NATIONAL TECHNICAL UNIVERSITY OF ATHENS
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DIPLOMA THESIS:

SHIP`S MAIN ENGINE OPERATION EVALUATION WITH THERMODYNAMIC SIMULATION MODEL

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THERMODYNAMIC SIMULATION MODEL

by

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ABSTRACT

During the lifetime of a ship`s main engine the original shop test and sea trials are the only available reference conditions, which are contributing in assessing its performance. In common practice, the measured engine performance parameters are corrected to either ISO or Shop Test conditions, in order to be compared with reference shop test values. The usage of a validated simulation model for predicting the engine operation, eliminates possible errors caused by the aforementioned corrections. Furthermore, the nominal performance figures produced by detailed simulation models can be used as reference, to compare with any shipboard measured actual performance data, for engine performance evaluation.

This thesis presents the methodology and procedure in setting up the 6RTA48-T main engine in "MOtor THERmodynamics" ('MOTHER') simulation software. Firstly, geometric and operational data were collected and afterwards, the simulation model was created and tuned using Shop Test data; finally, it was validated using Sea Trials data. That simulation model is able to simulate the engine operation at any possible operating point with minimum 50% load of Maximum Continuous Rating (MCR) and maximum 100% load of MCR respectively.

The simulations` results, allowed prediction of performance parameters, such as cylinder pressures, scavenge air pressure, brake power, turbocharger speed, within 3% of actual measured values at Shop Test and Sea Trials.

Amongst the available “M/E Performance Reports”, eight representative cases were selected for performance evaluation and condition assessment. Simulations in “Motor Thermodynamics” simulation software at the selected operating points allowed the prediction of all engine parameters and direct comparison to the onboard measured data that had been reported in “M/E Performance Reports”. Finally, the ISO correction methodology was implemented by correcting the measured data to Shop Test conditions and the results were compared with those acquired by the simulation model.

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CHAPTER 1

INTRODUCTION

1.1 Background

The operation of a marine diesel engine is constantly monitored by the engine room crew and the technical staff at the office of a shipping company. The engine room crew is responsible for the daily maintenance, work and efficient running of all machinery. All the maintenance works onboard are performed as per manufacturers` recommendations and reports are sent to the Technical Department for evaluation.

- Main engine performance data are preferably recorded, when weather conditions are favorable, in “M/E Full Performance Report” and sent to the Technical Department for evaluation. The “M/E Full Performance Report” contain extensive data which vary among shipping companies. The most appropriate data in order to perform the evaluation, are the following:Maximum and compression pressure in cylinders
- Scavenge air pressure and temperature
- Exhaust gas temperature before and after turbocharger
- Fuel consumption
- Main engine and turbocharger rotational speed
- Barometric pressure and ambient temperature

The “M/E Full Performance Report” can be divided in sections according to the kind of recorded data. The following sample list summarizes the sections and explains what data could be reported in each section.

- Various Ship Data. Data such as, ship`s draught, ship`s speed, wind direction, swell height, barometric pressure and propeller apparent slip, could be recorded.
- Various M/E Data. Data such as, M/E revolutions per minute (rpm), load indicator position, fuel consumption, scavenge air pressure and T/C`s rpm, could be included.
- Pressures. Here are recorded the maximum and compression pressure for each cylinder and other pressures related to the fuel and oil supply system such as, fuel oil pressure after filter, bearing oil pressure; and pressure losses across the air cooler and the compressor suction.
- Temperatures. Here, data such as, exhaust gas temperatures, piston cooling inlet and outlet temp., jacket cooling inlet and outlet temp. are recorded for each cylinder. Moreover, T/C`s cooling temperature at inlet and outlet, fuel`s temp., engine`s lub oil temp., air cooler`s cooling water temp. at inlet and outlet and sea water temperature, could be recorded as well.
- Fuel oil. Basic fuel properties are reported, such as, calorific value, viscosity, density, sulpher, vanadium and aluminum plus silicon content.
- Cylinder oil. Here, the type of the oil and its consumption are reported.

Although, a shipping company could include in the “M/E Full Performance Report” more data compared to the already explained above, its purpose remains the same: the evaluation of the engine operation in order to ensure smooth, uninterrupted and cost-effective operation.

The most common evaluation method that is used, lies in the simple idea of comparing the same parameters when they are measured at the same ambient conditions. To be more specific, the basis of all the parameters that are measured during the engine`s lifetime, is the "Shop Test" measurements. During the "Shop Test" the engine is being tested at specific loads by the manufacturer in order to test its performance and the fulfillment of the owners’ requirements. All the parameters, that are being measured thoroughly at the different loads, are recorded; these measurements, at these particular ambient conditions, are the basis of all the future performance evaluations. According to the common evaluation procedure the data that are measured onboard, are corrected using some empirical correlations and factors, in order to meet the Shop Test`s ambient conditions. Thus, the comparison is possible as both refer to same conditions. Another alternative, is to correct both Shop test data and the measured onboard data, to ''ISO reference conditions''. The procedure of the above mentioned correction methodologies is provided by the manufacturers.

Although, in practice the measured data are corrected either at Shop Test conditions or ISO conditions, this method does not provide an indication about the future engine`s performance.

Consequently, an alternative method in performance evaluation in order to overcome the ambient conditions` corrections, is the use of a computer based model of a particular engine for comparison of its operation profile with the actual data recorded onboard. The model used in the present thesis is called “MOTHER” (MOtor THERmodynamics). Such simulation programs have been mainly used by engines` manufactures in the design procedure but they are not often used for performance evaluation when the engines are in service.

MOTHER is an engine simulation and performance prediction software of the so-called thermodynamic control volume type. It considers the engine as a series of interconnected volumes, assuming spatial uniformity of fluid properties and constant rate of change of parameters in each control volume at any instant or computational time step. The ancestry of the basic model lies in the engine models developed in the UK and USA in the 1960`s and further at Imperial College, Univ. of London in the 1970`s. The kernel of the MOTHER program in its present form was developed at the NTUA (National Technical University of Athens).

As the NTUA cooperate with shipping companies, the computer model has been used for performance prediction for over 50 engine types and its accuracy between the measured and the predicted data, has been validated to be better than 3% over the whole operating envelope of an engine [1],[2]. This Thesis continues the previous efforts by simulating the operation of another engine type which is the SULZER Main Engine RTA-48T with 6 cylinders, in order to evaluate the engine performance at several operating points.

1.2 Previous Work

As mentioned above, the NTUA has successfully performed engine simulations of different engine types. The last completed project was held in association with DANAOS Shipping Company between 2008 and 2010 [3], in which two engines were selected for the simulation with the MOTHER program. The selected engines types were the 8K90MC-C and 12K98MC-C MAN B&W; the whole work can be summarized below:

i. Parametric runs for performance prediction were possible at different loads and speeds in contrast with the reference points.
ii. The validated model could be used for creating reference conditions to compare with recorded service data.

1.3 Motivation

From the beginning of manufacturing the first marine diesel engine until today, constant research is made in order to improve its performance and reliability. There are two parts which interact between them, the one is the engine manufacturer and the other the operator. From the part of the operator, the constant service of the engine is vital. A tool that could contribute in monitoring the installation, is the computer based simulation of its operation in order to predict its performance and create reference points which could be compared directly with measured data. Thus, from this perspective, the engine simulation is valuable when its accuracy is proven.

The completed project for performance prediction that was described in paragraph (1.2), was utilized for very large two-stroke engines installed in containerships. Apart from the fact that the previous project motivated this Thesis, a simulation of the RTA-48T has not been performed yet. Consequently, a bulk carrier built in 2004 with the SULZER 6RTA-48T main engine installed and operated by the A.M. Nomikos TWMA SA, was selected, and the following factors contributed in accomplishing the project:

a. Engine geometric data such as exhaust valve lift, inlet port size, cylinder dimensions and other, were available.
b. Shop Test and Sea Trial data were available too, and
c. Main Engine Performance reports from 2006 up to 2010 were also available. Particularly, these reports contain the most essential data in order to perform the simulation and the evaluation of the engine`s performance.

1.4 Objectives

The primary objectives of this work are:

i. To set up the 6RTA-48T engine`s simulation model in "MOTHER" software.
ii. To calibrate the model at a manner that it could predict the already measured data from Shop Test and Sea Trials.
iii. To make parametric runs at operating points where Performance Reports exist.
iv. To conduct a comparison between the measured service data with those acquired by the simulation model.
v. To provide an assessment of the influence of fuel type over the M/E Performance.
vi. To examine the reliability of using the MOTHER software in existing operating points with respect to the well-known method of ISO-corrections, in order to evaluate the overall main engine performance.

1.5 Thesis Outline

This chapter introduces the content of the thesis, reviews previous project, and outlines the motivation and objectives of this work.

Chapter 2 provides the basic thermodynamic principles which affect the engine operation and performance, as well.

Chapter 3 describes and shows major parts of the M/E, with emphasis to those that are required by the simulation model.

Chapter 4 introduces the methodology followed for the simulation. The simulation`s model setting-up was started with one cylinder and after evaluating its output the next step of the simulation procedure was possible, i.e. to add the other 5 cylinders. Simulation results and diagrams are also presented and compared with Shop Test data.

Chapter 5 presents the simulation results of the six cylinders simulation model and a prime energy balance calculation.

Chapter 6 describes the final adjustments of the engine`s simulation model. The results of calculated performance parameters are presented and compared with those acquired at Shop Test.

Chapter 7 presents the results of the engine`s simulation at Sea Trials. These Chapter`s scope is the validation of the simulation model through the reliable calculation of reported data.

Chapter 8 is related with the simulation of selected cases among the available data. Data from M/E Performance Reports have been used in "MOTHER" in order to simulate the actual engine operation. After each simulation is completed, the results are evaluated. Moreover, a comparison between the simulation-based results and those acquired by the ISO-correction methodology is also provided for each case separately.

Chapter 9 presents the conclusions that arise from these thesis. In addition, several ideas are provided in case this work will be continued.

Apart from Chapters, six Appendices are also included in order to provide additional and appropriate information.

Appendix I includes the Shop Test measurements, as reported at the official Shop Test report. Several measurements` data that were not needed for the simulation procedure, were omitted.

Appendix II includes the official reported Sea Trials results`.

Appendix III provides the engine`s layout and selected photos of the 6RTA48-T engine in order to have a better understanding about the engine and the space in the engine room.

Appendix IV provides the analytical calculations of several values that were required as input for the simulation model.

Appendix V includes the M/E Performance Data for the selected cases that were simulated, as reported by the Ship.

Finally, Appendix VI provides a rough description of the ISO correction methodology regarding the evaluation of the engine operation. Details are not included due to confidentiality of the data.

CHAPTER 2

BASIC CONCEPTS

2.1 Introduction

This chapter`s objective is to provide the reader brief information about thermodynamic and operating principles of diesel engines which are the basis for this thesis.

2.2 Diesel cycle

The thermodynamic diesel cycle is graphically shown in Fig. 2.1. The ideal diesel cycle consists of an isentropic compression process (1-2), a constant-pressure combustion process (2-3) followed by an isentropic expansion process (3-4), and finally a constant volume exhaust process (4-1). The application of two-stroke mechanical cycle can be related to Fig. 2.1(A) as compression stroke (1-2) and power stroke (2-3-4). Exhaust and scavenging take place in the constant volume process (4-1), theoretically, when the piston is at the bottom-dead-center (BDC) position.

The characteristic feature of the diesel cycle is injection of the fuel, theoretically starting at the end of the compression stroke (i.e. top dead center – TDC), and continuing at such a rate that the burning process proceeds at constant pressure (2-3). Normally only air is in the cylinder during the compression stroke. To initiate combustion, the temperature of compression, T_2, must be above the auto-ignition temperature of the fuel [4].

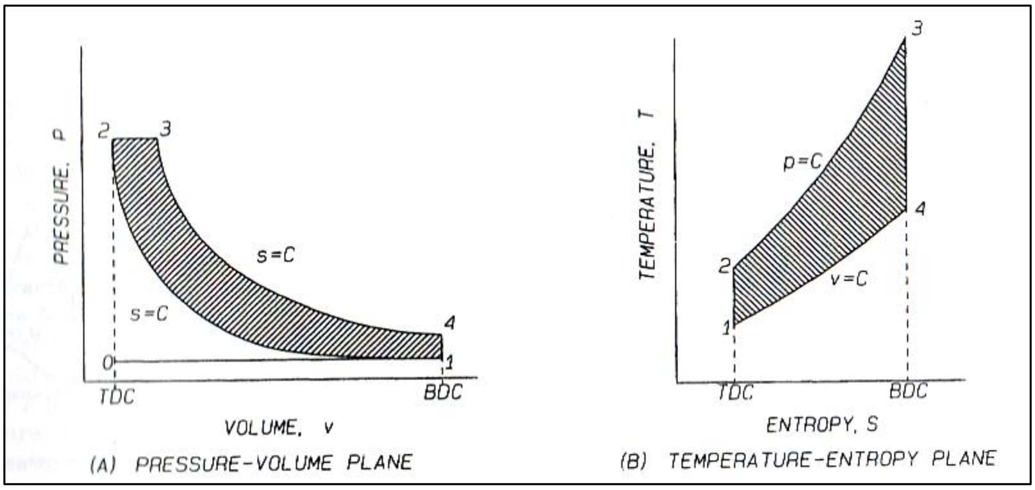


Fig. 2. 1 Diesel cycle

2.3 Diesel cycle thermal efficiency

For a first approximation it is common practice to analyze the diesel cycle using the concepts of the air-standard cycle. In the air-standard cycle the mass and properties of the fuel as injected and the working fluid are assumed to have the properties of air throughout. Furthermore, the heat added is assumed to be the energy released by the reaction of the fuel with the air. Although gas (i.e. air) tables should be used, it is customary to assume constant specific heats for the air.

For isentropic compression:

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{k-1} = T_1 r_c^{k-1} \tag{2.1}$$

where k is the ratio of specific heat at constant pressure $\left(c_p\right)$ of air to the specific heat at constant volume $\left(c_v\right)$ of air. For air the value of k is 1.4. The term r_c is called compression ratio.

The thermal efficiency of the air-standard diesel cycle is:

$$\eta = \frac{W}{Q_A} = \frac{Q_A - Q_R}{Q_A} \tag{2.2}$$

where the work W, is the difference between the heat added, Q_A, and the heat rejected, Q_R. The heat rejection process is a constant-pressure process and consequently $Q_R = U_4 - U_1$. Accordingly the cycle efficiency can be expressed as:

$$\eta = 1 - \frac{Q_R}{Q_A} = 1 - \frac{u_4 - u_1}{h_3 - h_2} = 1 - \frac{c_v \left(T_4 - T_1\right)}{c_p \left(T_3 - T_2\right)} \Rightarrow$$

$$\eta = 1 - \frac{\left(T_4 - T_1\right)}{k\left(T_3 - T_1\right)} \tag{2.3}$$

where u, h is the internal energy and enthalpy accordingly. Application of the isentropic compression equation (2.1) and Charles` Law, $T_3/T_2 = V_3/V_2$, allows T_3 to be written in terms of T_1:

$$T_3 = T_2 \left(\frac{V_3}{V_2} \right) = T_1 r_c^{k-1} \left(\frac{V_3}{V_2} \right) = T_1 r_c^{k-1} r_k \tag{2.4}$$

The ratio $r_k = V_3/V_2$ is called the fuel cutoff ratio. By using the converse of equation (2.1) for isentropic expansion and noting $V_4 = V_1$, T_4 can be expressed as T_1:

$$T_4 = T_3 \left(\frac{V_3}{V_4} \right)^{k-1} = T_3 \left(\frac{r_k V_2}{V_1} \right)^{k-1} = T_1 r_c^{k-1} r_k \left(\frac{r_k V_2}{V_1} \right)^{k-1} \Rightarrow$$

$$T_4 = T_1 r_k^k \tag{2.5}$$

Substituting equations (2.1), (2.4), and (2.5) into equation (2.3) yields:

$$\eta = 1 - \frac{T_1 r_k^k - T_1}{k\left(T_1 r_c^{k-1} r_k - T_1 r_c^{k-1}\right)} = 1 - \frac{1}{r_c^{k-1}}\left(\frac{r_k^k - 1}{k\left(r_k - 1\right)}\right) \tag{2.6}$$

Diesel engines generally have compression ratios above 13:1. Although the power of a diesel cycle increases with increased cutoff ratio, the thermal efficiency of the diesel cycle decreases [4].

2.4 Mean Effective Pressure

As noted above the efficiency of the diesel cycle depends upon the compression ratio, r_c, and the cutoff ratio, r_k. A quantity of special interest to designers and operators of diesel engines is the work, W, done on the piston divided by the displacement volume, $V_1 - V_2$. The dimensions of this quotient are those of pressure. The essence of this quality is an effective constant pressure which, if exerted on the piston for the entire power stroke, would yield work equal to the work of the cycle. This quality is called mean effective pressure, MEP:

$$MEP = \frac{W}{V_1 - V_2} \tag{2.7}$$

The MEP represents the height of a rectangle whose enclosed area is equal to the enclosed area of the *P-V* diagram (1-2-3-4-1) for the diesel cycle illustrated by Fig. 2.1(A). Since it represents the work the gas does on the piston acting through the stroke, it can be related to the power produced by an engine by the expression.

$$P = \frac{p_e L A n}{1000} \tag{2.8}$$

P = power, kW
p_e = MEP, bar
L = length of stroke, m
A = area of piston, m^2
n = number of power strokes per second

In slow-speed and in larger medium-speed engines, the MEP can be obtained by attaching an indicator or an IMEP meter to each cylinder of the engine. The pressure so determined is called indicated mean effective pressure, IMEP. The IMEP is a measure of the energy applied to the engine by the working fluid. A correlative to the IMEP is the brake mean effective pressure, BMEP. The BMEP is found by measuring the engine power with a dynamometer and solving the equation (2.8) for the mean effective pressure. If the BMEP is divided by the IMEP, the result is the mechanical efficiency of the engine.

Examining the P-V diagram for a diesel cycle shows that varying the cutoff ratio, r_k, has a profound effect on the area under the curve and, therefore the mean effective pressure and power produced by the engine [4].

2.5 Actual Diesel Cycle

The initial pressure in the cylinder at the BDC position, P_1, is the charge pressure. In the air-standard analysis, the compression stroke is assumed to commence immediately, as soon as the piston moves away from BDC. The compression process, 1-2, is then assumed to occur adiabatically and isentropically, that is, without friction, turbulence, or heat transfer. The pressure reached at TDC, P_2, is the compression pressure, and the corresponding temperature is the terminal compression temperature. The constant volume process, 2-3, and the constant pressure process, 3-4, are heat-addition process representing the heat released in combustion during the initial process in which fuel burns as it is injected, while the piston begins its descent. The constant pressure, which is maintained from 3 to 4 in the air-standard cycle, is called the maximum pressure. At 4, the addition of heat ceases and the air is assumed to expand isentropically, forcing the piston down to the BDC at 5. A constant volume heat-rejection process, 5-1, representing the exhaust of the hot gases and the charging of the cylinder of fresh air, completes the cycle.

The dashed trace in Fig.2.2 shows the pressure-volume relationship actually achievable in an cycle with the same heat input, and in a cylinder of equal dimensions. The largest discrepancy from the two traces results from the fact that the compression stroke in the actual cycle cannot begin until the air ports or valves are closed, by which time the piston will have traveled 10% to 20% of its stroke [4].

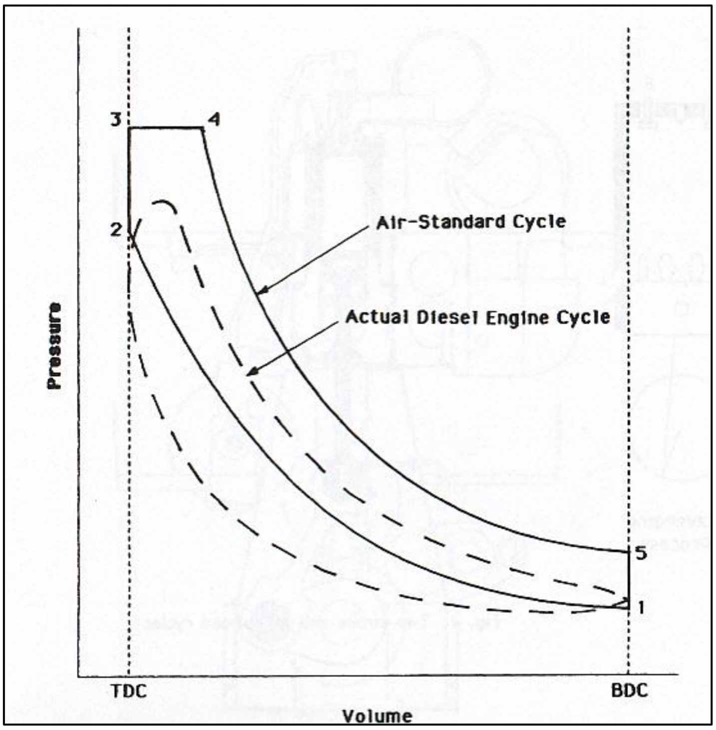


Fig. 2. 2 Pressure – Volume diagrams

2.5.1 Net work and heat addition

The work done and heat transferred in each process in the idealized cycle and be calculated using basic thermodynamic principles. The difference between the heat added (Q_A) during the cycle and the heat rejected (Q_R), illustrates the first Law of Thermodynamics:

$$W_{net} = Q_A - Q_R \tag{2.9}$$

From equation (2.9), the net work increases as the heat added is increased relative to the heat rejected. However, there is an upper limit on the heat that can be added in the cycle, which derived from the amount of air needed for the combustion of the fuel, that is, the air-fuel ratio. This ratio can be calculated from the chemical composition of the fuel and under ideal circumstances where there is just enough oxygen present with no excess, is called stoichiometric air-fuel ratio. For petroleum fuels, the stoichiometric air-fuel ratio is 14 to 15, but because conditions for combustion in diesel cylinders are far from ideal and the exact chemical composition of the fuel is mostly unknown, the actual mass ratio of the air trapped at the start of compression to fuel required for maximum output is generally in the range of 25 to 30.

The maximum mass of fuel that could be burnt in a diesel cylinder during one cycle is equal to the mass of air trapped divided by the trapped air-fuel ratio, AF, i.e.:

$$m_{fuel} = \frac{m_{air}}{AF} \tag{2.10}$$

The upper limit of heat added to the cycle is equal to that released by the mass of fuel, based on the lower calorific value, LCV, of the fuel:

$$Q_A = m_{fuel} LCV \tag{2.11}$$

The limit on heat (fuel) that may be added per cycle manifests itself in engine operation by the tendency of engine to exhaust smoke when loaded to the limits of their output at any given RPM. The smoke indicates that more fuel is being injected per cycle than can be completely burned in the air in the cylinder. Accordingly, a theoretical limit exists to the output of an engine at any RPM, based on the air available in the cylinders, and is independent of the mechanical strength of engine components [4],[5].

2.5.2 Mean indicated pressure, mean effective pressure and brake horse power

As mentioned in (2.3.2), the pressure in the cylinder, when averaged over the entire cycle, including the compression stroke as well as the power stroke, is the mean indicated pressure, i.e. IMEP. Graphically, with reference to Fig. 2.2, IMEP can be calculated as proportional to the area enclosed within the pressure-volume trace, by the length of the trace, which represents the displacement volume.

$$IMEP = \frac{W_i}{V_{disp.}} \tag{2.12}$$

The brake mean effective pressure (BMEP) in the cylinder can be calculated as mentioned in (2.3.2). It should be mentioned that engine manufacturers provide specific methodology for the calculation of the engine output. In any case, the brake horse power (BHP) is related to the brake mean effective pressure (BMEP), as shows the following equation:

$$BHP = \frac{V_h\left(BMEP\right)}{1000}\frac{N}{60} \tag{2.13}$$

BHP = brake horse power, kW
V_h = cylinder disp. volume, m^3
$BMEP$ = brake mean effective pressure, bar
N = engine rotational speed, RPM

Because the aforementioned limit on heat addition also limits the IMEP and therefore, the BMEP, and since the torque is directly proportional to the BMEP, diesel engines are torque limited [4],[5].

2.5.3 Efficiency

The thermal efficiency has be mentioned above in the equation (2.2). The mechanical efficiency is the ratio of IMEP to the BMEP and shows the fraction of power which is lost due to friction of the engine components:

$$n_{mech.} = \frac{BMEP}{IMEP} \tag{2.14}$$

2.6 Combustion

Fuel combustion does not take place at the tip of the injector, but rather at a distance away from it. This delay occurs because the individual fuel droplets must diffuse through the hot cylinder contents for a sufficient time to heat, vaporize, mix with air, and finally ignite. The combustion process in a diesel cylinder is considered to occur in four phases, which begin during the compression process and end during the expansion process, as shown in Fig. 2.3. The four phases are the ignition-delay period (when no combustion occurs), the rapid combustion period (which begins with ignition), the steady-combustion period, and the afterburning period [4].

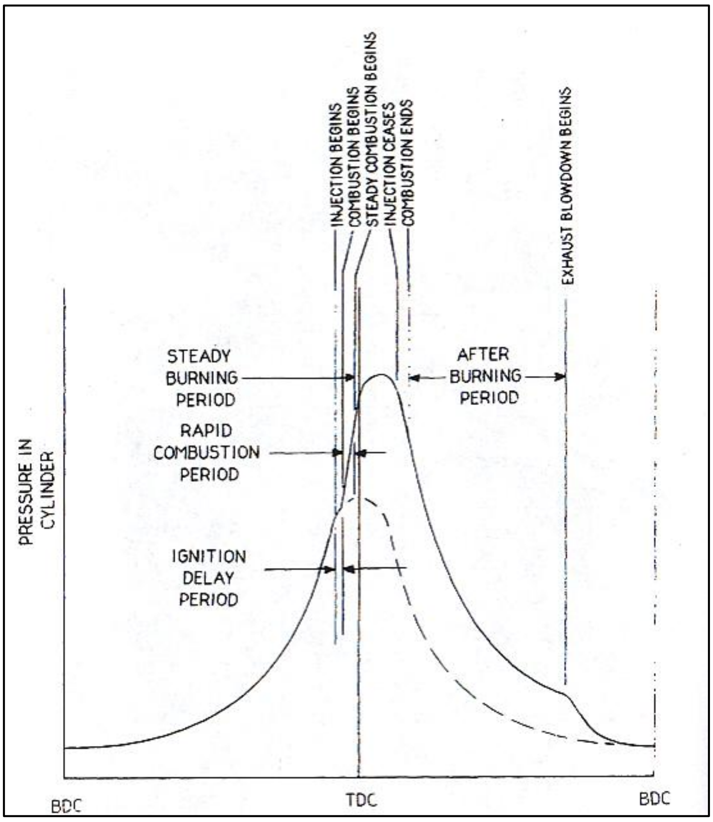


Fig. 2. 3 Phases of combustion

2.6.1 The ignition-delay period

The ignition-delay period is the interval between injector opening and the start of ignition. During this phase the first droplets to enter the cylinder are heated by the surrounding charge of compressed air as they disperse and vaporize. Until ignition occurs, there is no increase in the cylinder pressure above what it would have been had injection not occurred.

The ignition-delay period is primarily a function of the ignition quality of the fuel, which is related to its chemical composition. Fuels of lower ignition quality require more preparation time, and the delay period is therefore longer.

2.6.2 The rapid-combustion period

During the rapid-combustion period, the fuel that has accumulated in the cylinder during the delay period ignites and burns rapidly. Because the accumulated fuel has already mixed with the charge air, this phase is sometimes called the premixed combustion period. The rapid combustion is accompanied by a sharp rise in cylinder temperature and pressure. If the pressure rises too sharply, the combustion becomes audible, a phenomenon known as diesel knock.

2.6.3 The steady-combustion period

Once combustion has been established in the cylinder, the ignition of further fuel droplets entering the cylinder lags the injection rate by the time required for the fuel to mix, heat, and vaporize. Because the droplets burn as they diffuse into the cylinder, this phase is sometimes called the diffusion combustion period. This period ends shortly after the injector closes, when the last of the fuel has burned. The cylinder pressure usually peaks just beyond the TDC position and near the middle of the steady-combustion period. The cylinder pressure then declines as the expansion process proceeds.

2.6.4 The afterburning period

If all of the fuel has burned cleanly and completely by the end of the steady-combustion period, the pressure profile will be smooth through the expansion stroke. Typically, however, there will be some pressure fluctuations resulting from the combustion of incompletely burned fuel or of intermediate combustion products, and some delayed chemical reactions. It is during this period that soot and other pollutants are produced.

2.7 Limits of Engine Performance

Limits defining the operating envelope for an engine are identified in Fig. 2.4. Engine rpm is limited by the mechanical stress on running gear, by bearing loads, and by wear rates of piston rings and cylinder liners. The turbocharger overspeed limit is usually determined by the turbine blade root strength. The setting of the engine overspeed governor or trip is usually 115% to 120% of rated rpm. Except when an engine that drives a fixed-pitch propeller is run at a modest overspeed on trials in order to maximize the load, operation beyond rated rpm is unusual [4].

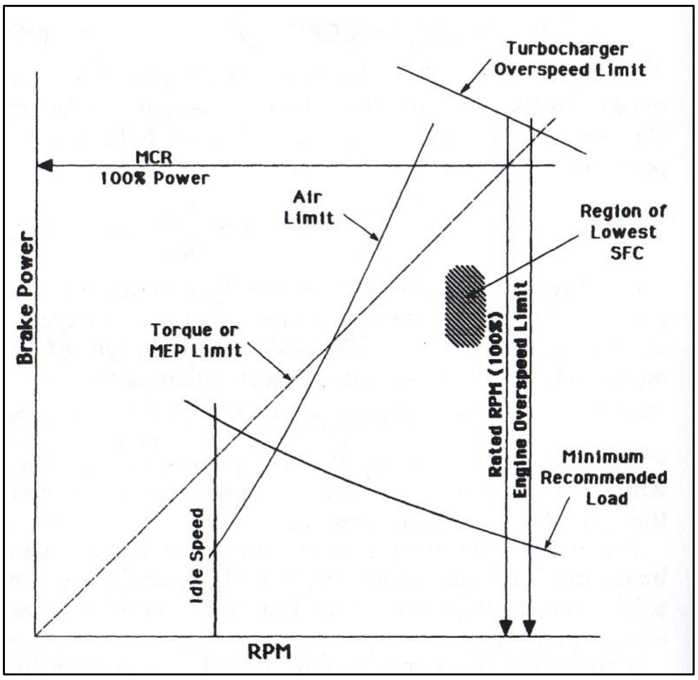


Fig. 2. 4 Engine Performance limits

Operation beyond the limits of rated power (MCR) and rated MEP constitutes an overload. Overload operation results in increased stresses and higher temperatures for the cylinder heads and liners and for the piston crowns, and is a condition often referred to as high thermal load. However, overload operation causes increased mechanical loadings on other engine components as well. Cylinder liner and piston ring wear can be expected to increase, and the lives of bearings and exhaust valves to be reduced, as a result of engine overload. The most convenient indication of high thermal load is the cylinder exhaust temperature since it is correlated well with MEP and is measured at quite often intervals.

The air limit in turbocharged engines is set by applying a margin to the surge limit of the turbocharger. Operation to the left of the air limit curve in Fig. 2.4 incurs an increased likelihood of combustion chamber and turbine fouling and smoke emission.

The engine minimum speed (for engines that are directly connected to propellers) is typically 25% to 40% of rated rpm. At lower engine speeds, the leakage of air passed the piston rings during compression stroke can result in low temperatures at the time of injection, erratic combustion, and an accumulation of unburned fuel and carbon in the exhaust system.

The bottom of the operating envelope is a light-load limit. Sustained light-load operation, particularly with heavy or sulfur-bearing fuels, is not recommended because combustion chamber temperatures at low loads are likely to be too low to effectively burn some fuel constituents. Fouling of the combustion areas, exhaust path, and turbine can result. In addition, the low temperatures toward the bottom of the cylinder liners can cause condensation of sulfur compounds in the exhaust, leading to sulfuric acid attack.

Limited operation outside the operating envelope will generally result in decreased component durability, which is reflected in increased requirements for inspection and maintenance. A catastrophic failure of a properly maintained engine under these conditions is unlikely because of the design margins, and because periodic scheduled inspections reveal such effects as burning, cracking, or distortion in time for component renewal. Most manufacturers consider a certain amount of operation beyond the envelope in the directions of high MEP, high power, and with marginal combustion air inevitable, and take this into account in their service recommendations. Statements permitting limited operation in these regions, limited to perhaps one hour in ten or twelve, or a cumulative total of perhaps 500 to 2000 hours per year, are typical [4].

2.8 Shop Tests and Sea Trials

The construction of a ship is concluded by a broad array of tests to demonstrate that the ship meets contract requirements. Those of tests that are directly related to the Main Engine are Shop Tests and Sea Trials.

Tests are preferably scheduled as early as feasible during the ship construction process, because early testing allows more time to evaluate and develop resolutions for design or material problems with a minimal disruption to ship construction. Shop tests for purchased equipment are advantageously conducted at the manufacturer's facility, where any corrections or adjustments can be expeditiously handled.

Shop tests are conducted for purposes such as confirming that assemblies are correctly built, verifying strength and tightness requirements, and demonstrating that controls and safety devices are functional and properly adjusted.

Particularly, the effective output (BHP) at various loads of the 6RTA-48T at Shop Test, was determined by a water brake. The Shop trial was carried out with marine diesel oil and several other tests were accomplished, such as:

- Starting and astern test
- Running test for 100%, 85%, 75% and 50% load.
- Governor test
- Fuel consumption measurement evaluated with 42,700 kJ/kg low calorific value.

All these measurements and adjustments, were recorded and filed; these data are used as reference for the overall evaluation of the engine`s condition during its lifetime.

Before the ship`s delivery, Sea Trials are conducted to demonstrate the performance and adequacy of those aspects of the ship that cannot be tested ashore. These tests are categorized in two parts, the Hull part and the Machinery Part. Sea Trials Hull part includes data of the ship performance during several tests (such as deck machinery test, steering gear tests etc) , whereas the Machinery part includes Main engine performance data. During Sea Trials of the 6RTA48-T, heavy fuel oil was used for the main engine. Sea trials included a series of tests for all the equipment installed in the ship; as far as the machinery part is concerned, speed trial and endurance test was carried out. During speed trial the contract ship speed was achieved at normal engine output. The endurance test lasted four hours and main engine was operating at normal output (NCR), which was measured by a power meter. Apart the already mentioned tests, other tests (in machinery part) were also conducted, such as:

- Main engine starting test
- Minimum shaft speed test
- Automation and remote control test
- Crash stop astern test
- Diesel generators auto changing test

The data collected during all the tests, were published officially to the owner by the builder up to the ship`s delivery.

CHAPTER 3

THE 6RTA48-T MAIN ENGINE

3.1 Introduction

This chapter is intended to provide a brief description of the diesel main engine 6RTA48-T with emphasis on the engine parts that are simulated in the MOtor THERmodynamics program.

3.2 General description

The RTA engine is single acting, which means that the combustion exists only at the one end of the cylinder, two-stroke Diesel engine of crosshead design with exhaust gas turbocharging and uniflow scavenging. Tie rods bind the bedplate, columns and cylinder block together. Crankcase and cylinder block are separated from each other by a partition which incorporates the sealing gland boxes for the piston rods [6].

The exhaust gases flow from the cylinders through the exhaust valves into an exhaust gas manifold. The exhaust gas turbocharger work on the constant pressure charging principle. The exhaust valves are opened hydraulically and closed pneumatically [6].

