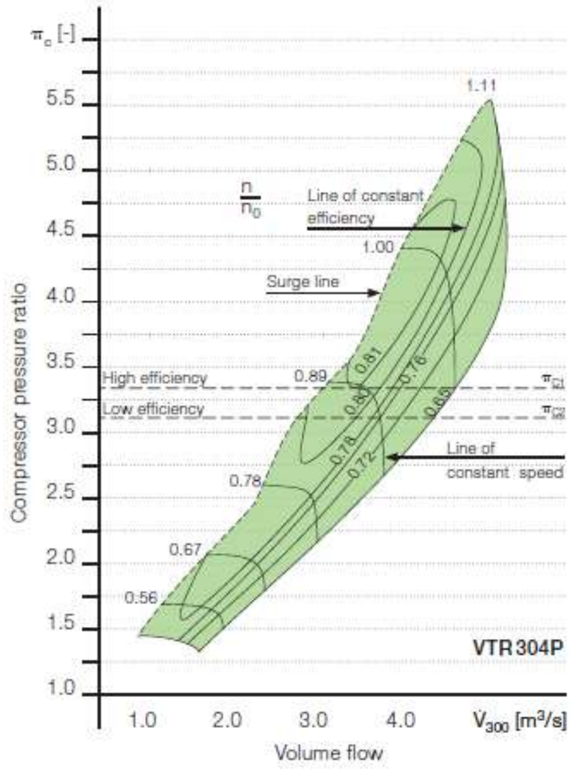


Figure.2. 10 Compressor map
(Turbo Magazine,1992-1996)

The figure shown that from compressor map we can know the characteristic of compressor. Dashed line shown the compressor is working at surge point, when the operating line intersect with that line, it means compressor working at high pressure ratio but low of volumetric air flow, work of the compressor is on sufficiently condition. Line of constant speed that can be seen at the graphic, when the engine revolution is at the constant value point, every volumetric air flow and pressure ratio is at the fluctuate value point. And the efficiency of turbocharger also can be seen at the graphic, the value point of turbocharger efficiency is the intersection between the compressor island and the operating line of engine-turbo condition.

The operating line of compressor can be defined with making a design point of compressor. The benchmark value point to make an operating line is the important thing that must be considered. The plotting of operating line is important to consider, it can be done by looking for the data of pressure ratio and volumetric air flow in each of engine revolutions. It can be assured that the revolutions of engine are affecting the value of operating line design point.

## 2.5 Turbine

The turbocharger turbine, which consists of a turbine wheel and a turbine housing, converts the engine exhaust gas into mechanical energy to drive the compressor. The gas, which is restricted by the turbine's flow cross-sectional area, results in a pressure and temperature drop between the inlet and outlet. This pressure drop is converted by the turbine into kinetic energy to drive the turbine wheel.

In the developement as a main part of turbocharger components, turbine are generally divided at two type of form. Radial turbine type, use for internal combustion at power output 500 kW – 4900 kW. The second once is the axial turbine type, axial turbines are used on medium-slow speed engines. They are perfectly capable of accepting the exhaust gas from engines running on heavy fuel oil. The turbine retains its high effeciency over a very long period of time, the more so when reasonable maintenance is provided. The axial turbine is able to supply an adequate output with good efficiency to drive the compressor from low presure ratios upwards, thus assuring good part-load perfomance of the engine. The latter is specially important for fixed-pitch propeller drives. Turbochargers with axial turbines are found on ships, in diesel power stations and on diesel locomotives, dredger, and other vehicles large diesel engine.

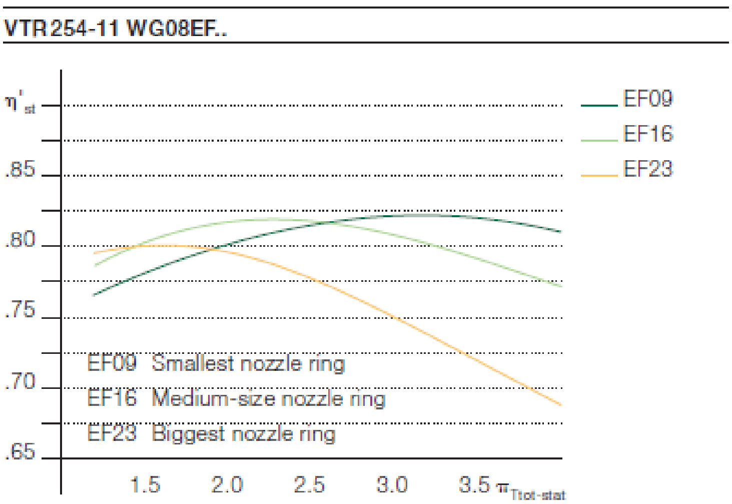


Figure.2. 11 Example of turbine map
(Turbo Magazine,1992-1996)

The turbine's characteristic behavior is determined by the specific flow cross-section, the throat cross-section, in the transition area of the inlet channel to the volute. By reducing this throat cross-section, more exhaust gas is dammed upstream of the turbine and the turbine performance increases as a result of the higher pressure ratio. A smaller flow cross-section therefore results in higher boost pressures.

The turbine's flow cross-sectional area can be easily varied by changing the turbine housing.

Besides the turbine housing flow cross-sectional area, the exit area at the wheel inlet also influences the turbine's mass flow capacity. The machining of a turbine wheel cast contour allows the cross-sectional area and, therefore, the boost pressure, to be adjusted. A contour enlargement results in a larger flow cross-sectional area of the turbine.

Turbines with variable turbine geometry change the flow cross-section between volute channel and wheel inlet. The exit area to the turbine wheel is changed by variable guide vanes or a variable sliding ring covering a part of the cross-section.

In practice, the operating characteristics of exhaust gas turbocharger turbines are described by maps showing the flow parameters plotted against the turbine pressure ratio. The turbine map shows the mass flow curves and the turbine efficiency for various speeds. To simplify the map, the mass flow curves, as well as the efficiency, can be shown by a mean curve.

For high overall turbocharger efficiency, the co-ordination of compressor and turbine wheel diameters is of vital importance. The position of the operating point on the compressor map determines the turbocharger speed. The turbine wheel diameter has to be such that the turbine efficiency is maximized in this operating range.

## 2.6 Grinevetsky Mazing Method

The actual cycle of an internal combustion engine is calculated with a view to determining the basic thermodynamics parameters of the working cycle, as well as the mean indicated pressure and the specific fuel consumption. Grinevetsky (Russia) and later perfected by his follower, Professor E. K. Mazing. (Petrovsky, 1976)

1. Charging Process (Petrovsky, 1976)
   - Calculation for temperature of air at the turbocharger outlet ($T_{sup}$), based on The Marine Internal Combustion Book page. 28;

     $$Tsup = To \times \left(\frac{Psup}{Po}\right)^{\frac{n-1}{n}}, \text{°K} \quad [2.2]$$

     Where :
     $To$ : Ambient temperature, temperature of the outside air in °K.
     $Psup$ : Pressure of air at the turbocharger outlet.
     $Po$ : Absolute pressure, 1 atm.
     $n$ : exponent of the polythropic compression lene of the turbocharger, 1,7-2 for centrifugal blower.
   - Calculation for temperature of air through the intercooler ($T^{i}_{sup}$), based on the Marine Internal Combustion Book page. 203;

     $$T'sup = Tsup - \Delta Tcool, \text{°K} \quad [2.3]$$

     Where :
     $Tsup$ : Temperature of air at the turbocharger outlet.
     $\Delta Tcool$: 60 °C based on engine logbook of MV. Meratus Palembang.

   - Calculation of air temperature in the end process of charging ($T_a$), based on the Marine Internal Combustion Book page. 29;

     $$Ta = T'\text{sup} + \Delta tw + \left(\frac{\gamma r.Tr}{1 + \gamma r}\right) \quad [2.4]$$

     Where :
     $T'sup$ : Temperature of air through the intercooler.
     $\Delta tw$ : With a value at 10-15 °C for Turbocharged Diesel Engine
     $\gamma r$ : Scavenging characteristic value, 0,03-0,04 for four-stroke diesel engine.
     $Tr$ : Rate temperature on diesel engine, 700-800 °C

   - Calculation of air pressure in the end process of charging ($P_a$), based on the Marine Internal Combustion Book page. 27;

     $$Pa = (0{,}90 - 0{,}95)\, Psup, \text{atm} \quad [2.5]$$

2. Compression Process (Petrovsky, 1976)
   - Calculation for compression temperature ($T_c$), Based on the Marine Internal Combustion Book page. 32;

     $Tc = Ta \times \varepsilon^{n1-1}$ , °K [2.6]

     Where :
     $Ta$ : Air temperature in the end process of charging.
     $\varepsilon$ : Compression Ratio.
     $n1$ : Polytropic exponent with value, 1,34-1,39

   - Calculation for Pressure in the end of compression process ($P_c$), based on the Marine Internal Combustion Book page. 32;

     $Pc = Pa \times \varepsilon^{n1}$ [2.7]

     Where :
     $Pa$ : Air pressure in the end process of charging.
     $\varepsilon$ : Compression Ratio.
     $n1$ : Polytropic exponent with value, 1,34-1,39

3. Combustion Process (Petrovsky, 1976)
   - Calculation for increasing of pressure point (λ), Based on the Marine Internal Combustion Book page. 44-45;

     $\lambda = \frac{Pz}{Pc}$ [2.8]

     Where :
     $Pz$ : Pressure at the end of combustion combustion process
     $Pc$ : Pressure at the end of compression process

   - Calculation for actual air value (*Lt*), based on the Marine Internal Combustion Book page. 38;

     $Lt = \alpha \times Vd$ [2.9]

     Where :
     $\alpha$ : Excess air coefficient, 1,7-2 for high-speed diesel
     $Vd$ : Theoritical air value

- The total quantity of moist combustion gases, based on the Marine Internal Combustion Book page. 39;

$Mg = MCO_2 + MH_2O + MN_2 + MO_2$, moles [2.10]

Where :
$CO_2$ : 44.0095 g/mol and 1 gram($CO_2$) = 0,0227 moles.
$H_2O$ : 18.01528 g/mol and 1 gram($H_2O$) = 0,0555 moles.
$N_2$ : 28.0134 g/mol and 1 gram($N_2$) = 0,0357 moles.
$O_2$ : 31.9988 g/mol and 1 gram($O_2$) = 0,0313 moles.

- Coefficient of molar change ($\mu_o$), based on the Marine Internal Combustion Book page. 40;

$\mu o = \frac{Mg}{Lt}$ [2.11]

Where :
$Lt$ : Actual air value
$Mg$ : The total quantity of moist combustion gases.

- Coefficient of molar change for residual gas ($\mu$), based on the Marine Internal Combustion page. 40;

$\mu = \frac{\mu o + \gamma r}{1 + \gamma r}$ [2.12]

Where :
$\mu o$ : Coefficient of molar change.
$\gamma r$ : Scavenging characteristic value, 0,03-0,04 for four-stroke diesel engine.

- Calculation for increasing of pressure point ($\lambda$), Based on the Marine Internal Combustion Book page. 44-45;

$\lambda = \frac{Pz}{Pc}$ [2.13]

- Calculation for combustion temperature ($T_z$), based on the Marine Internal Combustion Book page. 45;

  $Tz = \frac{\lambda}{\mu} \times Tc$ °C [2.14]

  Where :
  $\lambda$ : Calculation for increasing of pressure point.
  $\mu$ : Coefficient of molar change for residual.
  $Tc$ : Calculation for compression temperature.

- Calculation for prelimanary expansion ratio (ρ), based on the Marine Internal Combustion Book page. 50;

  $\rho = \frac{\mu}{\lambda} \times \frac{Tz}{Tc}$ [2.15]

  Where :
  $\mu$ : Coefficient of molar change for residual gas.
  $\lambda$ : Increasing pressure point.
  $Tc$ : Compression temperature.
  $Tz$ : Combustion temperature.

4. Expansion process (Petrovsky, 1976)
   - Calculation for the degree of subsequent expansion (δ), based on the Marine Internal Combustion Book page. 52;

     $\delta = \frac{\varepsilon}{\rho}$ [2.16]

     Where :
     $\varepsilon$ : Compression Ratio.
     $\rho$ : Prelimanary expansion ratio.

## 2.7 Parameter of Power Calculation

According to Mazing and Sineutsky at petrovsky, 1976. In finding the expression for the mean indicated pressure in an engine employing the basic mixed cycle the compression and expansion of the gases are to follow polytropic curves with the mean exponents *n1* and *n2*. The general expression for the theoritical of mean indicated pressure (Pit) has the following form:

$$Pit = \frac{Pc}{\varepsilon - 1}\ \left[(\rho - 1) + \rho\left(1 - \frac{1}{\delta^{n2-1}}\right)\frac{1}{n2-1} - \left(1 - \frac{1}{\varepsilon^{n1-1}}\right)\ \frac{1}{n1-1}\right], \text{kg/cm}^2 \qquad [2.17]$$

Where :
$Pc$ : Pressure at the end of compression process.
$\varepsilon$ : Compression Ratio.
$\mu$ : Coefficient of molar change for residual gas.
$\lambda$ : Increasing pressure point.
$\delta$ : The degree of subsequent expansion.
$n_1$ : Polytropic exponent with value, 1,34-1,39
$n_2$ : Polytropic exponent for expansion, the value is between 1,15-1,3.

To obtain the proper value of the mean indicated pressure the value of (Pit) should be corrected for the rounding off of the sharp angles in the basic indicator diagram which will make its form approach that of the actual indicator diagram. The corrected mean indicated pressure of a four-stroke or two-stroke engines has the following form:

$$Pi = \varphi\ \times Pit, \text{kg/cm}^2 \qquad [2.18]$$

Where :
$Pit$ : The theoritical of mean indicated pressure value.
$\varphi$ : Is the correction factor of a diagram for four-stroke engine (0,95-0,97).
Mean effective pressure occurs during the mechanical efficiency of shaft and main engine. Mean effective pressure ($P_e$) is a value when $P_{it}$ and $P_i$ already been calculated. We can obtain the mean effective pressure from:

$$Pe = Pi\ \times\ \eta m, \text{kg/cm}^2 \qquad [2.19]$$

Where :
$Pi$ : The corrected mean indicated pressure.
$\eta m$ : Mechanical efficiency for turbocharged four-stroke engine (0,8-0,88).

Brake horse power is the value of actual engine power which giving an energy to the engine’s drive shaft. The value can also be found by measuring with a dynamo meter and break the power. BHP in marine sector is a value that marine diesel engine can be produced, the BHP performance of marine diesel engines have a different characteristic for each type from many engine-manufacturer.

$$Nb = \frac{Pe \times Vd \times n \times i}{0{,}45 \times z}, \text{ HP} \qquad [2.20]$$

Where :

$Pe$ : Mean effective pressure,

$Vd$ : Volume displacement, $m^3$

$n$ : Speed of crankshaft, rpm

$i$ : Number number of cylinders

$z$ : For a four-stroke engine (2)

# CHAPTER III
# RESEARCH METHODOLOGY

The research methodology is the main framework of conducting research. The methodology covers all process of research for addressing the research problem, containing data collection, data processing, and results of analysis. The research methodology is presented in the form of a flow chart which composes of research steps from the beginning to the completion of the research. The flow charts are to show the research steps as well as to convince the readers that the research is academically accountable, therefore from the methodology, the research quality could be judged.

## 3.1 Literature Study

Literature study is the initial process to provide theoretical foundation of the concerned research. Through reviewing references which vary from books, published papers, websites and other sources, the research is sufficiently supported by conceptual foundation based on previous studies of the relevant field to the research topic. In addition through a literature study, the research gap which leads to further new research can be identified.

## 3.2 Data Collection

In this step, the required data for analysis is collected which consists of engine logbook, engine's specifications, turbocharger specifications of every selected type to be used for the projection of operating line within the compressor map.

Table. 3. 1 ABB VTR 304 Specification

| | |
|---|---|
| Manufacturer | ABB |
| Type | VTR-304 |
| RPM | 22500-2700 |
| Pressure Ratio | ¾ |
| Flowrate (m^3/s) | 1,95-4,00/2,80-5,20 |
| Overall Dimensions (mm): | |
| Length | 1144 |
| Wide | 768 |
| High | 189 |
| Weight (kg) | 565 |

The table of 3.1 is an example of a turbocharger specification, making operating lines of turbo-engine characteristic is critical important to consider the data from turbocharger specification, we can calculate the operating line from thermodynamics process of engine. The other turbochargers specification can be seen at the attachment.

Table. 3. 2 Daihatsu 8-8dk specifications

| | |
|---|---|
| Merk | Daihatsu |
| Type | 8-8DK |
| Cylinder | 8 |
| Bore | 280 mm |
| Stroke | 390 mm |
| Max. Output | 2500 kW |
| Max. RPM | 750 RPM |
| Ratio Compression | 13 |
| Firing Order | 1-3-2-5-8-6-7-4 |
| Standard Pressure (MPa) : | |
| LO ( Main Bearing) | 0,4-0,5 |
| FO (Engine Inlet) | 0,2-0,3/0,5-0,6 DO/FO |
| Cooling Water ( Engine Inlet) | 0,1-0,2 |
| Standard Temperature (◦C) : | |
| LO (Inlet) | 50-60 |
| Cooling Water ( Inlet) | 65-70 |
| Cooling Water (Outlet) | 70-75 |
| Exhaust Gas | 480 |
| Valve Timing : | |
| Opening Inlet | 50◦BTDC+180◦+35◦ABDC |
| Compression | 180◦+15◦ABDC |
| Pengkabutan | 10◦BTDC+10◦ATDC |
| Exhaust Open | 55◦BBDC+50◦ATDC |

Table 3.2 shown main engine specification, the data to be considered as a basic point to using grinevetsky mazing method. Main engine data is the first step to doing this bachelor thesis. The thermodynamics process in main engine is

connected each other to the turbocharging technology. The more complete specification of main engine can be seen at the attachment.

Table. 3. 3 Engine logbook of MV. Meratus Palembang

| RPM | 600 |
|---|---|
| Pressure (kg/cm^2) : | |
| Cooling Water (Engine Inlet) | 0,1-0,2 Mpa |
| LO ( Main Bearing) | 5,4-5,6 |
| FO ( Engine Inlet) | 6,0-6,5 |
| Temperature (◦C) | |
| Cooling Water (Engine Inlet) | 47-51 |
| Cooling Water (Engine Outlet) | 56-60 |
| LO ( Inlet ) | 53-56 |
| LO ( Outlet ) | 63-65 |
| Exhaust Gas (◦C) : | |
| Cylinder 1 | 375-380 |
| Cylinder 2 | 370-380 |
| Cylinder 3 | 370-380 |
| Cylinder 4 | 370-380 |
| Cylinder 5 | 370-375 |
| Cylinder 6 | 375-385 |
| Cylinder 7 | 365-380 |
| Cylinder 8 | 370-380 |

Engine logbook in table 3.3 is used as a comparison of main engine specification data and engine logbook data. The different value in some data, is to be selected where to be input in grivetsky mazing method to get a good basic design point. The more specific data can also be seen in the attachment.

## 3.3 Main Engine-Turbocharger Calculation

The collected data then calculated by using the *grinevetsky mazing method*. The result of this step is the value of volumetric of air flow rate which supplies the combustion chamber. The value of pressure ratio of a turbocharger and engine is calculated based on the nature of and the characteristics of thermodynamic which connects each other, so that value of operating line is resulted and it is used for projection of turbocharger compressor map.

## 3.4 Determined Turbocharger Efficiency

To determine the efficiency of turbocharger is by reading the compressor map. The turbocharger characteristic is shown at the compressor map which crosscuts the trend line of the value of operating line. Within the compressor map characteristic, there is a part so called as a compressor island, so that the level of efficiency can be determined, by identifying the cutting point in every compressor island.

## 3.5 Turbocharged Main Engine Power Calculation

With the supply of lower air pressure and air temperature due to turbocharger function, the calculation is continued by determining the composition of combustion process, through calculating the mole amount of $CO_2$, $H_2O$, $N_2$, and $O_2$. Those molecules are resulted from the combustion of engine which influences the value of the power output.

## 3.6 Data Validation

The data resulted from the above calculation is not necessarily valid, therefore this step is called as validation process. If the resulted data is not valid, it would be recalculated. If the data is already valid then it can be continued to the next process.

## 3.7 Conclusion and Recommendation

This step is to draw the conclusion and the recommendation of this research. This final process is to comprehensively analyze the result, by exploring the better knowledge of the role of turbocharger to improve the engine performance, so that it can be used as a reference to further study the dynamic of engine operation impacted by varies of turbocharger characteristics.

## 3.8 Flow Chart

The flow chart of the steps of this research is presented in Figure 3.1 as follows.

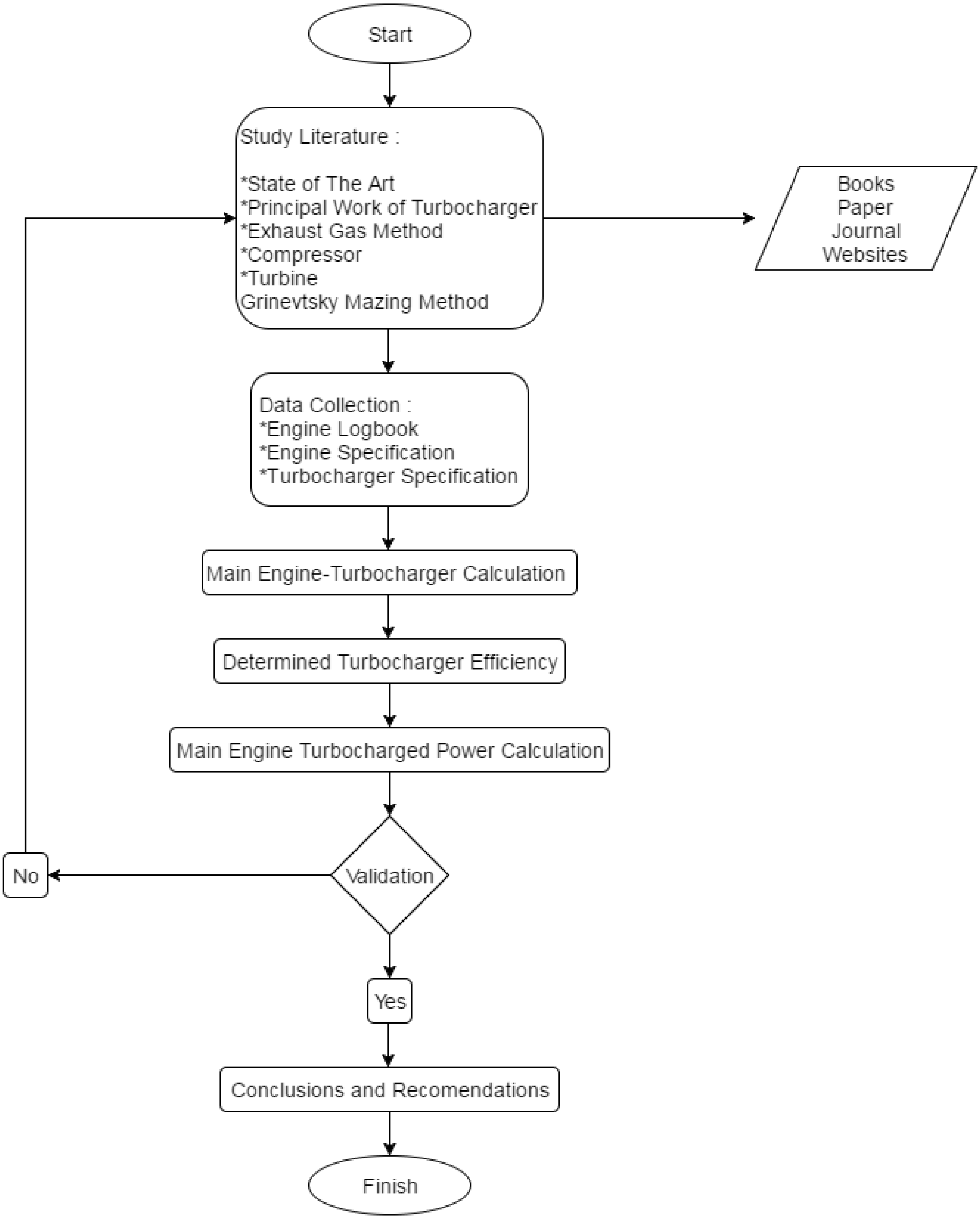


Figure. 3. 1 Flow Chart

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# CHAPTER IV
# Result and Discussion

## 4.1 Basic Data For Calculation

- Ship Particular

  MV. Meratus Palembang is a Multi Purpose Ship equipped for carriage container-heavy cargo. Ship is operated by Meratus Line, the data of ship is collected from Meratus Line as a shipping company.

  1. Name : MCP Altona
  2. Type : Container&Heavy Cargo
  3. LOA : 117 m
  4. LWL : 112,23
  5. Beam : 19,7 m
  6. Height : 8,5 m
  7. Draught : 6,562 m (tropical draught)
  8. Class : BKI&GL
  9. Built : 9 December 2006,
     China-Shandong Huanghai Shipbuilding Co. Ltd

- Engine Specifications

  MV. Meratus Palembang is a Multi Purpose Ship equipped for carriage container-heavy cargo. Ship is operated by Meratus Line, the data of ship is collected from Meratus Line as a shipping company.

  1. Manufacturer : Daihatsu
  2. Type : 8-8dk
  3. Total Cylinder : 8
  4. Bore : 280 mm
  5. Stroke : 390 mm
  6. Max. output : 2500 kW
  7. Max. RPM : 750 RPM
  8. Max. Cylinder pressure : 155 Bar
  9. Compression Ratio : 13,3
  10. Firing Order : 1-4-7-6-8-5-2-3

- Propeller Particular

  Propeller particular contains main data of turbocharger. The data is used for making a base point of engine propeller characteristic. Furthermore data of propeller particular will showing the effect to engine load, due to maintaining the ship design speed.

Table 4. 1 PropellerParticular

| | |
|---|---|
| Type | B5-60 |
| Db | 3,3 m |
| P/D | 0,76 |
| N | 189,8 rpm |
| Ae/A0 | 0,6 |
| Z | 5 |

- Turbocharger Specifications
  Turbocharger specifications contain the value of pressure ratio and volumetric flow rate, compressor map from manufacturer is a result from those two data.

Table 4. 2 ABB 304 Turbocharger specifications

| | |
|---|---|
| Merk | ABB |
| Type | VTR-304 |
| RPM | 22500-2700 |
| Pressure Ratio | 3/4 |
| Flow rate (m^3/s) | 1,95-4,00/2,80-5,20 |
| Overall Dimensions (mm): | |
| Length | 1144 |
| Wide | 768 |
| High | 189 |
| Weight (kg) | 565 |

*Table 4. 3 ABB 354 Turbocharger Specifications*

| | |
|---|---|
| Merk | ABB |
| Type | VTR-354 |
| RPM | 189000-228000 |
| Pressure Ratio | 3/4 |
| Flowrate (m^3/s) | 3,00-6,30/4,20-7,80 |
| Overall Dimensions (mm): | |
| Length | 1627 |
| Wide | 1159 |
| High | 1134 |
| Weight (kg) | 1680 |

Table 4. 4 ABB 321 Turbochaarger specificatons

| | |
|---|---|
| Merk | ABB |
| Type | VTR-321 |
| RPM | 27900-32100 |
| Pressure Ratio | 3/4 |
| Flowrate (m^3/s) | 1,7-3,8/2,4-4,2 |
| Overall Dimensions (mm): | |
| Length | 1144 |
| Wide | 768 |
| High | 189 |
| Weight (kg) | 565 |

## 4.2 EPM Calculation

Lewis, Edward V., Principles of Naval Architecture, Volume II Resistance, Propulsion, and Vibration, the Society of Naval Architects and Marine Engineers, NJ, 1988. Holtrop Method;

1. Total Resistance

   Calculation for total resistance ($R_T$), reference from Principal Naval Architecture page. 93;

$$R_T = \frac{1}{2}.\rho.{V_S}^2.S_{tot}.[C_F.(1+k)+C_A]+\frac{R_W}{W}.W \qquad [3.1]$$

2. Reynolds Number

Calculation for reynolds number ($R_n$), based on reynold number formulation;

$$\mathrm{R_n} = \frac{\upsilon \times L}{\gamma} \qquad [3.2]$$

Where :

$\upsilon$ : is the maximum velocity of the object relative to the fluid (m/s).

$L$ : is a characteristic linear dimension, travelled length of the fluid; hydraulic diameter when dealing with river systems (m).

$\gamma$ : 0,9242x10$^{-6}$ for temperature 15°C.

3. Calculation Frictional Coefficient ($C_F$)

For the holtrop method, $C_F$ that is used is from ITTC 1957, reference from Principal Naval Architecture page. 59 with the following formula:

$$C_F = \frac{0{,}075}{(\log Rn - 2)^2} \qquad [3.3]$$

4. Ship Correlation Allowance ($C_A$)

Calculation for Ship Correlation Allowance ($C_A$) for ($T/L_{WL}$>0,04), the formula stated on Principal Naval Architecture page. 93;

$$C_A = 0{,}006\ (L_{WL} + 100)^{-0{,}16} - 0{,}00205 \qquad [3.4]$$

Where :

$T/L_{WL}$ = 0,0585

5. Viscous Resistance Coefficient($C_V$)

Calculation for Viscous Resistance Coefficient($C_V$), reference from Principal Naval Architecture page. 162;

$$C_V = (1 + k)C_F + C_A \qquad [3.5]$$

Where :

$(1 + k)$: from the ship particular data with further consideration of calculation the value is 5,075.

$C_F$ : Frictional coefficient.

$C_A$ : Ship correlation allowance.

6. Wake Fraction Coefficient (*w*)

Calculation for Wake Fraction Coefficient (*w*), reference from Principal Naval Architecture page. 163;

$$w = 0{,}3095\ C_B + \ 10\ C_V\ C_B - 0{,}23\ D/\sqrt{B\ T} \qquad [3.6]$$

Where :

$C_B$ : Coefficient Block of Ships

$\mathrm{C_V}$ : Viscous resistance coefficient

$\mathrm{D}$ : Diameter of propeller

$\mathrm{B}$ : Length of ship i the midship position

$\mathrm{T}$ : Draft of ship in the maximum load

7. Froude Number ($F_n$)

Calculation for reynolds number ($R_n$), based on reynold number formulation;

$$\mathrm{F_n} = \frac{v}{\sqrt{g\ L}} \qquad [3.7]$$

Where :

$v$ : is the maximum velocity of the object relative to the fluid (m/s).

$L$ : is a characteristic linear dimension, travelled length of the fluid; hydraulic diameter when dealing with river systems (m).

$g$ : 9,81 m/s$^2$ for the mean value of gravity on earth.

8. $M_1$ and $M_2$

Calculation for $M_1$ and $M_2$ coefficient, reference from Principal Naval Architecture page. 92;

m1 = 0,01404(L/T)-1,7525($\nabla^{1/3}$/L)-4,7932(B/L)-C5 [3.8]

m2 = C6 X 0,4 e$^{-0{,}034}$ X (Fn^-3,29) [3.9]

Where :

$C5$ : for CP<0,8 the value is 1,172042.

$C6$ :for $L^3/\nabla \leq 512$ the value is -1,69385.

9. Rw/W

The calculation for Rw/W is formulated in from Principal Naval Architecture page. 92-93;

$$\frac{R_W}{W} = C_1 C_2 C_3 e^{m1xFn^d + m2\cos(\lambda\,\mathrm{Fn}^\wedge -2)} \quad [3.10]$$

Where :

$C_{1,2,3}$ : The value is 1

$d$ : The value is -0,9 for Fn≤0,4

λ : coefficient related to L/B ratio, for L/B<12, so λ as following is (1,446$C_p$ – 0,03 $L$/B)

10. Displacement Weight (W)

The calculation for Rw/W is formulated in Principal Naval Architecture page. 65;

$$W = \rho . g . \nabla \quad [3.11]$$

Where :

$\rho$ : The value for sea water is 1,025 kg/$m^3$.

$g$ : 9,81 m/$s^2$ for the mean value of gravity on earth.

∇ : Displacemet of ship.

11. Coefficient of Resistance ($C_T$)

The calculation for Coefficient of Resistance ($C_T$) is formulated in Reference Principal Naval Architecture page. 65;

$$C_T = \frac{Rt}{0,5 \times \rho \times S \times V^2} \quad [3.12]$$

Where :

$S$ : Wetted hull surface.

$\rho$ : The value for sea water is 1,025 kg/ $m^3$.

V : Velocity of ship.

12. Deadrise at mid-chne length (β)

The calculation for Coefficient of Resistance (β) is formulated in Reference Principal Naval Architecture page. 65;

$$\beta = \frac{0{,}5\times C_T \times S}{(1-t)\times(1-w)^2 \times D^2} \qquad [3.13]$$

Where :

$C_T$ : Coefficient of Resistance.

$t$ : Trhust-deduction fraction.

$w$ : Wake fraction.

$D$ : Diameter of ship.

Table 4. 5 KQ, KT, J in various speed design

| V (Knots) | KT | KQ | J |
|---|---|---|---|
| 6,9 | 0,15482 | 0,02307 | 0,341 |
| 7,6 | 0,16546 | 0,02449 | 0,355 |
| 8,3 | 0,17466 | 0,02584 | 0,367 |
| 9 | 0,17942 | 0,02637 | 0,374 |
| 9,7 | 0,19526 | 0,02702 | 0,392 |
| **10,4** | **0,20682** | **0,02891** | **0,405** |
| 11,1 | 0,22213 | 0,02901 | 0,421 |

From this parameter we can find the value of KT, KQ, J. The value of KT will affecting the Resistance of ship in every speed. KQ value will obtaining the power output that need by ship to moved from its resistance. J is the value of advance coefficients that determining the value of propeller revolutions. Every value can be difined from the open water graphic.

As table 4.5 shown that the data KQ, KT, J is calculated based on the holtrop method. Every value depend on its (Vs), the slower the speed the KQ is linear according the data. Its prove, at the slow speed engine running the power output of engine that need to thrust the speed resistance become less than the higher speed. This characteristic summarized on the open water diagram. Every type of propeller and ratio of pitch/diameter have their own characteristic, so in every type of propeller the open water diagram is differences.

Table 4. 6 Parameter of engine-propeller Characteristic

| V (Knots) | Nprop (m/s) | Q | DHP (kW) | SHP (kW) | BHPscr (kW) | BHP (%) | Revs (%) |
|---|---|---|---|---|---|---|---|
| 6,9 | 2,39 | 52,90 | 795,08 | 811,31 | 836,40 | 33,46 | 75,58 |
| 7,6 | 2,53 | 63,00 | 1002,90 | 1023,37 | 1055,02 | 42,20 | 80,06 |
| 8,3 | 2,68 | 74,33 | 1251,20 | 1276,73 | 1316,22 | 52,65 | 84,65 |
| 9 | 2,85 | 86,04 | 1542,33 | 1573,81 | 1622,48 | 64,90 | 90,16 |
| 9,7 | 2,93 | 93,37 | 1722,47 | 1757,62 | 1811,98 | 72,48 | 92,78 |
| **10,4** | **3,05** | **107,74** | **2064,23** | **2106,36** | **2171,51** | **86,86** | **96,36** |
| 11,1 | 3,13 | 114,13 | 2246,66 | 2292,51 | 2363,41 | 94,54 | 99,00 |

Table 4.6 showing every parameter to make an engine-propeller characteristic. The value KQ from the open water diagram B5-76 propeller type in ship design speed (Vs=10,4 knots) is 0,02891. From that value we can calculate the Torque power ($Q = KQ \times \rho \times n^2 \times D^5$), the Delivered Horse Power ($DHP = 2 \times \pi \times Q \times n$), the power in main shaft ($SHP = DHP/\eta_s$), and the value of Brake Horse Power ($BHP = SHP/\eta_G$). The percent of BHP and revolutions indicated the engine working at some range of power output load and the speed flywheel also at some range in percent units.