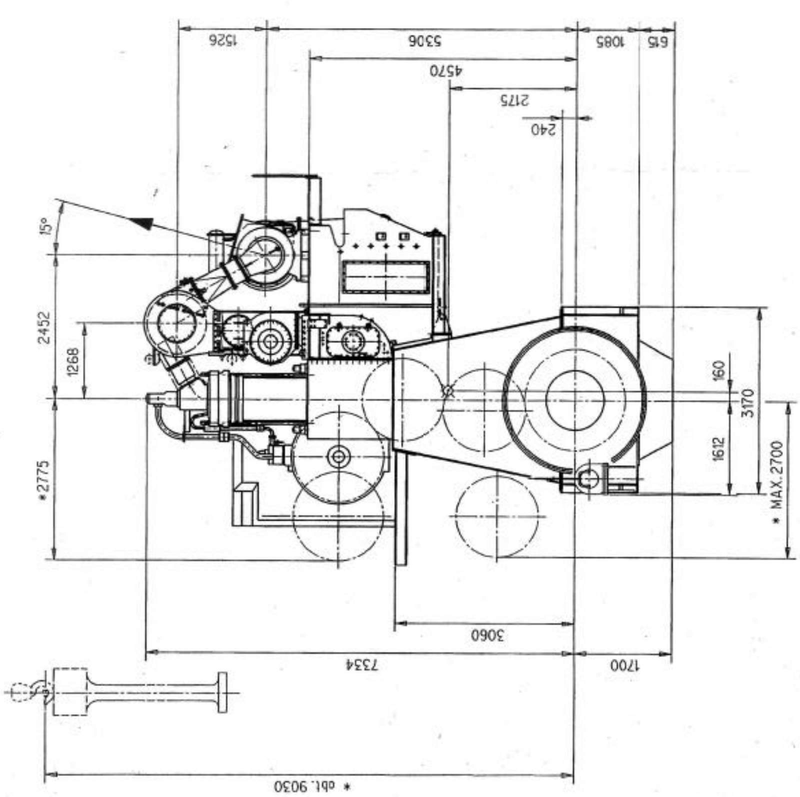
The charge air delivered by the turbocharger flows through air cooler and water separator into the air receiver. It enters the cylinders via valve groups though the scavenge ports when the pistons are nearly the BDC. At low loads independently driven auxiliary blowers supply additional air to the scavenging air space [6].

The cylinders and cylinder heads are fresh water cooled. The air-cooler is using sea water for cooling the charge air. The working pistons are cooled by bearing lubricating oil [6].

The camshaft is driven by gear wheels from the crankshaft [6].

The engine outline drawing (Fig. 3.1) is attached at the following page [7].

Longitudinal, cross section plans and photos of the installation, are attached at Appendix III.



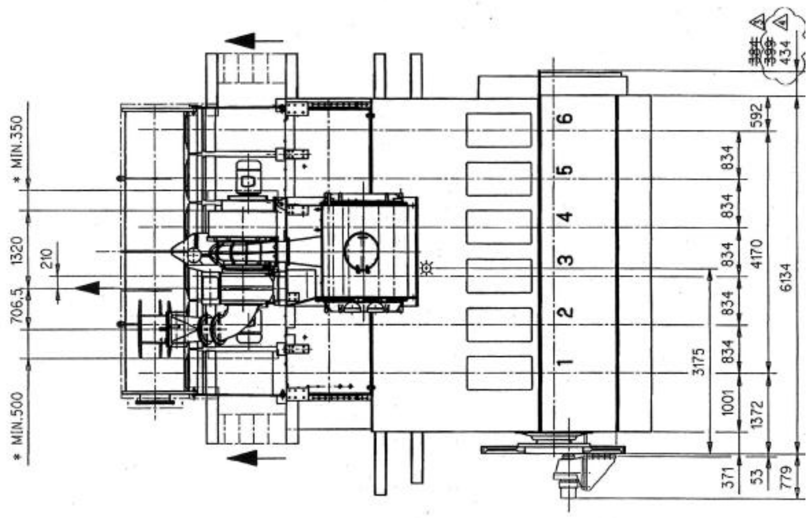


Fig. 3. 1 Engine Outline

3.3 Fuel Injection Valve

In each cylinder cover two injection valves are fitted. The fuel oil is supplied to the injection system through the booster fuel pump. The fuel quantity required for the injection flows through connection 'BH' and the bore to nozzle body 3. The high fuel pressure lifts nozzle needle 4 off its seat against the adjustable force of compression spring 9 and injection into the combustion space results [6]. The fuel injection arrangement is shown at Fig. 3.2.

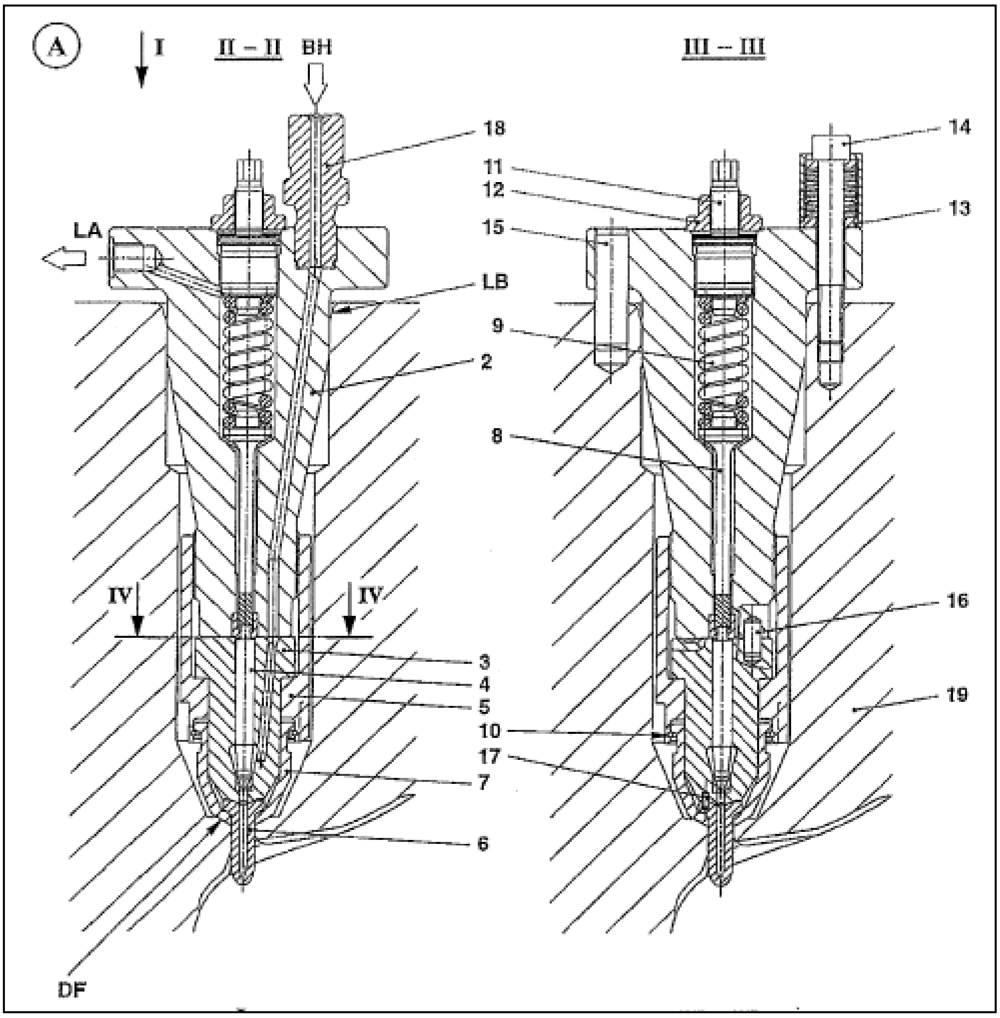


Fig. 3. 2 Fuel Injection valve

Key to illustration:

1 Fuel injection valve
2 Nozzle holder
3 Nozzle body with needle seat
4 Nozzle needle
5 Cap nut
6 Nozzle tip
7 Retaining sleeve
8 Tappet
13 Tension washer cage with cup springs
14 Screw
15 Dowel pin for nozzle holder
16 Dowel pin for nozzle body
17 Dowel pin for nozzle tip
18 Double nipple
19 Cylinder head

9 Compression spring
10 Snap ring
11 Spring tensioner
12 Collar nut

BH Fuel feed (high pressure)
LA Leakage fuel drain
LB Leakage fuel outlet (gap)
DF Sealing face

3.4 Exhaust valve

The exhaust valve cage is screw fastened in the centre of the cylinder cover. It consists of the following parts: valve drive 1, valve cage 2, exhaust valve spindle 3, valve seat 14 and air spring 'LF' (see illustration below). The exhaust valve opening is controlled by an actuator pump which presses hydraulic oil through the hydraulic oil connection 12 into valve drive housing 1. Piston 9 is moved downwards in cylinder 8. The exhaust valve spindle 3, with air spring piston 5 fastened to it, is also pushed downwards against pressure in the air spring 'LF' [6]. The exhaust valve opens and this is happening at 132 degrees [9] in crank angle. The exhaust gas outflow hits the rotation wings 15, thereby rotating the exhaust valve spindle. The exhaust valve`s spindle stroke is 71mm at the fully open position and its diameter is 48mm.

The exhaust valve closes at 250.8 in crank angle degrees, at the time that the hydraulic oil pressure from the actuator pump diminishes so the spindle 3 is pressed upwards by the pressure in the air spring 'LF' acting on the air spring piston. The hydraulic oil in valve drive 1 is pressed back to the actuator pump [6].

An illustration of the exhaust valve is attached at the next page.

Key to illustration:

1 Valve drive
2 Valve cage
3 Exhaust valve spindle
4 Valve guide bush
5 Air spring piston
6 Pressure flange
7 Air spring cylinder
8 Hydraulic cylinder
9 Hydraulic piston
10 Damper
11 Valve cone pieces (two-part)
12 Hydraulic oil connection
13 Vent screw with coarse filter
14 Valve seat
15 Rotation wings
16 Cylinder cover
17 Exhaust valve
18 Fuel injection valve
19 Locating pin

HO Hydraulic oil
LO Leakage oil
LE Air inlet to air spring
OV Oil supply to valve guide
LF Air spring
EB Inlet bore to air spring
AG Exhaust gas from cylinder
KA Cooling water outlet
LS Leakage oil collecting space
VB Connecting bore

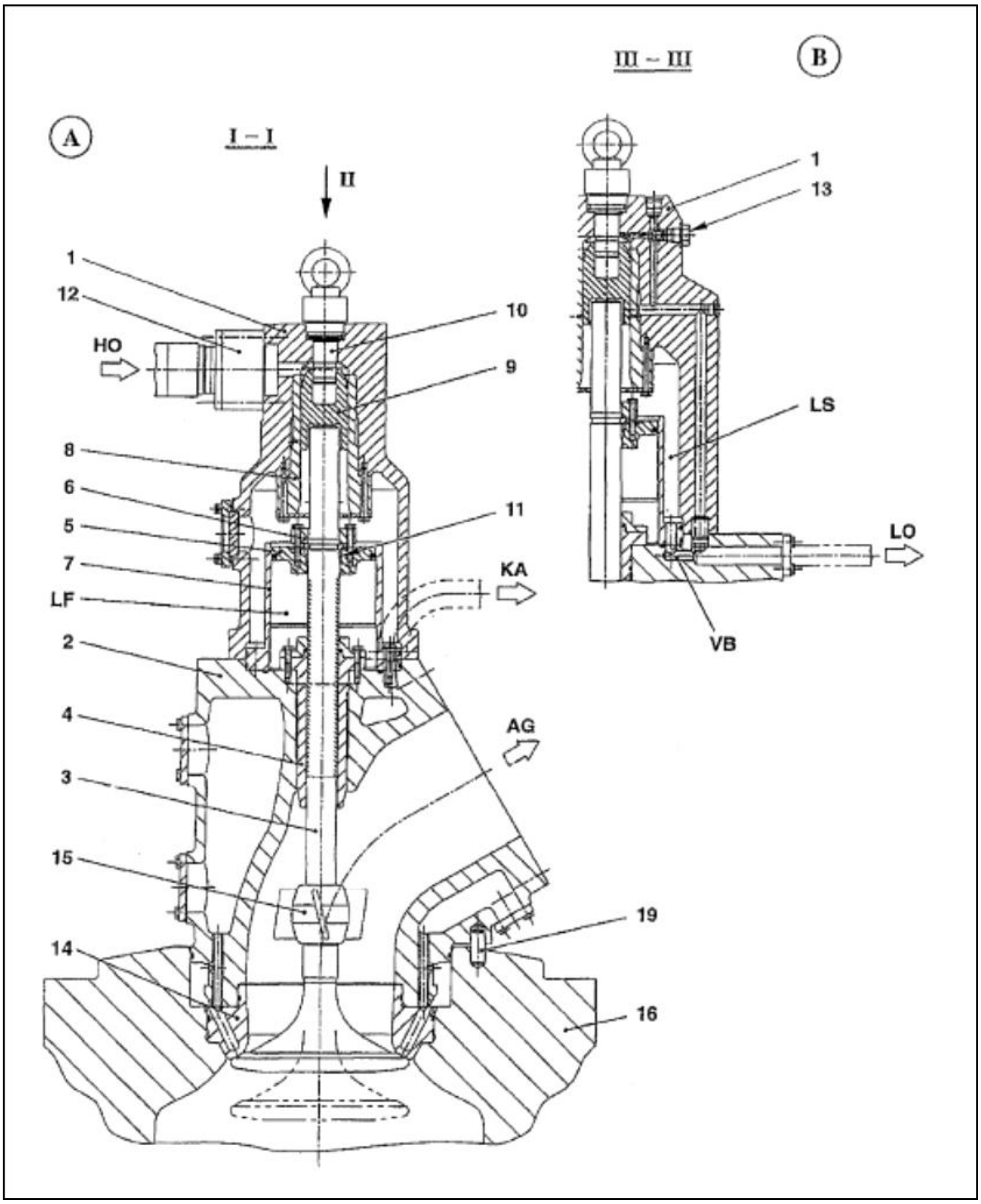


Fig. 3. 3a Exhaust valve

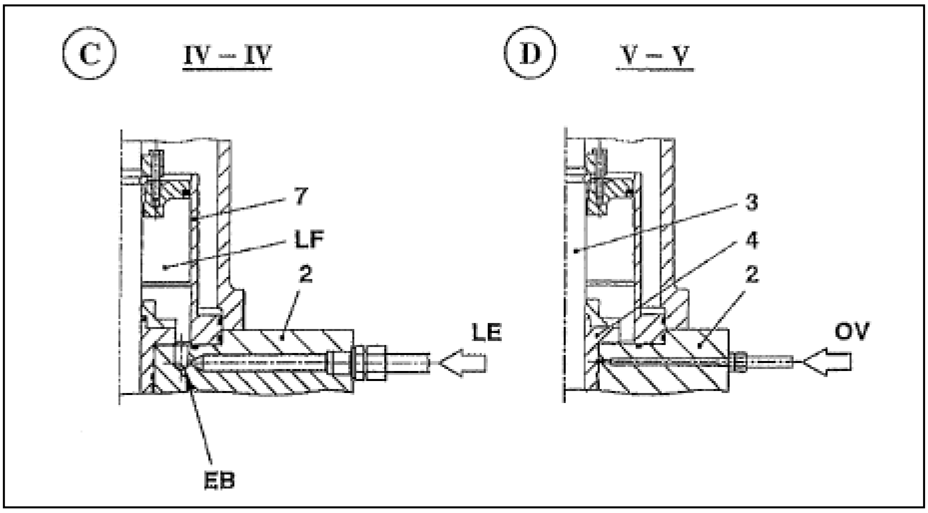


Fig. 3. 4b Exhaust valve

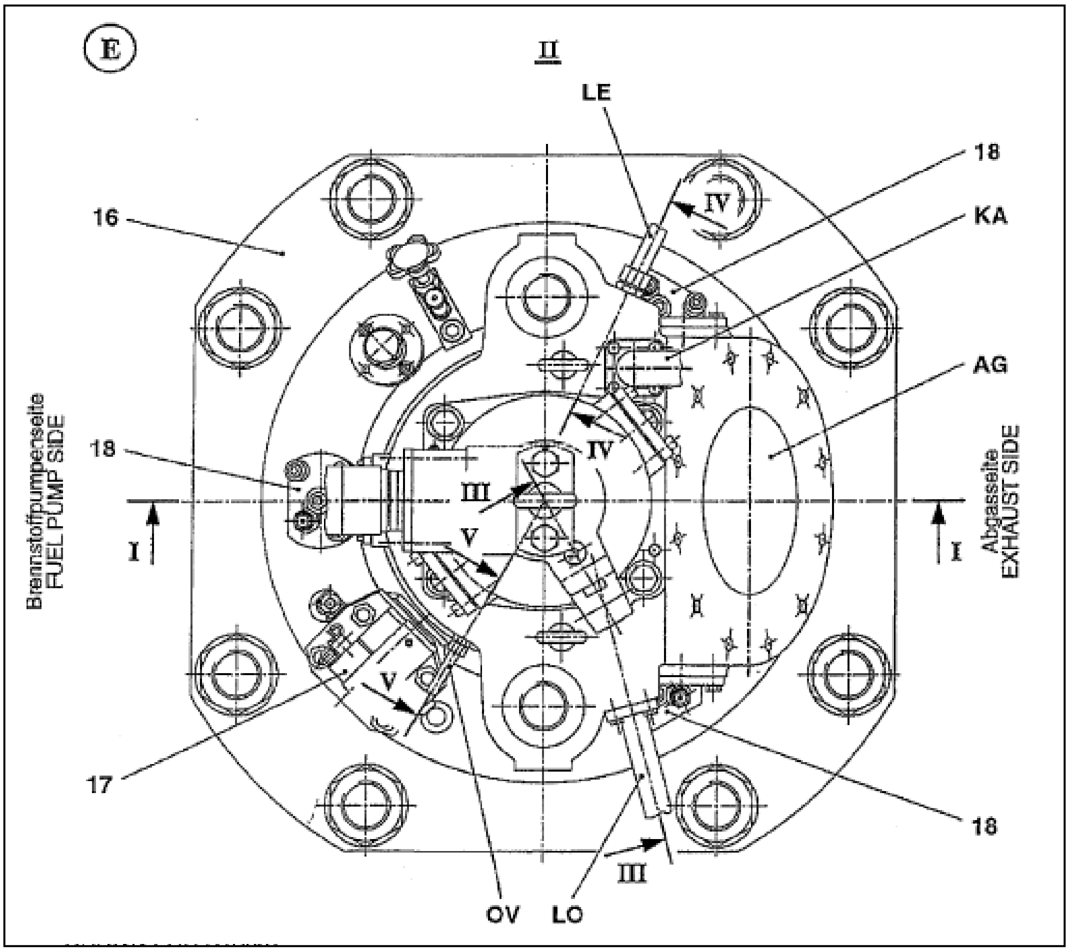


Fig. 3. 5c Exhaust valve

3.5 Scavenge ports

Scavenge ports are situated at the lower part of cylinder liner. Each cylinder liner has 30 scavenge ports through which fresh air fills the combustion chamber. The ports open at 154 crank angle degrees and close at 206 degrees respectively. The BDC is 169mm below the port base. The exact calculation of these data is shown in Appendix IV. The liner`s and ports` outline is shown at Fig. 3.4 [6].

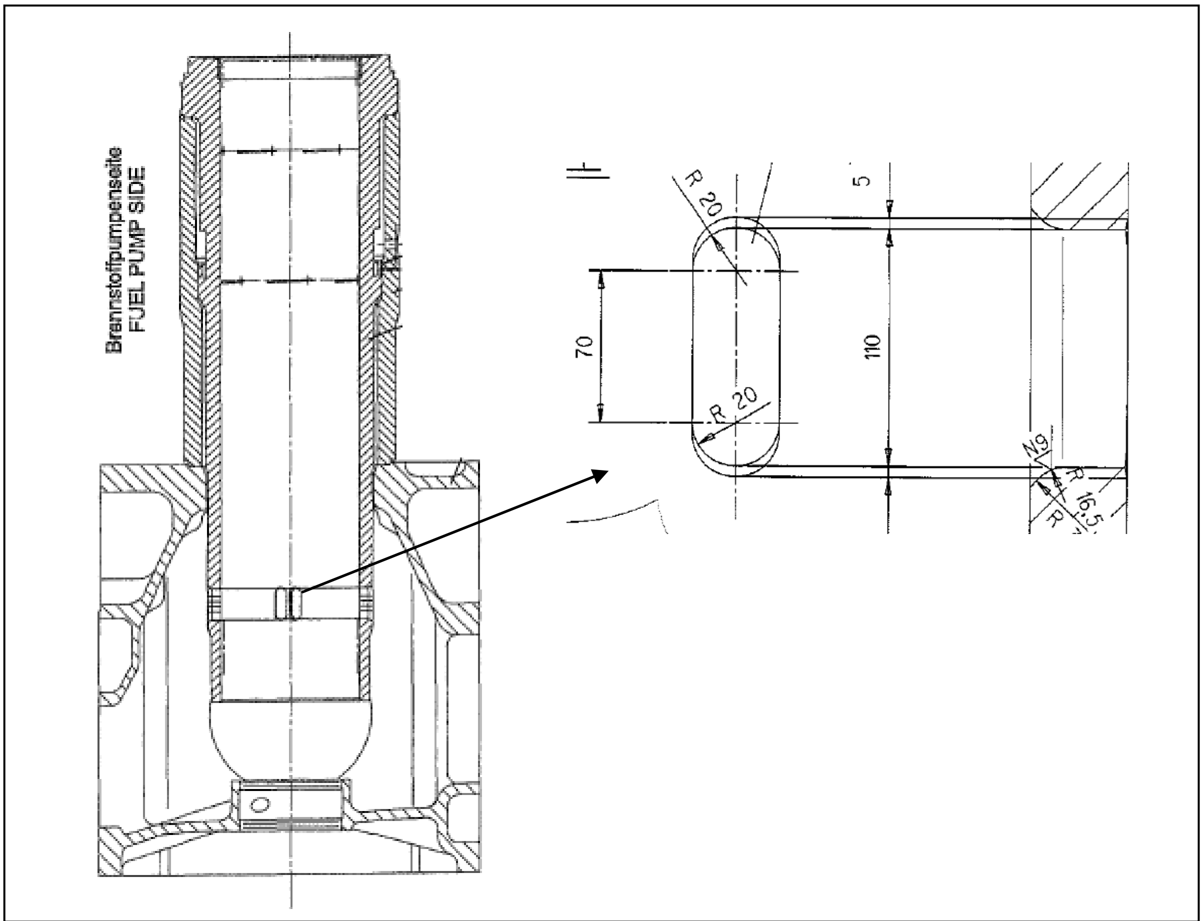


Fig. 3. 6 Cylinder Liner and Scavenge Ports

3.6 Combustion chamber

The compression chamber`s volume determines the compression ratio and consequently the compression pressure. This volume is adjustable through a compression shim that could be placed the piston rod and the crosshead pin. The following illustration shows this position [6].

Key to illustration:

1 Piston crown
2 Top piston ring
2a Piston rings
3 Piston skirt
4 Spray plate
5 Piston rod
6 Oil pipe to spray plate
7 Compression shim
8 Crosshead pin
9 Elastic bolt
EN Inlet slot (groove)
KO Piston cooling oil
OR Oil return from working piston
SO Spray oil

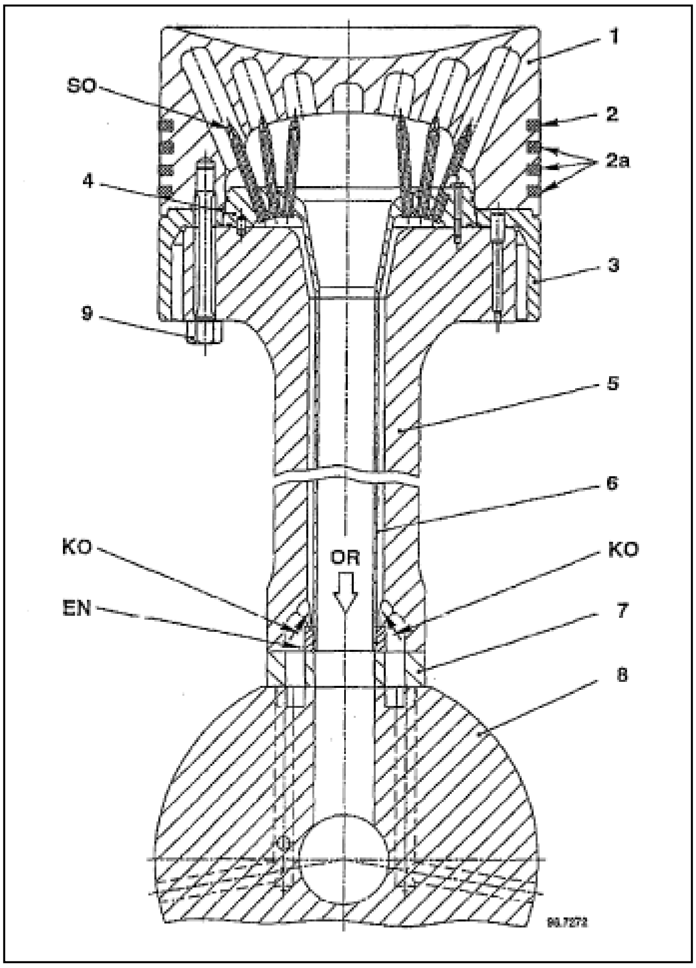


Fig. 3. 7 Working Piston

The 6RTA48-T`s compression shim thickness is 2.5mm.

Fig. 3.6, at the following page, shows the relation between the shim thickness and the compression ratio, and Fig. 3.7 illustrates the combustion chamber [8]. Distances are not included due to confidentiality of the data.

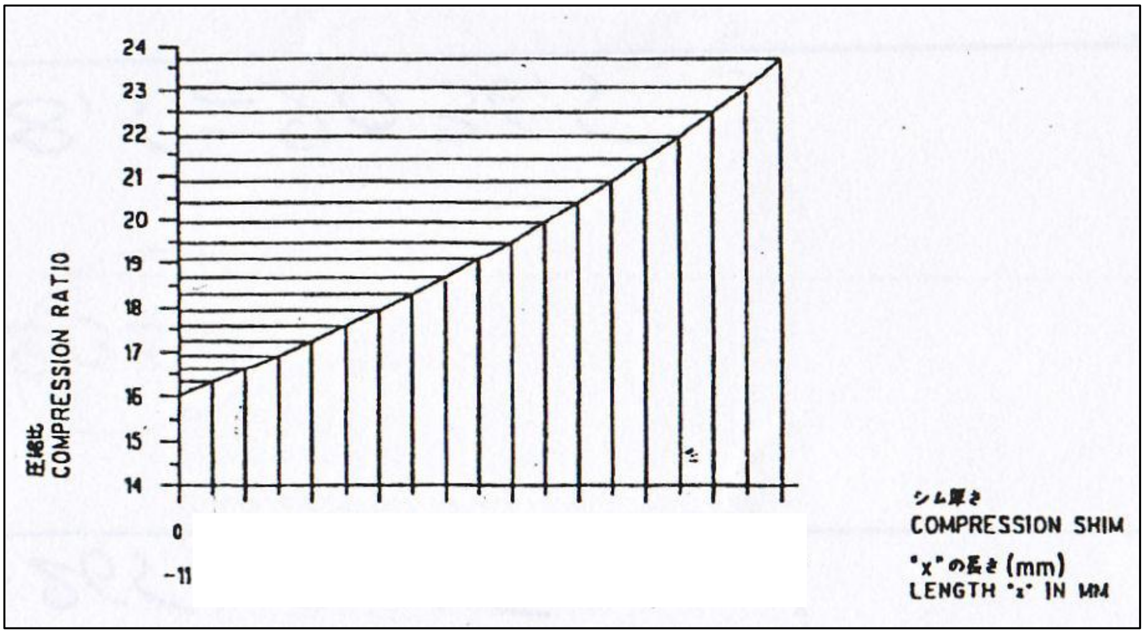


Fig. 3. 8 Compression ratio vs. Shim thickness

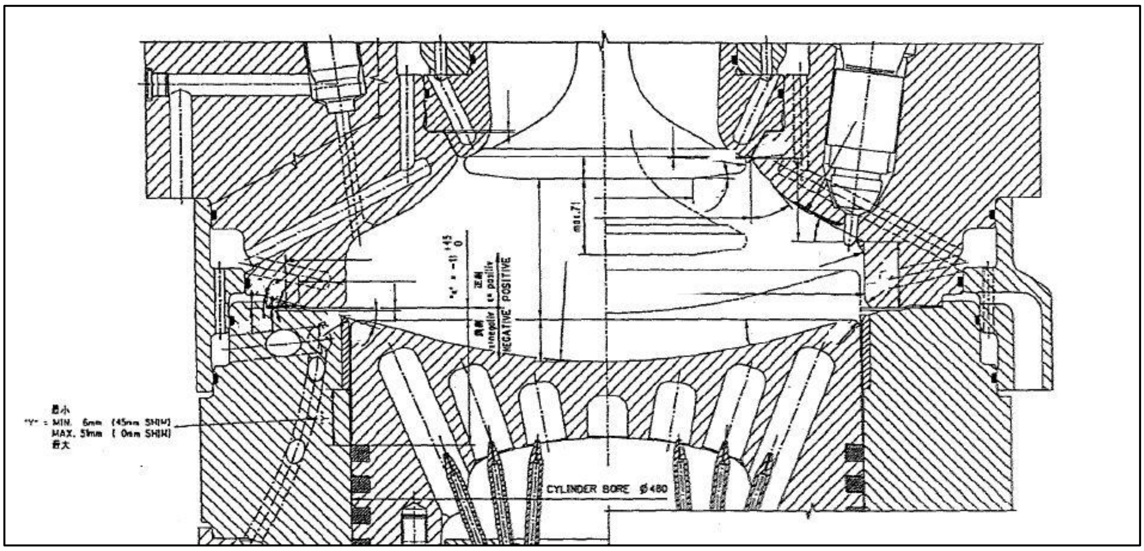


Fig. 3. 9 Combustion chamber

3.7 Inlet and exhaust receiver

Both the inlet and the exhaust receiver are situated at the exhaust side of the engine [6]. The compressed air passes through the air cooler and the water separator and fills the inlet receiver. Then as scavenge ports open, the charge air enter the combustion chamber and the compression stroke begins. The inlet receiver has a total volume of 3.70m^3.

The exhaust gases from each cylinder are collected in the exhaust receiver and afterwards they expand at the turbocharger`s turbine before they leave to the funnel. The purpose of exhaust receiver is to convert the gases kinetic energy (when the leave the cylinder) into pressure (before they enter to the turbocharger). The exhaust receiver`s volume is approximately 3.90m^3.

Volume calculations for both inlet and exhaust receivers are included in Appendix IV.

Here below are shown the inlet and exhaust receiver`s drawing.

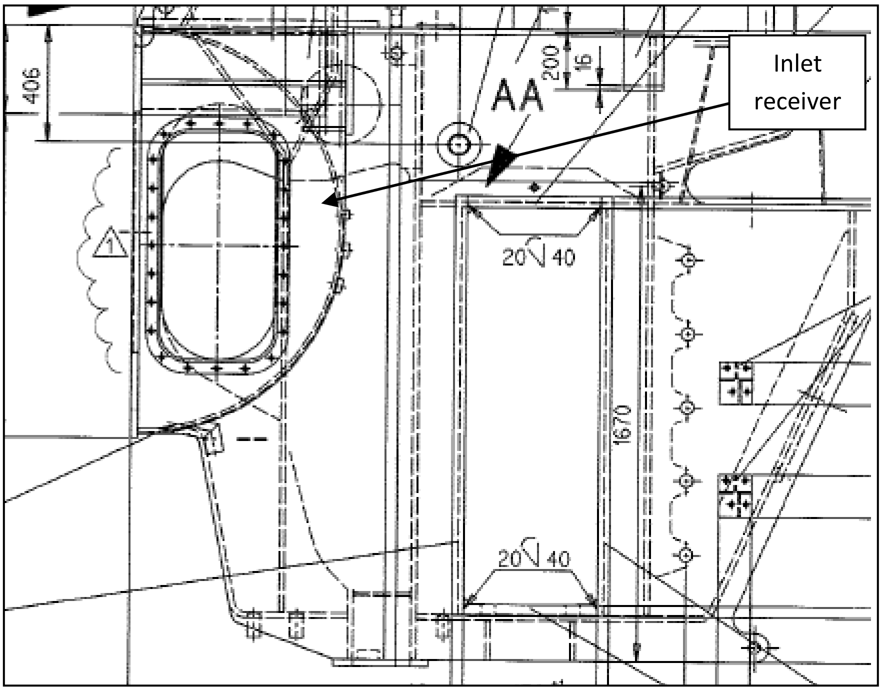


Fig. 3. 10 Inlet receiver

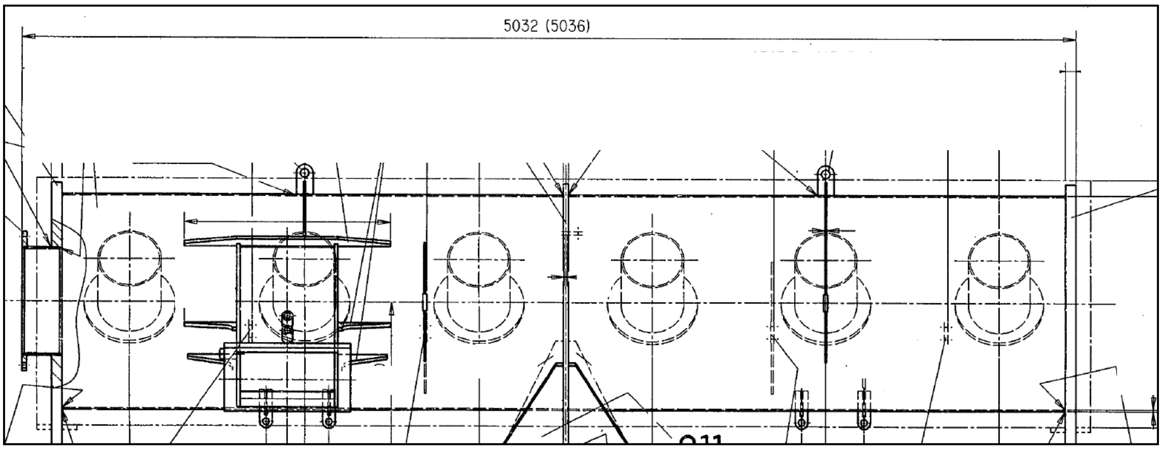


Fig. 3. 11 Exhaust receiver

3.8 Air cooler

The air cooler is installed downstream the compressor and its purpose is to reduce the charge air temperature in order to succeed higher air density compared to those without air cooler. It should be noted that as the air is compressed, its temperature rises significantly. Thus, the air cooler has an important role for the overall engine performance. The RTA48-T` air cooler uses sea water as cooling mean which passes through an array of tubes; these tubes have been positioned vertically to the air flow. The result of such an arrangement is to succeed high cooling surface at the minimum width and height. The following drawing shows the air cooler`s layout [11].

Key to illustration.