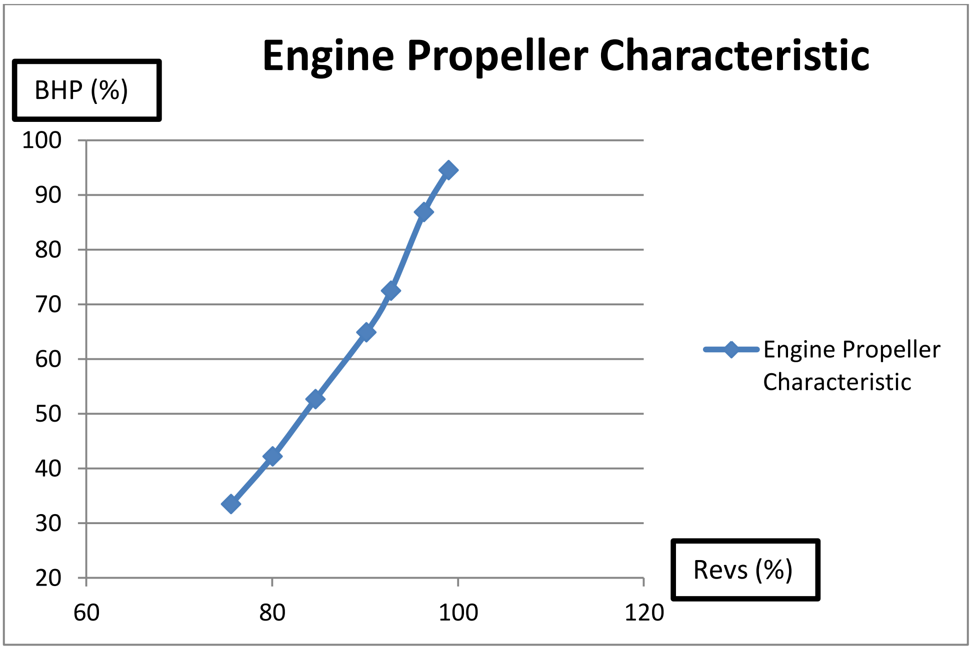


Figure. 4. 1 Graphic of engine-propeller characteristic

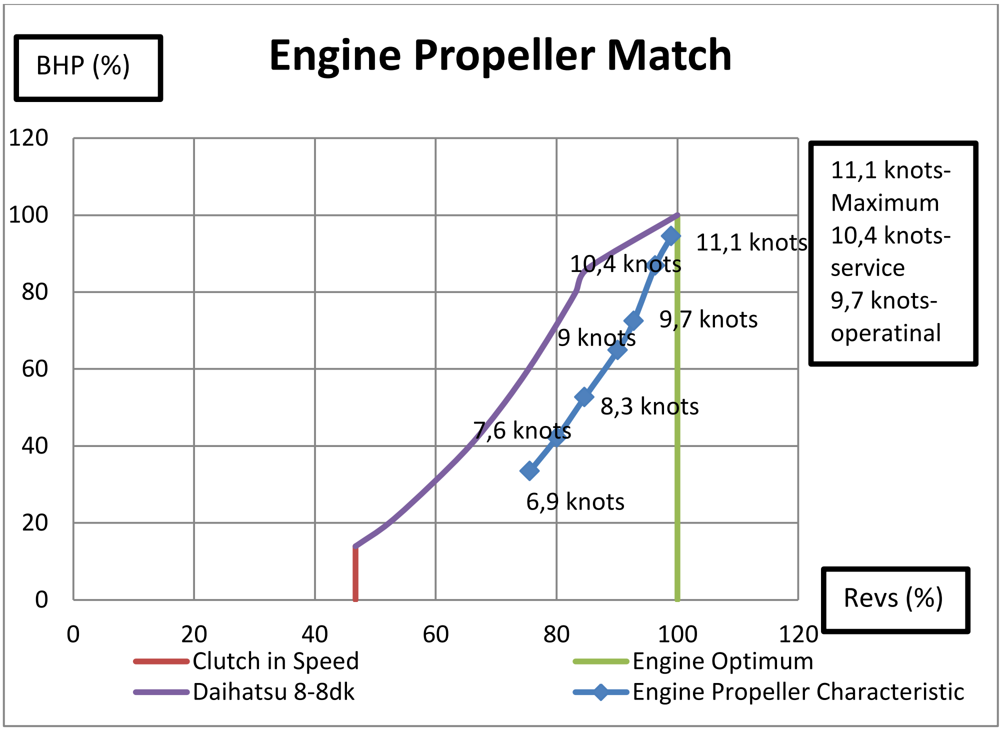


Figure. 4. 2 Graphic of engine-propeller match

Figure 4.2 is the graphic of engine propeller match, envelope line showing Working Load of Daihatsu 8-8dk Operational Characteristic and Revolutions of Daihatsu 8-8dk. The line of engine-propeller is plotted to the envelope line to sowing the trend line of engine propeller match. If the line of engine-propeller is throughout the envelope line it is indicated that the propeller is not necesarry applicable to the engine load operational characteristic.

The calculation of engine-propeller characteristic must be done to know the load of main engine in various design speed. This one is critically consider to know the effect to the turbocharger in various speed (dynamic main engine operating conditions). So further study about selection of turbochargers on dynamic main engine conditions can be continue.

### 4.3 Grinevetsky Mazing Method Calculation Example :

1. Charging Process

- Calculation for temperature of air at the turbocharger outlet ($T_{sup}$), based on The Marine Internal Combustion Book page. 28;

$$Tsup = To \times \left(\frac{Psup}{Po}\right)^{\frac{n-1}{n}}, \text{°K} \quad [3.14]$$

$$Tsup = 318 \times \left(\frac{3}{1}\right)^{\frac{1,7-1}{1,7}} = 499,91 \text{ °K}$$

Where :

$To$ : Ambient temperature, temperature of the outside air in °K.

$Psup$ : Pressure of air at the turbocharger outlet.

$Po$ : Absolute pressure, 1 atm.

$n$ : exponent of the polythropic compression turbocharger, 1,7-2 for centrifugal blower.

- Calculation for temperature of air through the intercooler ($T^{i}_{sup}$), based on the Marine Internal Combustion Book page. 203;

$$T'\,sup = Tsup - \Delta Tcool, \text{°K} \quad [3.15]$$

$$T'\,sup = 519,92 - 60 = 439,91, \text{°K}$$

Where :

$Tsup$ : Temperature of air at the turbocharger outlet.

$\Delta Tcool$ : 60 °C based on engine logbook of MV. Meratus Palembang.

- Calculation of air temperature in the end process of charging ($T_a$), based on the Marine Internal Combustion Book page. 29;

$$Ta = T'\,\text{sup} + \Delta tw + \left(\frac{\gamma r.Tr}{1+\gamma r}\right) \quad [3.16]$$

$$Ta = 439,91 + \left(\frac{0,03 \times 700}{1+0,03}\right) = 462,05, \text{°K}$$

Where :

$T'\,sup$ : Temperature of air through the intercooler.

$\Delta tw$ : With a value at 10-15 °C for Turbocharged Diesel Engine

$\gamma r$ : Scavenging characteristic value, 0,03-0,04 for four-stroke diesel engine.

$Tr$ : Rate temperature on diesel engine, 700-800 °C

- Calculation of air pressure in the end process of charging ($P_a$), based on the Marine Internal Combustion Book page. 27;

$Pa = (0{,}90 - 0{,}95)\ Psup$, atm [3.17]

$Pa = 0{,}95\ x\ 3 = 2{,}85\ kg/cm^2$

$Pa = 307436{,}91\ pa$

- Calculation for Engine Displacement (Vd), based on the Marine Internal Combustion Book page. 26;

$Vd = \pi \times r^2 \times L$

$Vd = \pi \times 0{,}14^2 \times 0{,}39 = 0{,}024\ m^3$ [3.18]

Where :

R : $\frac{1}{2}$ $Dimensional\ Bore\ Stroke$ (m)

L : Length of Stroke (m)

- Calculation for actual combustion chamber value ($L$), based on the Marine Internal Combustion Book page. 38;

$L = \alpha \times Lo$ [3.19]

$L = 2 \times 0{,}024 = 0{,}0408\ m^3$

Where :

$\alpha$ : Diesel factor coefficient (1,7-2)

$Lo$ : the value can be replaced by engine displacement for calculating the actual air value in chamber room.

- Calculation for Volumetric efficiency (ην), based Basic Diesel Engine Theory : Peter Theory, the equation stated;

$\eta v = \frac{L}{Vd}$ [3.20]

$$\eta v = \frac{0{,}048}{0{,}024} = 2$$

Where :
$L$ : Actual combustion chamber value.
$Vd$ : Engine Displacement.

- Calculation for air density (ρ), based on ideal gas law of Dalton the equition is stated;

$$\rho\ = \frac{Pa}{R \times Ta} \qquad [3.20]$$

$$\rho\ = \frac{279488{,}1}{287{,}1\ \times 462{,}05} = 2{,}1069\ kg/m^3$$

Where :
$Pa$ : Pressure in the end process of charging (pa)
$R$ : Avagadro Number, 287,1 (J/kg)
$Ta$ : Temperature in the end process of charging.

- Calculation for mass flow rate of air in the combustion room (ṁ), based on the Diesel Engine Reference Book: 1984, the equation is stated;

$$\dot{\mathrm{m}} = \left(\frac{N}{2}\right) \times Vd\ \times\ \rho\ \times\ \eta v \qquad [3.21]$$

$$\dot{\mathrm{m}} = \left(\frac{12{,}5}{2}\right) \times 0{,}024\ \times\ 2{,}1069\ \times\ 2 = 0{,}6575,\ m^3/s$$

Where :

$N$ : Rotational speed engine (rps).
$Vd$ : Engine Displacement ($m^3$).
$\rho$ : Density of air (kg/$m^3$).
$\eta v$ : Volumetric Efficiency.

- Calculation for Volumetric air flow (*V*), based on the Diesel Engine Reference Book: 1984, it can be calculated by;

$$\dot{V} = \frac{m \times \sqrt{To}}{Psup}, m^3 \qquad [3.22]$$

$$\dot{V} = \frac{0,6575 \times \sqrt{318}}{3} = 3,9085,\ m^3$$

Where :

ṁ : Mass flow rate of air
$To$ : Ambient Temperature (K)
$Psup$ : Boost Pressure (kg/$cm^2$)

- Calculation for Volumetric air flow ($\dot{V}$), it is stated on the compressor map specifications for corrected Volumetric air flow;

$$\dot{V}\,300 = \dot{V} \times \left(\sqrt{\frac{300}{To}}\right) \qquad [3.23]$$

$$V\dot{3}00 = 3,9085 \times \left(\sqrt{\frac{300}{318}}\right) = 3,15\ m^3/s$$

Where :
$V$ : Volumetric flow rate.
$To$ : Temperature Inlet or ambient temperature (K).

2. Compression Process

- Calculation for compression temperature ($T_c$), Based on the Marine Internal Combustion Book page. 32;

$$Tc = Ta \times \varepsilon^{n1-1}\ , °K \qquad [3.24]$$

$$Tc = 469,3 \times 13,3^{\,1,34-1} = 1131,243\ , °K$$

Where :
$Ta$ : Air temperature in the end process of charging.
$\varepsilon$ : Compression Ratio.
$n1$ : Polytropic exponent with value, 1,34-1,39

- Calculation for Pressure in the end of compression process ($P_c$), based on the Marine Internal Combustion Book page. 32;

$Pc = Pa \times \varepsilon^{n1}$, kg/cm$^2$ [3.25]

$Pc = 2{,}95 \times 13{,}3^{1{,}34} = 94{,}72$ , kg/cm$^2$

Where :

$Pa$ : Air pressure in the end process of charging.

$\varepsilon$ : Compression Ratio.

$n1$ : Polytropic exponent with value, 1,34-1,39

3. Combustion Process

- Calculation for air quantity in combustion chamber at moles unit (n), based on ideal gas law stated;

$$n = \frac{Pa \times V}{R \times Ta} \quad [3.26]$$

$$n = \frac{2{,}95 \times 0{,}048}{287{,}1 \times 469{,}3} = 0{,}0517\text{, moles}$$

Where :

$Pa$ : Air pressure in the end process of charging (pa).

$V$ : Volume of combustion chamber Lt (m$^3$)

$R$ : Avogadro number 287,1 (J/kg K)

$Ta$ : Air temperature in the end process of charging.

- The total quantity of moist combustion gases, based on the Marine Internal Combustion Book page. 39;

$$Mg = MCO_2 + MH_2O + MN_2 + MO_2\text{, moles} \quad [3.27]$$

$$Mg = 0{,}0516 + 0{,}0516 + 0{,}059 + 0{,}0516 = 0{,}214 \text{ moles}$$

Where :

$CO_2$ : 44.0095 g/mol and 1 gram($CO_2$) = 0,0227 moles.

$H_2O$ : 18.01528 g/mol and 1 gram($H_2O$) = 0,0555 moles.

$N_2$ : 28.0134 g/mol and 1 gram($N_2$) = 0,0357 moles.

$O_2$ : 31.9988 g/mol and 1 gram($O_2$) = 0,0313 moles.

- Coefficient of molar change ($\mu_o$), based on the Marine Internal Combustion Book page. 40;

$$\mu o = \frac{Mg}{Lt} \quad [3.28]$$

$$\mu o = \frac{0{,}214}{0{,}048} = 4{,}45$$

Where :
$Lt$ : Actual air value.
$Mg$ : The total quantity of moist combustion gases.

- Coefficient of molar change for residual gas ($\mu$), based on the Marine Internal Combustion page. 40;

$$\mu = \frac{\mu o + \gamma r}{1 + \gamma r} \quad [3.29]$$

$$\mu = \frac{4{,}45 \ + \ 0{,}03}{1 + \ 0{,}03} = 4{,}35$$

Where :
$\mu o$ : Coefficient of molar change.
$\gamma r$ : Scavenging characteristic value, 0,03-0,04 for four-stroke diesel engine.

- Calculation for increasing of pressure point ($\lambda$), Based on the Marine Internal Combustion Book page. 44-45;

$$\lambda = \frac{Pz}{Pc} \quad [3.30]$$

$$\lambda = \frac{167{,}48}{94{,}72} = 1{,}77$$

Where :
*Pz* : Pressure at the end of combustion process / maximum cylinder pressure (167,48 kg/cm$^2$)
*Pc* : Pressure at the end of compression process

- Calculation for increasing of pressure point (Tz), Based on the Marine Internal Combustion Book page. 45;

$$Tz = \frac{\lambda}{\mu} \times Tc \qquad [3.31]$$

$$Tz = \frac{1,66}{4,35} \times (1131,24 - 273) = 348,59, ℃$$

Where :
$\lambda$ : Calculation for increasing of pressure point.
$\mu$ : Coefficient of molar change for residual.
$Tc$ : Calculation for compression temperature.

- Calculation for prelimanary expansion ratio (ρ), based on the Marine Internal Combustion Book page. 50;

$$\rho = 1$$

4. Expansion Process

- Calculation for the degree of subsequent expansion (δ), based on the Marine Internal Combustion Book page. 52;

$$\delta = \frac{\varepsilon}{\rho} = 13,3 \qquad [3.32]$$

$$\delta = \frac{13,3}{1} = 13,3$$

## 4.4 Example of Power calculation :

According to Mazing and Sineutsky at petrovsky, 1976. In finding the expression for the mean indicated pressure in an engine employing the basic mixed cycle the compression and expansion of the gases are to follow polytropic curves with the mean exponents *n1* and *n2*. The general expression for the theoritical of mean indicated pressure (Pit) has the following form:

$$Pit = \frac{Pc}{\varepsilon - 1}\left[(\rho - 1) + \rho\left(1 - \frac{1}{\delta^{n2-1}}\right)\frac{1}{n2-1} - \left(1 - \frac{1}{\varepsilon^{n1-1}}\right)\frac{1}{n1-1}\right], \text{kg/cm}^2 \quad [3.33]$$

$$Pit = \frac{94{,}72}{13{,}3 - 1}\left[\left(1 - \frac{1}{13{,}3^{\,1{,}15-1}}\right)\frac{1}{1{,}15-1} - \left(1 - \frac{1}{13{,}3^{1{,}34-1}}\right)\frac{1}{1{,}34-1}\right] = 35{,}47, \text{kg/cm}^2$$

Where :
$Pc$ : Pressure at the end of compression process.
$\varepsilon$ : Compression Ratio.
$\mu$ : Coefficient of molar change for residual gas.
$\lambda$ : Increasing pressure point.
$\delta$ : The degree of subsequent expansion.
$n_1$ : Polytropic exponent with value, 1,34-1,39
$n_2$ : Polytropic exponent for expansion, the value is between 1,15-1,3.

To obtain the proper value of the mean indicated pressure the value of (Pit) should be corrected for the rounding off of the sharp angles in the basic indicator diagram which will make its form approach that of the actual indicator diagram. The corrected mean indicated pressure of a four-stroke or two-stroke engines has the following form:

$$Pi = \varphi \times Pit, \text{kg/cm}^2 \quad [3.34]$$

$$Pi = 0{,}95 \times 35{,}47 = 33{,}69, \text{kg/cm}^2$$

Where :
$Pit$: The theoritical of mean indicated pressure value.
$\varphi$ : Is the correction factor of a diagram for four-stroke engine (0,95-0,97).
Mean effective pressure occurs during the mechanical efficiency of shaft and main engine. Mean effective pressure ($P_e$) is a value when $P_{it}$ and $P_i$ already been calculated. We can obtain the mean effective pressure from:

$$Pe = Pi \times \eta m, \text{kg/cm}^2 \quad [3.35]$$

$Pe = 33{,}69 \times 0{,}8 = 26{,}95$, kg/cm$^2$

Where :
$Pi$ : The corrected mean indicated pressure.
$\eta m$: Mechanical efficiency for turbocharged four-stroke engine (0,8-0,88).

Brake horse power is the value of actual engine power which giving an energy to the engine's drive shaft. The value can also be found by measuring with a dynamo meter and break the power. BHP in marine sector is a value that marine diesel engine can be produced, the BHP performance of marine diesel engines have a different characteristic for each type from many engine-manufacturer.

$$Nb = \frac{Pe \times Vd \times n \times i}{0{,}45 \times z}, \text{ HP} \qquad [3.36]$$

$$Nb = \frac{26{,}95 \times 0{,}024 \times 750 \times 8}{0{,}45 \times 2} = 2935{,}378 \text{ , HP}$$

Where :
$Pe$ : Mean effective pressure,
$Vd$ : Volume displacement, m$^3$
$n$ : Speed of crankshaft, rpm
$i$ : Number number of cylinders
$z$ : For a four-stroke engine (2)

### 4.5 Turbocharger VTR 304 :

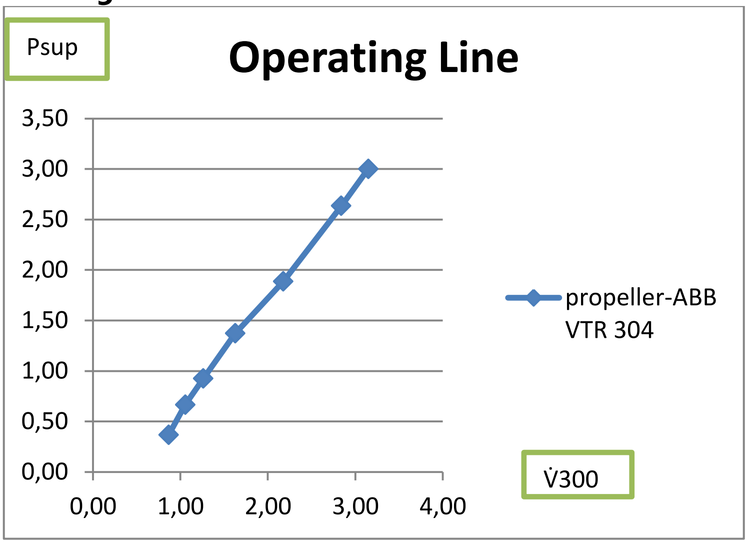


Figure. 4. 3 Graphic operating line ABB VTR 304

Figure 4.3 is the operating line of turbocharger due to the effect on dynamic main engine operating conditions. The ship design speed is affecting the pressure supply in combustion chamber, the volumetric flow rate variance is depends on the pressure supply of air to the combustion chamber.

*Table 4. 7 Turbocharger VTR 304 efficiency on dynamic ship speed conditions*

| No | V (knots) | Load (%) | Psup/Po kg/cm2 | V̇300 m3/s | η (%) |
|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 0,37 | 0,87 | - |
| 2 | 7,6 | 42,20 | 0,66 | 1,05 | - |
| 3 | 8,3 | 52,65 | 0,93 | 1,26 | - |
| 4 | 9 | 64,90 | 1,56 | 1,63 | 65 |
| 5 | 9,7 | 72,48 | 1,89 | 2,18 | 76 |
| **6** | **10,4** | **86,86** | **2,63** | **2,84** | **78** |
| 7 | 11,1 | 94,54 | 3,00 | 3,15 | 80 |

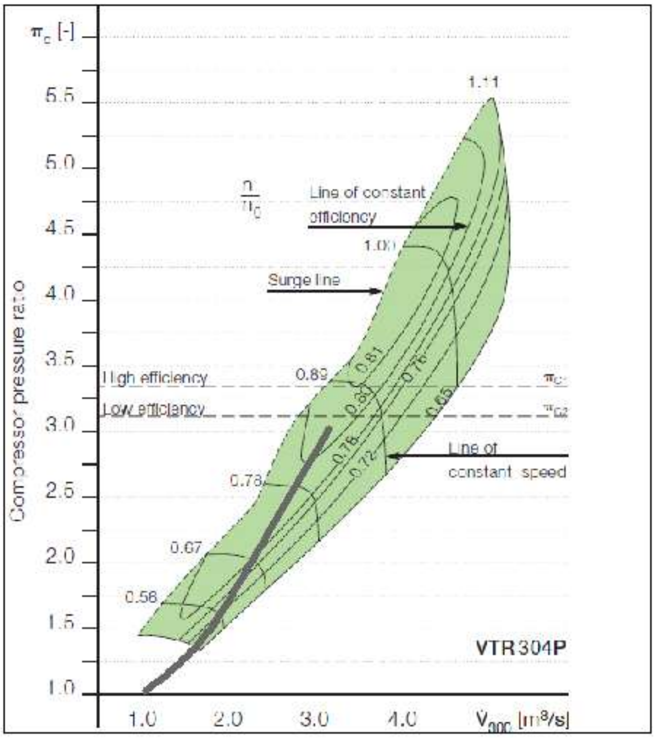


Figure. 4. 4 Compressor map VTR 304

The efficiency of turbocharger can be determined from the figure 4.4, the operating line of engine and turbocharger is plotted to tthe figure 4.4 from the intersection beetween compressor island and operating line it determined the turbocharger efficiency. The grey line is the engine-propeller and turbocharger characteristic. The figure of compressor map is how we see the engine-turbocharger match, the operating line in compressor map indicated the characteristic of turbocharger on main engine operational working load.

*Table 4. 8 Turbocharged Engine Power (VTR 304)*

| No | Psup kg/cm2 | Nb (kW) |
|---|---|---|
| 1 | 0,39 | 62,79034 |
| 2 | 0,85 | 202,6948 |
| 3 | 1,13 | 360,3463 |
| 4 | 1,41 | 713,1853 |
| 5 | 1,98 | 1313,762 |
| 6 | 2,83 | 2627,525 |
| 7 | 3,11 | 3220,595 |

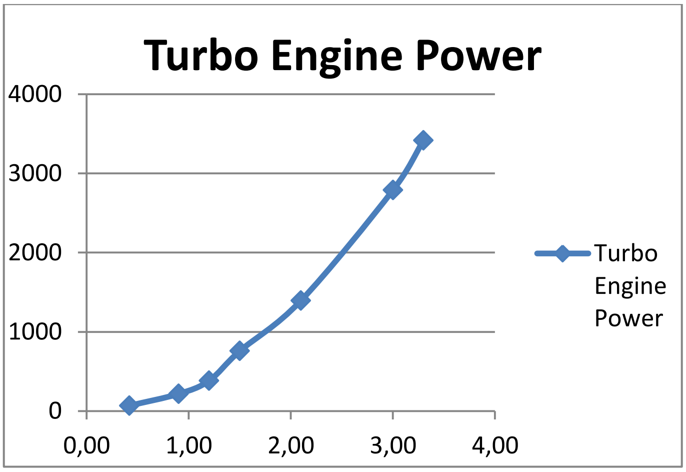


Figure. 4. 5 Turbocharged engine power (VTR 304)

Table 4.8 showing that pressure ratio has the main factor to increasing the value of power output, at 3,11 pressure ratio the power output of the maine engine become 3220,595 kW. This thing Happen because the value of combustion pressure is increasing due to the normal compression ratio engine is about 13,3 pressure and 3,11 point pressure is add by the turbocharger work at maximum load. The trendline of pressure supply and power output(brake horse power) is describe by the figure of 4.4, trendline indicated pressure supply is very affecting to the power output of main engine, the lower pressure supply to the combustion chamber, so engine power output also produced lower power.

## 4.6 Turbocharger VTR 321 :

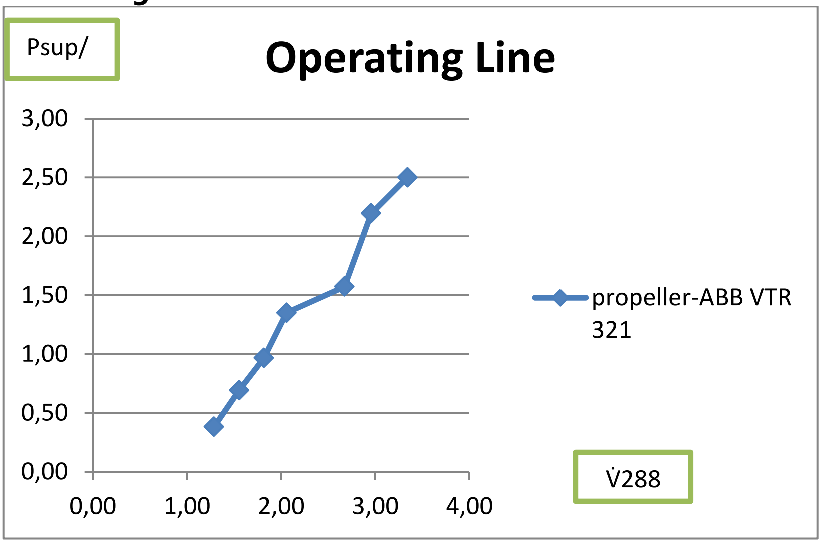


Figure. 4. 6 Graphic operating line VTR 321

Figure 4.6 is the operating line of turbocharger due to the effect on dynamic main engine operating conditions. The ship design speed is affecting the pressure supply in combustion chamber, the volumetric flow rate variance is depends on the pressure supply of air to the combustion chamber.

able 4. 9 Turbocharger VTR 321 efficiency on dynamic ship speed conditions

| No | V (knots) | Load (%) | Psup/Po kg/cm2 | V̇288 m3/s | η (%) |
|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 0,38 | 1,29 | - |
| 2 | 7,6 | 42,20 | 0,69 | 1,55 | - |
| 3 | 8,3 | 52,65 | 0,97 | 1,82 | - |
| 4 | 9 | 64,90 | 1,35 | 2,06 | 68 |
| 5 | 9,7 | 72,48 | 1,57 | 2,67 | 72 |
| **6** | **10,4** | **86,86** | **2,20** | **2,96** | **80** |
| 7 | 11,1 | 94,54 | 2,50 | 3,34 | 78 |

The efficiency of Turbocharger VTR 321 is provided in turbocharger compressor map with volumetric flow rate correction (V̇288), the calculation of corrected volumetric flow rate is ussually provided in compressor. The value of efficiency can be determined from the compressor island that located on the compressor map diagram. The value of efficiency in turbocharger showing how optimal the turbocharger working at some load engine that rely to the ship design speed, especially on dynamic main engine operating conditions.

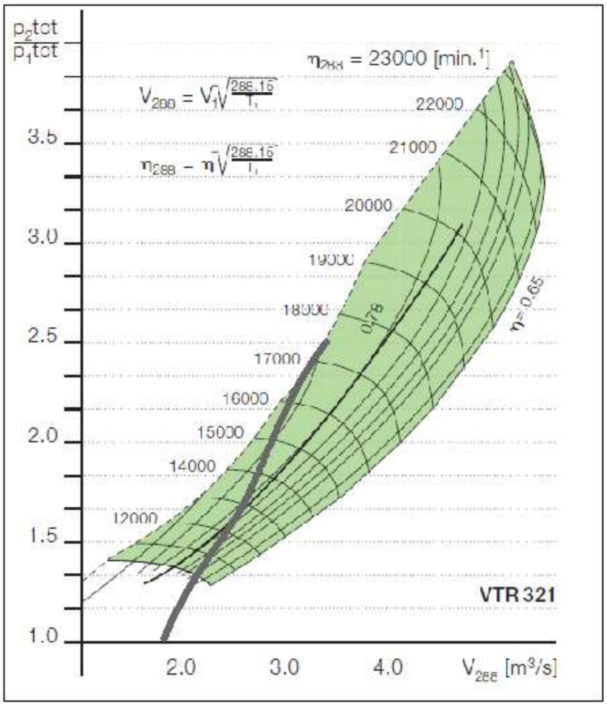


Figure. 4. 7 Compressor map VTR 321

On the compressor map of VTR 321 consist the value of turbocharger efficiency in compressor islands. The efficiency value can be obtained from the intersection between operating line of VTR 321 and the compressor island on the graphic. The efficiency of turbocharger indicated this device is working at some working area of turbocharger, the highest turbocharger efficiency is the one with the nearest point of surging line. The operating line in compressor map also showing turbocharger charateristic to the main engine operating conditions.

Table 4. 10 Turbocharged engine power (VTR 321)

| No | Psup kg/cm2 | Nb (kW) |
|---|---|---|
| 1 | 0,32 | 50,47455 |
| 2 | 0,68 | 162,9379 |
| 3 | 0,91 | 289,6674 |
| 4 | 1,14 | 573,3001 |
| 5 | 1,59 | 1056,079 |
| 6 | 2,27 | 2112,158 |
| 7 | 2,50 | 2588,903 |

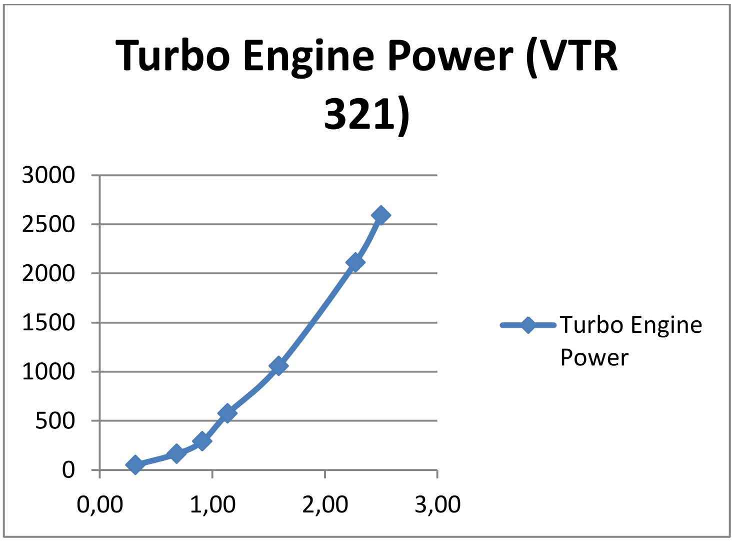


Figure. 4. 8 Turbocharged Engine Power (VTR 321)

Table 4.10 showing that pressure ratio has the main factor to increasing the value of power output, at 2,5 pressure ratio the power output of the maine engine become 2588,903 kW. This condition happen because the value of combustion pressure is increasing due to the normal compression ratio engine is about 13,3 pressure and 2,5 point pressure is add by the turbocharger work at maximum load. The trendline of pressure supply and power output(brake horse power) is describe by the figure of 4.8, trendline indicated pressure supply is very affecting to the power output of main engine, the lower pressure supply to the combustion chamber, so engine power output also produced lower power.

## 4.7 Turbocharger VTR 354 :

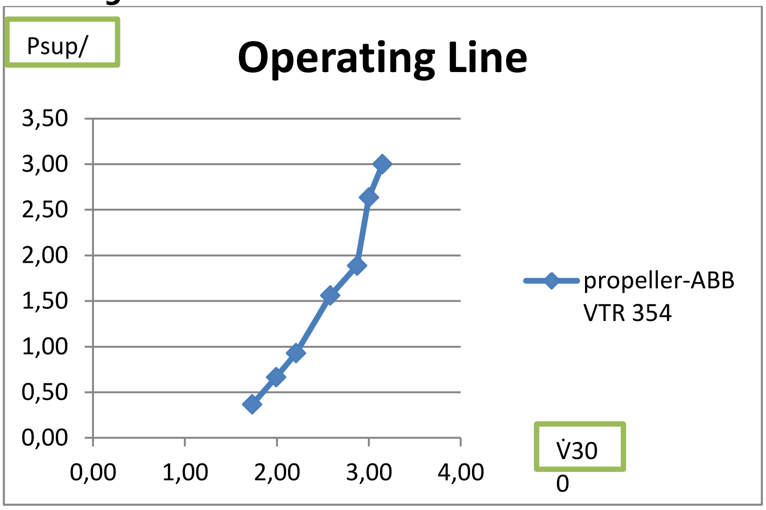


Figure. 4. 9 Graphic operating line ABB VTR 354

Figure 4.9 showing that operating line of turbocharger depend on the pressure ratio and volumetric flow rate produced by main engine operating conditions. Operating line of turbocharger is the characteristic of the turbocharger to main engine operational working conditions. The variance of pressure supply is effect from the dynamic main engine operating conditions.

After all operating line is the basic consideration to knowing the turbocharger-engine match. With operating line turbocharger can be analyzed the characteristic due to its operation of engine causes. Turbocharger characteristics depend to the engine operational condition.