1 Tube, Fin
2 Fixed tube plate
3 Floating tube plate
4 Nozzle header
5 Return header
6 Frame
7 Support beam
8 Nameplate
9 Manometer
10 Cock for manometer
11 Bush
12 Plug gasket
13 Plug

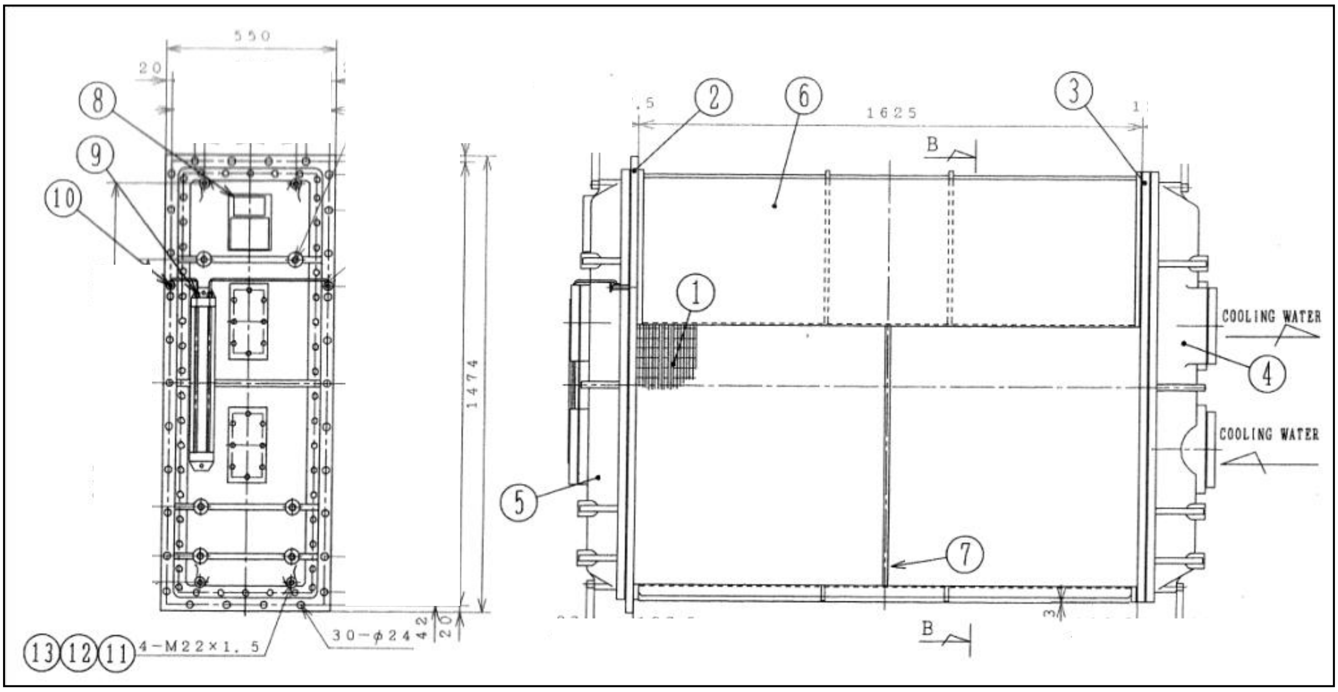


Fig. 3. 12 Air cooler

3.9 Turbocharger

The turbocharger (T/C) type installed to the 6RTA48-T main engine, is the TPL73-B12 (ABB). This type is one of a series of a novel design in turbochargers which was firstly introduced in 1999. Compared to older turbocharger types, this one provides higher compressor efficiencies and higher reliability. The following figure illustrates the t/c`s outline.

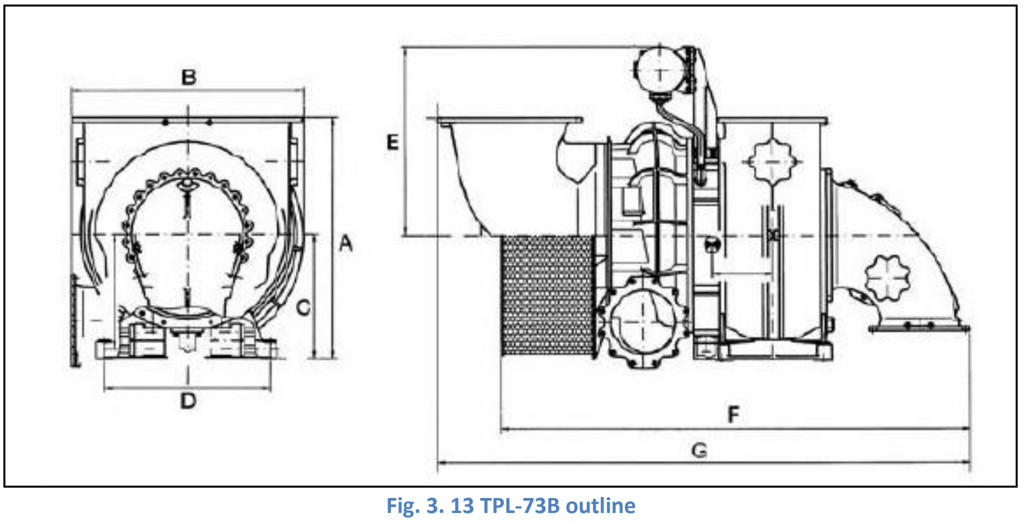


Fig. 3. 13 TPL-73B outline

Key to illustration.

A = 1192mm	D = 822mm	G = 2622mm
B = 1168mm	E = 976mm	Weight (*)= 2510kg
C = 616mm	F = 2287mm	(*) including filter silencer

Fig. 3.12 illustrates the compressor`s impeller diameter (dimension ‘A’) and the turboshaft`s length (dimension ‘B’).

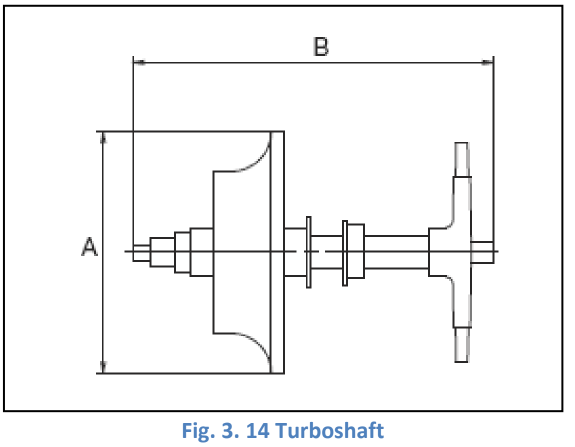


Fig. 3. 14 Turboshaft

Key to illustration.

A = 507mm	B = 963mm	Weight = 115kg

3.10 VIT and FQS

The injection timing is influenced by the fuel injection pumps and their relevant settings and also by the Variable Injection Timing (VIT) and the Fuel Quality Setting (FQS). VIT`s and FQS`s intention is to keep the 100% firing pressure in the range of CMCR (Contracted Maximum Continuous Rating) above 85%. VIT is controlled by a function which uses as input signal the scavenging air pressure and main engine speed, whereas the FQS is manually set according to the fuel quality in use. The fuel quality mechanism adjusts the injection timing by the value that is set manually. When using a fuel having poorer ignition qualities at the same injection timing, the maximum cylinder pressure will drop; on the another hand, the maximum cylinder pressure will rise with a fuel with better ignition quality. In case of a fuel with low ignition qualities, the maximum cylinder pressure can be raised by moving the FQS mechanism in the positive direction (i.e. ''+''), which means that the fuel with be injected earlier in crank angle degrees. By moving the FQS mechanism in the opposite direction (i.e. ''-''), as in case of a fuel with rich ignition quality, the fuel is injected later and the maximum pressure is then reduced. The reference graphs below illustrate the function of the VIT on the maximum pressure in relation with the engine output [12].

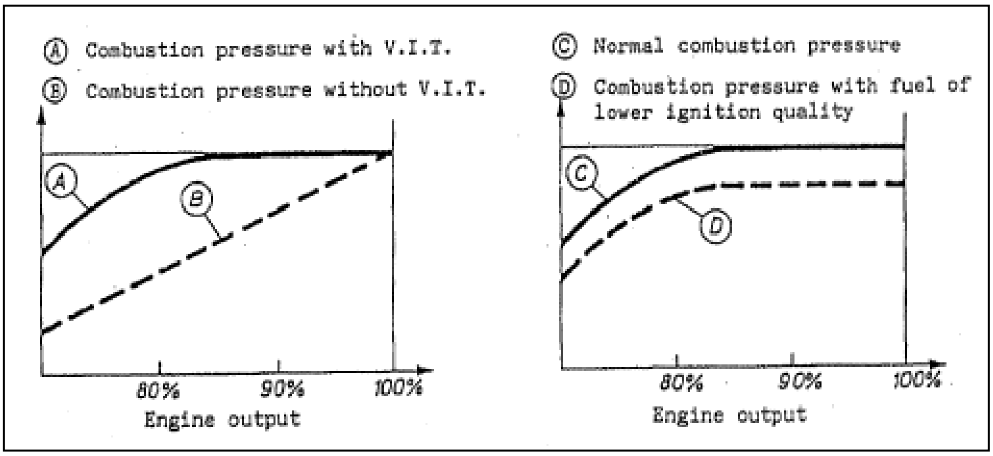


Fig. 3. 15 Function of the VIT

Furthermore the VIT & FQS mechanism allows the user to disable the VIT and set a desired value; such an adjustment is been made in case of a serious engine malfunction and helps the operator to determine the cause of the problem [8].

The scavenge air pressure and the engine revolution values are converted to dimensionless signals as shows the following illustration [12].

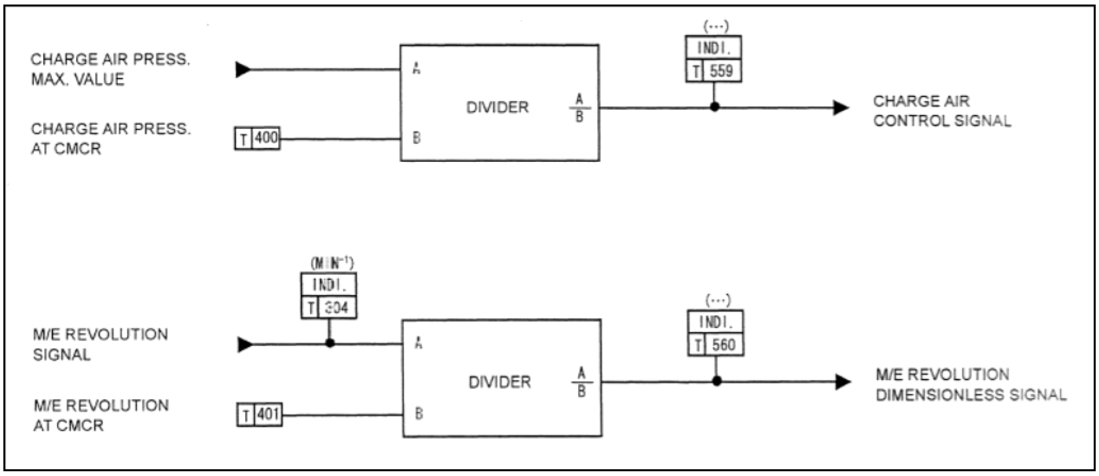


Fig. 3. 16 Charge air and engine speed signals

These signals are converted to crank angle degrees according to specific functions. Illustrations are attached below [12].

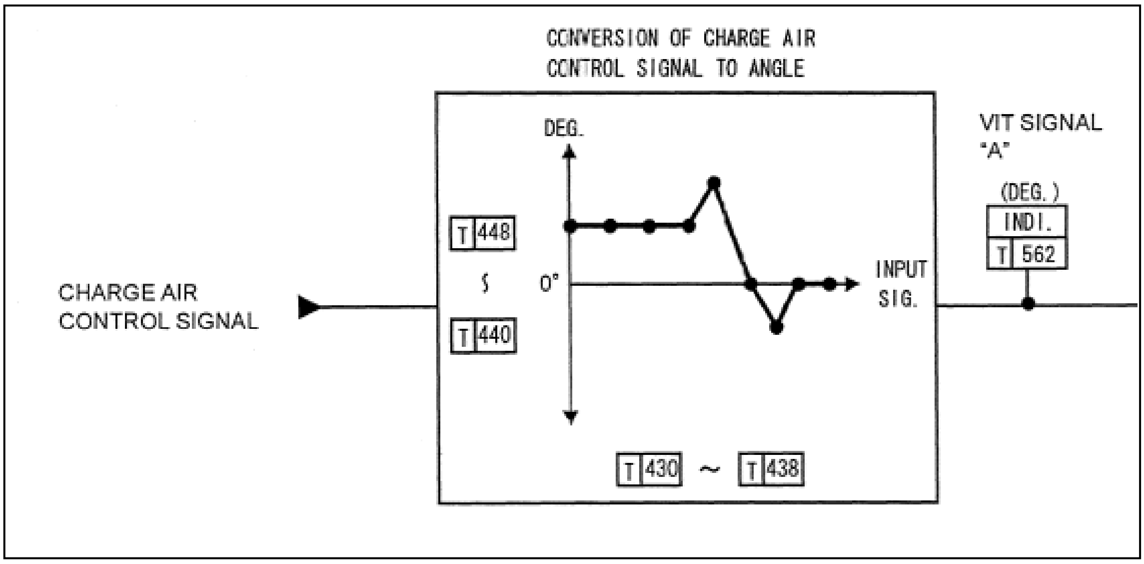


Fig. 3. 17 Charge air signal vs. Crank angle degrees

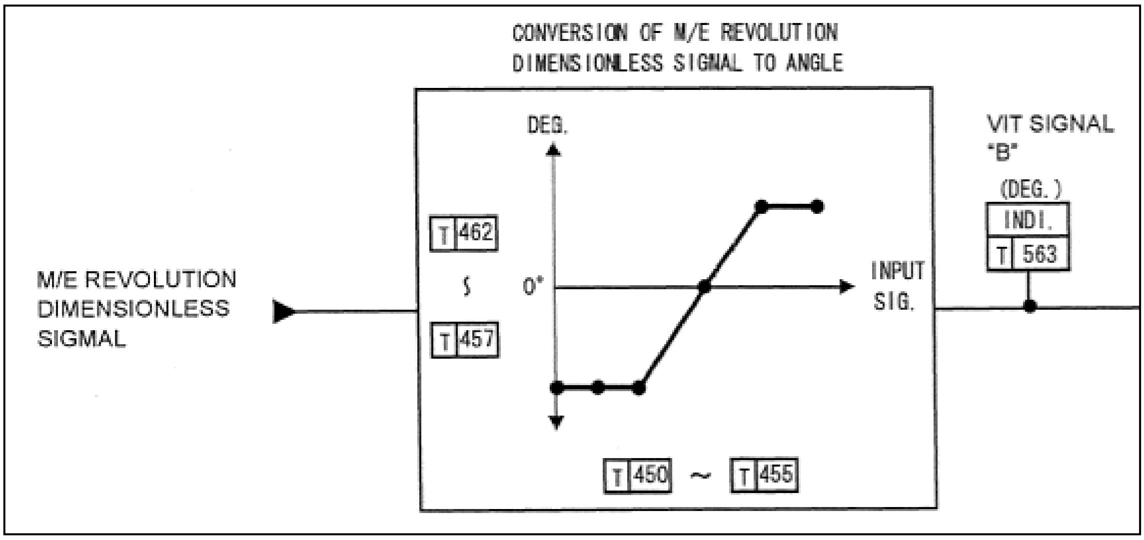


Fig. 3. 18 Engine speed signal vs. Crank angle degrees

As the illustrations show, two new signals are produced, the VIT ''A'' and ''B'' signal. The two signals are added and the result is the signal VIT ''C''. Afterwards a correction of the injection timing is made in order to fulfill low NOx emissions. This correction takes as input the charge air pressure dimensionless signal which is converted to crank angle degrees according to the function shown in Fig. 3.17, and the product is called VIT ''C4'' signal [12].

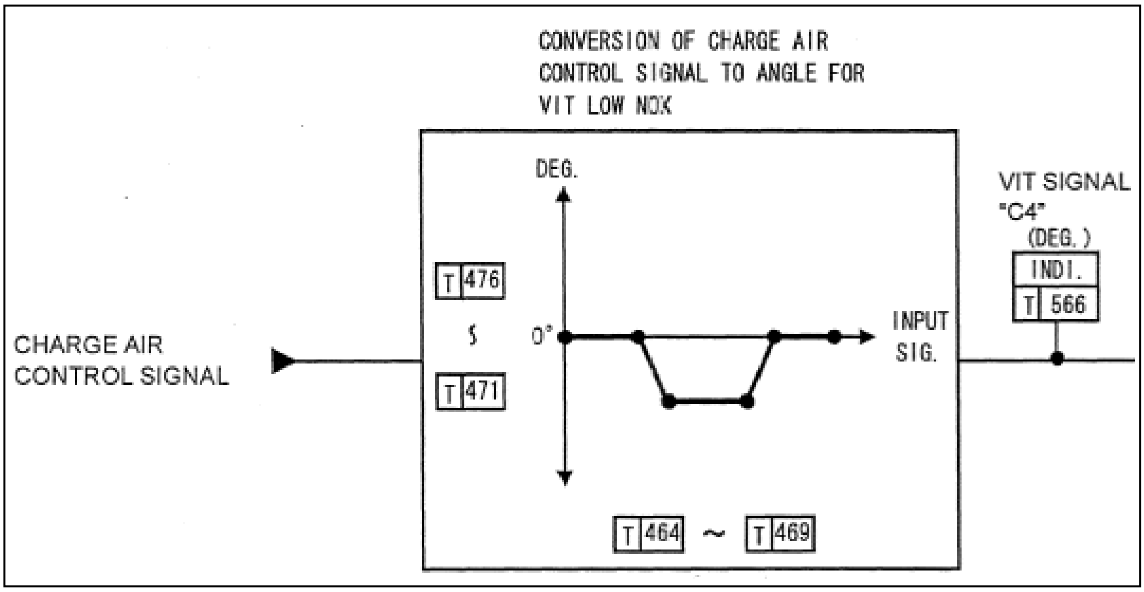


Fig. 3. 19 Conversion of charge air to angle for low NOx emissions

Finally, the signal ''C4'' is added to the signal ''B''; the sum is the injection angle that the VIT calculates and is referred as VIT signal ''D''. The FQS value is then added to the VIT signal ''D''.

The final injection timing is related with the injection pump settings. The RTA48-T`s fuel pumps have been set to pressurize the fuel towards the injector at 5deg before TDC. The VIT & FQS mechanism arranges an earlier or later injection timing according to the signal ''D''. The same principle as mentioned for the FQS, is followed; i.e. positive signal ''D'' means that this value is added to the 5deg, resulting in earlier injection time, and negative signal ''D'' means later injection time.

CHAPTER 4

SETTING UP THE ENGINE SIMULATION

4.1 Introduction

This Chapter aims to provide the reader the methodology that was followed in order to set up the engine`s simulation model. A brief description of the simulation-related factors is also included. At the end of this Chapter, calculations results are provided and evaluated as well.

4.1.1 The MOTor THERmodynamics software

The 'MOTor THERmodynamics' program uses geometric, combustion and heat transfer data in order to simulate the engine operation from a thermo-dynamical point of view. The MOTHER program considers the six-cylinder engine as a series of thermodynamic volumes, termed as flow receivers, interconnected via flow controllers, such as the exhaust valve, scavenge ports and others, and linked by mechanical elements, i.e. the crank shaft and the turbocharger`s shaft, for the transfer of the mechanical work. The Table 4.1 below shows both the thermodynamic and mechanical elements that were used for this particular engine simulation.

Thermodynamic elements		Mechanical elements
Flow receivers:	Flow controllers:	
Inlet receiver	Exhaust valve	Crank shaft
Cylinder	Scavenge ports	Turbocharger shaft
Exhaust receiver	Compressor	
Fixed fluid	Turbine	

Table 4. 1 Thermodynamic and mechanical elements used in MOTHER

Flow receivers are related with the calculation of the gas mixture properties (pressure P, temperature T, and equivalence ratio φ), whereas flow controllers are related with the calculation of mass and energy that flows through each element (pressure P, temperature T are calculated too).

As far as the flow receivers, is concerned, MOTHER considers the following general assumptions:

- Thermodynamic equilibrium and perfect gas behavior for the working medium
- Quasi-steady and one-dimensional flow through ports and valves at each instant
- Pressure wave interactions are ignored.

Apart from the above, it was considered that the gas mixture state (i.e. mixture of air and combustion products) is homogeneous, because the exact chemical analysis of the mixture of air, combustion products and fuel, is unknown.

As far as the flow controllers are concerned, it is assumed that the highly unsteady flow process through a valve or port are analyzed on a quasi-steady basis.

The use of mechanical elements is to connect the thermodynamic elements between them, in order to make possible the calculation of the produced mechanical work.

4.1.2 The simulation stages

The simulation of the 6RTA48-T was accomplished in stages through a specific trial-and-error procedure. Each simulation stage is described separately in a devoted Chapter.

The first simulation stage is related to the establishment of correct connections between the various elements (thermodynamic and mechanical) and the evaluation of the exhaust valve`s and scavenge ports` timing data. In order to simplify the simulation process, one acting cylinder is considered, whose aspiration is accomplished by a fixed fluid configuration. Separate fixed fluid elements simulate the ambient conditions; the inlet and exhaust receiver, by providing only the pressure P, temperature T, and equivalence ratio φ (the eq. ratio expresses the fuel to air ratio). The fixed fluid elements have fluid connections with the cylinder. Amongst the flow controllers, one exhaust valve and scavenge port element, is inserted and connected to the cylinder through a fluid connection. The turbocharger unit, is omitted for simplification. The last element that is mechanically connected with the cylinder, is the crank shaft. As the one acting cylinder has the appropriate fluid and mechanical connections, the thermodynamic model is capable to simulate the engine operation at the desired loads. The engine model was tested at the Shop Test loads (i.e. 50%, 75%, 85% & 100%) and the measured data were compared to the calculated figures.

The next configuration of the simulation model, is the addition of the remaining five cylinders and the establishment of the appropriate connections, as described in the above paragraph. Chapter 5 discusses the relevant procedure and provides the calculated data.

The last configuration of the simulation model, consists of the addition of the turbocharger unit, the inlet and exhaust receiver (Chapter 6). The simulation model was tested again in calculating the Shop Test measured data with accuracy better than 3%.

Sea Trials were used for the evaluation of the whole process (Chapter 7).

The 6RTA-48T`s main operating points are:

Maximum Continuous Rating (MCR)	7700kW at 117.0 rpm
Normal Continuous Rating (NCR)	6545kW at 110.8 rpm

Finally, it should be noted that Shop Test data have been recorded for 100%, 85%, 75% and 50% of load and Sea Trials for the 85% respectively.

4.2 Simulation with one (1) cylinder

At the first simulation stage, the simulation model consists of the following elements:

- One cylinder
- Scavenge ports
- Exhaust valve
- Fixed fluid inlet
- Fixed fluid outlet
- Crank shaft

The "Fixed fluid inlet" simulates the scavenge air conditions and the "Fixed fluid outlet" simulates the exhaust gas conditions after the turbine. The word "conditions" means the pressure *P*, temperature *T*, and the equivalence ratio φ. In general, it should be pointed out that wherever is required or calculated "gas pressure", in MOTHER program, it is the relative pressure which contains the barometric pressure.

The elements of the simulation model are connected as the Fig. 4.1 illustrates.

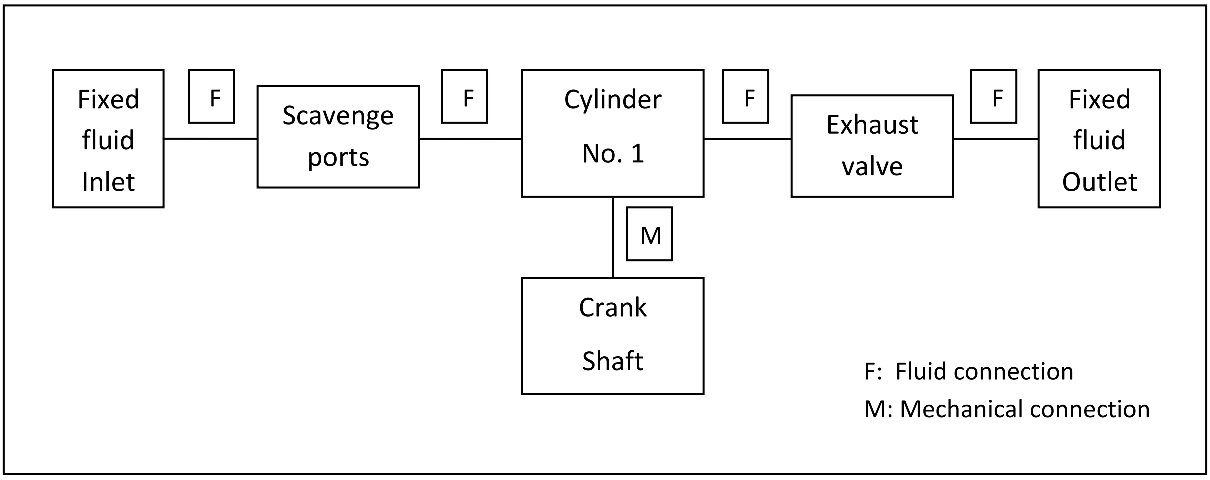


Fig. 4. 1 One cylinder model

The cylinder No. 1 was selected for the one cylinder simulation. The remaining five cylinders will be added when the one-cylinder model evaluation is completed (Chapter 5).

The engine is simulated as Fig. 4.1 illustrates; the scavenge air (corresponds to the compressed air after the Air Cooler, in the actual operation) of pressure *P*, temperature *T* and equivalence ratio φ, which is described by the Fixed fluid Inlet, passes through the scavenge ports into the cylinder. The combustion into the cylinder is simulated by specific models (which are described at 4.2.2); the combustion products through the exhaust valve, are released to the atmosphere. The exhaust gas properties after turbine, are simulated by Fixed Fluid Outlet. The cylinder produced work is simulated by the crank shaft element.

Each element requires a series of data as input before the simulation model could be able to calculate the desired operation points. The input data are described at the sections that follow. The engine`s operation point in MOTHER program is described by the following data:

- Engine speed (rounds per minute – RPM)
- Fuel injected per thermodynamic cycle (kg)
- Barometric pressure (bar)
- Scavenge air pressure (bar) and temperature (K)
- Exhaust gas pressure (bar) and temperature (K)

The execution of the MOTHER program with the one cylinder model, has the following targets:

- Evaluation of the scavenge ports data (port area as function of crank angle).
- Evaluation of the exhaust valve data (exhaust valve effective area as function of crank angle).
- Suitable adjustments of the combustion model (Chapter 4.2.1.5).
- Evaluation of the model output as far as the maximum pressure, compression pressure, mean effective pressure and brake power produced, are concerned, in conjunction with the Shop Test measurement. The Shop Test measurements` results are attached at the Appendix I – Shop Test.

The sections below describe the model`s elements (as shown in Fig. 4.1), their adjustments and the model`s output.

4.2.1 Cylinder

The cylinder`s operation simulation in the MOTHER interface requires the following data:

- Geometry data
- Heat transfer (gas – wall) data
- Heat transfer (wall – coolant) data
- Friction data
- Combustion data
- General data
- Scavenging data

4.2.1.1 Geometry data

Geometry data have been acquired from the engine manuals. These data will not be updated at the hole simulation process.

4.2.1.2 Heat transfer (gas – wall) data

The **Heat transfer (gas – wall) model** was adjusted at this stage of simulation and remained unchanged for the hole simulation process. The heat fluxes from the cylinder gas to the cylinder head, piston crown, the upper part and lower part of the liner, are calculated at each step of the simulation procedure by the relation:

$$q = h \cdot A \cdot \left(T_{gas} - T_{wall}\right) \tag{4.1}$$

where:

q	instantaneous heat flux	$[kW]$
h	gas cylinder instantaneous average heat tranfer coefficient	$\left[kW/m^2K\right]$
A	respective cylinder part wall gas side area	$\left[m^2\right]$
T_{gas}	instantaneous cylinder gas temperature	$[K]$
T_{wall}	instantaneous cylinder part wall surface temperature	$[K]$

Woschni model was selected for the calculation of heat transfer coefficient. This model is widely used for steady, turbulent heat transfer in the engine cylinders. The instantaneous average heat transfer coefficient of the cylinder gas is given by the following equation:

$$h = 0.00326 \cdot B^{-0.2} \cdot P^{0.8} \cdot T^{-0.55} \cdot w^{0.8} \tag{4.2}$$

where:

h	gas cylinder instantaneous average heat tranfer coefficient	$\left[kW/m^2K\right]$
B	cylinder bore	$[mm]$
P	cylinder gas pressure	$[kPa]$
T	cylinder gas temperature	$[K]$
w	average cylinder gas velocity	$[m/\sec]$

The cylinder gas velocity was considered to be constant, for simplification reasons.

4.2.1.3 Heat transfer (wall – coolant) data

The **Heat transfer (wall – coolant) model** that was selected, according to the available data, is the ‘Simple Model’, which is used for the calculation of the surface temperatures. This model requires less input data, compared to the ‘Full Model’ which is also available at the MOTHER interface. The model was set at this stage of simulation and remained unchanged for the hole process. According to the model, the surface temperature of a given component of the cylinder (head, upper/lower part of liner and piston) must be between the temperature of the cooling medium for that component and a theoretical temperature of the wall, which is calculated as follows:

$$T_{w,a} = \frac{\int hAT_g d\theta}{\int hAd\theta} \tag{4.3}$$

where:

Q	mean heat transfer from cylinder gas to wall of cylinder component	$[kW]$
h	cylinder gas heat transfer coefficient	$[mm]$
A	gas side area of cylinder	$[m^2]$
T_g	cylinder gas temperature	$[K]$
$T_{w,a}$	wall adiabatic temperature	$[K]$

The surface wall temperature, ranging from its wall cooling medium temperature T_c to its adiabatic wall temperature $T_{w,a}$, can be expressed as follows:

$$T_w = HTF \cdot T_c + (1 - HTF) \cdot T_{w,a} \tag{4.4}$$

where:

T_w	cylinder gas temperature	$[K]$
$T_{w,a}$	wall adiabatic temperature	$[K]$
T_c	cooling medium temperature	$[K]$
HTF	heat transfer factor $[0-1]$	

If the heat transfer factor is equal to 0, no heat will be transferred from the gas to wall (see eq. (4.4)), and when the heat transfer factor is equal to 1, the surface temperature will be equal to the coolant temperature and the maximum heat will be transferred from the gas to the wall.

4.2.1.4 Friction data

The **Friction** model that was selected, is the 'Model of Winterbone and Tennant'. This model assumes that the total engine losses vary linearly with the cylinder maximum pressure and with the rotational engine speed. The friction mean effective pressure is calculated by the following equation:

$$fmep = k_1 + k_2 P_{\max} + k_3 N_{M/E} \tag{4.5}$$

where:

$P_{\max}$	maximum cylinder pressure	$[Pa]$
$N_{M/E}$	rotational engine speed	$[rpm]$
k_1, k_2, k_3	constants	

The user is able to change values of the constants k_1, k_2, k_3 only. At this simulation stage the values of the constants were changed in order to succeed convergence between the calculated brake power and the measured at Shop Test, for each load (i.e. 50%, 75%, 85% & 100%). It is noted that the brake horse power is calculated when the friction power is subtracted by the indicated horse power. The final tuning of the friction model was accomplished at the last simulation stage, when the turbocharger unit has been added (refer to Chapter 6).

4.2.1.5 Combustion Data

The combustion simulation followed the simple idea of selecting firstly a simple model and afterwards a more comprehensive model. The 'simple' model is called 'S-Curve Model' and the other 'Woschni – Anisits'. Chapters 4.2.1.5.1 & 4.2.1.5.2 below, explain the methodology and the appropriate adjustments that were made at this simulation stage.

4.2.1.5.1 S-Curve Combustion Model

The ''S-Curve'' combustion model calculates the mass of fuel burnt at each time step for a given total fuel quantity and combustion duration. The model can simulate the three phases of the combustion process, which is the premixed combustion (first phase, i=0), the burnt fuel in the main combustion (second phase, i=1) and the burnt fuel in the last phase of combustion (third phase, i=2). The s-shaped mass fraction profile is specified by the function:

$$x_b = \frac{m_b}{m_{tot}} = \sum_{i=0}^{2} A_i \cdot \left(1 - e^{-a_i \cdot \left(\frac{\theta - \theta_i}{(\Delta\theta)_i} \right)^{m_i + 1}} \right) \qquad (4.6)$$

where:

m_b	burnt fuel	$[kg]$
m_{tot}	total fuel injected into each cylinder per cycle	$[kg]$
θ	crank shaft angle	$[\deg]$
θ_i	crank shaft angle at the start of combustion for each s-curve	$[\deg]$
$\Delta\theta_i$	combustion duration for each s-curve	$[\deg]$
α_i	model parameter for each s-curve	$[-]$
m_i	model parameter for each s-curve	$[-]$
A_i	weighting factor for each s-curve	$[-]$

It was assumed that the combustion happens at the first phase, in order to cope with a simple s-curve fuel mass fraction; consequently the "i" from equation 4.6 is equal to zero (i=0). In MOTHER interface the a_i, m_i, A_i parameters are adjustable. The parameters a_i, m_i were changed for each load in a way that will be explained below, whereas the A_0 remained unchanged and equal to 1, because the simple s-curve is used.

The model parameters, a_0 & m_0, affect differently the s-curve`s form, as long as the combustion profile. A reference s-curve is attached below (Fig. 4.2). The combustion duration (34 crank angle degrees (CA deg)) is the same for all cases and the injected fuel mass too. The figure 4.2 shows that the injection happens at 5 CA deg before top dead center (TDC) and the 100% of the injected fuel mass has been burnt at almost 28 CA deg after TDC. The combustion finishes at almost 29 CA deg after TDC. The influence of the s-curve parameters will be shown on the s-curve`s form (Fig. 4.3 and 4.4).

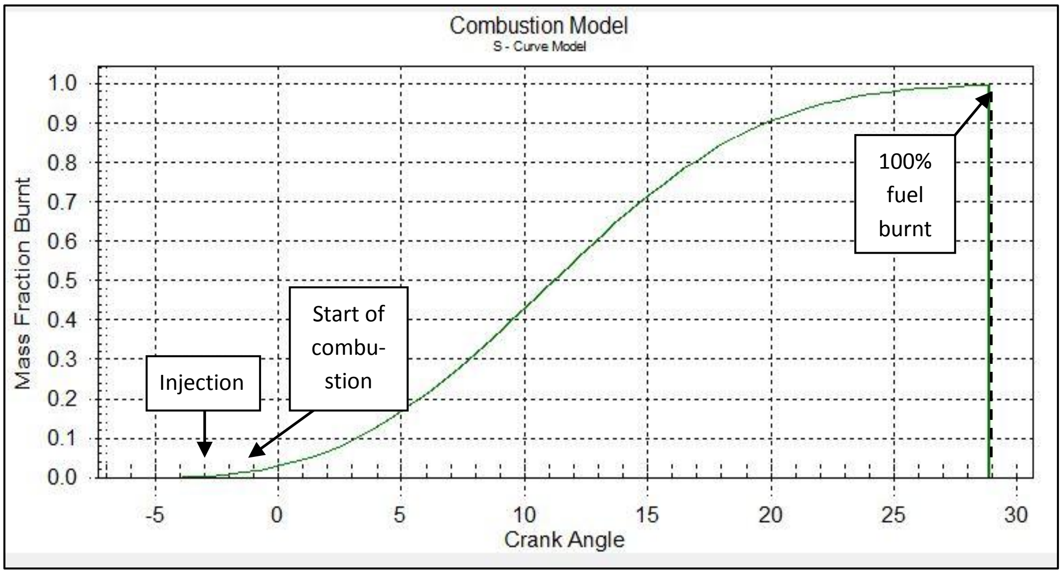


Fig. 4. 2 Reference combustion profile

If the parameter m_0 remains constant and the parameter a_0 increased compared to its initial value (Fig. 4.2), the s-curve will have a different form (Case 1). According to Fig. 4.3, the 100% of injected fuel will be burnt at almost 24 CA deg after TDC, instead of 28 CA deg at the reference condition. Consequently, the a_0 affects the time, in terms of crank angle, that the fuel is burnt completely. In general, it is obvious that the way (slowly or fast) that the fuel is burnt in the cylinder, affects its maximum pressure, which is also affected by the fuel properties and thus the combustion process. Several runs (execution of the program code) of the one cylinder model, showed that the increased a_0, leads to an increased maximum cylinder pressure (P_{max}). This is expected because the fuel is burnt 'earlier' compared to the reference case with lower a_0.