Table 4. 11 Turbocharger VTR 354 efficiency on dynamic ship speed conditions

| No | V (knots) | Load (%) | Psup/Po kg/cm2 | V̇300 m3/s | η (%) |
|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 0,37 | 1,73 | - |
| 2 | 7,6 | 42,20 | 0,66 | 1,99 | - |
| 3 | 8,3 | 52,65 | 0,93 | 2,21 | - |
| 4 | 9 | 64,90 | 1,56 | 2,58 | 78 |
| 5 | 9,7 | 72,48 | 1,89 | 2,87 | 82 |
| **6** | **10,4** | **86,86** | **2,63** | **3,00** | **Surging** |
| 7 | 11,1 | 94,54 | 3,00 | 3,15 | Surging |

The efficiency of Turbocharger VTR 354 is provided in turbocharger compressor map with volumetric flow rate correction ($\dot{V}300$). The value of efficiency can be determined from the compressor island that located on the compressor map diagram. On VTR 354 at the 10,4 knots and 11,1 knots ship speed the efficiency of turbocharger is troughout the surging line, it make the efficiency value on those point are surge.

From the below figure at 4.10, the compressor map of Turbocharger VTR 354 is plotted by its operating line. The line is the engine-propeller and turbocharger characteristic. The figure of compressor map is how we see the engine-turbocharger match, the operating line in compressor map indicated the characteristic of turbocharger on main engine operational working load. With the case on the dynamic main engine operating conditions, the surge point of turbocharger is proven in this type of turbocharger. It also can be conlclude that the pressure ratio is very relying to the speed of main engine

.

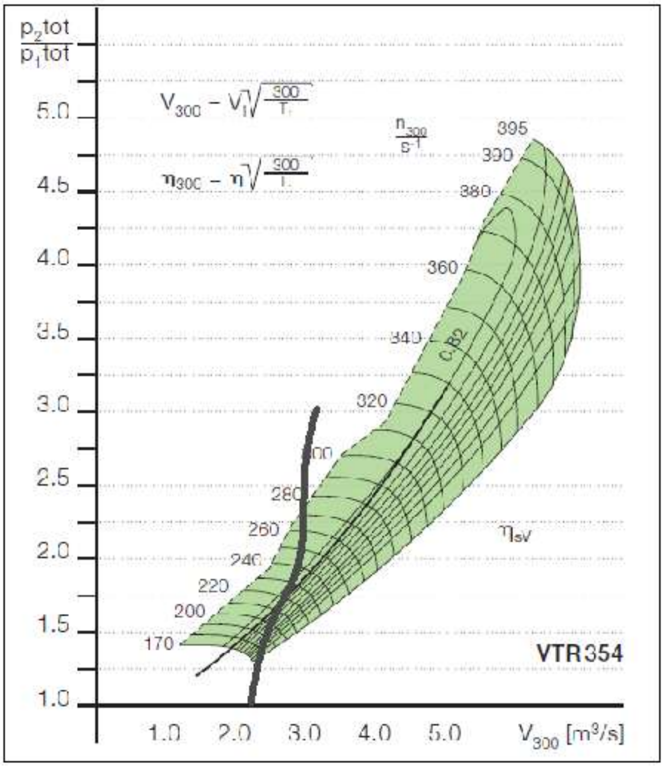


Figure. 4. 10 Compressor map VTR 354

Table 4. 12 Turbocharged engine power (VTR 354)

| No | Psup kg/cm2 | Nb (kW) |
|---|---|---|
| 1 | 0,39 | 62,79034 |
| 2 | 0,85 | 202,6948 |
| 3 | 1,13 | 360,3463 |
| 4 | 1,41 | 713,1853 |
| 5 | 1,98 | 1313,762 |
| 6 | 2,83 | 2627,525 |
| 7 | 3,11 | 3220,595 |

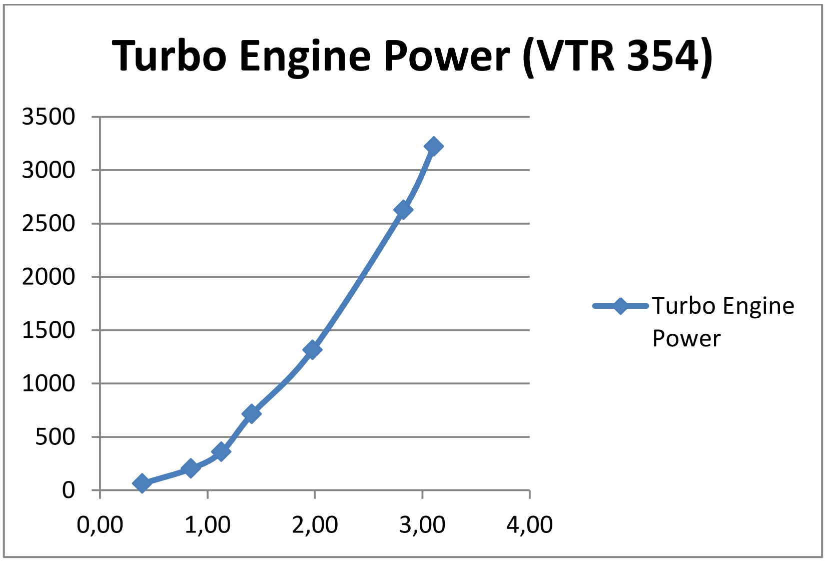


Figure. 4. 11 Turbocharged Engine Power (VTR 354)

The turbocharger of 354 make power output of engine is increasing we can seen from the graphic of 4.11 the trend line of turbocharged engine is become higher the power output tubocharged by VTR 354 is 3220,595 kW with the pressure supply is at 3,11 at the maximum load engine.

### 4.8 Pressure Ratio Against Volumetric Flow Rate

Table 4. 13 present the data which show pressure ratio will affecting the value of volumetric flow rate that supply the main engine in turbocharger VTR 304. In maximum pressure supply (3 $kg/cm^2$), the density of air supply become 2,1 $kg/m^3$ and the corrected volumetric flow rate become 3,15 $m^3/s$.

Table 4. 14 also presenting data of pressure ratio and volumetric flow rate in turbocharger VTR 321, obtained from a parameter of calculation. The density of air supply for the main engine become 1,9 kg/m$^3$ at maximum pressure supply (2,5 kg/cm$^2$), and the corrected volumetric flow rate become 3,34 m$^3$/s.

Table 4. 15 is the result from a parameter calculation, to obtained the value of pressure ratio and volumetric flow rate that supply the main engine in turbocharger VTR 354. The density of air supply at 2,1 kg/m$^3$ it is the same value with VTR 304 because the maximum pressure supply is also at 3 kg/cm$^2$, which is the value of volumetric flow rate is at the same point (3,15 m$^3$/s).

From the entire table of pressure ratio vs volumetric flow rate, density plays a role as a factor that can affecting the value of volumetric flow rate. From the basic knowledge of density is value between the mass of its fluids divided by its volume, and the value of volumetric flow rate is very relying to its volume in every second.  So the higher the value of density the value of volumetric flow rate become lower, the volumetric flow rate itself shows that engine work by given a flow rate of volume in every second depend to its density.

The pressure ratio which affecting the value of volumetric flow rate is depend to many factor that can be seen in the table, density become the main factor because is directly affected the value of volumetric flow rate, and the volumetric flow rate value is very clear to be correlated to the value of pressure ratio, the higher the pressure ratio the value of volumetric become higher also.

Table 4. 13 Pressure ratio vs volumetric flow rate (VTR 304)

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,37 | 0,37 | 318 | 60 | 210,01 | 150,01 | 180,59 | 0,35 | 34013,34 |
| 2 | 7,6 | 42,20 | 1 | 0,66 | 0,66 | 318 | 60 | 268,65 | 208,65 | 237,53 | 0,63 | 61857,34 |
| 3 | 8,3 | 52,65 | 1 | 0,93 | 0,93 | 318 | 60 | 308,02 | 248,02 | 275,75 | 0,88 | 86219,50 |
| 4 | 9 | 64,90 | 1 | 1,29 | 1,37 | 318 | 60 | 353,66 | 293,66 | 320,06 | 1,23 | 120600,14 |
| 5 | 9,7 | 72,48 | 1 | 1,89 | 1,89 | 318 | 60 | 412,94 | 352,94 | 377,61 | 1,79 | 175703,19 |
| 6 | 10,4 | **86,86** | 1 | 2,63 | 2,63 | 318 | 60 | 473,83 | 413,83 | 436,73 | 2,50 | 245396,38 |
| 7 | 11,1 | 94,54 | 1 | 3,00 | 3,00 | 318 | 60 | 499,91 | 439,91 | 462,05 | 2,85 | 279488,10 |

| No. | Vd m3 | Lt m3 | Hv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{300}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0037 | 0,0063 | 1,7 | 0,6560 | 0,0137 | 0,6714 | 0,865285 |
| 2 | 0,0055 | 0,0095 | 1,72 | 0,9071 | 0,0349 | 0,9382 | 1,054348 |
| 3 | 0,0074 | 0,0129 | 1,74 | 1,0891 | 0,0628 | 1,2097 | 1,261786 |
| 4 | 0,0105 | 0,0186 | 1,78 | 1,3125 | 0,1219 | 1,6790 | 1,625533 |
| 5 | 0,0154 | 0,0283 | 1,84 | 1,6207 | 0,2582 | 2,4415 | 2,17618 |
| 6 | 0,0216 | 0,0414 | 1,92 | 1,9571 | 0,5061 | 3,4262 | 2,839631 |
| 7 | 0,0240 | 0,0480 | 2 | 2,1069 | 0,6575 | 3,9085 | 3,14937 |

Table 4. 14 Pressure ratio vs volumetric flow rate (VTR 321)

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,38 | 0,38 | 318 | 60 | 213,67 | 153,67 | 184,15 | 0,36 | 35471,32 |
| 2 | 7,6 | 42,20 | 1 | 0,69 | 0,69 | 318 | 60 | 273,34 | 213,34 | 242,08 | 0,66 | 64508,85 |
| 3 | 8,3 | 52,65 | 1 | 0,97 | 0,97 | 318 | 60 | 313,39 | 253,39 | 280,96 | 0,92 | 89915,28 |
| 4 | 9 | 64,90 | 1 | 1,35 | 1,35 | 318 | 60 | 359,83 | 299,83 | 326,05 | 1,28 | 125769,65 |
| 5 | 9,7 | 72,48 | 1 | 1,57 | 1,57 | 318 | 60 | 383,07 | 323,07 | 348,61 | 1,49 | 146419,33 |
| 6 | 10,4 | **86,86** | 1 | 2,20 | 2,20 | 318 | 60 | 439,56 | 379,56 | 403,46 | 2,09 | 204496,98 |
| 7 | 11,1 | 94,54 | 1 | 2,50 | 2,50 | 318 | 60 | 463,75 | 403,75 | 426,94 | 2,38 | 232906,75 |

| No. | Vd m3 | Lt m3 | ηv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{288}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0031 | 0,0052 | 1,7 | 0,6709 | 0,0220 | 1,0283 | 1,28636 |
| 2 | 0,0055 | 0,0095 | 1,72 | 0,9282 | 0,0553 | 1,4246 | 1,554306 |
| 3 | 0,0080 | 0,0139 | 1,74 | 1,1147 | 0,0971 | 1,7936 | 1,816436 |
| 4 | 0,0111 | 0,0197 | 1,78 | 1,3436 | 0,1657 | 2,1893 | 2,058103 |
| 5 | 0,0154 | 0,0283 | 1,84 | 1,4629 | 0,2591 | 2,9397 | 2,672606 |
| 6 | 0,0203 | 0,0390 | 1,92 | 1,7654 | 0,4307 | 3,4986 | 2,956696 |
| 7 | 0,0240 | 0,0480 | 2 | 1,9001 | 0,5706 | 4,0701 | 3,343742 |

Table 4. 15 Pressure ratio vs volumetric flow rate (VTR 354)

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure Ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,37 | 0,37 | 318 | 60 | 210,01 | 150,01 | 180,59 | 0,35 | 34013,34 |
| 2 | 7,6 | 42,20 | 1 | 0,66 | 0,66 | 318 | 60 | 268,65 | 208,65 | 237,53 | 0,63 | 61857,34 |
| 3 | 8,3 | 52,65 | 1 | 0,93 | 0,93 | 318 | 60 | 308,02 | 248,02 | 275,75 | 0,88 | 86219,50 |
| 4 | 9 | 64,90 | 1 | 1,29 | 1,56 | 318 | 60 | 353,66 | 293,66 | 320,06 | 1,23 | 120600,14 |
| 5 | 9,7 | 72,48 | 1 | 1,89 | 1,89 | 318 | 60 | 412,94 | 352,94 | 377,61 | 1,79 | 175703,19 |
| 6 | 10,4 | **86,86** | 1 | 2,63 | 2,63 | 318 | 60 | 473,83 | 413,83 | 436,73 | 2,50 | 245396,38 |
| 7 | 11,1 | 94,54 | 1 | 3,00 | 3,00 | 318 | 60 | 499,91 | 439,91 | 462,05 | 2,85 | 279488,10 |

| No. | Vd m3 | Lt m3 | ηv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{300}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0074 | 0,0126 | 1,7 | 0,6560 | 0,0275 | 1,3427 | 1,73057 |
| 2 | 0,0105 | 0,0180 | 1,72 | 0,9071 | 0,0660 | 1,7721 | 1,991547 |
| 3 | 0,0129 | 0,0225 | 1,74 | 1,0891 | 0,1099 | 2,1170 | 2,208126 |
| 4 | 0,0166 | 0,0296 | 1,78 | 1,3125 | 0,1936 | 2,6666 | 2,58173 |
| 5 | 0,0203 | 0,0374 | 1,84 | 1,6207 | 0,3408 | 3,2228 | 2,872557 |
| 6 | 0,0228 | 0,0438 | 1,92 | 1,9571 | 0,5350 | 3,6219 | 3,001896 |
| 7 | 0,0240 | 0,0480 | 2 | 2,1069 | 0,6575 | 3,9085 | 3,14937 |

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# CHAPTER V
# Conclussion and Recomendations

## 5.1 Conclusions

1. Engine Turbo Matching

- On VTR 304 at 11,1 knots turbocharger have an efficiency at 80%, at the design speed (Vs) at 10,4 knots turbocharger have an efficiency at 78% and for operation speed 9,7 knots turbocharger have an effeciency at 76%.
- On VTR 321 at 11,1 knots turbocharger have an efficiency at 78 % and at the design speed (Vs) at 10,4 knots turbocharger have an efficiency at 80% and for operation speed 9,7 knots turbocharger have an effeciency at 72%.
- On VTR 354 at 11,1 knots turbocharger have a surge efficiency and at the design speed (Vs) at 10,4 knots turbocharger have a surge efficiency also and for operation speed 9,7 knots turbocharger have an effeciency at 82%.

2. Volumetric flow rate against Pressure Ratio

- The maximum pressure ratio for VTR 304 for the engine-propeller and turbocharger is at 3,0 with volumetric flow rate at 3,15 $m^3/s$, it shown the propeller load is make the volumetric flow rate increasing. This thing happen because at this load of propeller engine given a high load and the turbocharger supporting more energy than the engine normal load.
- The maximum pressure ratio for VTR 321 for the engine-propeller and turbocharger is at 2,5 with volumetric flow rate 3,34, it also the propeller load is make the volumetric flow rate increasing. This thing happen because at this load of propeller engine given a high load and the turbocharger supporting more energy than the engine normal load.
- The maximum pressure ratio for VTR 354 for the engine-propeller and turbocharger is at 3,00 with volumetric flow rate 3,15, it also the propeller load is make the volumetric flow rate increasing. This thing happen because at this load of propeller engine given a high load and the turbocharger supporting more energy than the engine normal load.

So the dynamic main engine operating conditions make the volumetric value increasing because turbocharger helps more.

## 5.2 Recommendations

Recommendations for selection of turbocharger on dynamic main engine operating conditions in MV. Meratus Palembang is VTR 304. Which the criteria are:

1. Turbocharger Efficiency for ship design speed (Vs) at 10,4 knots is 78%, and the various efficiency because operating engine on dynamic conditions have the best range for 304 is about: 65, 76, 78, 80(%).
2. Engine power output is completely fulfilled, at the maximum load of engine the power output become 3220,595 kW.
3. The operating line have the best range than VTR 321 and 354, the compressor island showing turbocharger effeciennyc, it shown in the compressor map the highest effeciency of turbocharger is the nearest to the surge line.

The further recommendation is to analyzing the other type of turbocharger from different turbo-manufacturer or can be replaced by VGT type. The characteristic of other type of turbocharger can be better than the selected turbocharger in this bachelor thesis. VGT type is a unique turbocharger which the pressure ratio and volumetric flow rate can be control by advance technology, so turbocharger can support main engine in high performance by controlable valve.

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http://ucship.com

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## Attachment

## Main Engine Specifications

| Model | | 6DK-28 | 8DK-28 |
|---|---|---|---|
| Type | | Vertical water-cooling direct injection type 4-cycle diesel engine | |
| Number of cylinders | | 6 | 8 |
| Cylinders bore | mm | 280 | |
| Piston stroke | mm | 390 | |
| Engine speed | $min^{-1}$ | * | |
| Output | kW | * | |
| | {PS} | * | |
| Ignition sequence | | 1-2-4-6-5-3 | 1-3-2-5-8-6-7-4 |
| Rotating direction | | Clockwise when seen from the flywheel | |
| Turbocharging method | | Turbocharged by exhaust gas turbine equipped with air cooler | |
| Starting method | | Compressed air (Starting valve type) | |
| Cooling method | Jacket | Fresh water | |
| | Cooler | Fresh (or Sea) water | |

Note : (1) Both the ignition sequence and the rotation direction show the data in the case of normal rotation respectively.

(2) Since the engine speed and the output, which are marked with ★, differ depending upon each specifications, be minded to enter the data after referring to the "Engine Specifications" and "Test Run Record".

| Item | | | Normal value | Alarm setting value (emergency stop value) | Reference |
|---|---|---|---|---|---|
| Pressure MPa {$kgf/cm^2$} | Starting air | Air tank | 2.0~3.0 {20~30} | 1.5 {15} | |
| | Control air | Air tank | 0.6~0.9 {6.0~9.0} | 0.6 {6.0} | |
| | Intake air | Air intake duct | | | Varies depending on the engine output |
| | Fuel oil | Engine inlet | Diesel fuel oil: 0.2~0.3(2~3) Heavy fuel oil: 0.5~0.6(5~6) | | |
| | Lubricating oil | Engine inlet (filter outlet) | 0.40~0.50 {4.0~5.0} | 0.25 (0.20) {2.5 (2.0)} | |
| | | Turbocharger inlet (filter outlet) | 0.06~0.15 {0.6~1.5} | | In case of MET turbocharger |
| | Cooling water | Jacket line (jacket inlet) | 0.25~0.35 {2.5~3.5} | | Consider static and dynamic pressure due to tank head and pipe resistance |
| | | Cooler line (cooler inlet) | 0.1~0.2 {1~2} | | |
| Temperature °C | Intake air | Air intake duct | 45~55 | | |
| | Exhaust gas | Cylinder outlet | | 500 | |
| | | Turbocharger inlet | | 600 | |
| | | Turbocharger outlet | | 500 | |
| | Lubricating oil | Engine inlet (cooler outlet) | 50~60 | 65 | |
| | Cooling water: Jacket line (fresh water) | Engine inlet | 65~70 | 80 | |
| | | Engine outlet | 70~75 | 85 (90) | |
| | Cooling water: Cooler line | Engine inlet | ~32 | | |

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| TYPE | RATED SPEEDR/MIN | PRESSURE RATIO | M3/SFLOWRATE | OVERALL DIMENSIONS LENGTH | WIDE | HIGH | KGWEIGHT |
|---|---|---|---|---|---|---|---|
| ABB/BBC VTR 161 | 35000-44000 | 2 | 0.50-1.10 | 837 | 543 | 542 | 230 |
| | | 2.5 | 0.62-1.50 | | | | |
| | | 3.2 | 0.75-1.70 | | | | |
| ABB/BBC VTR 201 | 28000-37000 | 2 | 0.85-2.00 | 961 | 460 | 515 | 250 |
| | | 2.5 | 1.00-2.30 | | | | |
| | | 3.2 | 1.25-2.40 | | | | |
| ABB/BBC VTR 251 | 22500-30000 | 2 | 1.15-3.20 | 1170 | 580 | 655 | 430 |
| | | 2.5 | 1.35-3.65 | | | | |
| | | 3.2 | 1.70-3.80 | | | | |
| ABB/BBC VTR 321 | 18000-23500 | 2 | 1.80-5.20 | 1422 | 725 | 843 | 870 |
| | | 2.3 | 2.10-5.80 | | | | |
| | | 3.2 | 2.65-5.90 | | | | |
| ABB/BBC VTR 401 | 14000-18500 | 2 | 2.80-8.00 | 1724 | 875 | 937 | 1500 |
| | | 2.5 | 3.40-9.20 | | | | |
| | | 3.2 | 4.30-9.70 | | | | |



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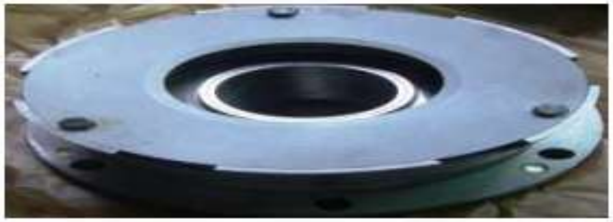
| TYPE | RATED SPEED R/MIN | PRESSURE RATIO | M3/SFLOWRATE | OVERALL DIMENSIONS LENGTH | WIDE | HIGN | KG WEIGHT |
|---|---|---|---|---|---|---|---|
| ABB/BBC VTR 214 | 38100-39420 | 3 | 2.20-3.20 | 1016 | 649 | 656 | 350 |
| | | 4 | 2.50-3.50 | | | | |
| ABB/BBC VTR 254 | 27900-32100 | 3 | 1.70-3.80 | 1144 | 768 | 780 | 565 |
| | | 4 | 2.40-4.20 | | | | |
| ABB/BBC VTR 304 | 22500-27000 | 3 | 1.95-4.00 | 1353 | 802 | 869 | 1020 |
| | | 4 | 2.80-5.20 | | | | |
| ABB/BBC VTR 354 | 18900-22800 | 3 | 3.00-6.30 | 1627 | 1159 | 1134 | 1680 |
| | | 4 | 4.20-7.80 | | | | |
| ABB/BBC VTR 454 | 14800-17400 | 3 | 5.10-10.00 | 2042 | 1464 | 1432 | 3250 |
| | | 4 | 6.40-12.50 | | | | |
| ABB/BBC VTR 564 | 13500-14100 | 3 | 7.50-19.00 | 2590 | 1802 | 1771 | 6400 |
| | | 4 | 11.00-20.50 | | | | |





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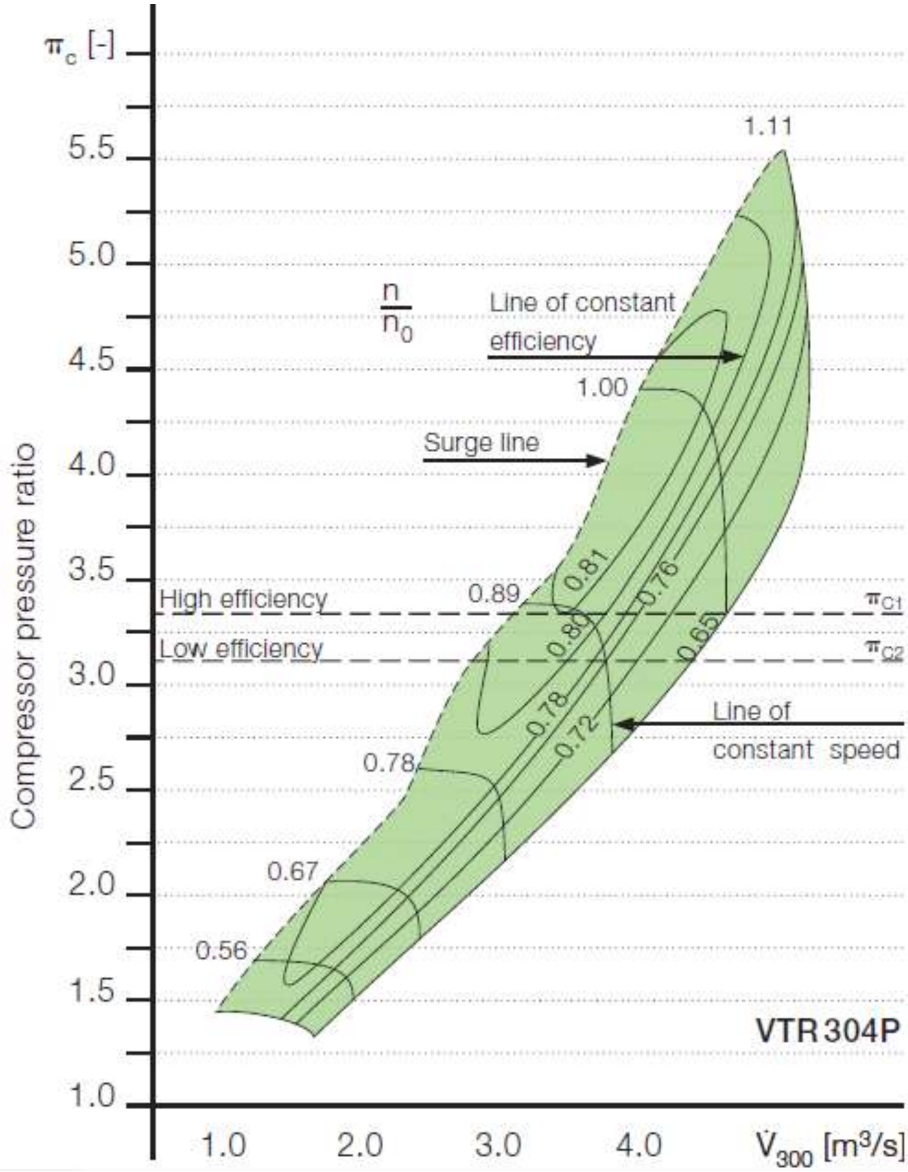
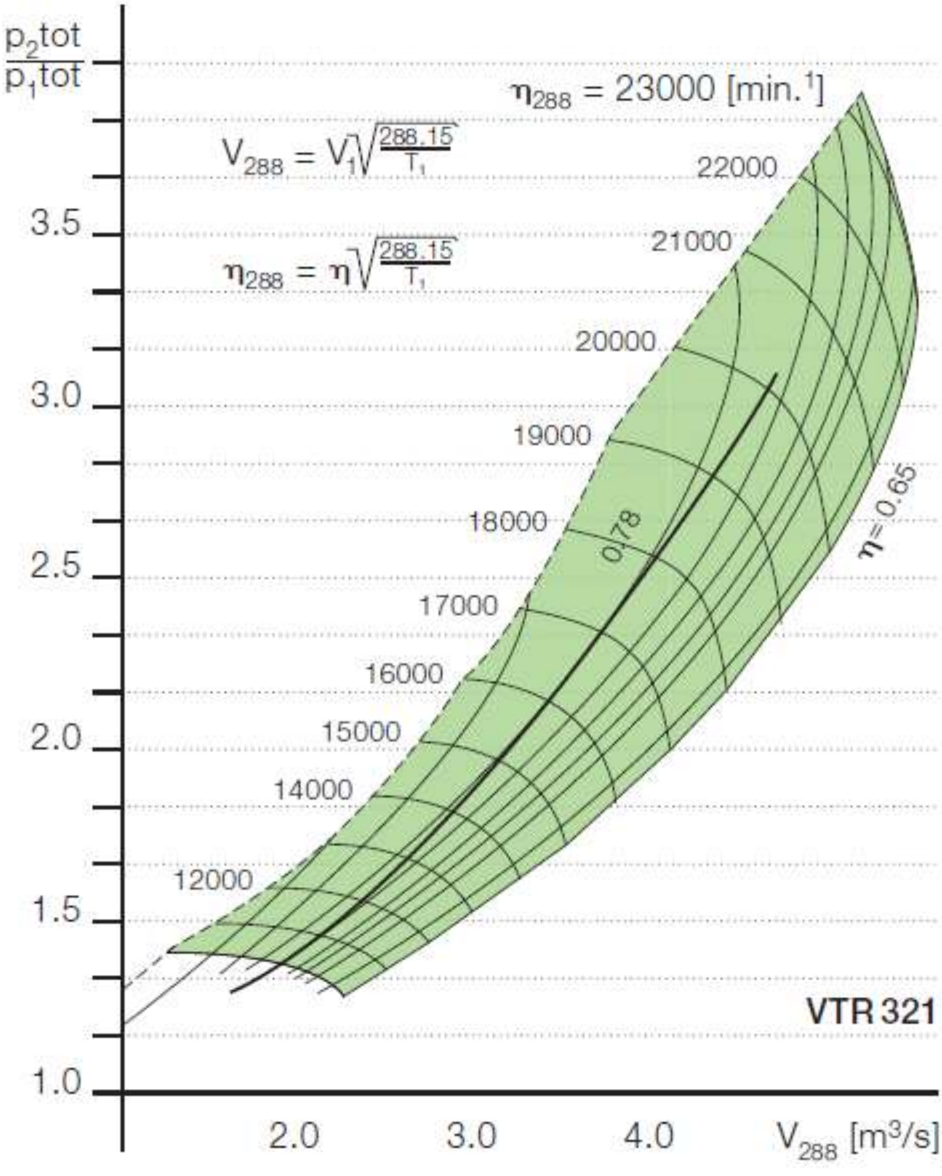
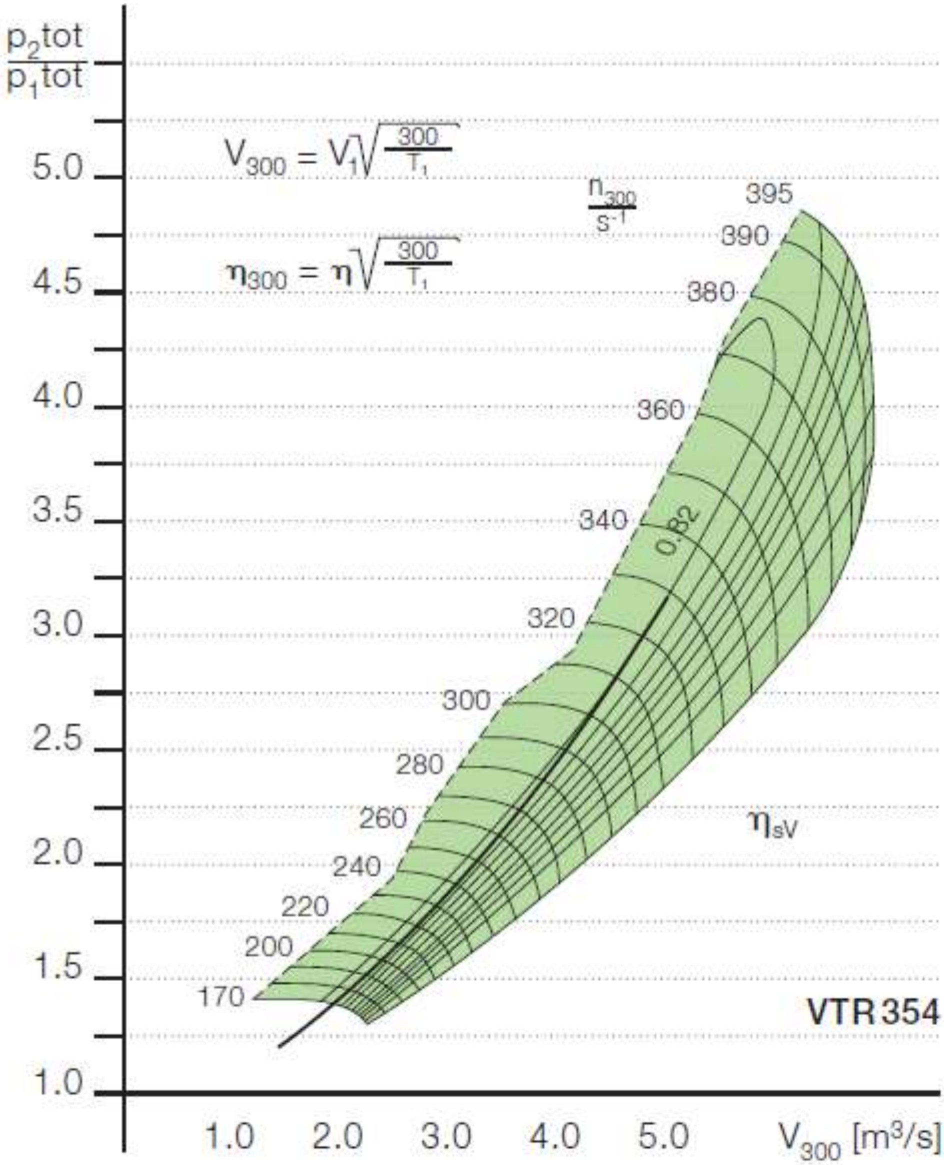


$\pi_c$ [-]
Compressor pressure ratio
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
1.0
2.0
3.0
4.0
$\dot{V}_{300}$ [m³/s]
$\frac{n}{n_0}$
1.11
1.00
0.89
0.78
0.67
0.56
Line of constant efficiency
Surge line
High efficiency
Low efficiency
$\pi_{C1}$
$\pi_{C2}$
0.81
0.76
0.80
0.65
0.78
0.72
Line of constant speed
VTR 304P



$\frac{p_2tot}{p_1tot}$
$\eta_{288} = 23000$ [min.-1]
$V_{288} = V_1\sqrt{\frac{288.15}{T_1}}$
$\eta_{288} = \eta\sqrt{\frac{288.15}{T_1}}$
22000
21000
20000
19000
18000
17000
16000
15000
14000
12000
0.78
$\eta = 0.65$
3.5
3.0
2.5
2.0
1.5
1.0
2.0
3.0
4.0
$V_{288}$ [m3/s]
VTR 321



p2tot / p1tot
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
V300 = V1 √(300/T1)
η300 = η √(300/T1)
n300 / s-1
395
390
380
360
340
320
300
280
260
240
220
200
170
0.82
ηsV
VTR 354
1.0
2.0
3.0
4.0
5.0
V300 [m3/s]

## Specifications of gear box MG32.35

| Rotation frequency | Rated gear ratio | Factual gear ratio | Transmit power kW (h.p.)/min$^{-1}$ |
|---|---|---|---|
| 400 – 900 | 2:1 | 1,9762:1 | 1,367 (1,860) |
| 400 – 1200 | 2,5:1 | 2,5082:1 | 1,0880 (1,470) |
| 400 – 1400 | 3:1 | 3,0466:1 | 0,889 (1,210) |
| 400 – 1600 | 3,5:1 | 3,4254:1 | 0,786 (1,070) |
| 400 – 1800 | 4:1 | 3,9524:1 | 0,684 (0,930) |
| 400 – 1800 | 4,5:1 | 4,4464:1 | 0,610 (0,830) |
| 400 – 1800 | 5:1 | 5,0206:1 | 0,537 (0,730) |
| 400 – 1800 | 5,5:1 | 5,5333:1 | 0,493 (0,670) |
| 400 – 1800 | 6:1 | 5,9286:1 | 0,456 (0,620) |

| MERATUS Asset Division | SHIP PARTICULAR ON THE BRIDGE<br>KM Meratus Palembang | No. Form : AST-SPE-4.2-01-44<br>Revision : 2<br>Revision Date : 19 Jun 2015<br>Page : 1/1 |
|---|---|---|