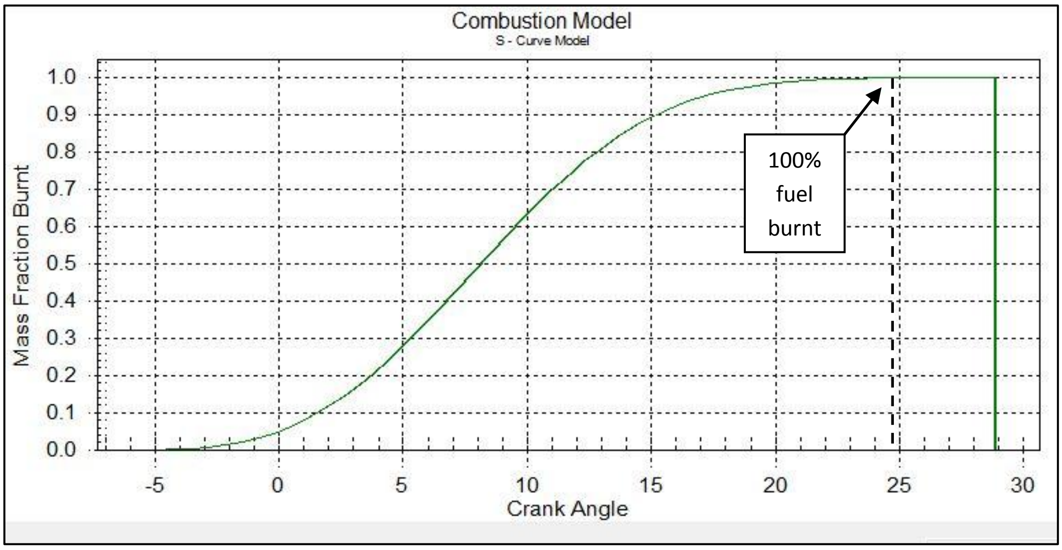


Fig. 4. 3 Combustion profile by increasing ''a0'' (''m0'' remains constant), Case 1.

On the other hand, if the parameter a_0 remains constant and the parameter m_0 increased compared to its reference value, the s-curve will have a different form too (Case 2, see Fig. 4.4). According to the reference figure, 4.2, the fuel starts to be burnt at 4 CA deg before TDC, whereas in case 2 the ignition happens at about the TDC. Thus, the increased value of m_0, retards the ignition and fuel is burnt in respectively less time, as the combustion duration is the same for all cases. Moreover, several runs of the model showed that this change in the s-curve form, lead the maximum pressure to decrease. The relative lower P_{max} (compared with the reference one) can be explained by the less time available to fuel to be burnt.

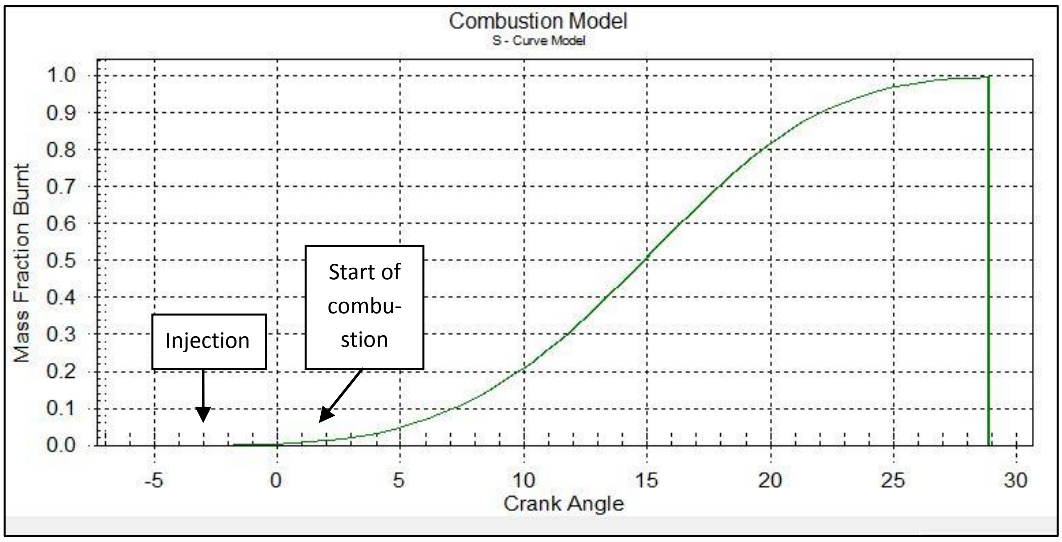


Fig. 4. 4 Combustion profile by increasing “m0” (“a0” remains constant), Case 2.

The combustion profile affects the cylinder maximum pressure, whereas the compression pressure is not affected by any change on the s-curve shape. This influence is rational because the compression pressure is dependent on scavenging ports`, exhaust valves` geometry and compression ratio.

The s-curve combustion model also requires the injection time in terms of crank angle, the mass of fuel injected and the combustion duration; both of these additional data are discussed below.

- **Injection Timing**

The injection timing data, as part of the required combustion data, are dependent on the engine load. It should be stated that the default injection timing for this particular engine happens before its top dead center (TDC), according to the DU-SULZER manufacturer. Furthermore, the injection timing is affected by the VIT and FQS system, as explained in Chapter 3.10. According to ''Setting Table sheet A'' [9] of Shop Test (refer to Appendix – I), the default injection time is designed at 5deg CA before TDC, with the VIT and FQS arranged at position zero (0). The actual (i.e. during engine operation) VIT and FQS positions differ from zero, consequently the injection timing is affected for each load of the engine. For that reason, the discrete VIT and FQS positions were recorded for each load during Shop Tests. According to the engine`s ''Technical File'' (a manual that provides additional information about the engine`s operation and design), if the VIT or FQS has negative value, the injection is later from reference value of -5 deg [8]. Positive FQS and VIT values indicate that the fuel is injected earlier than the initial injection setting angle. For a more detailed explanation of the VIT and FQS function, refer to Chapter 3.10. Table 4.2 below shows the injection timing calculation.

CALCULATION OF INJECTION TIMING AT SHOP TEST					
LOAD	INJ. SETTING	VIT	FQS	VIT+FQS	INJECTION
%	deg	deg	deg	deg	deg
100%	-5	-1.0	-0.5	-1.5	-3.5
85%	-5	+1.5	-0.5	+1.0	-6.0
75%	-5	+1.1	-0.5	+0.6	-5.6
50%	-5	-0.5	-0.5	-1.0	-4.0

Table 4. 2 Injection Timing (Shop Test)

For example at 100% Load:

VIT = -1^{o}
FQS = -0.5^{o}
(VIT+FQS)= -1.5^{o}
Initial Setting at Shop Test[1] = -5^{o}
Final injection timing[2] = -5+1.5 = **-3.5^{o}**

[1] The initial setting assumes that: (VIT + FQS) = 0^{o}
[2] The VIT and FQS positions are taken into account: (VIT+FQS)= -1.5^{o}

As the injection timing is known for Shop Test, the mass of the fuel injected is required for the simulation`s calculations, as well.

- **Fuel injected per cycle**

MOTHER requires the mass of fuel injected during one thermodynamic cycle in kilograms per cylinder. The relevant calculation procedure is attached in Appendix IV.

Table 4.3, shows the fuel mass injected per cylinder per thermodynamic cycle, with respect to the engine load at Shop Tests.

LOAD (%)	Fuel injected per cycle per cylinder [kg]
100%	0.03160
85%	0.02757
75%	0.02539
50%	0.02051

Table 4. 3 Fuel mass injected per thermodynamic cycle per cylinder

- **Combustion duration**

The duration of the combustion in degrees of crank angle, is undisclosed, for that reason it is estimated before any insertion is made into the s-curve model. The initial values of the combustion duration for 100% and 50% loads, were selected with reference to the sample MOTHER`s simulation model of a two-stroke engine. Afterwards, the initial values were slightly changed in order to achieve convergence between the calculated and measured cylinder compression and maximum pressure.

In practice, it has been proven that the combustion duration is affected by the load; in other words, the combustion duration increases when the engine load raises. Thus, the intermediate loads (i.e. 85% and 75%) of these simulation, should have intermediate values for their combustion duration, compared to the duration`s values of 100% and 50% of load.

The model`s measured pressures (i.e. compression and maximum) were compared with the mean measured values from Shop Test (Appendix I). This process contributes to the general evaluation of the settings that are made to the one cylinder model. The initial values are attached at the Table 4.4.

LOAD	COMBUSTION DURATION (deg)	LOAD	COMBUSTION DURATION (deg)
100%	40.5^o	75%	34.0^o
85%	36.5^o	50%	29.5^o

Table 4. 4 Comb. Duration

- **General comments for the S-curve model**

The combustion, for each load of Shop Test, can now be simulated by the s-curve model according to the data described above. However, the s-curve model has some disadvantages: the injection and combustion delay are not calculated. Furthermore, the combustion duration cannot be calculated for any desired engine load. In order to overcome these deficiencies, a more comprehensive combustion model is used and is based to the s-curve`s settings; this model is called "Woschni-Anisits combustion model".

4.2.1.5.2 Woschni – Anisits Combustion Model

The ''Woschni – Anisits'' combustion model is a phenomenological model that can be used for Direct Injection diesel engines combustion simulation. The model is based on the single ''S-curve'' combustion model and calculates the S-curve constants based on a ''Reference Point'', where the values of these constants are known by tuning the S-curve model. Thus, the application of this model, requires firstly the utilization of the s-curve model at a given operating point and then the calculation of values for input variables related to that point.

The reference point was chosen to be the 75% of Load at Shop Test because close to this operating point do exist some Performance data concerning the IHP; consequently, the verification of the model output is more accurate. Table 4.5 includes these data.

LOAD (%)	RPM	I.H.P. (kW)	BHP/IHP	DATE OF REPORT
73.0%	109.2	1001	93.7%	12-03-2008
72.0%	108.9	987	93.7%	10-02-2008
71.6%	110.0	981	93.7%	28-01-2008

Table 4. 5 IHP as reported from the ship

The verification of the simulation model`s output power is made through the Indicated Horse Power because the mechanical losses (i.e. friction) are not taken into account. Thus, the IHP, for the reference point, is estimated by assuming that the factor BHP/IHP from Table 4.5 is maintained the same for each load. The measured Brake Horse Power for 75% load, is divided by 6, in order to calculate the BHP per cylinder. Finally, the IHP is the product of the division of the "BHP per cylinder" with the factor "BHP/IHP". Table 4.6 shows the aforementioned calculation.

LOAD (%) [1]	RPM [2]	BHP (kW) [3]	BHP per cylinder (kW) [4]=[3]/6	BHP/IHP [5]	IHP (kW) [6]=[4]/[5]
75%	106.4	5807	967.83	93.7%	**1032.91**

Table 4. 6 IHP at 75% with regression

Due to the complexity of the model and the relatively large number of required input data, the form in the MOTHER interface is divided in 4 tabs, each one containing a thematic set of input data. The method that was followed to adjust this combustion model, is described below. It should be pointed out that the adjustment of the Woschni-Anisits model started from the reference point, the 75% load, and afterwards to other loads, the 100%, 85% and 50%.

- In the <u>first tab</u>, that contains the **main model parameters**, the required input data are:

  - The **"a" parameter** of the single S-Curve model:

$$x_b = \frac{m_b}{m_{tot}} = 1 - e^{-a \cdot \left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}} \tag{4.7}$$

where:

m_b	burnt fuel	$[kg]$
m_{tot}	total fuel injected into each cylinder per cycle	$[kg]$
θ	crank shaft angle	$[\deg]$
θ_0	crank shaft angle at the start of combustion	$[\deg]$
$\Delta\theta$	combustion duration	$[\deg]$
α	constant (input to the model)	$[-]$
m	constant (calculated by the model)	$[-]$

As Woschni-Anisits combustion model is based on the s-curve model, the already selected values of the “a” parameter for each load (50%, 75%, 85% and 100%), were used as input for the Woschni-Anisits model. The model was adjusted by adding the remaining input data.

  - The **Start of Static Injection** (SOI) in crank angle degrees, which is the same as in the S-Curve model, depending on the load (Table 4.2).

  - The **Mass of Fuel injected** per cycle of the S-Curve model (m_{tot}) (Table 4.3).

- The <u>second tab</u>, that contains parameters for the calculation of **Start of Combustion Angle θ_0**:

$$\theta_0 = SOI + \Delta\varphi_{IND} + \Delta\varphi_{IGD} \tag{4.8}$$

where:

SOI	crank angle at start of static injection	$[\deg]$
$\Delta\varphi_{IND}$	injection delay (between delivery start of the injection pump and start of injection)	$[\deg]$
$\Delta\varphi_{IGD}$	ignition delay	$[\deg]$

The required data are:

- The **Injection delay** at the reference point ($\Delta\varphi_{IND,REF}$), selected to be zero for all the loads. Thus, the injection delay ($\Delta\varphi_{IND}$) for the reference load (i.e. 75%) is not calculated in the equation 4.8.

$$\Delta\varphi_{IND} = \Delta\varphi_{IND,REF}\left(\frac{n}{n_{REF}}\right)^{a_{IND}} \tag{4.9}$$

where:

$\Delta\varphi_{IND,REF}$	injection delay (between delivery start of the injection pump and start of injection) at Reference Point	$[\mathrm{deg}]$
n	engine speed	$[\mathrm{rpm}]$
n_{REF}	engine speed at the Reference Point	$[\mathrm{rpm}]$
a_{IND}	constant	$[-]$

- The **engine speed** (n_{REF}) at the reference point (i.e. 75%) is 106.4 RPM.

- The injection delay constant which has the default value, does not affect the combustion model as the injection delay is zero.

- The **ignition delay** is calculated by the following equation:

$$\Delta\varphi_{IGD} = 6\cdot n\cdot 10^{-3}\cdot\left[a_{IGD} + b_{IGD}\cdot e^{\frac{7.8}{6.9167\cdot R}}\cdot\left(1.01197\cdot p^{-0.7}\right) + c_{IGD}\cdot e^{\frac{7.8}{6.9167\cdot R}}\cdot\left(1.01197\cdot p^{-1.8}\right)\right] \tag{4.10}$$

where:

n	engine speed	$[rpm]$
a_{IGD}	constant	$[-]$
b_{IGD}	constant	$[-]$
c_{IGD}	constant	$[-]$
P	pressure averaged from start of injection until start of combustion	$[bar]$
T	temperature averaged from start of injection until start of combustion	$[K]$
R	gas constant	$[-]$

Regarding the model`s constants, the default values were selected:
$a_{IGD} = 0.39, b_{IGD} = 0.105, c_{IGD} = 3.12$.

The ignition delay is calculated by the MOTHER program through the equation 4.10. The values of pressure and temperature are calculated by the program, thus, the user is able to affect the ignition delay through the engine speed and the constants $a_{IGD}, b_{IGD}, c_{IGD}$.

- The <u>third tab</u>, contains parameters for the calculation of **Combustion Duration $\Delta\theta_0$**:

$$\Delta\theta_0 = \Delta\theta_{0,REF}\left(\frac{\lambda_{REF}}{\lambda}\right)^{a_{CD}}\left(\frac{n}{n_{REF}}\right)^{b_{CD}}\left(\frac{m_{fuel}}{m_{fuel,REF}}\right)^{c_{CD}} \tag{4.11}$$

where:

$\Delta\theta_{0,REF}$	combustion duration at the Reference Point	$[deg]$
λ	air to fuel equivalence ratio	$[-]$
λ_{REF}	air to fuel equivalence ratio at the Reference Point	$[-]$
n	engine speed	$[rpm]$
n_{REF}	engine speed at the Reference Point	$[rpm]$
m_{fuel}	fuel injected per cylinder and per cycle	$[kg]$
$m_{fuel,REF}$	fuel injected per cylinder and per cycle at the Reference Point	$[kg]$
a_{CD}	constant	$[-]$
b_{CD}	constant	$[-]$
c_{CD}	constant	$[-]$

The required input data are:

- The **Combustion Duration** $\Delta\theta_{0,REF}$, at the reference point (i.e. 75%), in crank angle degrees. This value is depicted by the s-curve model ($\Delta\theta_{0,REF} = 34\ CA\ \deg$ Table 4.4)

- The **Relative Air/Fuel Ratio** λ_{REF} at the reference point. The λ_{REF} was calculated by the program for the reference point, when the S-curve model was selected. This particular value was inserted in the relative tab for the utilization of the Woschni–Anisits model. The relative air/fuel ratio is defined as follows:

$$\lambda = \varphi^{-1} = \frac{(A/F)_{actual}}{(A/F)_{st}} \tag{4.12}$$

where:

$(A/F)_{actual}$	actual air/fuel ratio	$[-]$
$(A/F)_{st}$	stoichiometric ratio	$[-]$

- The model **constants** were the following: $a_{CD} = 0.6, b_{CD} = 0.5, c_{CD} = 0$ (default values were selected).

The correct adjustment of the combustion duration for the reference point (i.e. 75%) is significant, because MOTHER predicts combustion parameters of any load, according to these adjustments.

According to equation 4.11, when the input data (combustion duration, engine speed, relative air/fuel ratio and mass of fuel burnt) of the reference point are inserted, the calculated combustion duration (i.e. $\Delta\theta$) will be equal to the combustion duration at the reference point (i.e. $\Delta\theta_0$), because the ratios of the equation 4.11, are equal to one.

$$\frac{\lambda_{REF}}{\lambda_{75\%}} = \frac{n_{75\%}}{n_{REF}} = \frac{m_{fuel,75\%}}{m_{fuel,REF}} = 1 \tag{4.13}$$

Consequently, the combustion duration calculation of the reference point is cross-checked through the model`s output; in other words, whether the calculated value of combustion duration coincides with the input value ($\Delta\theta = \Delta\theta_0$). In case the two values do not converge, only the relative air/fuel ratio input value is checked if coincides with the calculated value. As stated above, the input value of the relative air/fuel ratio, is calculated by the MOTHER program and this particular value is inserted as input for the reference point, at the Woschni-Anisits model. Thus, a loop is being made by checking the combustion duration output and the relative air/fuel ratio for the reference point.

- The <u>fourth tab</u> contains parameters for the calculation of the S-curve **Shape parameter "m"**:

$$m = \left(m_{REF} + \Delta m\right) \cdot \left(\frac{\Delta\varphi_{IGD,REF}}{\Delta\varphi_{IGD}}\right)^{a_{VM}} \cdot \left(\frac{n_{REF}}{n}\right)^{b_{VM}} \cdot \left(\frac{P_{IVC} V_{IVC} T_{IVC,REF}}{P_{IVC,REF} V_{IVC,REF} T_{IVC}}\right)^{c_{VM}} - \Delta m \tag{4.14}$$

where:

m_{REF}	fuel injected per cylinder and per cycle at the Reference Point	$[kg]$
Δm	constant	$[-]$
$\Delta\varphi_{IGD}$	ignition delay	$[\deg]$
$\Delta\varphi_{IGD,REF}$	ignition delay at the Reference Point	$[\deg]$
n	engine speed	$[rpm]$
n_{REF}	engine speed at the Reference Point	$[rpm]$
P_{IVC}	pressure at start of close cycle	$[bar]$
$P_{IVC,REF}$	pressure at start of close cycle at the Reference Point	$[bar]$
T_{IVC}	temperature at start of close cycle	$[K]$
$T_{IVC,REF}$	temperature at start of close cycle at the Reference Point	$[K]$

V_{IVC}	cylinder volume at start of close cycle	$\left[m^3\right]$
$V_{IVC,REF}$	cylinder volume at start of close cycle at the Reference Point	$\left[m^3\right]$
a_{VM}	constant	$[-]$
b_{VM}	constant	$[-]$
c_{VM}	constant	$[-]$

The required data are:

- The constant Δm is equal to zero for large diesel engines, according to MOTHER user`s manual [13].
- The **s-curve shape parameter** ''m'' at the reference point.
- The **ignition delay** at the reference point, $\Delta\varphi_{IGD,REF}$, (i.e. 75%) in crank angle degrees, which is the time elapsed from start of dynamic injection to the start of combustion. The final value was selected through a repeating process; firstly, the constants b_{VM} and c_{VM} were set equal to zero in order to isolate the ignition delay from the influence of the other input data. A trial value for the ignition delay was selected between the range of 0.2 and 0.8 CA deg. Afterwards, it was checked the calculated ignition delay with the input value; in case of discrepancy, the input value was corrected in order to succeed convergence between input and calculated value. According to equation 4.14, the calculated shape parameter for the reference point, should be equal to the input shape parameter:

  $$m_{75\%} = m_{REF} \tag{4.15}$$
- **Pressure** ($P_{IVC,REF}$), **Volume** ($V_{IVC,REF}$) and **Temperature** ($T_{IVC,REF}$) at the start of closed cycle at the reference point. The closed cycle starts when the exhaust valve closes and the compression begins. The required values were selected through a loop process. Firstly, the constants a_{VM} and b_{VM} were set equal to zero, afterwards the closed cycle`s pressure, volume and temperature, values were inserted as input to the model. These initial values had been calculated when the s-curve model was utilized. In case the calculated ignition delay is not equal with the input value, the values of pressure and temperature are updated (the volume is same). The loop ends, when the calculated shape parameter converges to the initially set for the reference point (as per eq. 4.15).

According to the above description, each of the input ignition delay, pressure and temperature data, of the closed cycle, are evaluated separately (by setting the exponent equal to zero) in order to succeed convergence between the input shape parameter and the calculated (eq. 4.15) at the reference point. However, the exponents (a_{VM}, b_{VM}, c_{VM}) are values not equal to zero. The final calculation of the shape parameter contains one more loop in which the exponents have their real values:

$$a_{VM} = 0.5,\ b_{VM} = 0.3,\ c_{VM} = 1 \tag{4.16}$$

During this loop, both the ignition delay and the pressure, temperature of the closed cycle, are updated in order to succeed the convergence of calculated shape parameter and the input value, at the reference point, i.e. 75% load.

General remarks:

- The update of the already described parameters at the reference point, was possible through loop processes. The target, is the convergence between input and output parameters at the reference point, and also, the calculation of $P_{\max}, P_{comp}, IHP, BHP$ with the minimum possible error with those measured at Shop Test.
- When the reference point (i.e. 75%) was tuned correctly, the reference Woschni-Anisits parameters` were copied to the other loads. The s-curve parameters of each load were used in the relative tabs as explained above.
- The configuration of the Woschni-Anisits combustion model for the other engine loads (50%, 85% and 100%) lies on the specification of the s-curve "a" parameter, the start of static injection (SOI) and the fuel mass injected per cycle.
- The combustion model configuration for the one cylinder model was completed according to the aforementioned methodology. The selected values at this stage of simulation, are maintained and evaluated with the six cylinder model (refer to Chapter 5).

4.2.1.6 General Data

The thermodynamic model, requires several initial values, for each load, such as pressure, temperature and equivalence ratio, in order to be converged. Thus, the selection of these values should take into account the gas state inside the cylinder. Specifically, if the piston is at its TDC position, compression stroke has been finished and combustion begins; consequently, high pressures and temperatures are expected inside the cylinder.

At this stage of simulation, the one cylinder is considered to be at its TDC, where the initial values for the simulation model are attached at Table 4.7.

Initial Pressure	65	Bar
Initial Temperature	790	K
Initial Equivalence Ratio	0.022	-

Table 4. 7 Initial values for the one cylinder model

4.2.1.7 Scavenging Data

The scavenging model is of the control-volume type. The engine cylinder is divided into three zones, namely, a zone of pure air (zone I), a zone comprising a mixture of air and burnt gas (zone II) and a zone of burnt gas (zone III). The scavenging period of a cylinder is considered to commence when the cylinder inlet ports open and to finish when the exhaust valve closes.

Mass exchange between the zones is assumed to take place only in the following manner: scavenging air (from zone I, when it exists) and burnt gas (from zone III, when it exists) enter the mixing zone (zone II). In order to account for the mass flow rates from zones I and III to zone II, air and gas penetration coefficients are used.

The air penetration coefficient is defined by:

$$\mu_\alpha = \frac{\dot{m}_{I-II}}{\dot{m}_{inp}} \tag{4.17}$$

The gas penetration coefficient is defined by:

$$\mu_g = \frac{\dot{m}_{III-II}}{\dot{m}_{inp}} \tag{4.18}$$

where:

μ_α	air penetration coefficient	$[-]$
μ_g	gas penetration coefficient	$[-]$
$\dot{m}_{I-II}$	air mass flow rate from zone I to zone II	$[kg/s]$
$\dot{m}_{III-II}$	air mass flow rate from zone III to zone II	$[kg/s]$
$\dot{m}_{inp}$	air mass flow rate through inlet ports	$[kg/s]$

The selected values for the above penetration coefficients were the default by the program values, as they are suitable for the simulation of the 6RTA48-T engine.

$$\mu_a = 0.60,\ \mu_g = 0.45 \tag{4.19}$$

The penetration coefficients control the growth of the air zone and mixing zone and contribute to the reduction of the burnt gas zone, so that they effectively influence the engine breathing conditions and the heat transfer during the scavenging period. Their values are considered to be constants during the scavenging process.

4.2.2 Ports Data

The required input data for both exhaust valve and scavenge ports, are:

- The valve type, inlet or exhaust,
- The valve area in square meters as a function of crank angle, and
- The values of CA in degrees when Valve Opens and Closes respectively.

4.2.2.1 Exhaust Valve

Exhaust`s valve effective area data as a function of crank angle are required as input into the MOTHER interface. That effective area is calculated through a specific methodology due to its complex geometry, which is attached in Appendix IV. The exhaust valve opens at 132˚ and closes at 250.8˚ crank angle degrees.

Figure 4.8, illustrates the exhaust valve`s lift diagram.

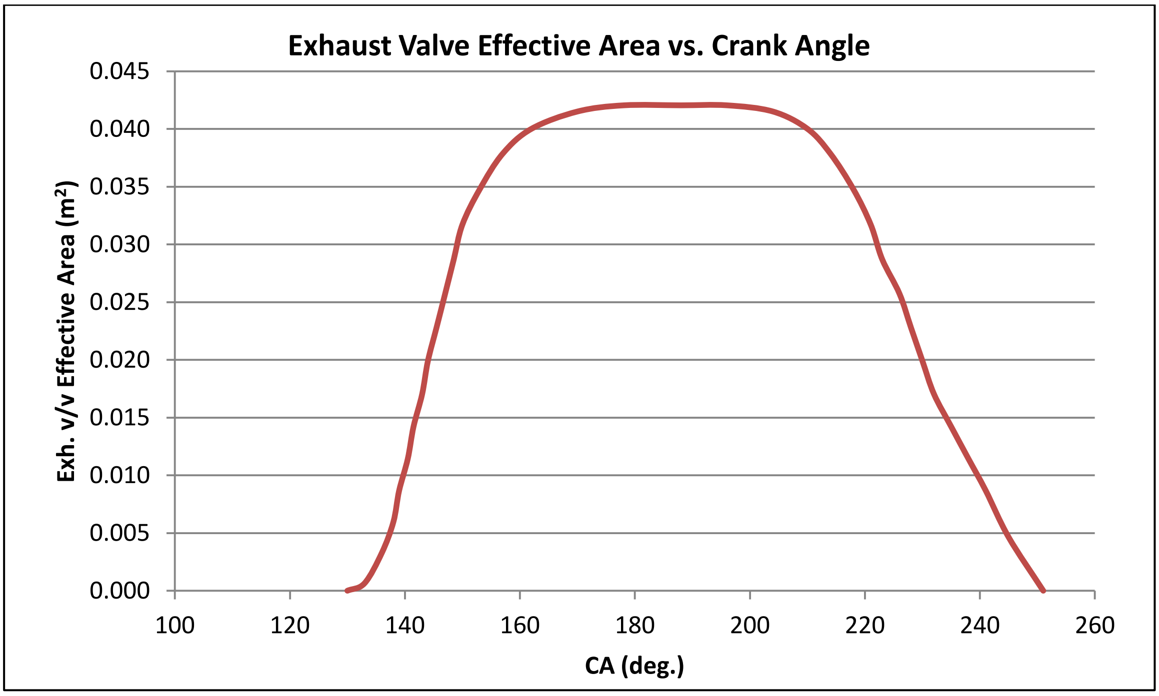


Fig. 4. 5 Exhaust valve lift diagram

4.2.2.2 Scavenge Ports

The scavenge ports` required input data are, the opening, closing angle in degrees and the ports` effective area in relation with the crank angle. The scavenge ports` effective area diagram is not included into any engine manual; for that reason a methodology is followed in order to calculate it. The calculation procedure is attached in Appendix IV. Scavenge ports open at 154° and close at 206°.

Figure 4.9 illustrates the scavenge ports` effective area with respect to the crank angle.

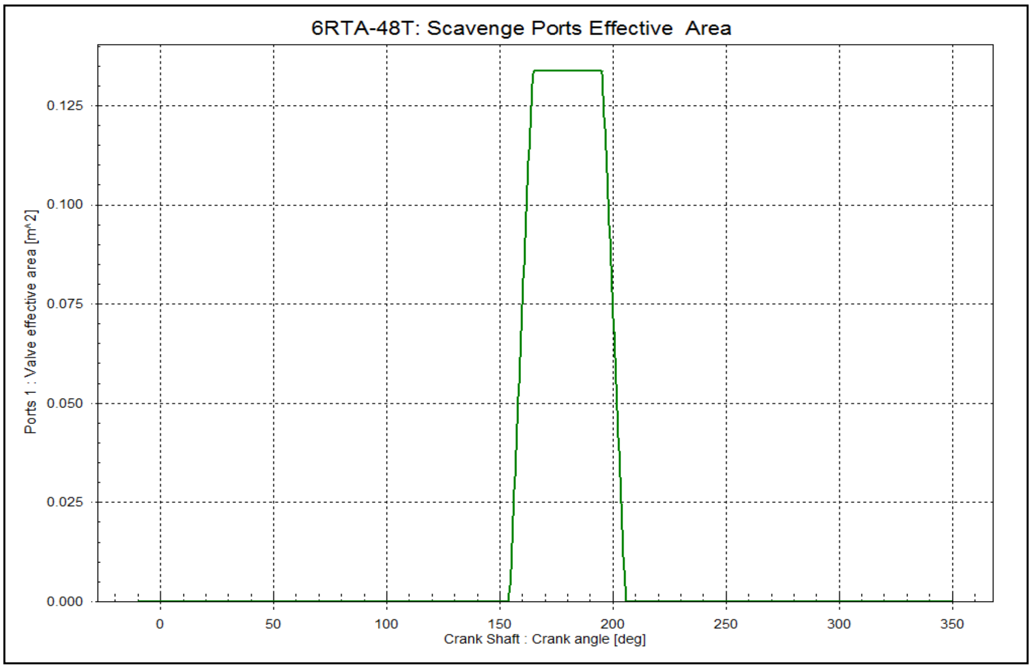


Fig. 4. 6 Scavenge port`s effective area vs. Crank Angle

4.2.3 Fixed fluid inlet and outlet

The fixed fluid element corresponds to an infinite size plenum (i.e. the inlet/exhaust receiver). The fixed conditions of the inlet and the exhaust receiver, as well, can be represented by the "Fixed Fluid" elements. The state of the "Fixed Fluid" elements is defined by pressure, temperature and equivalence ratio. Consequently, the inlet receiver is represented by the "Fixed Fluid Inlet" element and the exhaust receiver by the "Fixed Fluid Outlet" element.

The scavenging pressure, which is used as input for the fixed fluid inlet element, is the relative pressure reported at the Shop Tests sheet, plus the barometric pressure at the time of the measurement. During Shop Tests the barometric pressure was slightly different between each measurement at different loads (Appendix I).

The exhaust gas pressure, which is used as input for the fixed fluid outlet element, is the relative exhaust gas pressure after the cylinder plus the barometric pressure. The exhaust gas pressure is not reported at Shop Tests` measurements, thus, it was set manually by assuming that this pressure will be by 0.25 to 0.6 bar less than the scavenging pressure (Table 4.8). By changing the fixed fluid exhaust gas pressure, the compression pressure was affected, as expected. The final value for the exhaust gas pressure, at this simulation stage, was set, when the calculated compression pressure converged the measured for each load. The difference between the exhaust gas pressure and the scavenge air, was maintained between 0.25 and 0.60bar.

Table 4.8, shows the fixed fluid element input data regarding the scavenging and exhaust gas pressure for each load.

Load	P_{baro} [1]	$P_{scav.}$ **(measured)** [2]	$P_{scav.}$ **(total)** [3]=[1]+[2]	$P_{exh.}$ **(Fixed fluid)** [4]	$\Delta P = (P_{scav.(total)} - P_{exh.})$ [5]=[3]-[4]
[%]	[bar]	[bar]	[bar]	[bar]	[bar]
100%	1.0245	2.65	3.6745	3.0800	0.5945
85%	1.0250	2.06	3.0850	2.6900	0.3950
75%	1.0255	1.75	2.7755	2.5200	0.2555
50%	1.0255	0.97	1.9955	1.6700	0.3255

Table 4. 8 Scav. and exh. pressure

4.2.4 Crank Shaft

The crank shaft is represented by the shaft load element. The model assumes that the shaft load is a rotating disk having a moment of inertia, and it is connected (mechanically) to a shaft element.

The values of the engine speed at each load, were inserted as input to that element.

Table 4.9, shows the Shop Test`s engine speed at each load. It is noted that the Shop Test Report contains two values for the engine speed (Appendix I), the "actual" and the "planned"; the first value is the measured engine speed, which is used at this simulation, whereas the last value is the speed setting from the engine control room.

Load (%)	50%	75%	85%	100%
Engine Speed (rpm)	93.0	106.4	110.8	117.2

Table 4. 9 Engine Speed

The description of the appropriate input data for the one cylinder simulation model, has been finished.

The sub-chapters that follow, provide a summary of the methodology described above and the output of the one cylinder simulation model.

4.2.5 Work Flow charts

Figure 4.10(a) below shows how the engine simulation model is being set in the MOTHER interface, whereas Figure 4.10(b) illustrates a logic flow chart according to which the engine simulation was feasible.

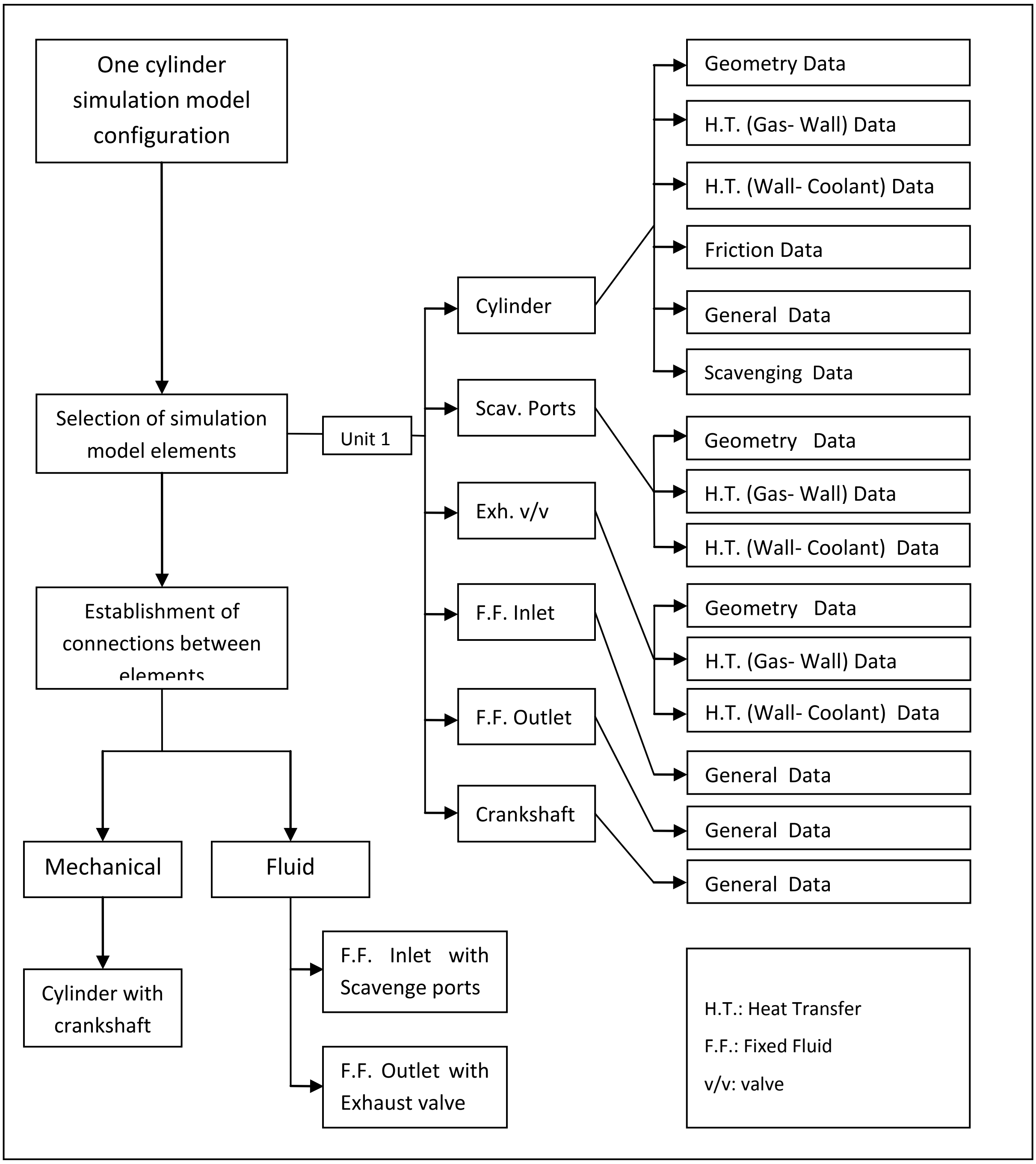


Table 4. 10(a) One cylinder simulation model chart

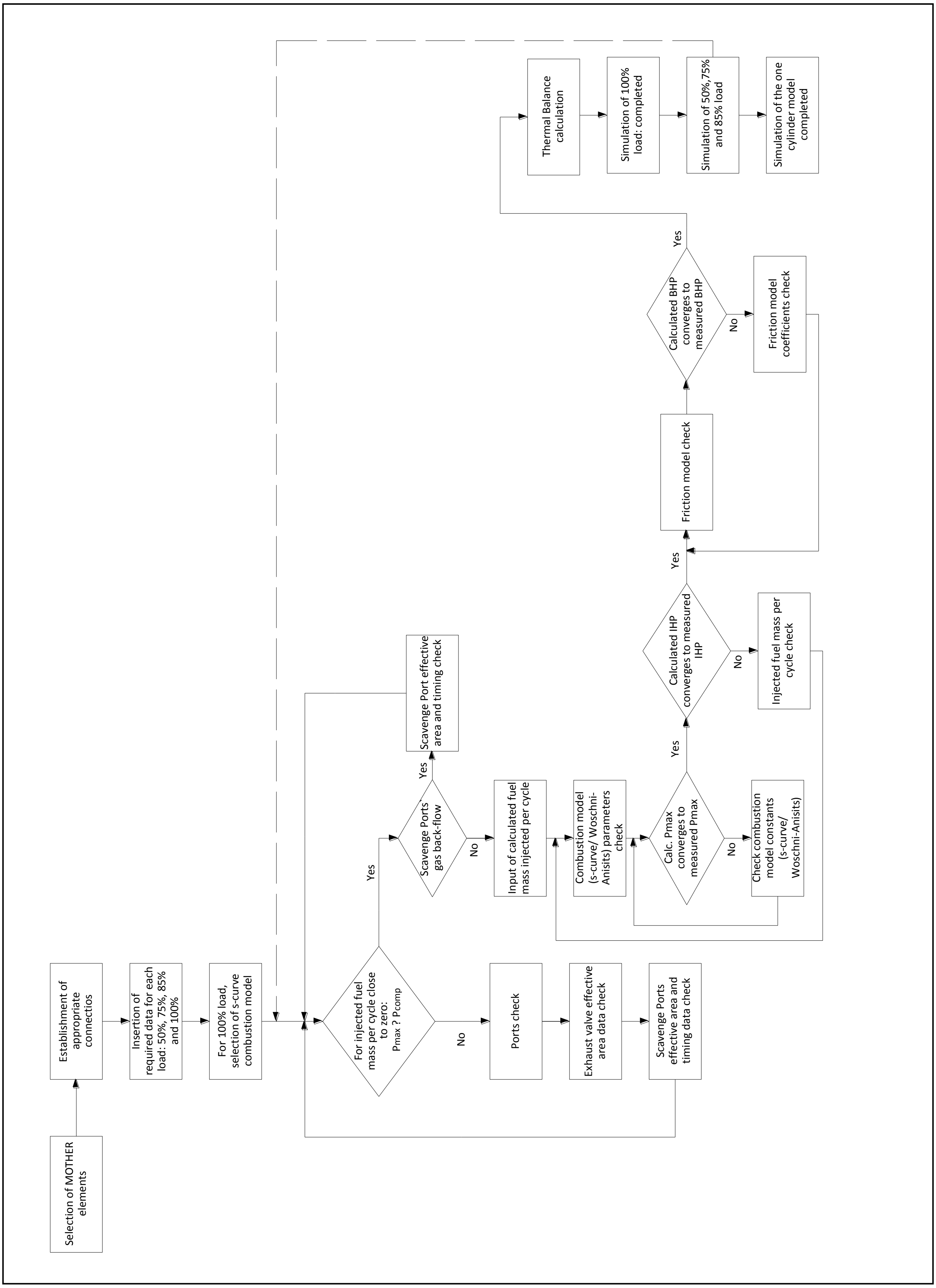


Fig. 4.10(b) One cylinder simulation flow chart

4.2.6 Results

The calculated values at this stage of the simulation with one acting cylinder, contribute to the evaluation of ports data, combustion model data and friction data.

The tables below show the results for compression, maximum pressure and brake horse power, for each load as calculated by 'MOTHER' and using the 'Woschni-Anisits' combustion model.

100% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	139.4	139.8	-0.3%
P_{comp}	bar	119.7	120.8	-0.9%
BHP	kW	1308.4	1291.5	1.3%

85% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	139.4	139.5	-0.1%
P_{comp}	bar	99.7	99.7	0.0%
BHP	kW	1115.6	1096.2	1.7%

75% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	130.1	130.0	-0.3%
P_{comp}	bar	89.6	89.7	-0.9%
IHP (*)	kW	1016.6	1032.9	-1.5%
BHP	kW	985.8	967.8	1.9%

50% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	99.8	99.7	0.1%
P_{comp}	bar	64.6	64.8	-0.3%
BHP	kW	653.1	648.7	0.7%

Table 4. 11 Results for the one cylinder configuration

(*) Although at Shop test report the IHP is not mentioned, the IHP that is compared with the calculated is based to data that found in performance reports (see 4.2.1.5.2).

The 75% Load was considered the reference load for the correct configuration of the Woschni-Anisits combustion model.

Thermal balance calculation

The purpose for calculating the thermal balance at this early stage of simulation, is the evaluation of the amount of energy which is given by the fuel and it is transformed to heat and brake power.

The fuel power can be calculated by multiplying the fuel mass injected per cycle by the calorific value. 'MOTHER' program assumes ISO fuel with caloric value 42700 kJ/kg. The calculation is made for the 75% Load. The following table shows the calculation of thermal balance.

Power	Calculation	Value	%
Fuel Power	$P_F = \dot{m}_f \Theta_u = 0.04611 kg/s\, x 42700\, kJ/kg \Rightarrow$	$P_f = 1968.90 kW$	100%
Total Heat Transfer Power [HT]	(as calculated by MOTHER program)	$HT = 127.55 kW$	6.48%
BHP	-»-	$BHP = 985.80 kW$	50.07%
Friction Power [fr]	-»-	$fr. = 30.80 kW$	1.56%
Exhaust Gas Power [P_{exh}]	$P_{exh.} = P_f - (HT. + BHP + fr.)$ (wasted power at exhaust gases is not calculated at the simulation with one cylinder because the T/C data don`t exist at this time)	$P_{exh.} = 824.70 kW$	41.80%

Table 4. 12 Thermal Balance calculation

The power distribution at Table 4.12, is within the acceptable margins for marine engines. It should be mentioned that the above analysis does not include the air cooler cooling power because the A/C is not included at this stage, thus the above results show a tendency only and they will updated when the simulation is completed.

Final remark

By considering the results in Tables 4.11 and 4.12, the simulation with one acting cylinder was completed successfully. The next stage is the re-evaluation of the above data with the 6-cylinder model, which is discussed at Chapter 5.

CHAPTER 5

SIMULATION WITH SIX (6) CYLINDERS

5.1. Introduction

The engine simulation continues by adding the other 5 cylinders to the one cylinder model. The new model has all the cylinders but the engine`s aspiration is controlled by the fixed fluid configuration instead of the T/C, which is added at the final stage of the simulation (Chapter 6). After several runs at each of the loads 50%, 75%, 85% & 100%, some minor corrections were made at Woschni-Anisits model parameters, friction model values and exhaust gas pressure (initial value). The mass flow through the exhaust valve and scavenge ports, was checked by plotting the relevant diagrams for each load.

5.2. Simulation Procedure

As mentioned in 5.1, the six cylinder model is based on the one cylinder model. The main difference between the two simulation models is the number of cylinders, whereas, the model`s structure illustrated in Fig. 4.10(a) remains the same.

Furthermore, regarding the cylinders` geometric data, each cylinder has discrete "Phase angle" value, which is determined by the crank shaft construction. Cylinder No. 1 is considered the reference cylinder. Consequently, it has a phase angle equal to zero. Fig. 5.1, on the right, illustrates the crankpin positions with the angle values between them. The annotations 'P1', 'P2', 'P3', etc., indicate the cylinder position (i.e. the phase angle) with respect to cylinder No.1. The cylinders` position is determined by the crankpin angle which is provided by the manufacturer and is attached at the Table 5.1.

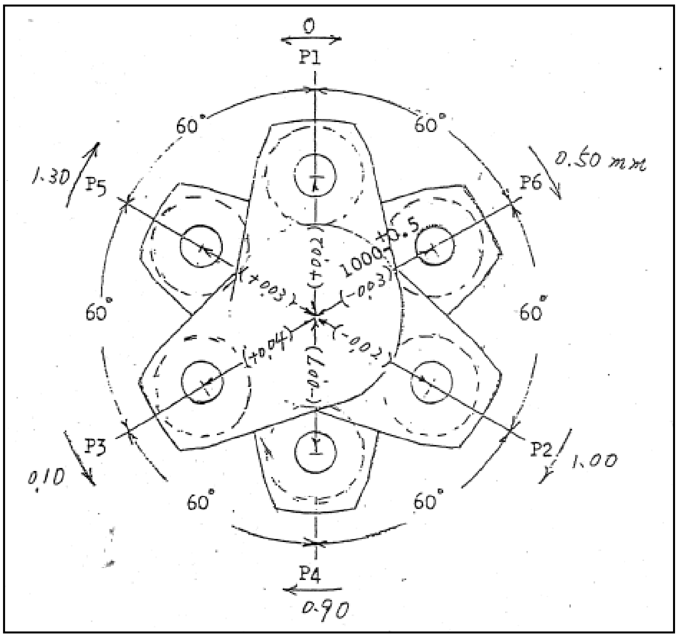


Fig. 5. 1 Crank shaft

Cylinder:	Cyl. 1	Cyl. 2	Cyl. 3	Cyl. 4	Cyl. 5	Cyl. 6
Crank Angle:	0^o	120^o	240^o	180^o	300^o	60^o

Table 5. 1 Crank angle of each cylinder

After the phase angle has been set for each cylinder, the appropriate connections between the cylinders were established. Each cylinder is connected mechanically with the crank shaft and through fluid connection with the Fixed Fluid Inlet and Outlet element. Fig. 5.2, below, illustrates the simulation procedure, similar to the aforementioned in Chapter 4.2.5.

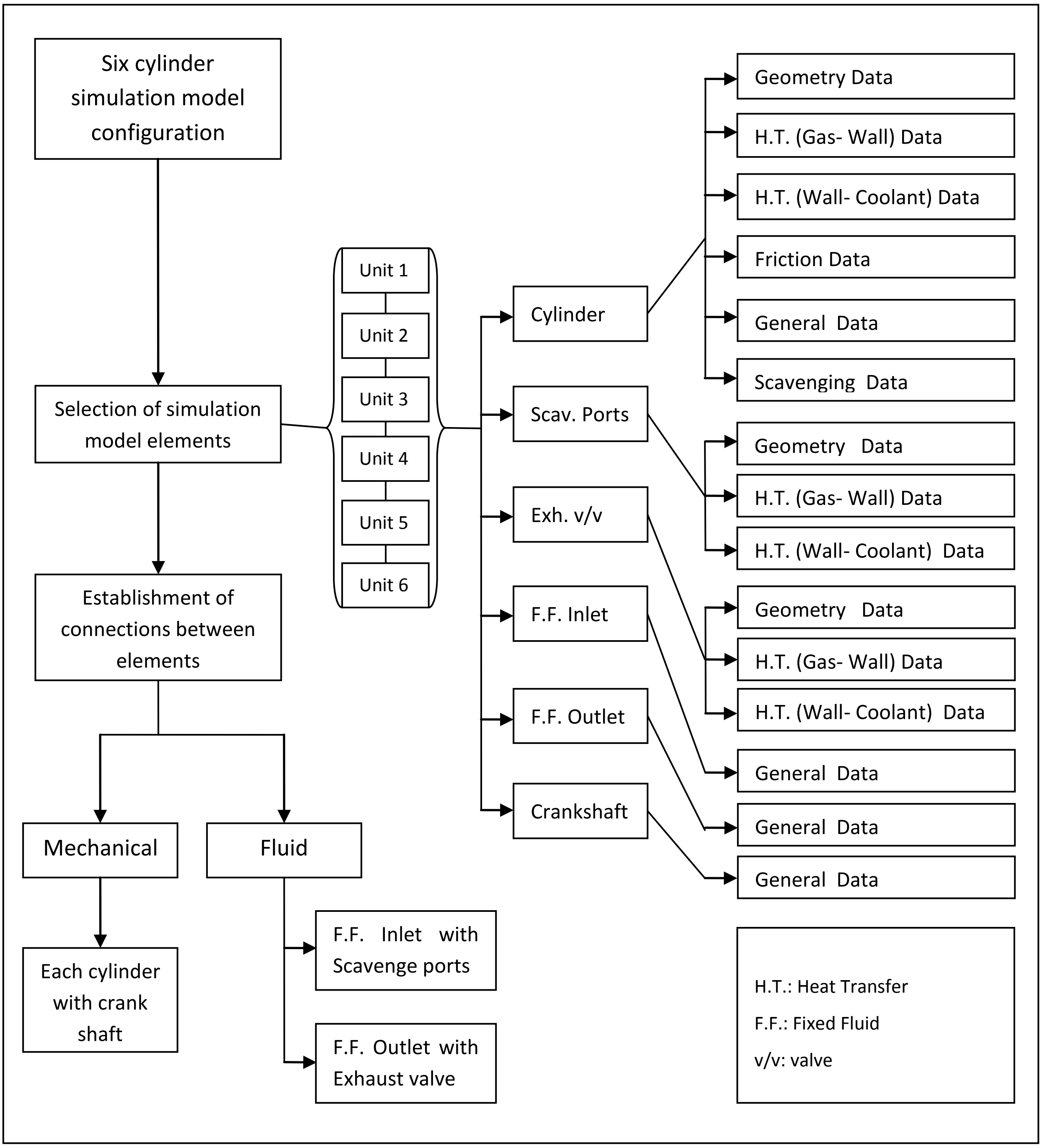


Fig. 5. 2 Six cylinder simulation model chart

The simulation procedure is terminated with the evaluation of the input data, by comparing the model calculated values with those measured at Shop Test. Minor corrections were made to the combustion model`s constants and friction model constants. Regarding the Fixed Fluid Outlet pressure (initial value), it was altered slightly in order to succeed convergence between the calculated compression and measured pressure at Shop Test.

5.3. Results

Table 5.1, below, shows the updated mean values for maximum and compression pressure and the brake power produced in each cycle, for the six cylinder model.

100% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	139.7	139.8	-0.07%
P_{comp}	bar	119.1	120.8	-1.40%
BHP	kW	7752	7749	0.04%

85% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	139.1	139.5	-0.3%
P_{comp}	bar	99.0	99.7	-0.7%
BHP	kW	6517	6577	-0.9%

75% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	129.9	130.0	-0.1%
P_{comp}	bar	89.5	89.7	-0.2%
BHP	kW	5773	5807	-0.6%

50% Load		Calculated	Shop Test	Deviation %
P_{max}	bar	99.7	99.7	0.0%
P_{comp}	bar	64.1	64.8	-1.0%
BHP	kW	3906	3892	0.4%

Table 5. 2 Results for the six cylinder model

According to Table 5.2, the calculated values have an error less than 3%, consequently, the simulation results fulfill the general target in succeeding deviations less than 3%.

5.4. Thermal balance calculation

Table 5.3, includes the thermal balance calculation for the six cylinder model. 'MOTHER' program cannot perform this calculation because the inlet and exhaust receivers have not been installed yet, consequently a rough calculation is being made manually. This calculation is helpful because it shows the magnitude of each value and the power percentage can be easily compared with other admissible percentages for marine engines.

Power	**Calculation**	**Value**	**%**
Fuel Power	$P_F = 6\dot{m}_f \Theta_u = 6x0.04611 kg/s\, x42700\, kJ/kg \Rightarrow$	$P_f = 11813 kW$	100%
Total Heat Transfer Power [HT]	(as calculated by MOTHER program)	$HT = 758 kW$	6.40%
BHP	-»-	$BHP = 5773 kW$	48.90%
Friction Power [fr]	-»-	$fr. = 315 kW$	2.70%
Exhaust Gas Power [P_{exh}]	$P_{exh.} = P_f - (HT. + BHP + fr.)$ (wasted power at exhaust gases is not calculated at the simulation with six cylinders because the T/C data don`t exist at this time)	$P_{exh.} = 4967 kW$	42.0%

Table 5. 3 Thermal balance calculation for the six cylinder model

The power percentages in Table 5.2 are within the admissible one for marine engines and they are close to that calculated in Table 4.12 for the one cylinder model.

Consequently, the six cylinder model simulation is completed with the relevant checks. The final stage of the RTA48-T simulation is discussed at Chapter 6.

CHAPTER 6

FINAL SIMULATION STAGE – SHOP TEST RESULTS

6.1. Introduction

The 6RTA48-T`s simulation was completed by adding (to the already existing six cylinder model) the T/C data, air cooler (A/C) data and the inlet and exhaust receiver data and by establishing the correct fluid and mechanical connections between the various elements. At this stage of simulation, the exhaust gas pressure is calculated automatically according to turbine data. Some data, such as initial values (i.e. cyl pressures & temperatures), Woschni-Anisits coefficients, friction model coefficients, have been updated in such a way that the engine model calculates the Shop Test measured data, with the minimum possible error. Due to the fact the turbocharger`s data are confidential, neither figure showing the compressor nor the turbine map, will be attached.

Fig. 6.1, below, illustrates the 6RTA48-T`s arrangement. The air entering to the compressor is simulated by a Fixed Fluid element in “MOTHER”`s interface, which is called ‘Engine Inlet’. With the same thought, the exhaust gas that exits the turbine towards the atmosphere, is simulated by another Fixed Fluid element, which is called ‘Engine Outlet’. All the elements used for the engine simulation, are shown in Fig. 6.2, where the simulation procedure is illustrated.

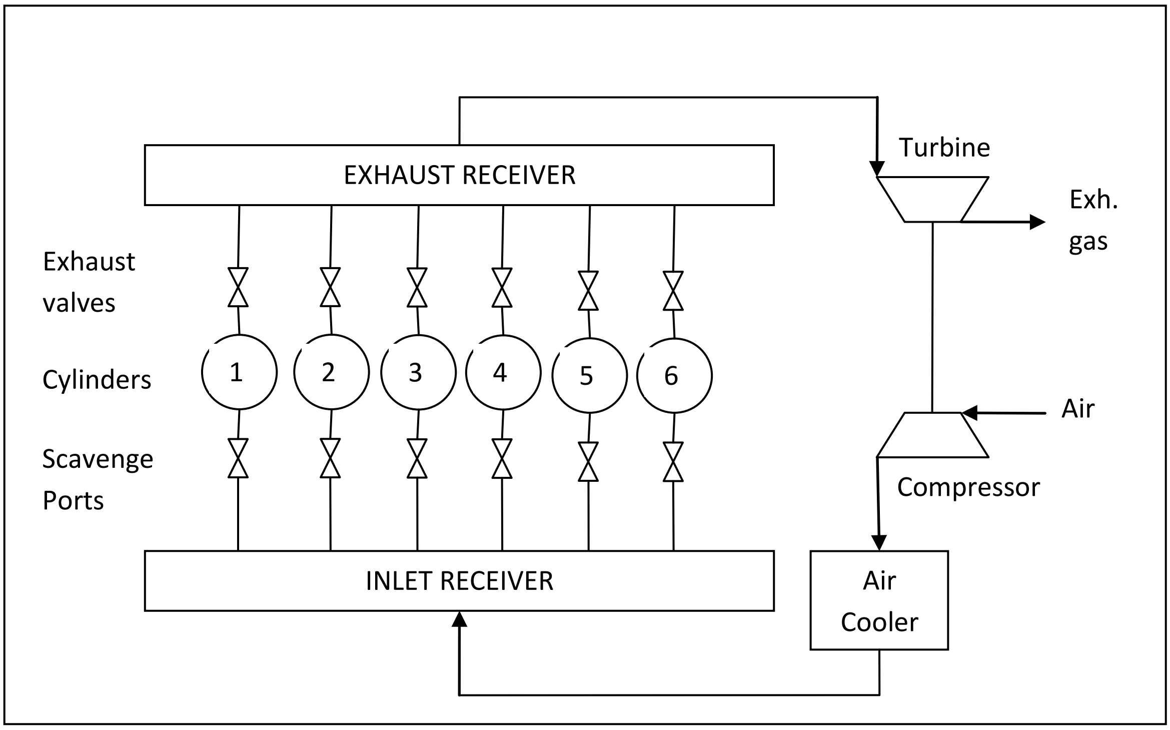


Fig. 6. 1 6RTA48-T arrangement

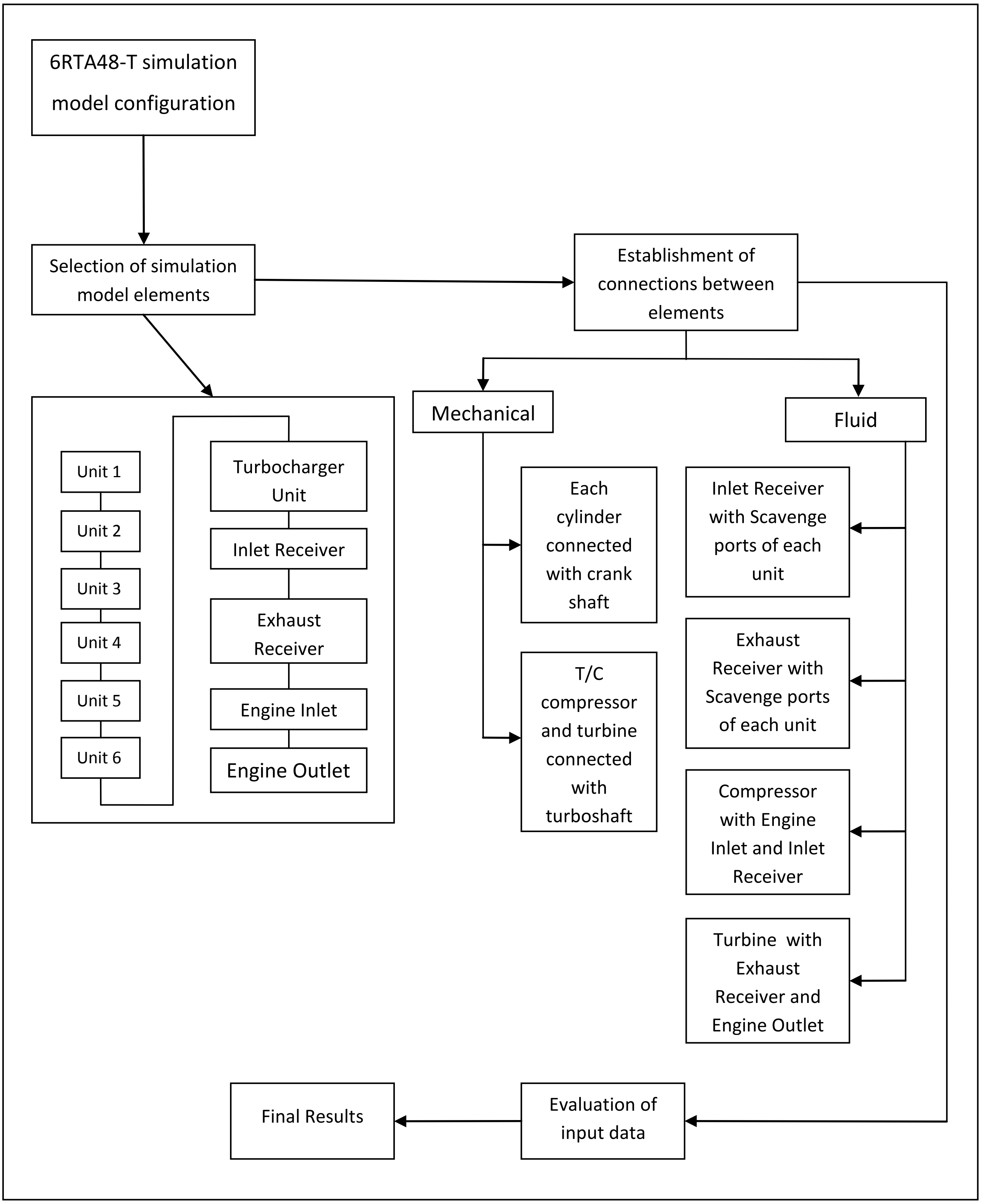


Fig. 6. 2 Simulation procedure

Several runs were made before the model converges with the minimum possible error and the final results arise. This chapter provides the description of the added elements and the simulation results of Shop Test for the following loads: 50%, 75%, 85% and 100%.

6.2. Combustion model

As stated in Chapter 4.2.1.5, the combustion`s model (i.e. Woschni-Anisits model) main input data, were the following:

- Model parameter “ a ”
- Start of static injection in CA degrees
- Mass of fuel injected per cycle per cylinder in kg.

The reference load was the 75% load, as described in Chapter 4. The Woschni-Anisits model parameters were evaluated according to the methodology described in Chapter 4.2.1.5.2.

The final values for the “ a ” parameter, together with the other required input data for the combustion model, are shown in Table 6.1, below.

Load			**50%**	**75%**	**85%**	**100%**
RPM			92.9	106.3	110.8	117.0
Load Indicator Position			5.9	6.9	7.2	7.9
L.I.xRPM			548.1	733.5	797.8	924.3
Woschni-Anisits “ a ” parameter	a	$[-]$	3.8	5.6	6.8	9.95
Start of static injection	SOI	$[CA\ \deg]$	-3.5	-6.0	-5.6	-4.0
Mass of fuel injected per cycle per cylinder	m_{tot}	$[kg]$	0.02000	0.02539	0.02757	0.0316

Table 6. 1 Combustion model parameters

In practice, the engine load is expressed by the Load Indicator`s position and the relevant engine speed. By multiplying the L.I. position with the engine speed, both factors are taken into account in estimating the engine load. The relevant methodology in estimating the engine output, is described in Chapter 8.4. However, the Woschni-Anisits “a” parameter relation with L.I.xRPM, from Shop Test simulation, is essential in order to calculate the “a” parameters of Sea Trials and Performance runs with given the value of L.I.xRPM.

Note: As per manufacturer`s methodology, in Table 6.1 the planned engine speed is used for the calculation of L.I.xRPM, instead of the achieved in Shop Test.

Figure 6.3 illustrates the " *a* " parameter as function of L.I.xRPM, as depicted from Table 6.1.

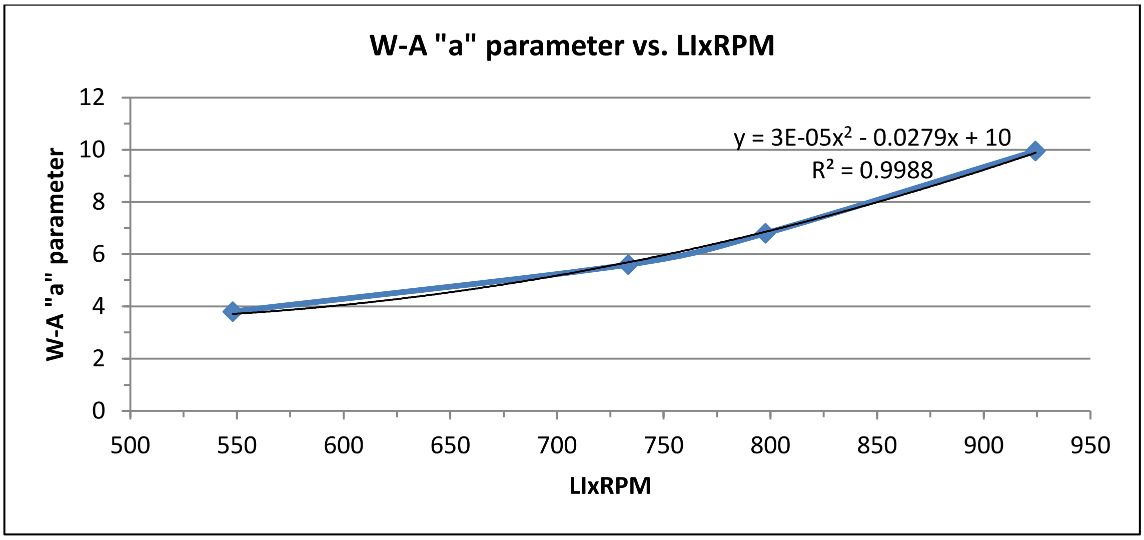


Fig. 6. 3 Woschni-Anisits "a" parameter vs. LIxRPM

Figure 6.4, illustrates the increase in fuel mass injected per cycle per cylinder, with respect to the engine load which is expressed as percentage of the Maximum Continuous Rating (MCR).

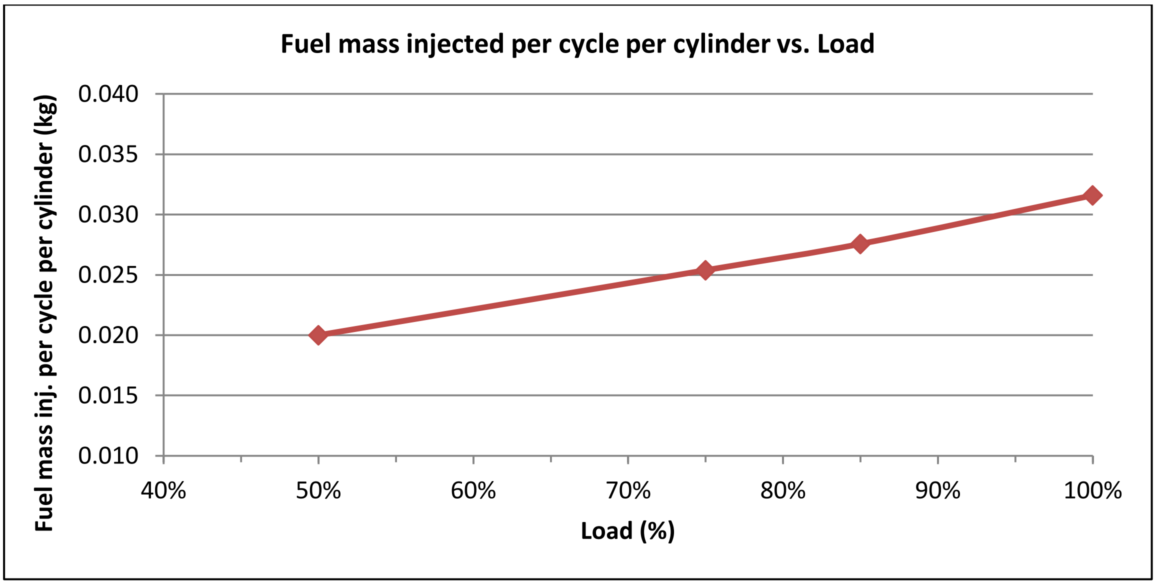


Fig. 6. 4 Fuel mass injected per cycle per cylinder vs. Engine Load

6.3. Friction model adjustments

Brake power prediction depends on how accurate the friction model is; consequently, the initial friction coefficients, that were selected in Chapter 4.2.1.4, were updated in order the calculated BHP to converge to the measured BHP value with the minimum possible error. The table below shows the final values for k1, k2 and k3 coefficients.

LOAD	M/E RPM	L.I.	L.I.xRPM	k1	k3	k2
50%	**92.9**	5.9	548.1	5000	630	0
75%	**106.3**	6.9	733.5	9000	630	0
85%	**110.8**	7.2	797.5	10000	600	0
100%	**117.0**	7.9	924.3	12500	400	0.003

Table 6. 2 Friction model coefficients

Figures 6.5, 6.6 and 6.7, below,, show the k_1, k_2 coefficients as function of L.I.xRPM and k_3 as function of the engine speed. The purpose of presenting the data in Table 6.2, as functions, will be clear in Chapter 8; nevertheless, it is possible to calculate the k_1, k_2, k_3 for any operating point which is described by the brake power and the engine speed.