## GENERAL INFO

| | |
|---|---|
| Previous name | MCP Altona |
| Owner | PT Mitrarejeki Investa |
| Operator | PT Meratus Line |
| Built | 9 December 2006 |
| Builder | Shandong Huanghai Shipbuilding Co. Ltd, China/ Hull No. HCY-47 |
| Kind of Ship | Multi purpose ship, equipped for carriage container-heavy cargo |
| Call Sign | POGI |
| Flag | Indonesia |
| Port of Registry | Surabaya |
| IMO-Number | 9371921 |
| Class | BKI & GL |
| Class/Reg. Number | BKI=17925 |
| MMSI-Number | 525025074 |
| Official Number | 2011 Ka No.4548 / L |
| AAIC | IA-25 |
| P & I | SOP |

## TONNAGES (T)

| | |
|---|---|
| Gross Tonnage | 5612 |
| Net Tonnage | 2916 |
| Ton Per Cm immersion | 19.05 |

## WEIGHTS (T)

| | |
|---|---|
| Light Ship | 3244.9 |
| Displacement (summer) | 11097.7 |
| Displacement (tropical) | |

## DIMENSIONS (m)

| | |
|---|---|
| Length over all | 117 |
| LBP | 110.03 |
| Beam | 19.7 |
| Depth to Maindeck | 8.5 |
| Highest point from keel (air draft) | 37 |

## MLC

| | |
|---|---|
| Safe Manning | 15 persons |
| Accomodation | |

| LOADLINES | FREEBOARD (mm) | DRAFT(m) | DWT (MT) |
|---|---|---|---|
| Tropical | 1938 | | |
| Summer | 2092 | 6.45 | 7852.8 |

## CARGO CAPACITIES

| | |
|---|---|
| Grain (m³) | |
| Bales (m³) | |
| Container | |
| Homogeneus @14ton | 371 |
| Reefers | 60 plugs (available onboard)<br>380 Volt 50Hz |

## COMMUNICATIONS WITH VESSEL

| | | |
|---|---|---|
| FBB | Phone | |
| Satcom-C | Telex | 463700673 |
| Satcom-C | Telex | 463700672 |
| FBB | Email | |

## MACHINERIES

| | |
|---|---|
| Type of Propulsion | |
| Main Engine | Daihatsu |
| Engine Model | 8DKM-28X2 |
| Engine Model / rpm | 2X2500 KW/7 500 Rpm |
| Auxilary Engines | |
| Engine Model | |
| Generators | |
| Emergency Generator | |
| Shaft Generator | |

## ANCILLIARIES

| | |
|---|---|
| Evaporator | |
| Boiler | |
| Ballast Pumps | |
| Live Saving Equipment | 2 units lifeboat @ cap.22 persons<br>2 units liferaft @ cap.20 persons<br>immersion suit 23 pcs |
| Provision Crane | |
| Deck Crane | |
| Bowthruster | |
| Steering Gear | |

## SPEED & CONSUMPTION

| | |
|---|---|
| Service Speed (knots) | 10.4 |
| Consumption IFO | 8.287 kL/day |

## TANK CAPACITIES (M³)

| | |
|---|---|
| Water Ballast (100%) | 3824 |
| Freshwater (100%) | 160 |
| Bilge Water | |
| IFO (100%) | 380 |
| MDO (100%) | 229 |
| Lubricating Oils | |

## CONTAINER STACKING LOAD

| | |
|---|---|
| Tank Top (20') | 60 |
| Tank Top (40') | 70 |
| Hatch cover (20') | 55 |
| Hatch cover (40') | 70 |

## DECK LOAD (Tons /m²)

| | |
|---|---|
| Tank top | 0 |
| Second deck | 0 |
| Upper deck | 0 |
| On hatch cover | 0 |

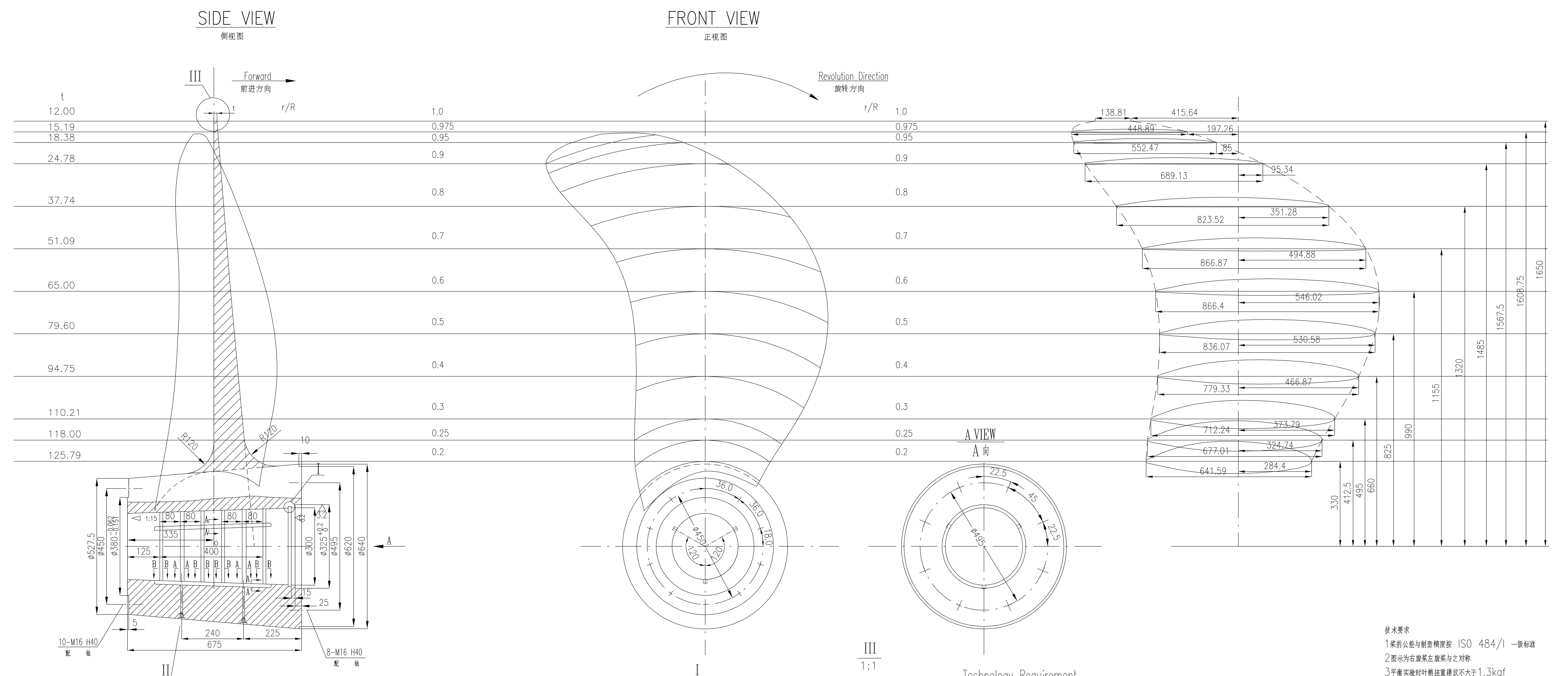
** All details and figures are about, given to the best of owner's knowledge and without guarantee*

**Total Ship Resistance Calculation**

| V (Knots) | V (m/s) | Va (m/s) | Rn | Cf | CA | 1+k | Cv | w | Fn | m2 |
|---|---|---|---|---|---|---|---|---|---|---|
| 6,9 | 3,54936 | 2,690547 | 4,E+08 | 0,001704 | 0,000496 | 5,075 | 0,009143 | 0,241963 | 0,10697 | -5,805E-24 |
| 7,6 | 3,90944 | 2,966763 | 5,E+08 | 0,001683 | 0,000496 | 5,075 | 0,009035 | 0,241128 | 0,117822 | -1,111E-17 |
| 8,3 | 4,26952 | 3,243208 | 5,E+08 | 0,001663 | 0,000496 | 5,075 | 0,008938 | 0,240381 | 0,128674 | -1,860E-13 |
| 9 | 4,6296 | 3,519859 | 6,E+08 | 0,001646 | 0,000496 | 5,075 | 0,00885 | 0,239706 | 0,139526 | -1,611E-10 |
| 9,7 | 4,98968 | 3,796697 | 6,E+08 | 0,00163 | 0,000496 | 5,075 | 0,00877 | 0,23909 | 0,150378 | -2,037E-08 |
| **10,4** | **5,34976** | 4,073707 | 6,E+08 | 0,001616 | 0,000496 | 5,075 | 0,008697 | 0,238525 | 0,16123 | -7,080E-07 |
| 11,1 | 5,70984 | 4,350875 | 7,E+08 | 0,001603 | 0,000496 | 5,075 | 0,008629 | 0,238004 | 0,172082 | -1,009E-05 |

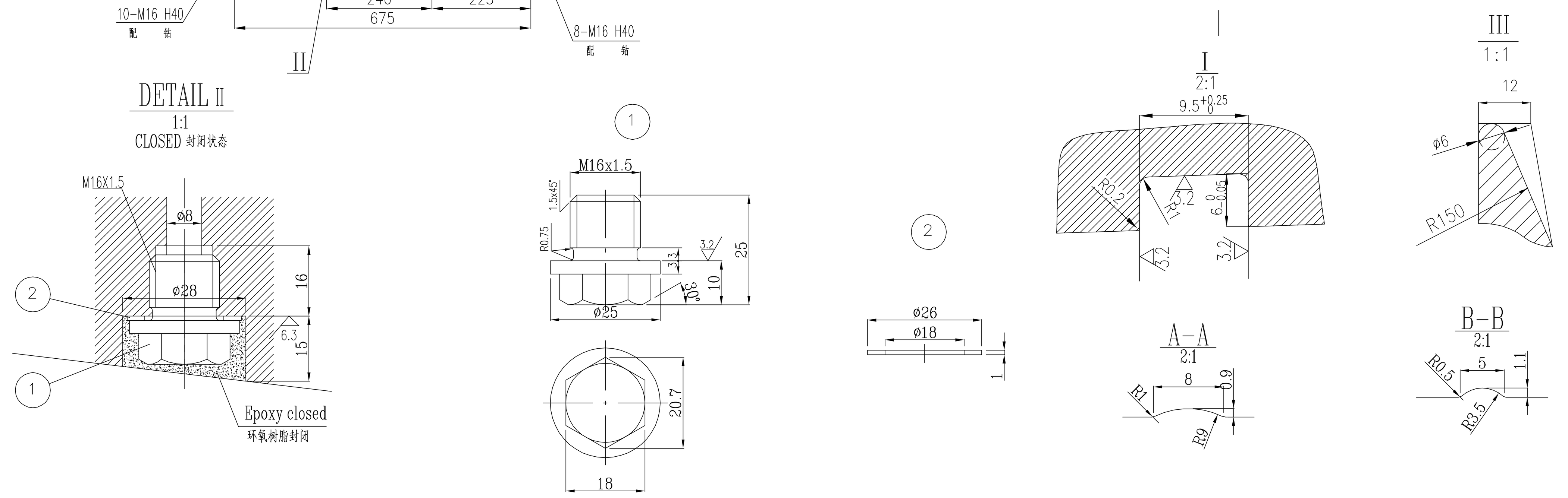
| RW/W | W | Rt | Rt/2 |
|---|---|---|---|
| 1,156E-07 | 1,123E+05 | 174,489 | 87,24472 |
| 4,374E-07 | 1,123E+05 | 209,214 | 104,6068 |
| 1,335E-06 | 1,123E+05 | 246,940 | 123,4699 |
| 3,455E-06 | 1,123E+05 | 287,712 | 143,8561 |
| 7,844E-06 | 1,123E+05 | 331,623 | 165,8117 |
| 1,603E-05 | 1,123E+05 | 378,822 | 189,4108 |
| 3,005E-05 | 1,123E+05 | 429,515 | 214,7577 |

SC4382HC-107-01




## PARTICULARS OF PROPELLER
螺旋桨主要参数

| | |
|---|---|
| type 叶型 | Monobloc |
| diameter 直径 | 3.30 m |
| pitch 螺距 | 2.508 m |
| pitch ratio 螺距比 | 0.760 |
| blade area ratio 盘面比 | 0.60 |
| number of blades 叶片数 | 5 |
| hub diameter ratio ?径比 | 0.18 |
| rake angle 后倾角 | 0° |
| revolution direction 旋转方向 | one levogyrate, one dextrogyrate 左右旋各一只 |
| rated power 主机额定功率MCR | 2500 kW x2 |
| speed of main engine 主机转速 | 750 rpm |
| ratio of gear 齿轮箱减速比 | 3.9524:1 |
| received power of propeller 螺旋桨设计点收到功率 | 2250 kW (each one) |
| speed of design point of propeller 螺旋桨设计点转速 | 189.8 rpm |
| class 船级 | CCS |
| material of propeller 材料 | Cu4 锰铝青铜 |
| mass of propeller 重量 | 3066 Kg |
| inertia moment of propeller 惯性矩 | 1416.9 kg·m |




Technology Requirement

1, Tolerance & manufacture precision accord with ISO 484/I grade 1 standard.

2, The drawing shows the dextrogyrate propeller, another is symmetrical.

3, The hanging weight on blade tip not more than 1.3 kgf during equilibrium test.

4, The surface roughness of blades should be: r/R ?.3 take 3.2

r/R < 0.3 take 6.3

5, The surface roughness of hub be showed on the drawing;

6, The chemical compositon and mechanical performance of the material should meet the requirement of related regulation in CCS "material and welding regulation' 2003 amend chapter 9 section 1

技术要求

1桨的公差与制造精度按 ISO 484/I 一级标准

2图示为右旋桨左旋桨与之对称

3平衡实验时叶梢挂重建议不大于1.3kgf

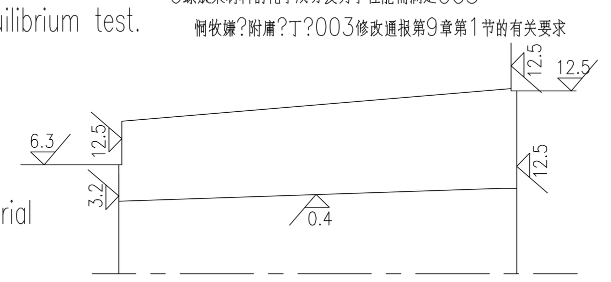
4桨叶表面光洁度在r/R ≥0.3取 3.2

r/R< 0.3取 6.3

5桨?表面光洁度如图所示

6螺旋桨材料的化学成分及力学性能需满足CCS

惘牧嫌?附庸?丁?003修改通报第9章第1节的有关要求



| ITEM NO. 序号 | MARK 代号 | NAME & SPEC 名称及规格 | QTY. 数量 | MATERIAL 材料 | SET 每件 WEIGHT(kg) 重量(kg) | TOTAL 共件 | REMARKS 附注 |
|---|---|---|---|---|---|---|---|
| 3 | | 螺旋桨 PROPELLER | 2 | Cu4 | | 3018 | |
| 2 | | 垫圈 GASKET | 4 | Non-amiant sealing material | | | |
| 1 | | 螺塞 SCREW PLUG | 4 | CuZn36 | | | |

| | |
|---|---|
| 600TEU MULTI-PUR. CON. VESSEL 600TEU 多用途集装箱船 | DETAIL DESIGN 详细设计 |
| PROPELLER PLAN 螺旋桨图 | SC4382HC-107-01 |

| MARKS 标记 | NUM. 数量 | REVISION NO. 修改单号 | SIGN 签字 | DATE 日期 |
|---|---|---|---|---|
| | | | | |

| | | | |
|---|---|---|---|
| DESIGNED 设绘 | | DATE 日期 | |
| CHECKED 校对 | | DATE 日期 | |
| VERIFIED 审核 | | DATE 日期 | |
| STANDARDIZE 标检 | | DATE 日期 | |
| APPROVED 批准 | | DATE 日期 | |