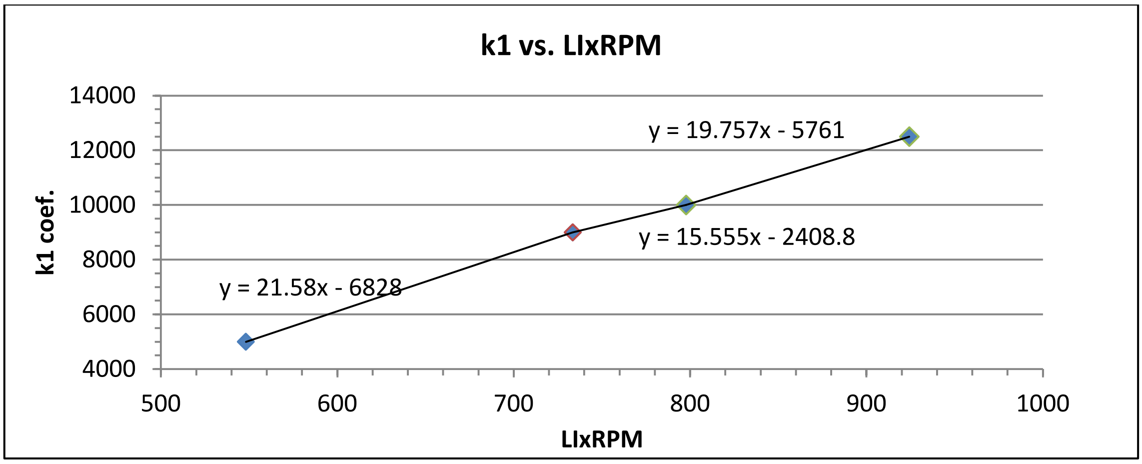


Fig. 6. 5 "k1" coef. vs. LIxRPM

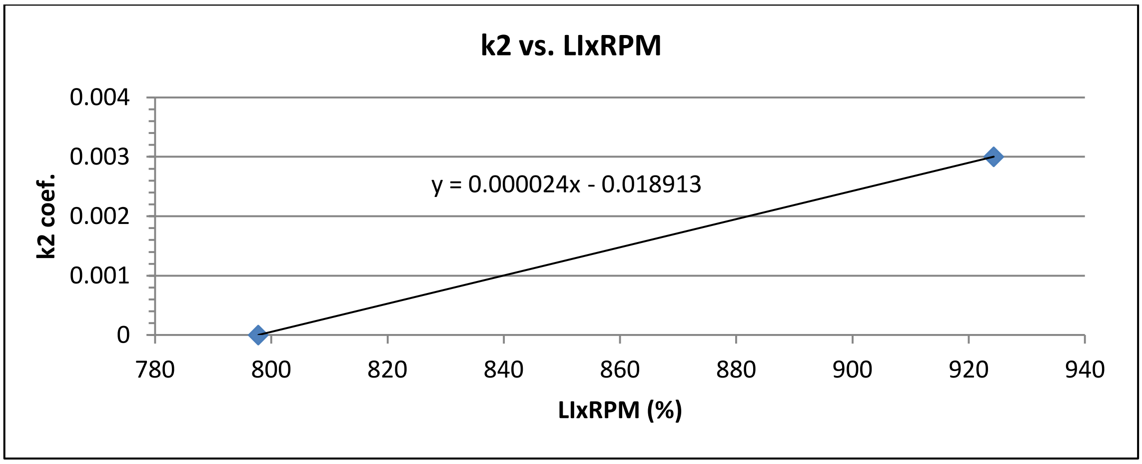


Fig. 6. 6 "k2" coef. vs. LIxRPM

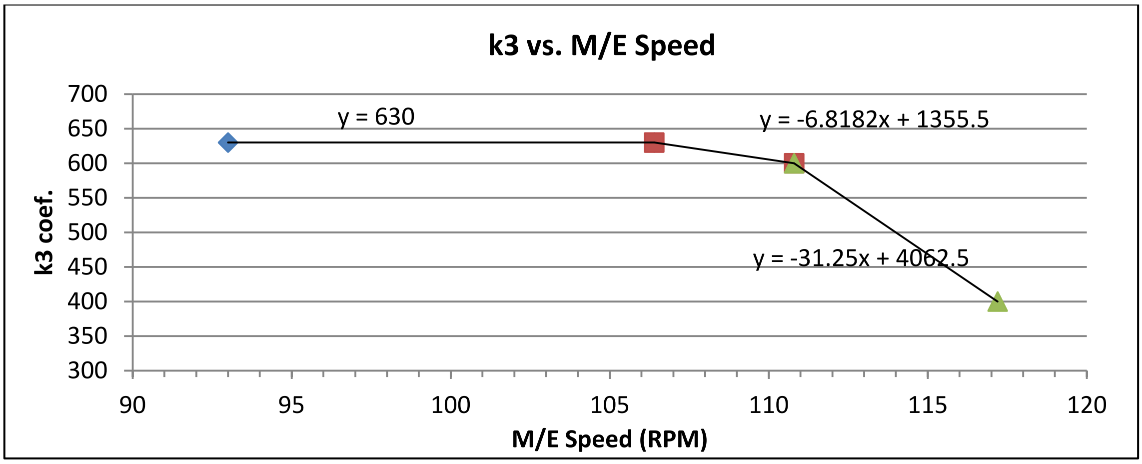


Fig. 6. 7 "k3" coef. vs. M/E Speed

6.4. Model add-ins

As stated above, the elements stated below were added to the existing engine 6-cylinder simulation model:

- Scavenge air receiver
- Exhaust receiver
- Turbocharger
- Engine Inlet
- Engine Outlet

The scavenge and exhaust receiver`s volume are 3.69m^3 and 3.89m^3 respectively (refer to Appendix IV). The volumes have been calculated from the dimensions given in the drawings supplied by the manufacturer (see Fig. 3.8 & 3.9).

6.4.1. Scavenge air receiver

The scavenge receiver`s simulation in MOTHER interface, is possible through the following data:

- Geometry data
- Heat transfer (gas- wall) data
- Heat transfer (wall- coolant) data
- General data

Regarding the **'Geometry data'**, only the receiver`s volume is required as input:

$$V_{Scav.\mathrm{Receiver}} = 3.69m^3 \tag{6.1}$$

The instantaneous **heat flux** from the plenum gas (i.e. scavenge air) to the plenum wall (i.e. scavenge receiver`s wall) is calculated at each step of the simulation procedure by the following relation:

$$q = h_{gas} \cdot A_{gas} \cdot \left(T_{gas} - T_{wall}\right) \tag{6.2}$$

where:

q	instantaneous heat flux	$[kW]$
h_{gas}	instantaneous spatial average heat transfer coefficient of plenum gas	$[kW/m^2K]$
A_{gas}	gas side plenum wall area	$[m^2]$
T_{gas}	instantaneous plenum gas temperature	$[K]$
T_{wall}	mean surface temperature of the plenum wall	$[K]$

The heat transfer coefficient, h_{gas}, of the plenum gas (i.e. scavenge air) is calculated by the Nusselt- Reynolds- Prandtl relationship, that follows:

$$Nu = C_1 \operatorname{Re}^{C_2} \operatorname{Pr}^{C_3} \tag{6.3}$$

where:
$C_1 = 0.022$
$C_2 = 0.760$
$C_3 = 0.400$

Regarding the **heat transfer** from the plenum wall to the coolant, the 'Simple Model' was used. According to this model, the plenum is regarded as a cylindrical duct and its gas side surface wall temperature is considered to be between the temperature of the cooling medium of the plenum and the plenum wall adiabatic temperature (as in case of the cylinder heat transfer wall to coolant- Simple Model, Chapter 4.2.1.3). The cooling mean for the scavenge air receiver is the air of the engine room. The surface temperature of the plenum wall can be calculated by the relation:

$$T_{PLG} = HTF_P \cdot T_{CPL} + \left(1 - HTF_P\right) \cdot T_{w,a} \tag{6.4}$$

where:

T_{PLG}	surface temperature of the plenum wall	$[K]$
T_{CPL}	plenum cooling medium temperature	$[K]$
$T_{w,a}$	plenum wall adiabatic temperature	$[K]$
HTF_P	plenum heat transfer factor, 0 or 1	$[-]$

The plenum wall adiabatic temperature is calculated by the relation:

$$T_{w,a} = \frac{\int hAT_g d\theta}{\int hAd\theta} \tag{6.5}$$

When the surface temperature of the plenum is equal to its adiabatic temperature, no heat will be transferred from the plenum gas to the plenum wall in an engine cycle, $HTF_P = 0$. When the surface temperature of the plenum wall is equal to its cooling medium temperature, the maximum heat will be transferred from the gas to the wall in an engine cycle, $HTF_P = 1$.

In the '**General Data'** tab (of MOTHER interface), initial values of the scavenge air pressure, P_{scav}, temperature, T_{scav}, and equivalence ratio, φ, are required. Thus, the measured values of pressure and temperature, from Shop Test for each load, are used as initial data. As far as the equivalence ratio, is concerned, its value is close to zero because the scavenge receiver contains only pure air (the equivalence ratio indicates the presence of unburnt fuel, i.e air to fuel ratio).

Table 6.3 shows these initial values for the inlet receiver (where 'Scav. Air Temp.' is the temperature after the A/C). It should be pointed out that the scavenging pressure has been measured through a manometer, which measures the pressure difference between two positions; the one is the scavenging receiver and the other is the barometric pressure. Thus, the measurement of P_{scav} is the relative scavenging pressure, in which the barometric pressure is not included. Consequently, the scavenging air pressure that is inserted in MOTHER as initial value, is the product of the addition of the measured pressure and the barometric pressure.

Load	**P_{baro}** [1]	**P_{scav} (measured)** [2]	**P_{scav} (total)** [3]=[1]+[2]	**Scav. Air Temperature** [4]	**Equivalence ratio** [5]
[%]	[bar]	[bar]	[bar]	[°C]	[-]
100%	1.0245	2.65	3.6745	44	0.001
85%	1.0250	2.06	3.0850	40	0.001
75%	1.0255	1.75	2.7755	39	0.001
50%	1.0255	0.97	1.9955	32	0.001

Table 6. 3 Initial values for the scavenge receiver

6.4.2. Exhaust gas receiver

The exhaust receiver is considered a plenum in MOTHER interface, and it is described by the following data (similar to Scavenge air receiver):

- Geometry data
- Heat transfer (gas- wall) data
- Heat transfer (wall- coolant) data
- General data

The governing equations which are used for the various calculations, related to the exhaust gas receiver, are the same with those described for the scavenge air receiver (Eq. 6.2 to 6.5).

Regarding **'Geometry data'**, the required input is the exhaust receiver`s volume, which is the following:

$$V_{exh.receiver} = 3.89m^3 \tag{6.6}$$

Regarding the **'Heat transfer (gas-wall) data'**, the Nusselt-Reynolds-Prandtl model was used; the model`s constants are the following:

$C_1 = 0.03$
$C_2 = 0.80$
$C_3 = 0.40$

Regarding the **'Heat transfer (wall- coolant) data'**, the simple model was used. The engine room temperature is the plenum`s coolant temperature.

Regarding the **'General data'**, the required exhaust receiver`s initial values selection is quite different compared to those of scavenge receiver`s. The exhaust gas temperature after cylinders, has been reported at Shop Test for each load, in contradiction with their pressures, which have not been reported. Consequently, the initial exhaust gas temperature (in MOTHER interface) is considered to be equal to the measured reported temperature ($T_{exh.\ after\ cyl}$) at Shop Test. Regarding the initial exhaust gas pressure, a loop procedure was utilized in order to estimate it. A trial exhaust gas pressure value was evaluated with the calculated value by the model. In case the calculated value did not converge to the selected value, the latter was updated by substituting it by the calculated value. The first trial exhaust gas pressure was acquired from the six cylinder model of the previous simulation stage (from 'Fixed Fluid Outlet' element, refer to Chapter 5.1). The required initial values are attached at Table 6.4 below.

Load	**Initial Exh. Gas pressure** [1]	**Initial Exh. Gas Temperature** [2]	**Initial Equivalence ratio** [3]
[%]	[bar]	[^{o}C]	[-]
100%	3.21	367	0.279
85%	2.76	338	0.266
75%	2.49	327	0.260
50%	1.80	303	0.245

Table 6. 4 Exhaust receiver`s initial value

6.4.3. Turbocharger and Air Cooler

The turbocharger unit, in MOTHER interface, is described by the following elements:

- Compressor
- Turbine

- Turboshaft

6.4.3.1. Compressor

The compressor is described by the following data:

- General Data
- Heat Transfer (Gas- Wall) data
- Heat Transfer (Wall- Coolant) data

Regarding the **General Data** tab in MOTHER interface, the compressor map data and impeller diameter were defined. The compressor map represents the compressor speed and efficiency as functions of the compressor air corrected mass flow with respect to pressure ratio. Using the instantaneous values of the compressor pressure ratio and the compressor speed, the mass flow rate and the compressor efficiency can be calculated. For any given pair of pressure ratio and turbocharger speed, the efficiency and the mass flow rate of the compressor are calculated by interpolation between adjacent points. A sample compressor map, as it appears in MOTHER interface, is shown in Fig. 6.8. Because the turbocharger`s (TPL73 B-12) compressor map is confidential, only the sample map is attached below.

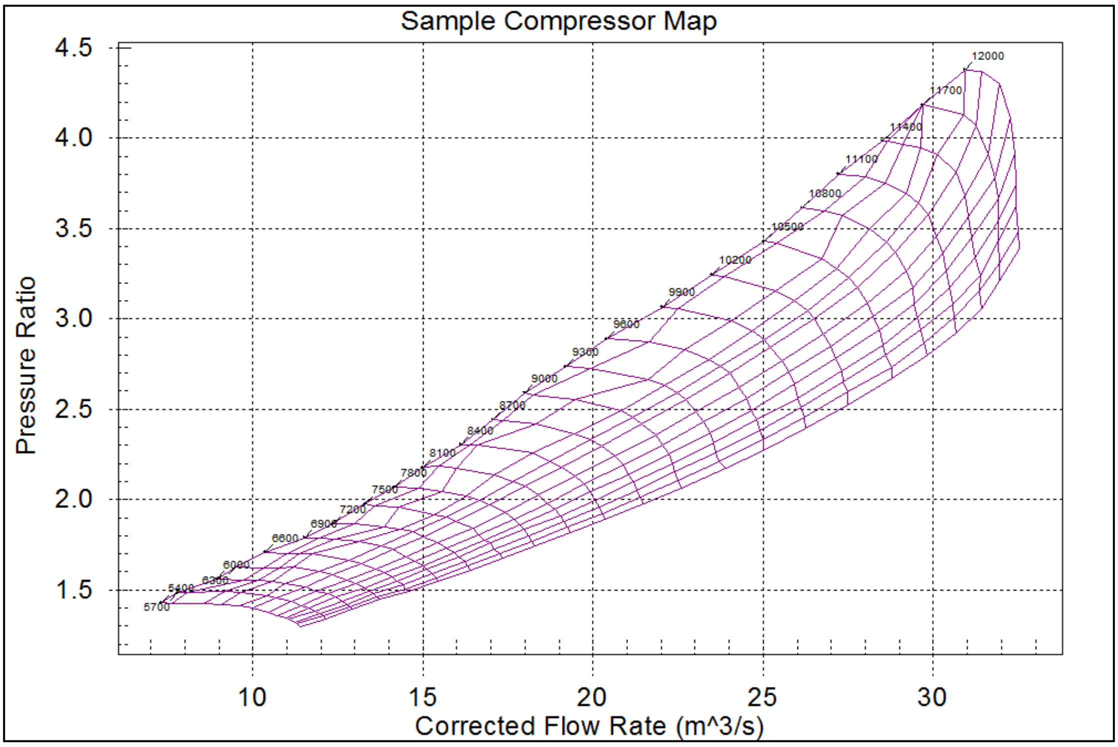


Fig. 6. 8 Sample Compressor Map (not the 6RTA48-T)

The compressor pressure ratio, is the ratio between the pressure of the flow receiver connected downstream of the compressor and the pressure of the flow receiver connected upstream of the compressor. As the air cooler is connected downstream of the compressor, the pressure drop of the air passing through the air cooler is taken into consideration.

MOTHER uses the first law of thermodynamics in order to calculate the work required to drive the compressor. Thus, the work transfer rate to the compressor is calculated by the following equation:

$$\dot{W}_c = \dot{m}_c \left(h_{02} - h_{01} \right) + q_c \tag{6.7}$$

where:

$\dot{W}_c$	work tranfer rate to the compressor	$[kW]$
$\dot{m}_c$	compressor air mass flow rate	$[kg/s]$
h_{01}	total specific enthalpy of the air entering the compressor	$[kJ/kg]$
h_{02}	total specific enthalpy of the air exiting the compressor	$[kJ/kg]$
q_c	heat transfer rate (from the compressor gas to wall)	$[kW]$

The work transfer rate to the compressor is the power that has to be provided to drive the compressor impeller. Thus, the compressor impeller torque is delivered by the following equation (Eq. 6.8):

$$T_c = \frac{\dot{W}_c}{\omega} \tag{6.8}$$

where:

T_c	compressor impeller torque	$[kNm]$
ω	compressor impeller angular velocity	$[rad/s]$

Since $\omega = 2\pi N/60$, the equation 6.8 becomes:

$$T_c = 30 \cdot \frac{\dot{W}_c}{\pi N} \tag{6.9}$$

where:

N	rotational speed of the compressor impeller	$[rpm]$

Regarding the **Heat Transfer (Gas-Wall) data**, the Nusselt-Reynolds-Prandlt relationship was used (as in Eq. 6.3) in order to calculate the heat transfer coefficient of the gas (i.e. fresh air) passing through the compressor.

$$Nu = C_1 \operatorname{Re}^{C_2} \operatorname{Pr}^{C_3} \tag{6.10}$$

where:

$C_1 = 0.022$

$C_2 = 0.800$

$C_3 = 0.400$

Regarding the **Heat Transfer (Wall- Coolant) data**, the insulated model was selected, according to which, no heat is transferred from the compressor wall to the coolant mean.

Air Cooler (A/C)

The Air Cooler is always connected downstream of the compressor (Fig. 6.1). The Air Cooler\`s versatility is described in Chapter 3.8; its operation results in increasing the air\`s density, causing more mass of air to enter to the cylinders, thus, enabling more fuel to be burnt. In MOTHER interface, the cooling mean\`s temperature (i.e. water temp.) and the A/C\`s efficiency, are required. The A/C efficiency is given by the following formula:

$$n_{A/C} = \frac{T_{air\ in} - T_{air\ out.}}{T_{air\ in} - T_{water\ in}} \tag{6.11}$$

The A/C efficiency is calculated by the already measured temperatures at Shop Test. Table 6.5 shows this calculation according the equation 6.11.

LOAD:		50%	75%	85%	100%
Air temp. bfr. A/C	[^{o}C]	100	140	155	185
Air temp. aft. A/C	[^{o}C]	32	39	40	44
A/C inlet water temp.	[^{o}C]	21	21	21	21
Air Cooler Efficiency	**[-]**	**0.8608**	**0.8487**	**0.8582**	**0.8598**

Table 6. 5 Air Cooler Efficiency calculation

Fig. 6.9 illustrates the Air Cooler Efficiency with respect to the Engine Load, as calculated by the equation 6.11.

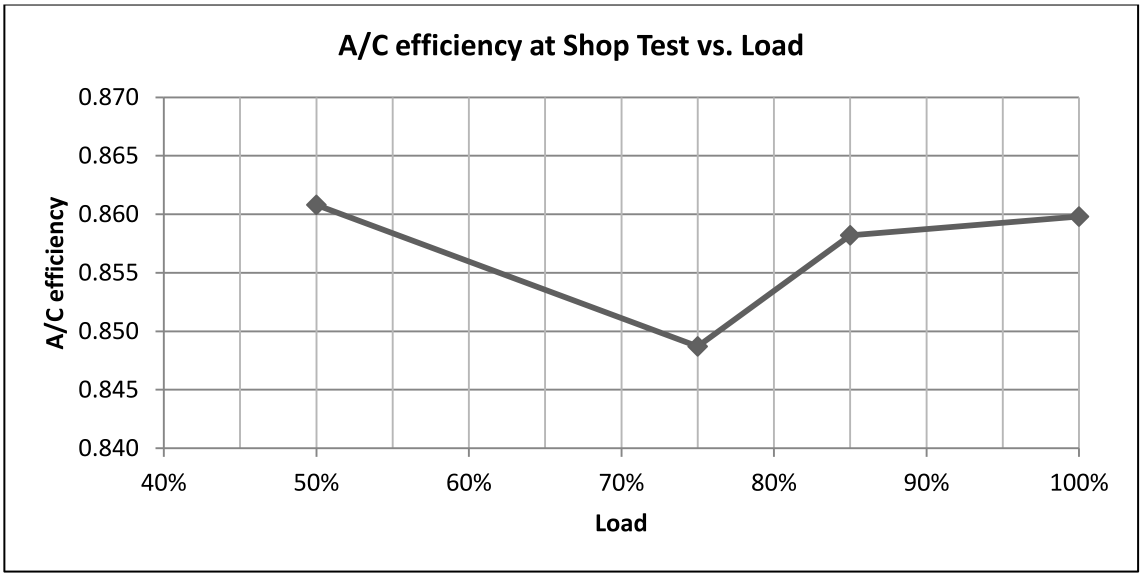


Fig. 6. 9 Air Cooler efficiency at Shop Test

In MOTHER interface, the air cooler efficiency is calculated by the following equation.

$$n_{A/C} = C_1 + C_2 \cdot \dot{m}_{air} + C_3 \cdot \dot{m}_{air}^2 \tag{6.12}$$

where:

$n_{A/C}$	air cooler efficiency	$[-]$
$\dot{m}_{air}$	air mass flow rate	$[kg/s]$
C_1, C_2, C_3	constants	$[-]$

Thus, there are two options in expressing the air cooler efficiency:

- Constant $C_1 \neq 0$, and $C_2 = C_3 = 0$: consequently, the A/C efficiency is inserted manually.
- Constants C_1, C_2 and $C_3 \neq 0$: consequently, the A/C efficiency is calculated by MOTHER as a function of air mass flow rate (Fig. 6.10).

According to Fig. 6.9, the A/C efficiency raises from 75% to 100% of load, whereas at 50% it has the higher value. Thus, a combination of the above mentioned options was made for the A/C efficiency calculation after considering the A/C efficiency form (Fig. 6.9). At 50% load the A/C efficiency was inserted manually ($C_1 = 0.8608,\ C_2 = C_3 = 0$), whereas a function of the air mass flow with the A/C efficiency, was used for the other loads. The air mass flow rate is calculated by MOTHER; Fig. 6.10 illustrates the A/C efficiency as function of the air mass flow, beginning from the 75% load.

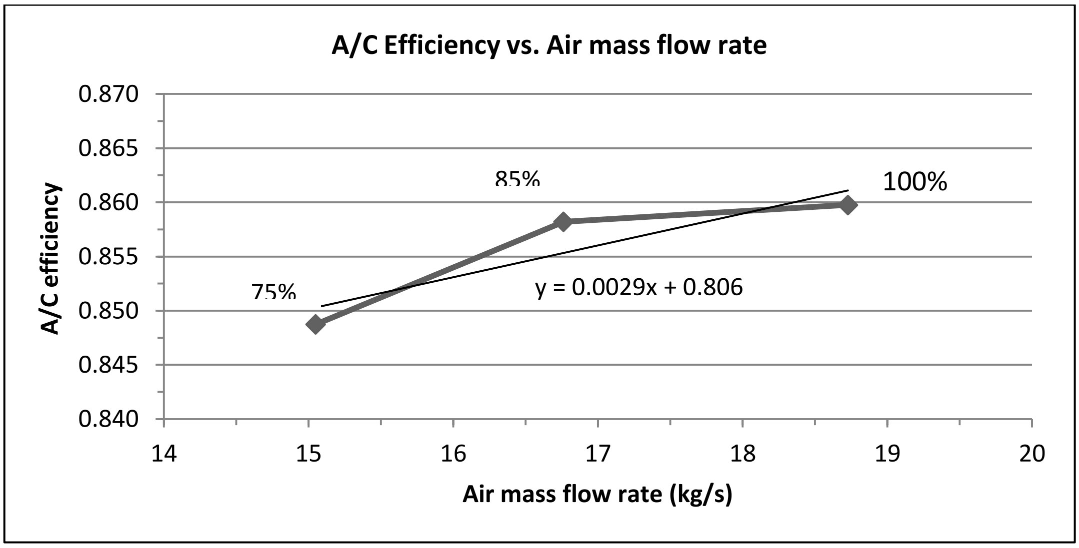


Fig. 6. 10 A/C efficiency vs. Air mass flow rate

Summarizing, the input data in MOTHER interface, regarding the A/C efficiency, are the following:

- For 50% load: $C_1 = 0.8608,\ C_2 = C_3 = 0$

- For 75%, 85% and 100%: $C_1 = 0.806,\ C_2 = 0.0029,\ C_3 = 0$; (constants C_1, C_2, C_3 are depicted from equation at Fig. 6.10)

Except for the A/C`s efficiency and the coolant temperature, the '*Pressure loss coefficient'* is required to be calculated and to be entered at the relevant tab in MOTHER interface. The pressure drop through the air cooler is depended on the size and detailed design and could be expressed as a function of the mass flow rate of the in-flowing air:

$$\Delta P_{cool} = K \rho_{in} \frac{u_a^2}{2} \tag{6.13}$$

$$m = \rho_{in} A u_a \text{ and } \rho_m = \frac{p_{in}}{RT_{in}} \tag{6.14}$$

where:

ΔP_{cool}	air cooler pressure drop	$[Pa]$
K	pressure loss coefficient (constant)	$[-]$
u_a	velocity of the air entering the air cooler	$[m/s]$
ρ_{in}	density of the air entering the air cooler	$\left[kg/m^3\right]$
m	mass flow rate of the air passing through the air cooler	$[kg/s]$
A	air cooler inlet area	$\left[m^2\right]$
R	gas constant	$[-]$
T_{in}	temperature of the air entering the air cooler	$[K]$
P_{in}	pressure of the air entering the air cooler	$[Pa]$

Combining the equations 6.13 and 6.14, the pressure drop through the air cooler can be written in the following form:

$$\Delta P_{cool} = K \left(\frac{m}{A}\right)^2 \frac{RT_m}{2P_{in}} \tag{6.15}$$

The pressure loss coefficient values were selected through a trial-and-error procedure for each testing load. A trial value was initially selected; afterwards, the model`s calculated 'Pressure Drop' was compared with the measured value at Shop Test. In case the two values do not converge, the constant is altered until a convergence is succeeded. In case the calculated and the measured 'Pressure Drop' values converge, the loop procedure finishes. The final values for the 'Pressure loss coefficient' are reported in relation with the engine load. The 'Pressure Drop' across the air cooler is interdependent with the 'Pressure loss coefficient', but the 'Pressure Drop' is related with the 'Air mass flow rate' (Eq. 6.15), consequently, the 'Pressure loss coef.' could be a function of the 'Air mass flow rate'. Table 6.6 shows the aforementioned relation.

Load	Pressure loss coefficient [-]	Air mass flow rate [kg/s]
100%	33.5	18.75
85%	33.9	16.76
75%	36.5	15.05
50%	43.7	10.68

Table 6. 6 Pressure loss coefficient vs. Air mass flow rate

Fig. 6.11, illustrates the data in Table 6.6 as plotted in the MOTHER program.

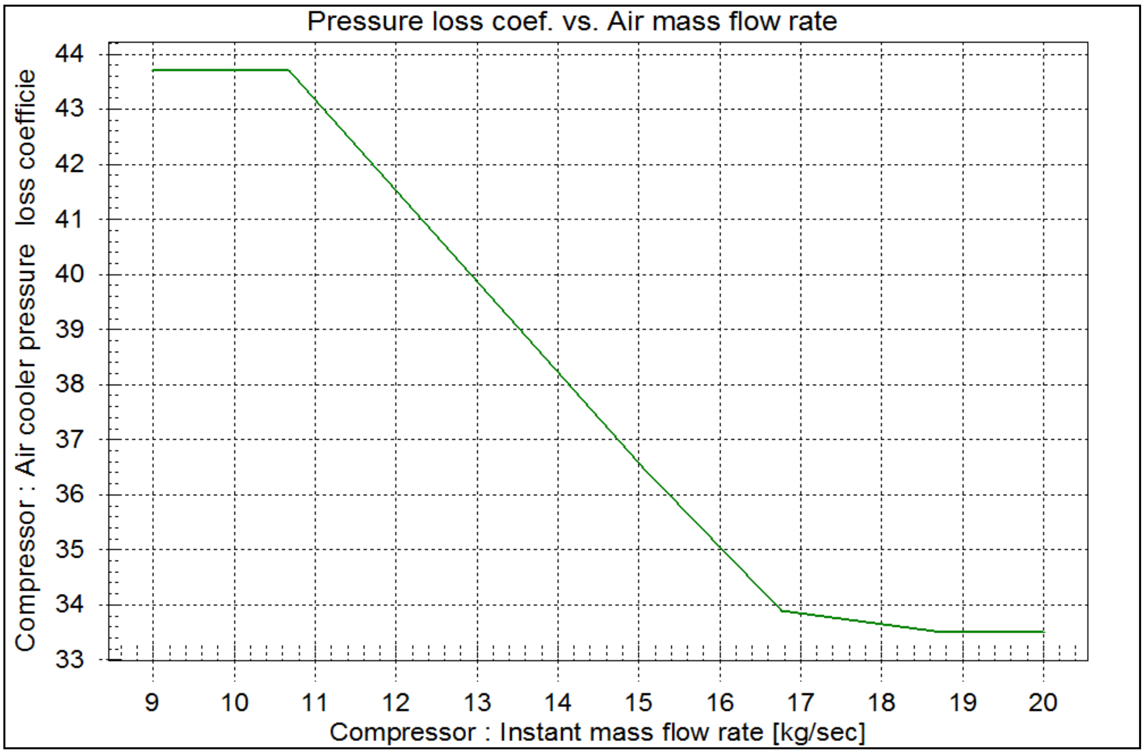


Fig. 6. 11 Pressure loss coefficient vs. Air mass flow rate

Apart from the above, the air cooler model requires the calculation of the equivalent a/c area, which is actually the minimum flow area of the air cooler used for the calculation of the a/c pressure loss. The equivalent area calculation, which value is 1.5m^2, and the air cooler design are attached at Appendix IV.

6.4.3.2. Turbine

The Turbine`s simulation, in MOTHER`s interface, is similar to the compressor`s, as reported at Chapter 6.3.3.1, and it is utilized by the following data:

- General Data

- Heat Transfer (Gas- Wall) data
- Heat Transfer (Wall- Coolant) data

Regarding the **General Data** tab, the turbine map data and the turbine`s wheel diameter were inserted. The Turbine Map represents experimentally obtained data regarding the swallowing capacity and efficiency with respect to the turbine`s shaft rotational speed. Using the instantaneous values of the turbine pressure ratio and the turbine shaft rotational speed, the volume flow parameter and the turbine efficiency are calculated by interpolation between adjacent points.

The turbine`s pressure ratio is defined as the ratio between the pressure of the flow receiver connected upstream of the turbine and the pressure of the flow receiver connected downstream of the turbine.

Due to the fact that the turbine map data are confidential, the turbine map is not attached here.

The work transfer rate from the turbine is calculated by the equation 6.16 by applying the first law of thermodynamics.

$$\dot{W}_t = \dot{m}_t \left(h_{03} - h_{04} \right) + q_t \tag{6.16}$$

where:

$\dot{W}_t$	work tranfer rate from the turbine	$[kW]$
$\dot{m}_t$	turbine air mass flow rate	$[kg/s]$
h_{03}	total specific enthalpy of exh. gas entering the turbine	$[kJ/kg]$
h_{04}	total specific enthalpy of exh. gas exiting the turbine	$[kJ/kg]$
q_t	heat transfer rate (from the turbine gas to wall)	$[kW]$

The work transfer rate from turbine, is the power delivered by the turbine to the turbocharger shaft to drive the compressor connected to the turbine. Thus, the turbine wheel torque is derived by the following equation (similar to the Eq. 6.9):

$$T_t = 30 \cdot \frac{\dot{W}_t}{\pi N} \tag{6.17}$$

where:

N	rotational speed of the turbine wheel	$[rpm]$

Regarding the **Heat Transfer (Gas-Wall) data**, the Nusselt-Reynolds-Prandlt relationship was used (in the same way as in compressor) in order to calculate the heat transfer coefficient of the gas (i.e. exhaust gas) passing through the turbine.

$$Nu = C_1 \operatorname{Re}^{C_2} \operatorname{Pr}^{C_3} \tag{6.18}$$

where:

$C_1 = 0.04$
$C_2 = 0.80$
$C_3 = 0.40$

Regarding the **Heat Transfer (Wall- Coolant) data**, the simple model was selected, in order to simulate the heat flow from the turbine`s wall to the coolant mean. The heat transfer here is modeled in the same way as in case of the exhaust receiver in Chapter 6.4.1 where the Eq. 6.4 and 6.5 are the governing equations of the model. Moreover, the simple model requires the turbine`s coolant temperature, which has been acquired by the Shop Test report (Appendix I), where is referred as “T/C oil outlet”.

6.4.3.3. Turboshaft

The compressor and the turbine are interconnected with the turbocharger shaft. Its rotational speed is required in order to complete the simulation of the turbocharger unit in MOTHER interface. However, the polar moment of inertia and a coefficient of convergence are required as well. The calculation of the exact value of the polar moment of inertia required data which were not available, consequently, a trial value has been inserted. The engine`s simulation was not affected by this value. The coefficient of convergence was set equal to one (1).

6.4.4. Engine Inlet

The ambient conditions, at each engine testing, are been simulated by the Fixed Fluid element, which is called ‘Engine Inlet’. The values of barometric pressure, ambient temperature and equivalence ratio, are inserted at the relevant tab in MOTHER interface. The equivalence ratio for the ‘Engine Inlet’ is close to zero because there no fuel presence in the air entering the compressor.

6.4.5. Engine Outlet

The ambient conditions after the turbine, at each engine testing, have been simulated by the Fixed Fluid element, which is called ‘Engine Outlet’. In the same way as in the ‘Engine Inlet’ element, the values of barometric pressure, ambient temperature and equivalence ratio, have been inserted at the relevant tab in MOTHER interface.

According to the aforementioned paragraphs, the RTA48-T`s model configuration has been completed and the engine operation at Shop Test can now be simulated. Next section provides the simulation`s results values in contrast with the measured values.

6.5. Input Data Synopsis

The following list provides a summary of the input values to the thermodynamic model for simulating Shop Test.

- Main engine rotational speed [rpm]
- Scavenging air pressure [bar] (as initial value)
- Barometric pressure [bar]
- Scavenging air temperature [K] (as initial value)
- Fuel injected per cycle per cylinder [kg]
- Start of static injection [CA deg]
- M/E jacket cooling outlet (F.W.) temperature for each cylinder [K]
- M/E piston cooling outlet (L.O.) temperature for each cylinder [K]
- Turbocharger rotational speed [rpm]
- Turbine coolant (L.O.) outlet temperature [K]
- Air cooler coolant water (S.W.) temperature [K]
- Exhaust valve coolant (F.W.) temperature [K]
- Exhaust gas temperature [K] (as initial value)
- Exhaust receiver`s coolant temperature [K] (i.e. room temp. during test)
- Inlet receiver`s coolant temperature [K] (i.e. room temp. during test)

6.6. Results – Shop Test

The following values which have been measured at Shop Test, are compared with the 'MOTHER'-calculated values.

- Engine power
- Fuel consumption
- T/C speed
- Scavenge air pressure
- Pressure drop across A/C
- Temperature before A/C (i.e. "Temp. aft. Blower" as stated at Shop Trial Report)
- Temperature after A/C
- A/C efficiency
- Exhaust gas temperature after cylinder
- Maximum cylinder pressure
- Combustion pressure

6.6.1. Cylinder pressures

Calculated compression and maximum pressure have been compared with the measured values at Shop Test for each cylinder at the following loads: 50%, 75%, 85% & 100%; the error percentages between the calculated by MOTHER values and the measured at Shop Test values respectively, are also calculated.

Note: *positive* error percentages means that the calculated value is greater that the measured at Shop Test and *negative* the opposite.

50% Load Pmax, Pcomp		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error (%)	Calculated	Measured	Error (%)
Aver.	bar	*100.23*	*99.7*	*0.53%*	*66.72*	*64.8*	*2.96%*
1		100.4	99	1.4%	66.9	64	4.5%
2		100.1	100	0.1%	66.5	65	2.3%
3		100.3	101	-0.7%	66.8	66	1.2%
4		100.3	99	1.3%	66.7	64	4.2%
5		100.1	99	1.1%	66.8	64	4.4%
6		100.2	100	0.2%	66.6	66	0.9%

Table 6. 7 Pmax, Pcomp for 50% Load

75% Load Pmax, Pcomp		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error (%)	Calculated	Measured	Error (%)
Aver.	bar	*129.91*	*130*	*-0.07%*	*89.04*	*89.7*	*-0.74%*
1		130.2	130	0.2%	89.1	90	-1.0%
2		130.0	130	0.0%	89.1	90	-1.0%
3		129.9	130	-0.1%	89.2	90	-0.9%
4		129.7	129	0.6%	89.0	89	0.0%
5		130.0	130	0.0%	89.0	88	1.1%
6		129.6	131	-1.1%	88.8	91	-2.4%

Table 6. 8 Pmax, Pcomp for 75% Load

85% Load Pmax, Pcomp		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error	Calculated	Measured	Error
Aver.	bar	*139.0*	*139.5*	*-0.36%*	*99.0*	*99.7*	*-0.70%*
1		139.1	139	0.1%	99.0	99	0.0%
2		139.1	140	-0.6%	99.0	100	-1.0%
3		139.1	140	-0.6%	99.0	100	-1.0%
4		138.7	139	-0.2%	98.8	99	-0.2%
5		139.0	139	0.0%	98.9	100	-1.1%
6		139.0	140	-0.7%	98.9	100	-1.1%

Table 6. 9 Pmax, Pcomp for 85% Load

100% Load Pmax, Pcomp		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error	Calculated	Measured	Error
Aver.	bar	*139.1*	*139.8*	*-0.5%*	*118.83*	*120.8*	*-1.63%*
1		139.2	139	0.4%	118.9	119	-0.1%
2		139.0	140	-0.7%	118.8	121	-1.8%
3		139.1	140	-0.6%	118.9	122	-2.5%
4		138.7	139	-0.9%	118.7	121	-1.9%
5		139.2	139	0.1%	118.7	120	-0.9%
6		139.1	140	-0.6%	118.8	122	-2.6%

Table 6. 10 Pmax, Pcomp for 100% Load

- **Cylinder pressures**

Figure 6.10, below, illustrates the pressure inside the cylinders as a function of the engine load. Moreover, this figure illustrates the correlation of the mean calculated maximum (and compression) pressure with the mean measured maximum (and compression) pressure at Shop Test.

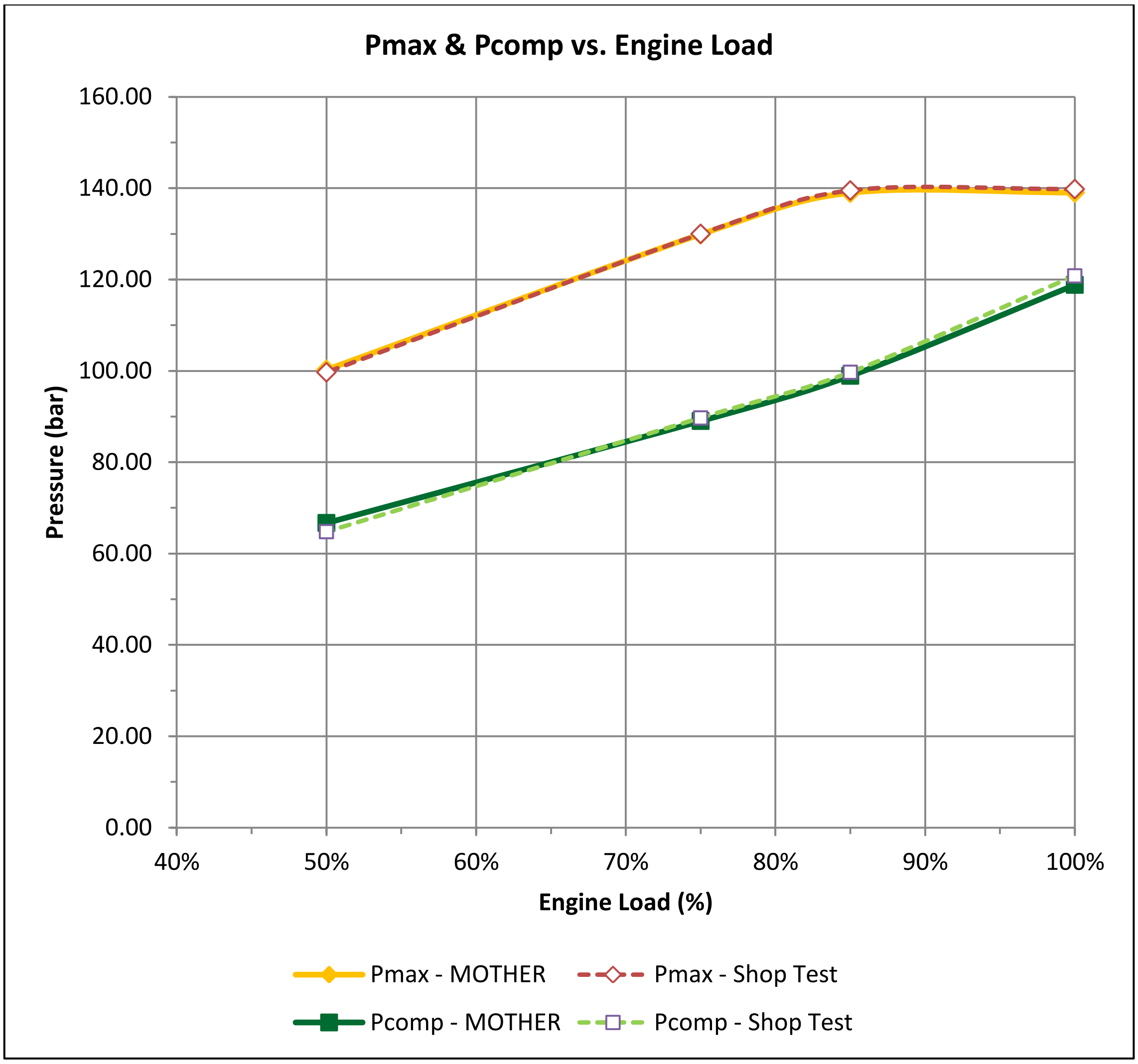


Fig. 6. 12 Pmax & Pcomp vs. Load

The error percentages for the calculated of maximum and compression pressure, are illustrated at Fig. 6.13 and 6.14.

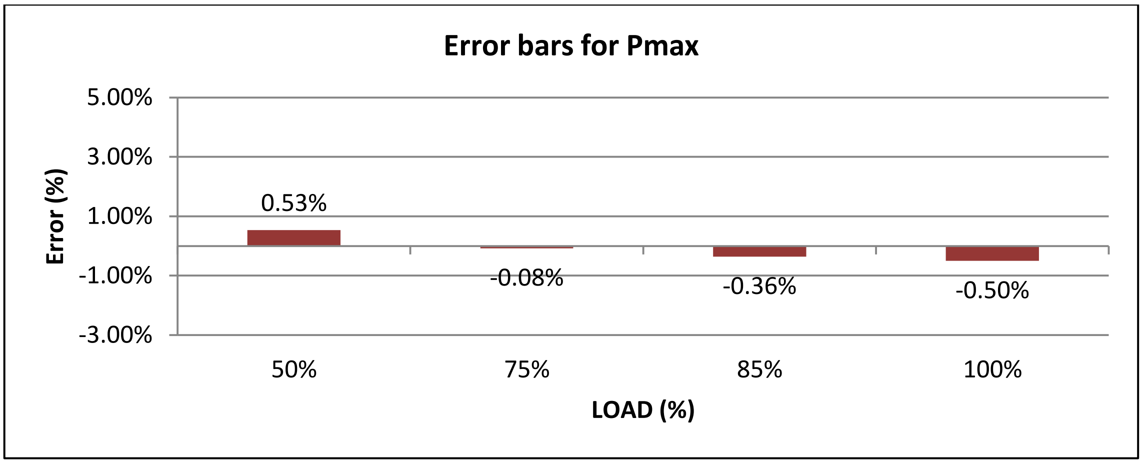


Fig. 6. 13 Error bars for Pmax

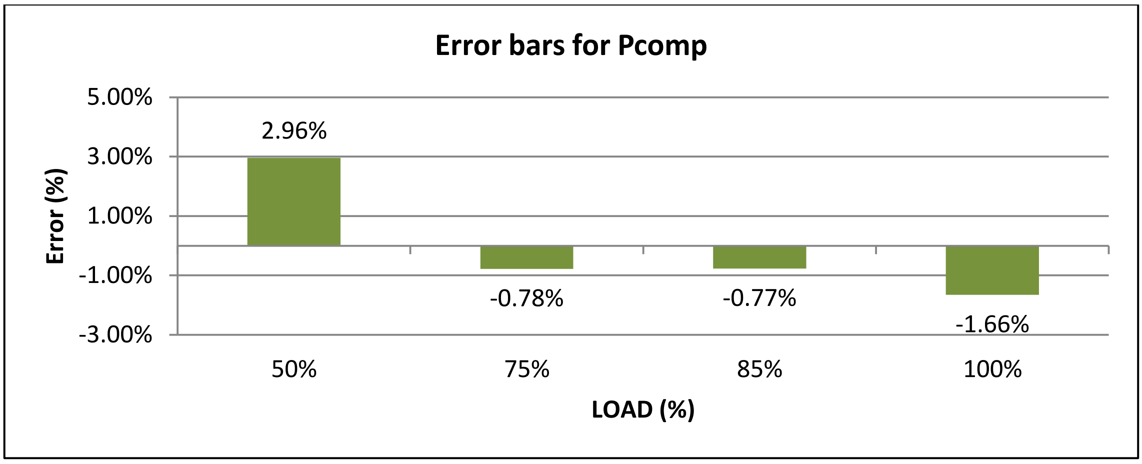


Fig. 6. 14 Error bars for Pcomp

Tables 6.7 to 6.11 and Fig. 6.10 and 6.12 show clearly that the measured pressures are predicted by the simulation model with an error margin less that 3%.

6.6.2. Presentation of various figures

The presentation of the remaining data will be accomplished by attaching comparative figures and afterwards the relative tables. There are two ways of presenting the power curves, the first as a function of power and the other as a function of the engine speed (i.e. RPM).

- **Mean Effective Pressure**

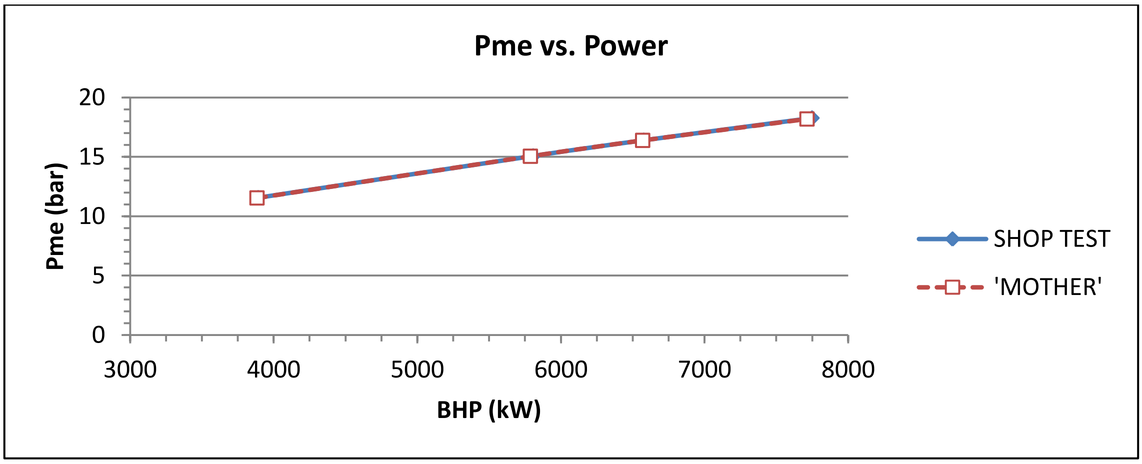


Fig. 6. 15 Mean eff. pressure vs. Engine Power

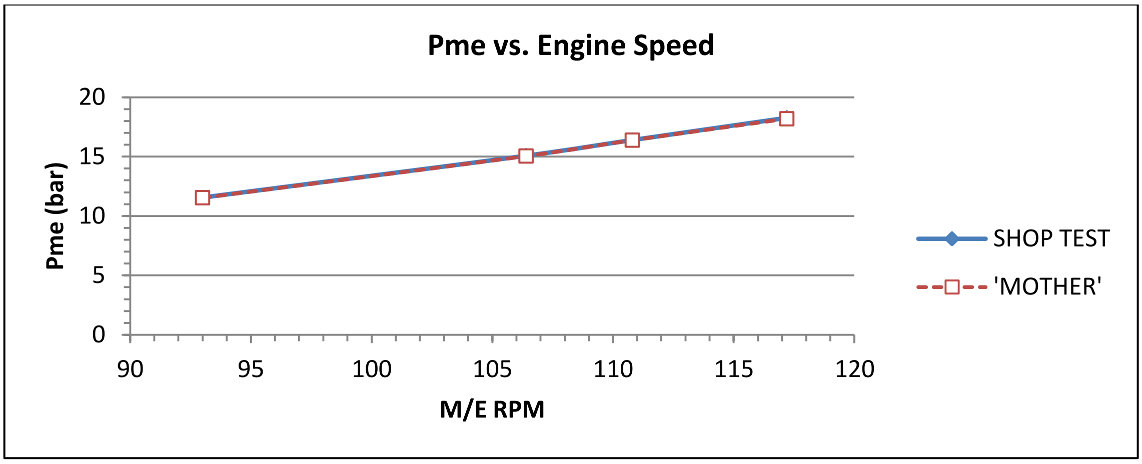


Fig. 6. 16 Mean eff. pressure vs. Engine Speed

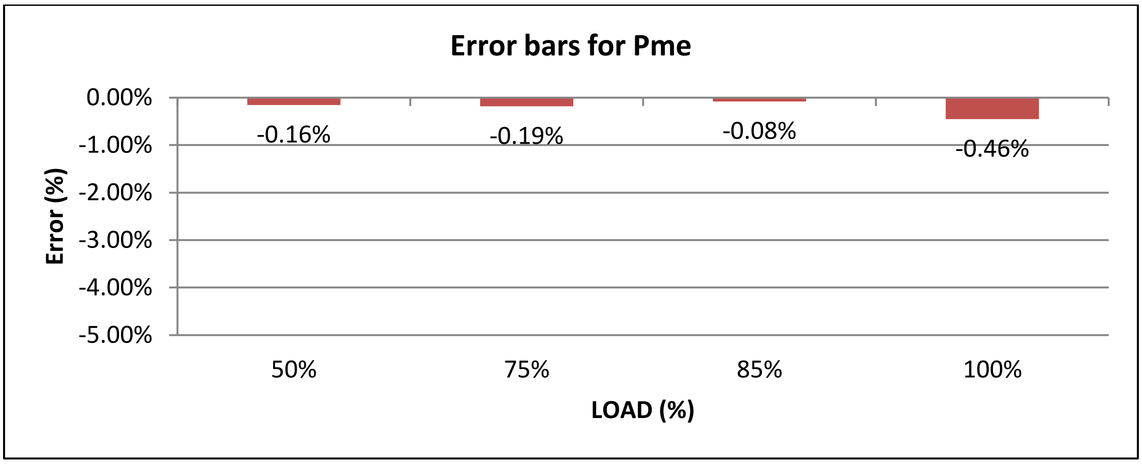


Fig. 6. 17 Error bars for Mean eff. pressure

The mean effective pressure is calculated with the maximum possible precision.

- **Brake Power**

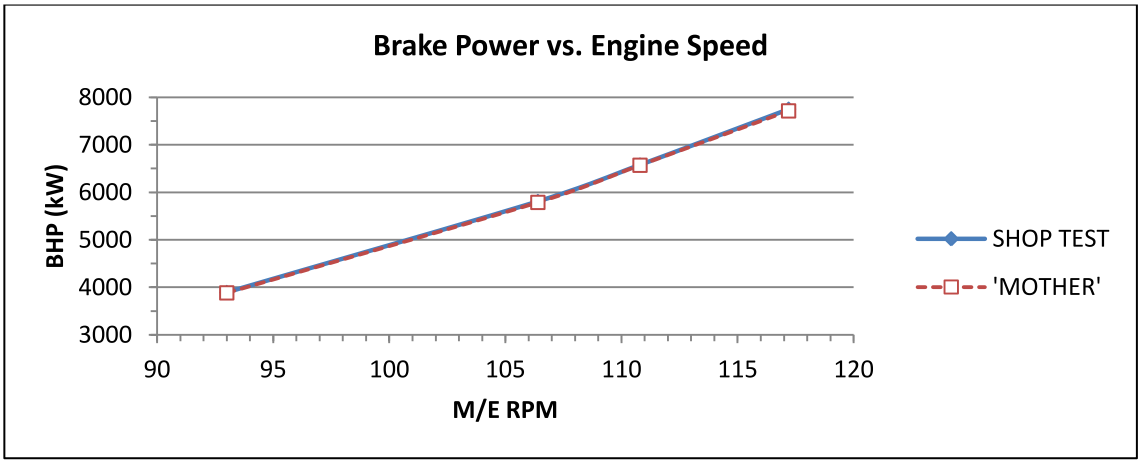


Fig. 6. 18 Brake Power vs. Engine Speed

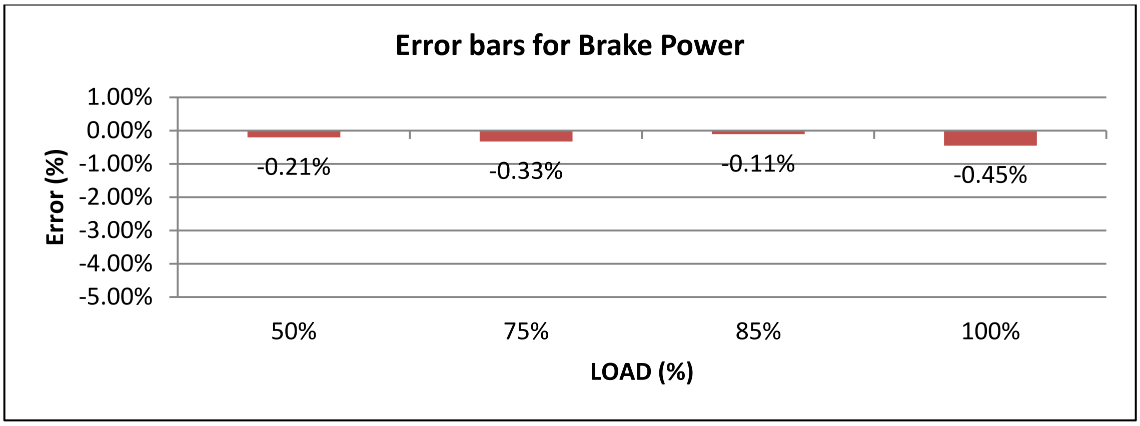


Fig. 6. 19 Error bars for Brake Power

Brake Power has been predicted satisfactory as the error is less than 1%. The capable coefficients` selection at the friction model, contributes to these results.

- **Turbocharger rotational speed**

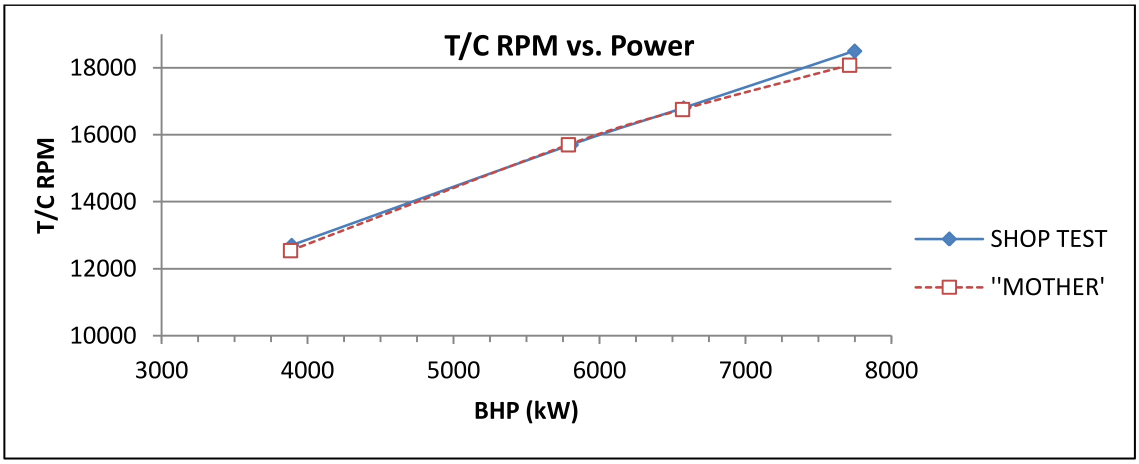


Fig. 6. 20 T/C RPM vs. Power

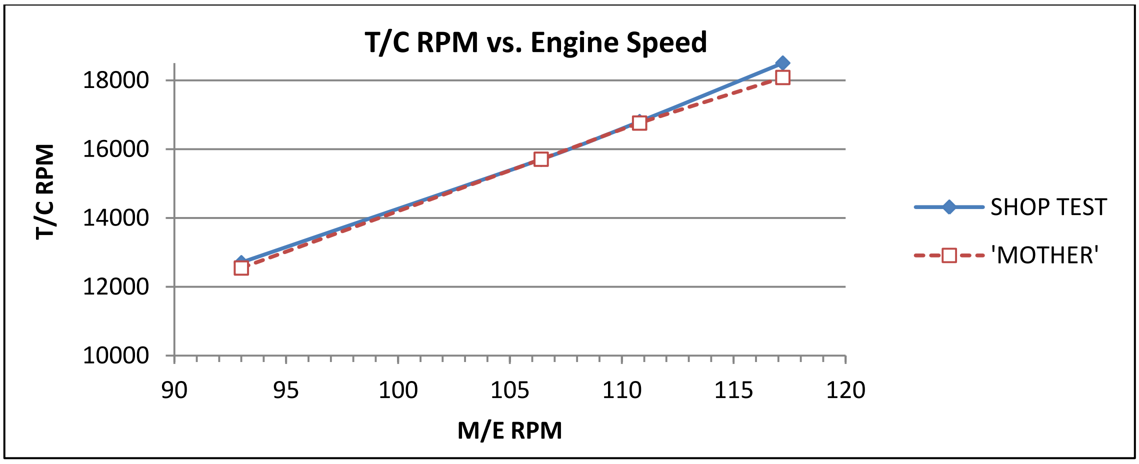


Fig. 6. 21 T/C RPM vs. M/E RPM

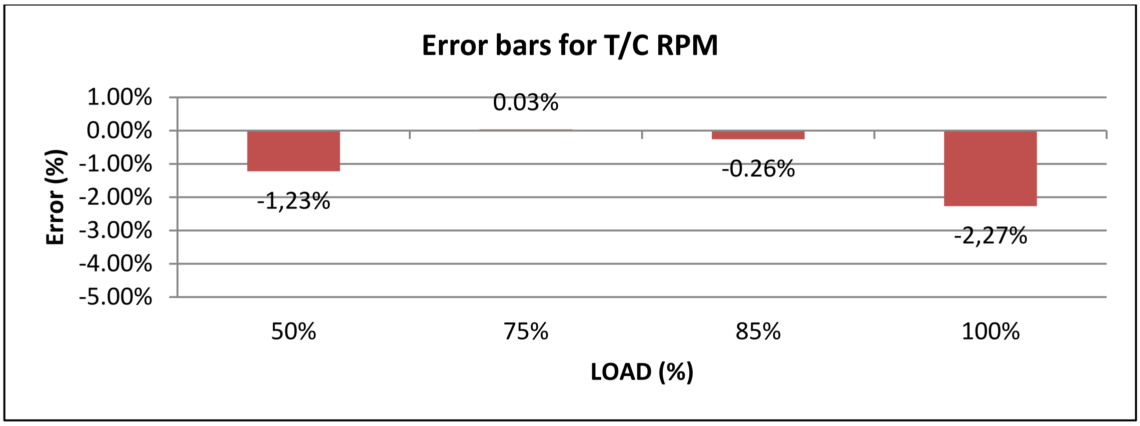


Fig. 6. 22 Error bars for T/C RPM

Although the measured (at Shop Test) T/C speed is inserted to the relevant tab in MOTHER interface, at the end of the simulation, the model converges to a different value. The calculated value for the turbocharger`s rotational speed is depending on the compression map data. The errors in Fig. 6.22 are within the acceptable limits (less than 3%) and the convergence is satisfying.

- **Scavenge Air Pressure**

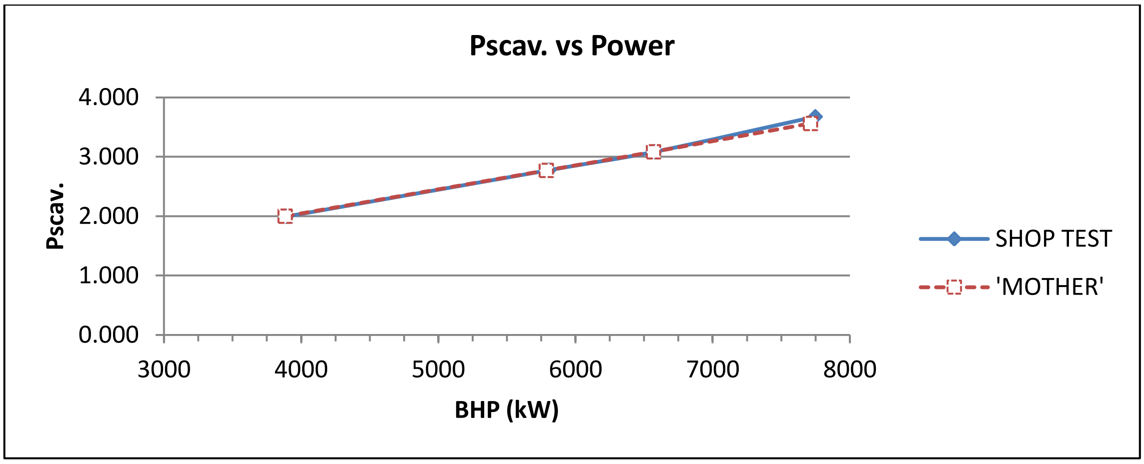


Fig. 6. 23 Scav. pres. vs. Power

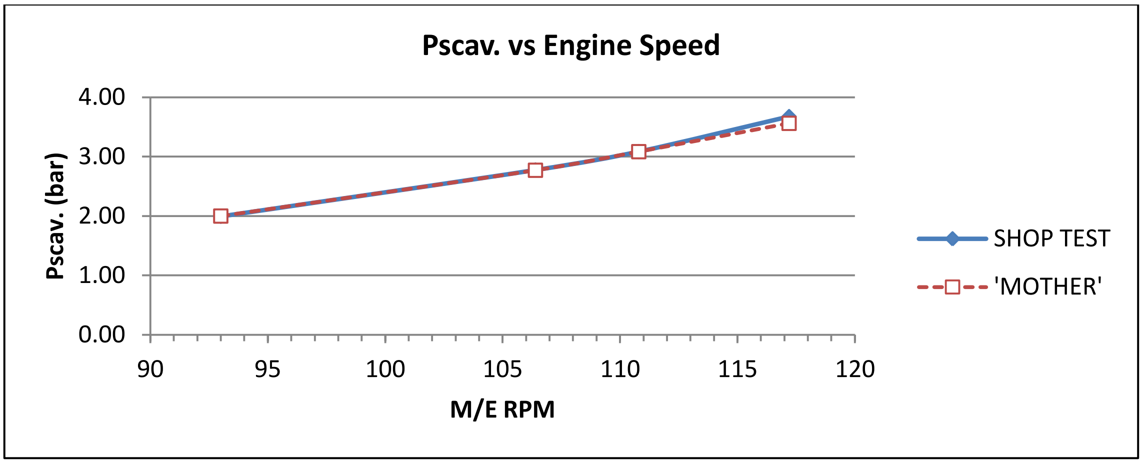


Fig. 6. 24 Scav. pres. vs. M/E RPM

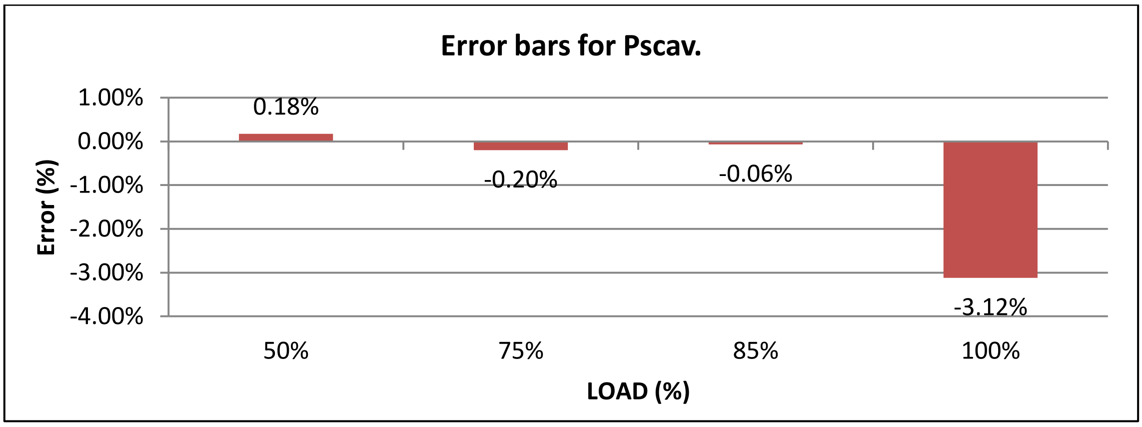


Fig. 6. 25 Error bars for Scav. pressure

Figures 6.23 and 6.24, illustrate the total scavenge pressure in relation to the engine load and engine speed respectively. The Shop Test report and the engine performance reports do not take into account the barometric pressure in the scavenge air pressure value; whereas, MOTHER calculated scavenge pressure, includes the barometric pressure. Consequently, Figures 6.23 & 6.24, illustrate the total scavenge pressure which is the product of the relative scavenge pressure plus the barometric pressure.

The maximum deviation from the measured data is -3.12% and lies at the 100% load, as Fig. 6.25 shows. This deviation can be explained by the reduced T/C speed that the model calculated at the same load (Fig. 6.22). Scavenge air pressure is related with the T/C speed when the other factors that affect the charge air pressure are considered to be the same; thus, higher T/C speed indicates higher scavenge air pressure and an increased air flow rate, whereas lower T/C speed indicates exactly the opposite relation.

■ **Exhaust gas temperature**

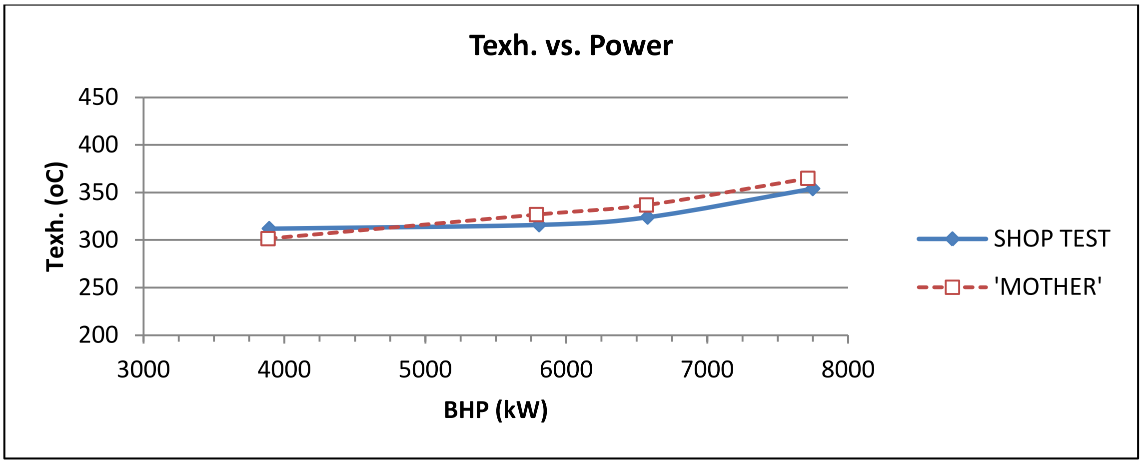


Fig. 6. 26 Exhaust gas temp. vs. Power

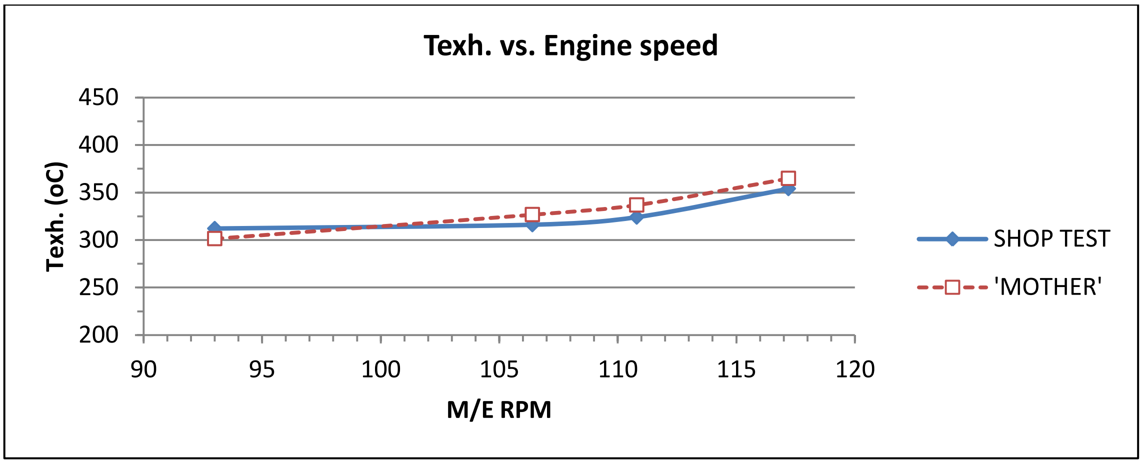


Fig. 6. 27 Exhaust gas temp. vs. M/E speed

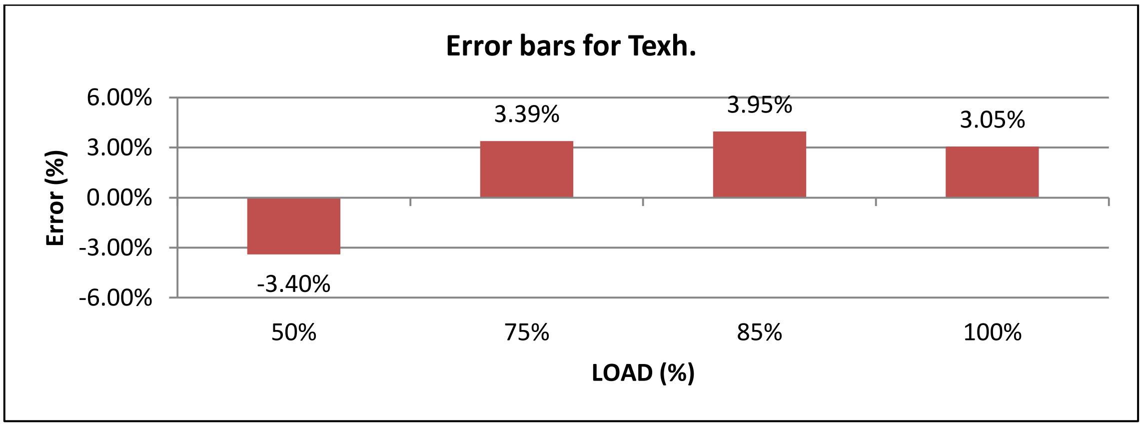


Fig. 6. 28 Error bars for exhaust gas temp.