| COR. MARKS 图样标记 | WEIGHT 重量 | SCALE 比例 |
|---|---|---|
| | 3068kg | 1 : 10 |

| PAGE 页数 | 1 / 2 | TOT. AREA 总面积 | 0.5 $m^2$ |
|---|---|---|---|

SHANDONG HUANGHAI SHIPBUILDING CO. LTD
山东黄海造船有限公司

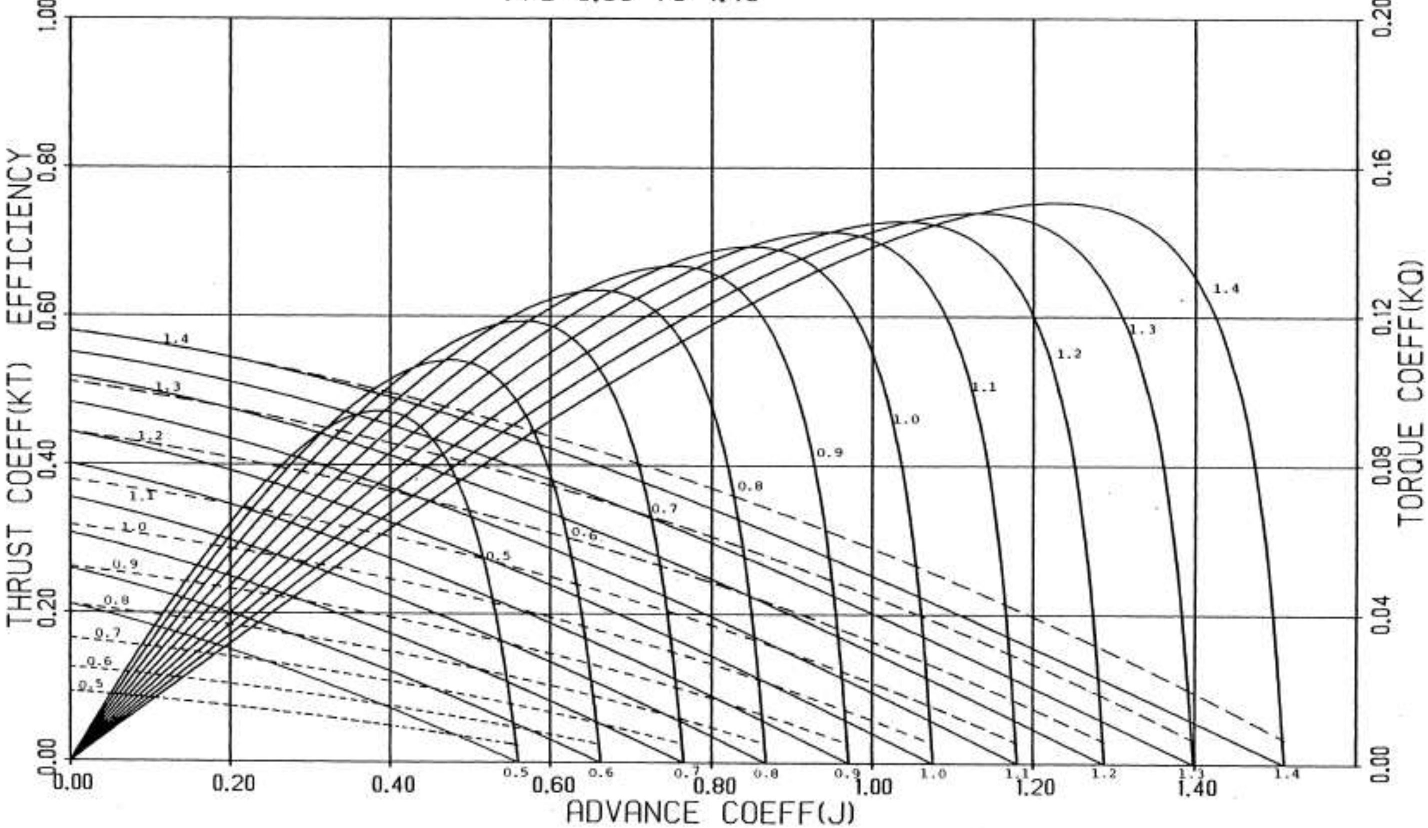
面积: 0.375m²

会 签

旧底图登记号

底图登记号

FIGURE 55. WAGENINGEN B-SERIES PROPELLERS
FOR 5 BLADES AE/AO= 0.600
P/D=0.50 TO 1.40



**VTR 354 Pressure Ratio and Volumetric Flow rate Calculation**

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,37 | 0,37 | 318 | 60 | 210,01 | 150,01 | 180,59 | 0,35 | 34013,34 |
| 2 | 7,6 | 42,20 | 1 | 0,66 | 0,66 | 318 | 60 | 268,65 | 208,65 | 237,53 | 0,63 | 61857,34 |
| 3 | 8,3 | 52,65 | 1 | 0,93 | 0,93 | 318 | 60 | 308,02 | 248,02 | 275,75 | 0,88 | 86219,50 |
| 4 | 9 | 64,90 | 1 | 1,29 | 1,56 | 318 | 60 | 353,66 | 293,66 | 320,06 | 1,23 | 120600,14 |
| 5 | 9,7 | 72,48 | 1 | 1,89 | 1,89 | 318 | 60 | 412,94 | 352,94 | 377,61 | 1,79 | 175703,19 |
| 6 | 10,4 | **86,86** | 1 | 2,63 | 2,63 | 318 | 60 | 473,83 | 413,83 | 436,73 | 2,50 | 245396,38 |
| 7 | 11,1 | 94,54 | 1 | 3,00 | 3,00 | 318 | 60 | 499,91 | 439,91 | 462,05 | 2,85 | 279488,10 |

| No. | Vd m3 | Lt m3 | ηv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{300}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0074 | 0,0126 | 1,7 | 0,6560 | 0,0275 | 1,3427 | 1,73057 |
| 2 | 0,0105 | 0,0180 | 1,72 | 0,9071 | 0,0660 | 1,7721 | 1,991547 |
| 3 | 0,0129 | 0,0225 | 1,74 | 1,0891 | 0,1099 | 2,1170 | 2,208126 |
| 4 | 0,0166 | 0,0296 | 1,78 | 1,3125 | 0,1936 | 2,6666 | 2,58173 |
| 5 | 0,0203 | 0,0374 | 1,84 | 1,6207 | 0,3408 | 3,2228 | 2,872557 |
| 6 | 0,0228 | 0,0438 | 1,92 | 1,9571 | 0,5350 | 3,6219 | 3,001896 |
| 7 | 0,0240 | 0,0480 | 2 | 2,1069 | 0,6575 | 3,9085 | 3,14937 |

**VTR 354 Power Output Calculation**

| No | Load (%) | Tc (K) | Pa kg/cm2 | Pc kg/cm2 | n (mol) | Mg (mol) | $\mu_0$ | $\mu$ | $\lambda$ | Tz (◦C) |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 13,94 | 451,0503 | 0,37 | 12,0037 | 0,0025 | 0,01046 | 1,665379 | 1,64599918 | 6,508824 | 704,0698 |
| 2 | 30,00 | 639,2632 | 0,81 | 25,83294 | 0,0058 | 0,02383 | 2,499409 | 2,455737291 | 3,298115 | 491,9004 |
| 3 | 40,00 | 726,7197 | 1,07 | 34,44391 | 0,009 | 0,03727 | 2,897798 | 2,842522218 | 2,667815 | 425,8332 |
| 4 | 50,00 | 802,0618 | 1,34 | 43,05489 | 0,0161 | 0,06684 | 3,208237 | 3,143919713 | 2,374875 | 399,6462 |
| 5 | 70,00 | 929,5981 | 1,88 | 60,27685 | 0,0256 | 0,10623 | 3,748947 | 3,668880315 | 2,012879 | 360,2332 |
| 6 | **100,00** | 1085,545 | 2,69 | 86,10979 | 0,0439 | 0,18194 | 4,395165 | 4,296276523 | 1,749511 | 330,8811 |
| 7 | 110,00 | 1131,243 | 2,95 | 94,72076 | 0,0517 | 0,214 | 4,453804 | 4,353207756 | 1,768145 | 348,593 |

| $\rho$ | $\delta$ | Pit (kg/cm2) | Pi (kg/cm2) | Pe (kg/cm2) | Nb (HP) | Nb (kW) |
|---|---|---|---|---|---|---|
| 1 | 13,3 | 4,494689 | 4,269955 | 3,415964 | 84,169 | 62,7903 |
| 1 | 13,3 | 9,672933 | 9,189286 | 7,351429 | 271,71 | 202,695 |
| 1 | 13,3 | 12,89724 | 12,25238 | 9,801905 | 483,04 | 360,346 |
| 1 | 13,3 | 16,12155 | 15,31548 | 12,25238 | 956,01 | 713,185 |
| 1 | 13,3 | 22,57018 | 21,44167 | 17,15333 | 1761,1 | 1313,76 |
| 1 | 13,3 | 32,24311 | 30,63095 | 24,50476 | 3522,2 | 2627,52 |
| 1 | 13,3 | 35,46742 | 33,69405 | 26,95524 | 4317,2 | 3220,59 |

**VTR 321 Pressure Ratio and Volumetric Flow rate Calculation**

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,38 | 0,38 | 318 | 60 | 213,67 | 153,67 | 184,15 | 0,36 | 35471,32 |
| 2 | 7,6 | 42,20 | 1 | 0,69 | 0,69 | 318 | 60 | 273,34 | 213,34 | 242,08 | 0,66 | 64508,85 |
| 3 | 8,3 | 52,65 | 1 | 0,97 | 0,97 | 318 | 60 | 313,39 | 253,39 | 280,96 | 0,92 | 89915,28 |
| 4 | 9 | 64,90 | 1 | 1,35 | 1,35 | 318 | 60 | 359,83 | 299,83 | 326,05 | 1,28 | 125769,65 |
| 5 | 9,7 | 72,48 | 1 | 1,57 | 1,57 | 318 | 60 | 383,07 | 323,07 | 348,61 | 1,49 | 146419,33 |
| 6 | 10,4 | **86,86** | 1 | 2,20 | 2,20 | 318 | 60 | 439,56 | 379,56 | 403,46 | 2,09 | 204496,98 |
| 7 | 11,1 | 94,54 | 1 | 2,50 | 2,50 | 318 | 60 | 463,75 | 403,75 | 426,94 | 2,38 | 232906,75 |

| No. | Vd m3 | Lt m3 | ηv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{300}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0031 | 0,0052 | 1,7 | 0,6709 | 0,0220 | 1,0283 | 1,28636 |
| 2 | 0,0055 | 0,0095 | 1,72 | 0,9282 | 0,0553 | 1,4246 | 1,554306 |
| 3 | 0,0080 | 0,0139 | 1,74 | 1,1147 | 0,0971 | 1,7936 | 1,816436 |
| 4 | 0,0111 | 0,0197 | 1,78 | 1,3436 | 0,1657 | 2,1893 | 2,058103 |
| 5 | 0,0154 | 0,0283 | 1,84 | 1,4629 | 0,2591 | 2,9397 | 2,672606 |
| 6 | 0,0203 | 0,0390 | 1,92 | 1,7654 | 0,4307 | 3,4986 | 2,956696 |
| 7 | 0,0240 | 0,0480 | 2 | 1,9001 | 0,5706 | 4,0701 | 3,343742 |

**VTR 321 Power Output Calculation**

| No | Load (%) | Tc (K) | Pa kg/cm2 | Pc kg/cm2 | n (mol) | Mg (mol) | $\mu_0$ | $\mu$ | $\lambda$ | Tz (◦C) |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 13,94 | 407,4404 | 0,30 | 9,64928 | 0,0022 | 0,00931 | 1,482019 | 1,467979189 | 8,096977 | 741,5371 |
| 2 | 30,00 | 579,471 | 0,65 | 20,76603 | 0,0051 | 0,02114 | 2,216486 | 2,18105455 | 4,102855 | 576,5129 |
| 3 | 40,00 | 659,4081 | 0,86 | 27,68803 | 0,008 | 0,03302 | 2,567204 | 2,52155724 | 3,318762 | 508,5733 |
| 4 | 50,00 | 728,2723 | 1,08 | 34,61004 | 0,0143 | 0,05917 | 2,840273 | 2,786672918 | 2,954345 | 482,6657 |
| 5 | 70,00 | 844,8433 | 1,51 | 48,45406 | 0,0227 | 0,09396 | 3,31595 | 3,24849511 | 2,504021 | 440,7911 |
| 6 | **100,00** | 987,3822 | 2,16 | 69,22008 | 0,0388 | 0,16079 | 3,884341 | 3,800331396 | 2,176391 | 409,1157 |
| 7 | 110,00 | 1029,151 | 2,38 | 76,14209 | 0,0456 | 0,18909 | 3,935388 | 3,849890975 | 2,199572 | 432,0144 |

| $\rho$ | $\delta$ | Pit (kg/cm2) | Pi (kg/cm2) | Pe (kg/cm2) | Nb (HP) | Nb (kW) |
|---|---|---|---|---|---|---|
| 1 | 13,3 | 3,613094 | 3,43244 | 2,745952 | 67,66 | 50,4745 |
| 1 | 13,3 | 7,77567 | 7,386886 | 5,909509 | 218,42 | 162,938 |
| 1 | 13,3 | 10,36756 | 9,849181 | 7,879345 | 388,29 | 289,667 |
| 1 | 13,3 | 12,95945 | 12,31148 | 9,849181 | 768,5 | 573,3 |
| 1 | 13,3 | 18,14323 | 17,23607 | 13,78885 | 1415,7 | 1056,08 |
| 1 | 13,3 | 25,9189 | 24,62295 | 19,69836 | 2831,3 | 2112,16 |
| 1 | 13,3 | 28,51079 | 27,08525 | 21,6682 | 3470,4 | 2588,9 |

**VTR 304 Pressure Ratio and Volumetric Flow rate Calculation**

| No. | Vs | Load (%) | Po kg/cm2 | Psup kg/cm2 | Pressure ratio | To K | Δt cool ◦C | Tsup K | Ti sup K | Ta K | Pa kg/cm2 | Pa satuan pa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 6,9 | 33,46 | 1 | 0,37 | 0,37 | 318 | 60 | 210,01 | 150,01 | 180,59 | 0,35 | 34013,34 |
| 2 | 7,6 | 42,20 | 1 | 0,66 | 0,66 | 318 | 60 | 268,65 | 208,65 | 237,53 | 0,63 | 61857,34 |
| 3 | 8,3 | 52,65 | 1 | 0,93 | 0,93 | 318 | 60 | 308,02 | 248,02 | 275,75 | 0,88 | 86219,50 |
| 4 | 9 | 64,90 | 1 | 1,29 | 1,37 | 318 | 60 | 353,66 | 293,66 | 320,06 | 1,23 | 120600,14 |
| 5 | 9,7 | 72,48 | 1 | 1,89 | 1,89 | 318 | 60 | 412,94 | 352,94 | 377,61 | 1,79 | 175703,19 |
| 6 | 10,4 | **86,86** | 1 | 2,63 | 2,63 | 318 | 60 | 473,83 | 413,83 | 436,73 | 2,50 | 245396,38 |
| 7 | 11,1 | 94,54 | 1 | 3,00 | 3,00 | 318 | 60 | 499,91 | 439,91 | 462,05 | 2,85 | 279488,10 |

| No. | Vd m3 | Lt m3 | ηv | ρ kg/m3 | ṁ kg/s | V̇ m3/s | $\dot{V}_{300}$ m3/s |
|---|---|---|---|---|---|---|---|
| 1 | 0,0037 | 0,0063 | 1,7 | 0,6560 | 0,0137 | 0,6714 | 0,865285 |
| 2 | 0,0055 | 0,0095 | 1,72 | 0,9071 | 0,0349 | 0,9382 | 1,054348 |
| 3 | 0,0074 | 0,0129 | 1,74 | 1,0891 | 0,0628 | 1,2097 | 1,261786 |
| 4 | 0,0105 | 0,0186 | 1,78 | 1,3125 | 0,1219 | 1,6790 | 1,625533 |
| 5 | 0,0154 | 0,0283 | 1,84 | 1,6207 | 0,2582 | 2,4415 | 2,17618 |
| 6 | 0,0216 | 0,0414 | 1,92 | 1,9571 | 0,5061 | 3,4262 | 2,839631 |
| 7 | 0,0240 | 0,0480 | 2 | 2,1069 | 0,6575 | 3,9085 | 3,14937 |

**VTR 304 Power Output Calculation**

| No | Load (%) | Tc (K) | Pa kg/cm2 | Pc kg/cm2 | n (mol) | Mg (mol) | $\mu_0$ | $\mu$ | $\lambda$ | Tz (◦C) |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 13,94 | 451,0503 | 0,37 | 12,0037 | 0,0025 | 0,01046 | 1,665379 | 1,64599918 | 6,508824 | 704,0698 |
| 2 | 30,00 | 639,2632 | 0,81 | 25,83294 | 0,0058 | 0,02383 | 2,499409 | 2,455737291 | 3,298115 | 491,9004 |
| 3 | 40,00 | 726,7197 | 1,07 | 34,44391 | 0,009 | 0,03727 | 2,897798 | 2,842522218 | 2,667815 | 425,8332 |
| 4 | 50,00 | 802,0618 | 1,34 | 43,05489 | 0,0161 | 0,06684 | 3,208237 | 3,143919713 | 2,374875 | 399,6462 |
| 5 | 70,00 | 929,5981 | 1,88 | 60,27685 | 0,0256 | 0,10623 | 3,748947 | 3,668880315 | 2,012879 | 360,2332 |
| 6 | **100,00** | 1085,545 | 2,69 | 86,10979 | 0,0439 | 0,18194 | 4,395165 | 4,296276523 | 1,749511 | 330,8811 |
| 7 | 110,00 | 1131,243 | 2,95 | 94,72076 | 0,0517 | 0,214 | 4,453804 | 4,353207756 | 1,768145 | 348,593 |

| $\rho$ | $\delta$ | Pit (kg/cm2) | Pi (kg/cm2) | Pe (kg/cm2) | Nb (HP) | Nb (kW) |
|---|---|---|---|---|---|---|
| 1 | 13,3 | 4,494689 | 4,269955 | 3,415964 | 84,169 | 62,7903 |
| 1 | 13,3 | 9,672933 | 9,189286 | 7,351429 | 271,71 | 202,695 |
| 1 | 13,3 | 12,89724 | 12,25238 | 9,801905 | 483,04 | 360,346 |
| 1 | 13,3 | 16,12155 | 15,31548 | 12,25238 | 956,01 | 713,185 |
| 1 | 13,3 | 22,57018 | 21,44167 | 17,15333 | 1761,1 | 1313,76 |
| 1 | 13,3 | 32,24311 | 30,63095 | 24,50476 | 3522,2 | 2627,52 |
| 1 | 13,3 | 35,46742 | 33,69405 | 26,95524 | 4317,2 | 3220,59 |

# BIOGRAPHY

The author was born in Bogor, January 1, 1994, is the second sons of Suroso Familly. Author was studied in SDN Sukadamai 3 Bogor, SMPIT Ummul Quro Bogor, and graduated from SMA Swasta Islam Darul Hikam Bandung. The author continue to Double Bachelor Degree program with a major in Marine Engineering FTK - ITS through the Double Degree in 2013. Registered with the Student Registration Number 4213101021. Author takes a field of study in Marine Power Plant (MPP), Department Marine Engineering-FTK ITS Surabaya. During the study authors was an active member in the laboratory Marine Power Plant (MPP) Department Marine Engineering FTK - ITS Surabaya. Also authors had many activities during the study in ITS, authors was a Head of Shorinji Kempo ITS Organization 2014/2015, a Head of Department Internal Affairs-BEM FTK ITS 2015/2016, and active in many social activities arround the campus.

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