Although the Fig. 6.26 & 6.27 have the same form, 'MOTHER' calculations for the exhaust gas temperatures deviate from the measured values by an average ±3%. By considering the complexity of the predicting these temperatures, the results are satisfying.

- **Specific fuel consumption**

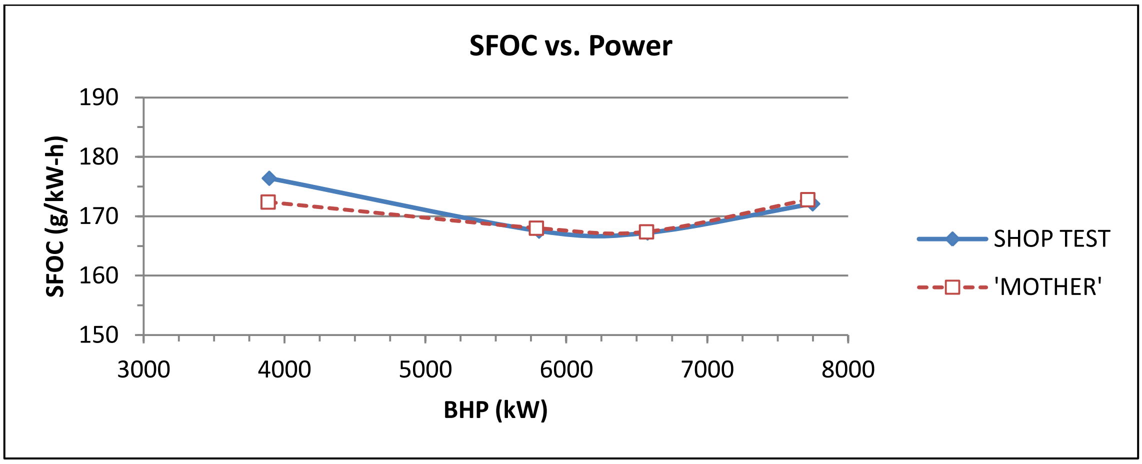


Fig. 6. 29 Specific fuel consumption

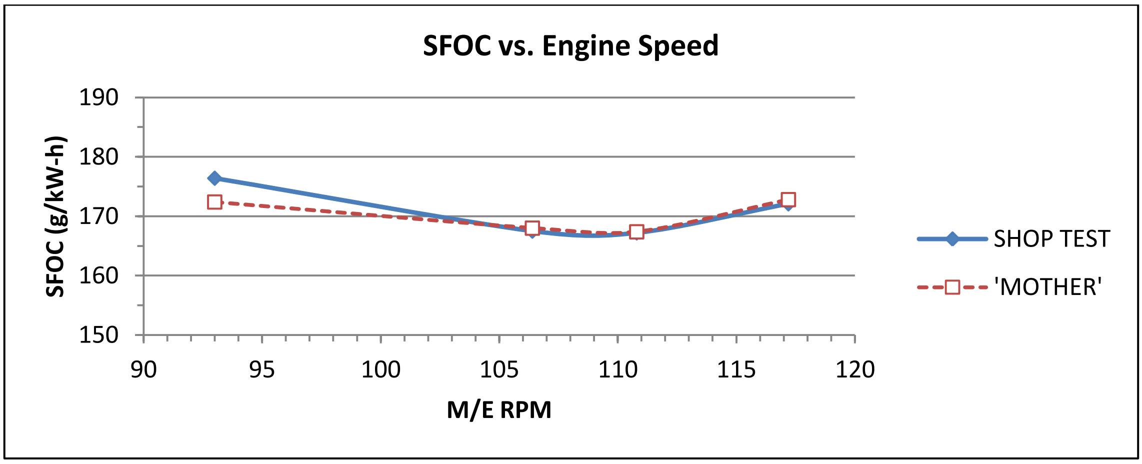


Fig. 6. 30 Specific fuel consumption vs. M/E speed

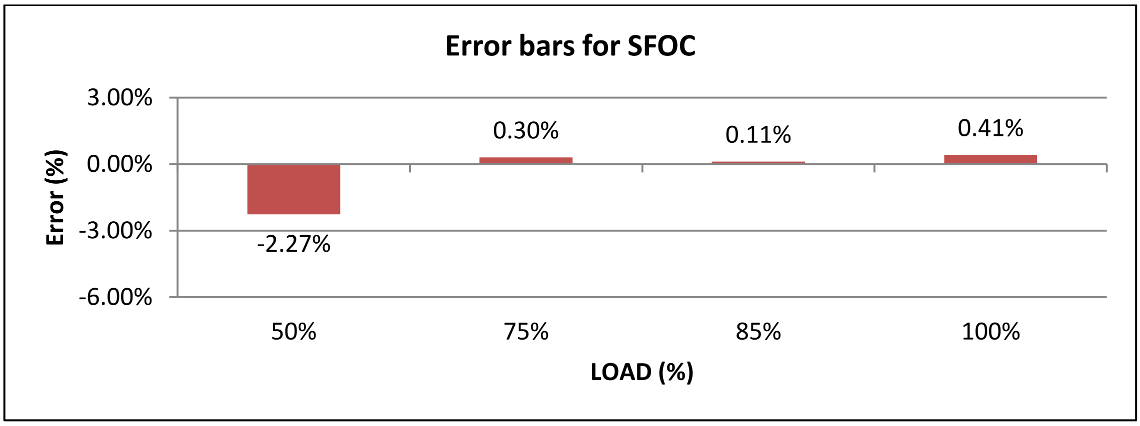


Fig. 6. 31 Error bars for Specific fuel consumption

The fuel used at Shop Test is Diesel Oil. The results are satisfying as the consumption is predicted with accuracy better than 3%.

- **Pressure drop across Air Cooler**

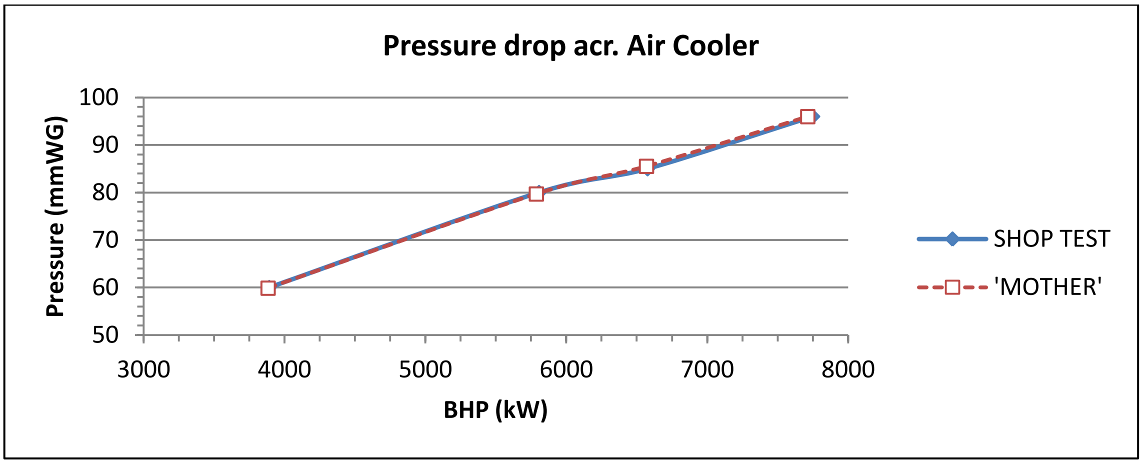


Fig. 6. 32 Pressure drop across Air Cooler

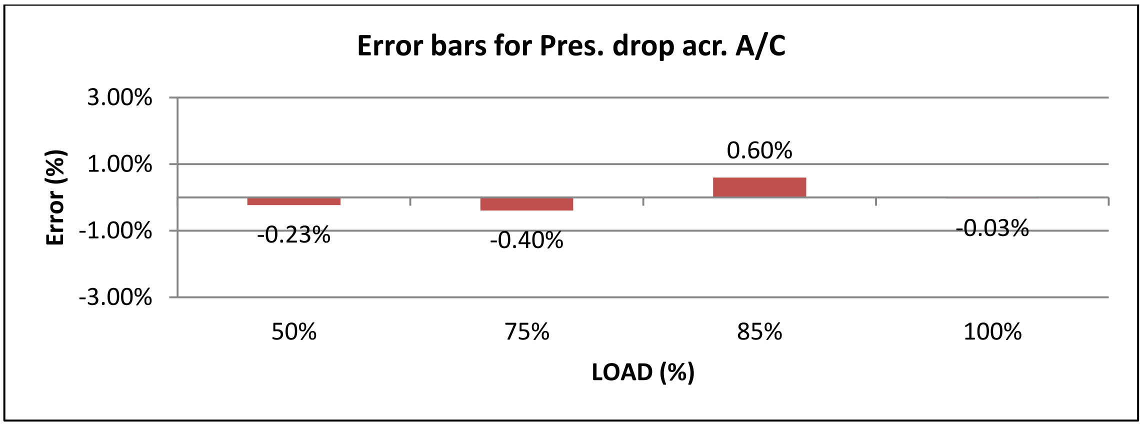


Fig. 6. 33 Error bars for Pres. drop acr. A/C

The pressure drop across the A/C raises at higher loads. As discussed at Chapter 6.4.3 the pressure drop was inserted manually by trial-and-error adjustment of the 'pressure loss coefficient'.

- **Air Cooler Efficiency**

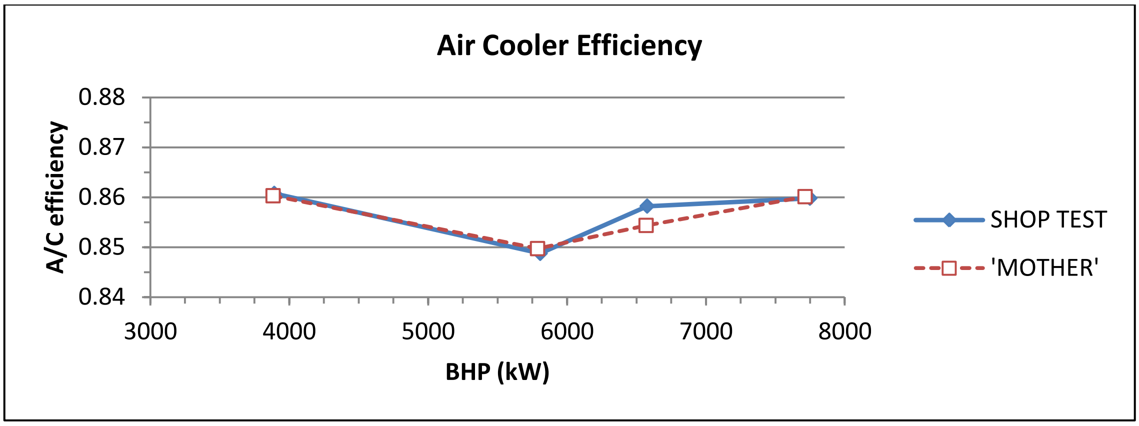


Fig. 6. 34 A/C efficiency

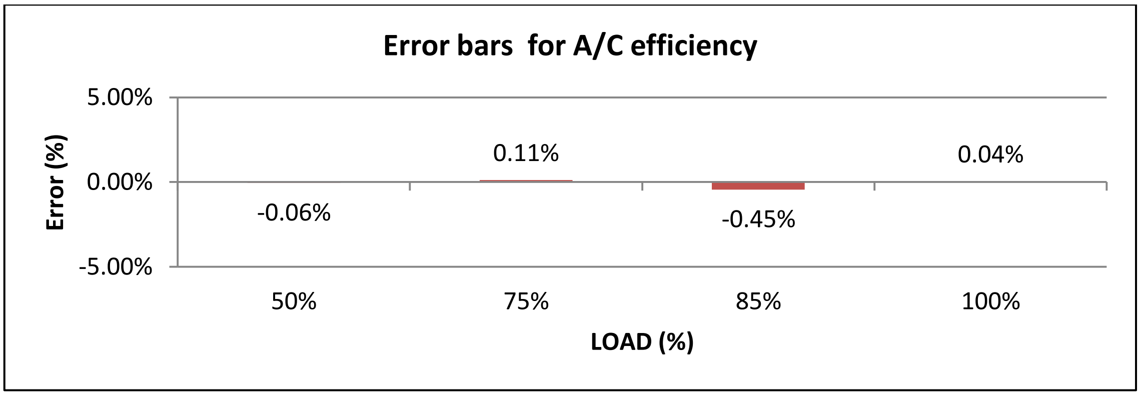


Fig. 6. 35 Error bars for A/C efficiency

The measured A/C cooler efficiency, according to Fig. 6.34, increases between the 75% and 100% load, whereas the maximum value is at 50% load. The same pattern follows the calculated A/C efficiency. The temperatures (Eq. 6.11) that affect the efficiency are showed below.

- **Temperature after compressor (before Air Cooler)**

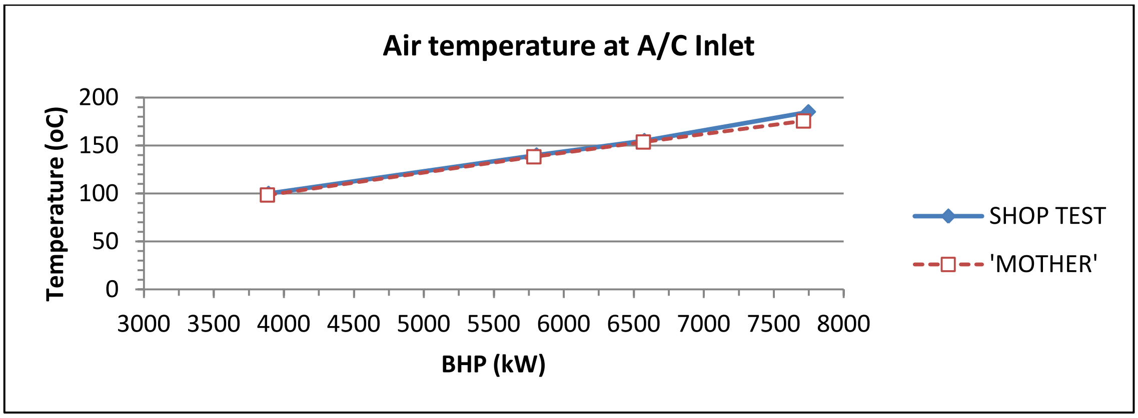


Fig. 6. 36 Temperature before A/C

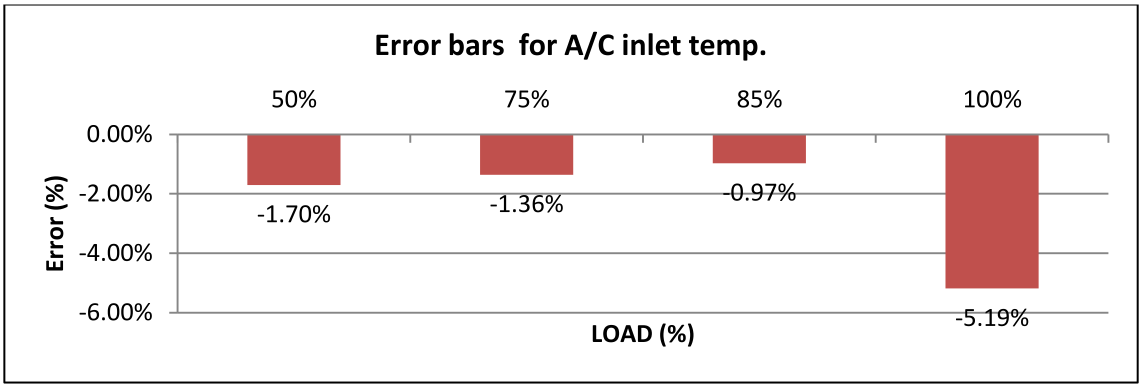


Fig. 6. 37 Error bars for temperature bef. A/C

- **Temperature after Air Cooler**

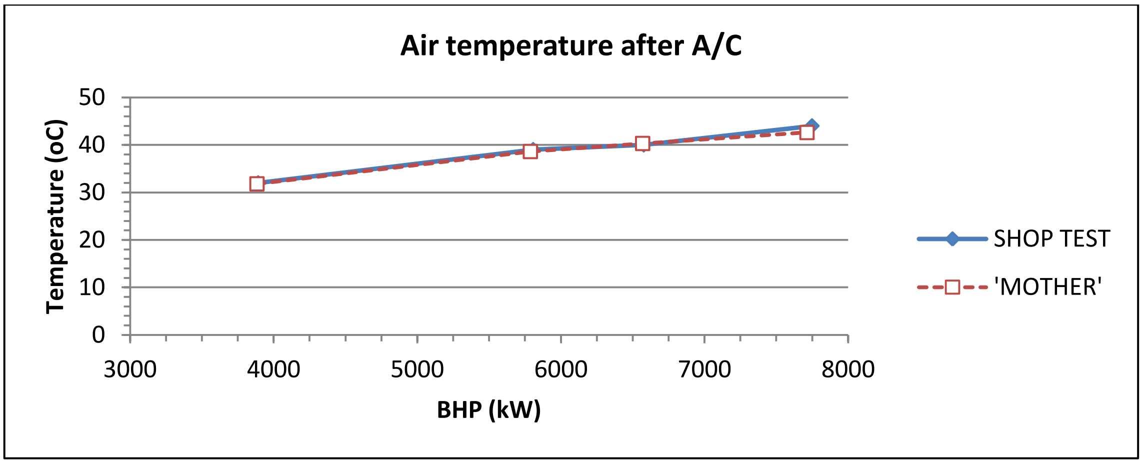


Fig. 6. 38 Temperature after A/C

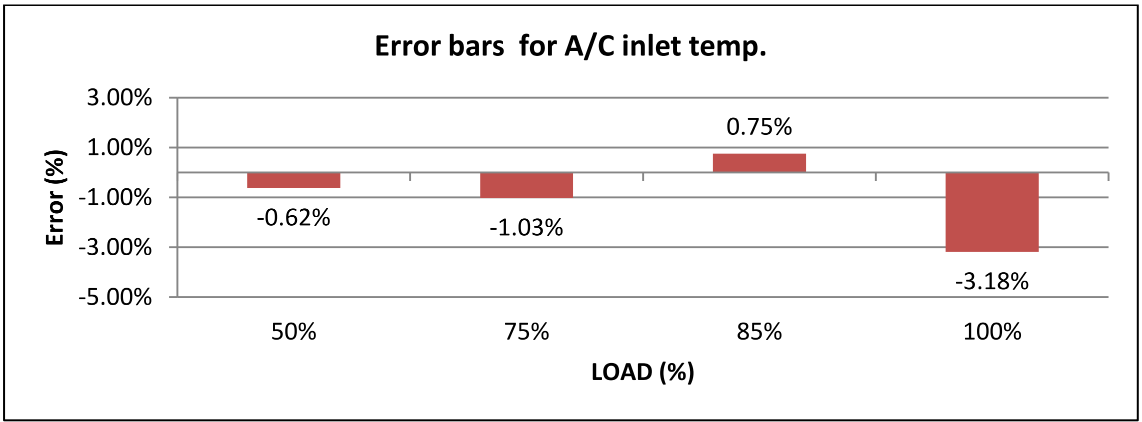


Fig. 6. 39 Error bars for temperature after A/C

6.6.3. Arithmetic results

The tables below include the data that 'MOTHER' calculated, as introduced in Chapter 6.5.

50% LOAD at 93 RPM			MEASURED	CALCULATED	ERROR (%)
	T/C Rev.		12700.0	12544	*-1.23%*
PRESSURES	Scavenge air press. (incl. Pbaro)	bar	1.9955	1.9990	*0.18%*
	Scavenge air press. (relative)		0.9700	0.9735	*0.36%*
	Barometric Pressure		1.0255	-	-
	Pmax (aver. value)		99.70	100.23	*0.53%*
	Pcomp (aver. value)		64.80	66.72	*2.96%*
	BMEP		11.56	11.54	*-0.16%*
	FMEP (aver. value)		-	0.64	-
	Pres. Drop In Air Cooler		0.0058838	0.00587011	*-0.23%*
	Pres. in exh. gas receiver		-	1.80	-
TEMPERATURES	Air temp. before A/C	°C	100.0	98.3	*-1.70%*
	Air temp. after A/C		32.0	31.8	*-0.62%*
	A/C coolant water		21.0	21.0	-
	Exh. Gas temp. aft. cyl. (aver. value)		312.0	301.4	*-3.40%*
	Exh. Gas temp. aft. Turbine		265	-	-
MISC.	Compressor pressure ratio	-	-	1.9600	-
	Compressor corrected air flow rate	m^3/s	-	8.8850	-
	Compressor efficiency	-	-	0.8120	-
	Turbine efficiency	-	-	0.8124	-
	Air Cooler efficiency	-	0.8608	0.8603	*-0.06%*
	Indicated Horse Power (IHP)	kW	-	4098	-
	Brake Horse Power (BHP)	kW	3892	3883.9	*-0.21%*
	SFOC	g/kW-h	176.4	172.4	*-2.27%*

Table 6. 11 Results for 50% load

75% LOAD at 106.4 RPM			MEASURED	CALCULATED	ERROR (%)
	T/C Rev.		15700.0	15705	*0.03%*
PRESSURES	Scavenge air press. (incl. Pbaro)	bar	2.7755	2.7700	*-0.20%*
	Scavenge air press. (relative)		1.7500	1.7445	*-0.31%*
	Barometric Pressure		1.0255	-	-
	Pmax (aver. value)		130.0	129.90	*-0.08%*
	Pcomp (aver. value)		89.7	89.00	*-0.78%*
	BMEP		15.08	15.05	*-0.19%*
	FMEP (aver. value)		-	0.76	-
	Pres. Drop In Air Cooler		0.0078500	0.00781395	*-0.46%*
	Pres. in exh. gas receiver		-	2.474	-
TEMPERATURES	Air temp. before A/C	°C	140.0	138.1	*-1.36%*
	Air temp. after A/C		39.0	38.6	*-1.03%*
	A/C coolant water		21	21	-
	Exh. Gas temp. aft. cyl. (aver. value)		316	326.7	*3.39%*
	Exh. Gas temp. aft. Turbine		246	-	-
MISC.	Compressor pressure ratio	-	-	2.71	-
	Compressor corrected air flow rate	m^3/s	-	12.526	-
	Compressor efficiency	-		0.830	
	Turbine efficiency	-		0.829	
	Air Cooler efficiency	-	0.8487	0.8497	*0.11%*
	Indicated Horse Power (IHP)	kW	-	6080.5	-
	Brake Horse Power (BHP)	kW	5807	5787.8	*-0.33%*
	SFOC	g/kW-h	167.5	168	*0.3%*

Table 6. 12 Results for 75% load

85% LOAD at 110.8 RPM			MEASURED	CALCULATED	ERROR (%)
	T/C Rev.		16800.0	16756.2	*-0.26%*
PRESSURES	Scavenge air press. (incl. Pbaro)	bar	3.085	3.083	*-0.06%*
	Scavenge air press. (relative)		2.0600	2.0580	*-0.10%*
	Barometric Pressure		1.025	-	-
	Pmax (aver. value)		139.5	139.0	*-0.36%*
	Pcomp (aver. value)		99.7	98.9	*-0.77%*
	BMEP		16.40	16.39	*-0.08%*
	FMEP (aver. value)		-	0.765	-
	Pres. Drop In Air Cooler		0.008335	0.008385	*0.60%*
	Pres. in exh. gas receiver		-	2.760	-
TEMPERATURES	Air temp. before A/C	°C	155.0	153.5	*-0.97%*
	Air temp. after A/C		40.0	40.3	*0.75%*
	A/C coolant water		21.0	21.0	
	Exh. Gas temp. aft. cyl. (aver. value)		324.0	336.8	*3.95 %*
	Exh. Gas temp. aft. Turbine		244	-	-
MISC.	Compressor pressure ratio	-	-	3.02	-
	Compressor corrected air flow rate	m^3/s	-	13.945	-
	Compressor efficiency	-	-	0.824	-
	Turbine efficiency	-	-	0.829	-
	Air Cooler efficiency	-	0.8582	0.8543	*-0.45%*
	Indicated Horse Power (IHP)	kW	-	6876.3	-
	Brake Horse Power (BHP)	kW	6577.0	6569.6	*-0.11%*
	SFOC	g/kW-h	167.2	167.4	*0.11%*

Table 6. 13 Results for 85% load

100% LOAD at 117.2 RPM			MEASURED	CALCULATED	ERROR (%)
	T/C Rev.		18500	18080.2	*-2.27%*
PRESSURES	Scavenge air press. (incl. Pbaro)	bar	3.6745	3.5600	*-3.12%*
	Scavenge air press. (relative)		2.6500	2.5355	*-4.32%*
	Barometric Pressure		1.0245	-	-
	Pmax (aver. value)		139.8	139.1	*-0.50%*
	Pcomp (aver. value)		120.8	118.8	*-1.63%*
	Pme		18.27	18.19	*-0.46%*
	FMEP (aver. value)		-	1.02	-
	Pres. Drop In Air Cooler		0.009414	0.00941	*-0.03%*
	Pres. in exh. gas receiver		-	3.20	-
TEMPERATURES	Air temp. before A/C	°C	185.0	175.4	*-5.19%*
	Air temp. after A/C		44.0	42.6	*-3.18%*
	A/C coolant water		21	21	-
	Exh. Gas temp. aft. cyl. (aver. value)		354.0	364.8	*3.05%*
	Exh. Gas temp. aft. Turbine		258	-	-
MISC.	Compressor pressure ratio	-	-	3.49	-
	Compressor corrected air flow rate	m^3/s	-	15.610	-
	Compressor efficiency	-	-	0.8199	-
	Turbine efficiency	-	-	0.8136	-
	Air Cooler efficiency	-	0.8598	0.8601	*0.04%*
	Indicated Horse Power (IHP)	kW	-	8145.8	-
	Brake Horse Power (BHP)	kW	7749	7714	*-0.45%*
	SFOC	g/kW-h	172.1	172.8	*0.41%*

Table 6. 14 Results for 100% load

6.7. Other figures

At this Chapter are attached various figures of values that are calculated by the simulation model and they are not reported at the Shop Test final report.

- **Friction Mean Effective Pressure (FMEP)**

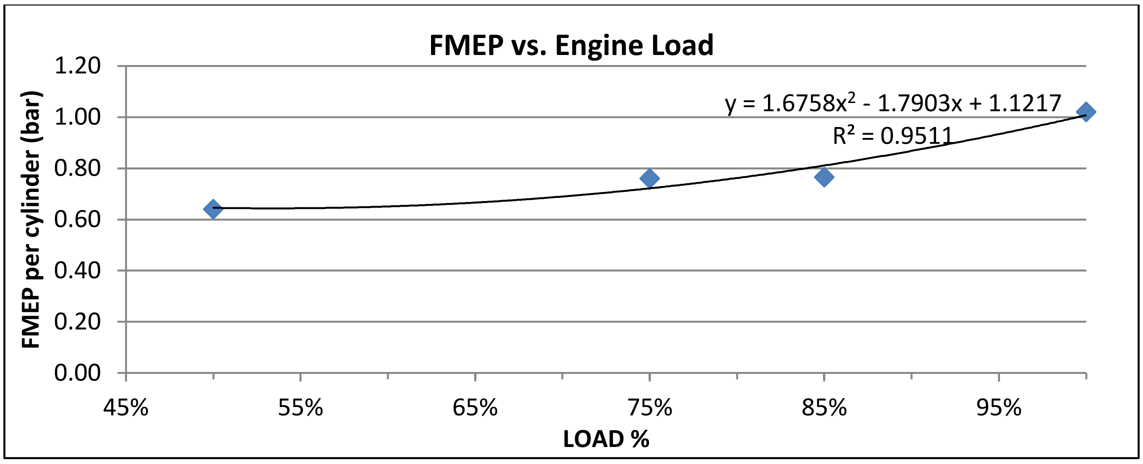


Fig. 6. 40 FMEP at SHOP Test vs. Load

The FMEP raises as the engine load raises; this rational form is common for marine engines. Friction, at higher engine loads, is raising because of the higher velocity of the engine moving parts, i.e. piston, connecting rod and crankshaft have increased velocity at high loads compared with lower loads. Fig. 6.40 shows this relation between friction and engine load.

- **Compressor pressure ratio**

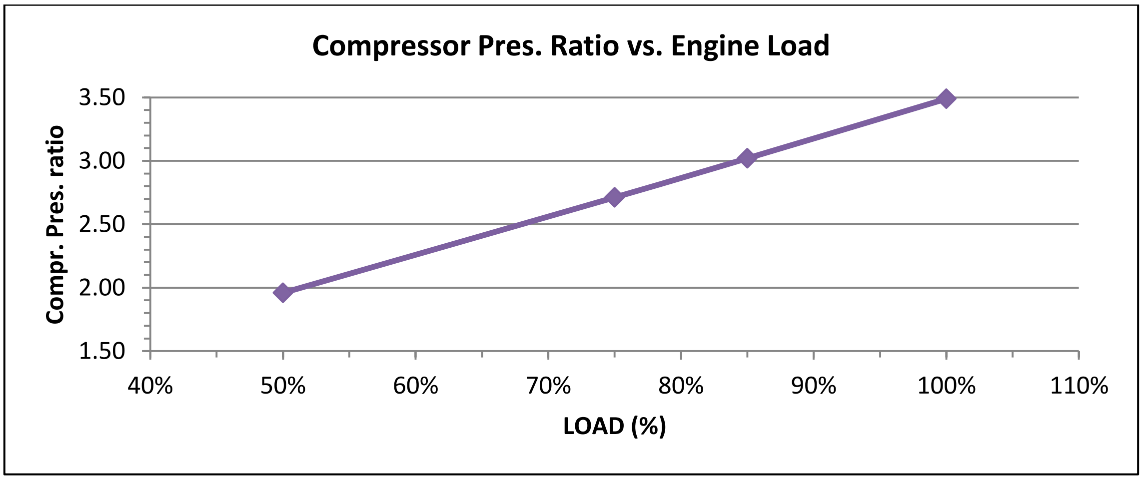


Fig. 6. 41 Compressor pressure ratio as calculated by 'MOTHER'

At higher loads the compressor pressure ratio raises, which means that the scavenge air pressure is raising with same barometric pressure. This is rational because at higher loads the turbocharger rotational speed is higher and consequently the air delivery pressure is higher compared to lower engine loads.

6.8. Energy balance

The energy balance is calculated for each load by the simulation model. The results are illustrated at the figures 6.42 to 6.45.

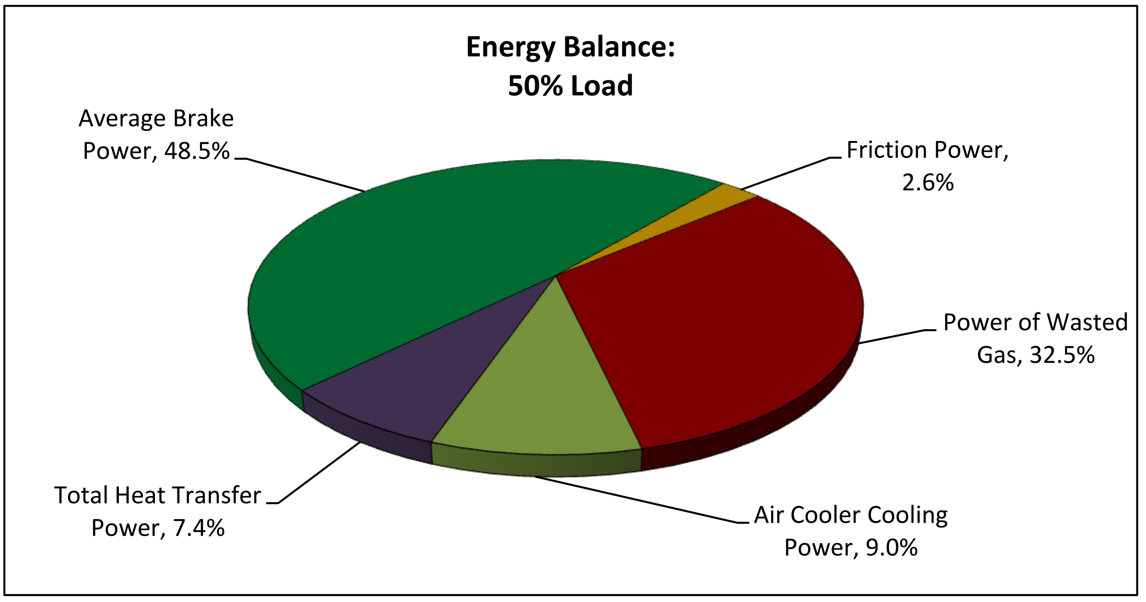


Fig. 6. 42 Energy Balance for 50% Load

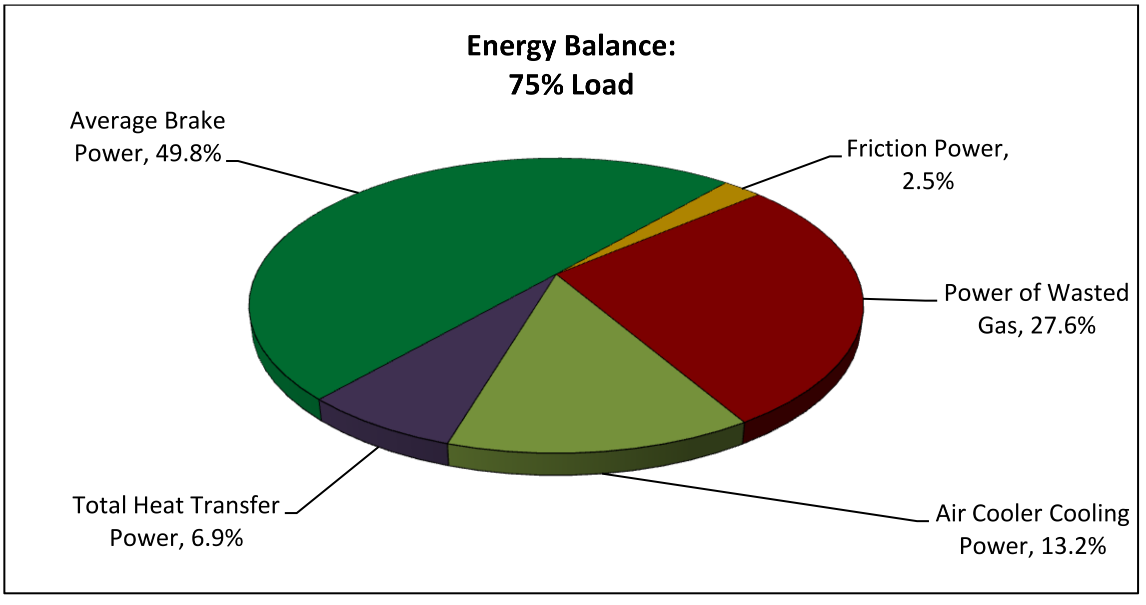


Fig. 6. 43 Energy Balance for 75% Load

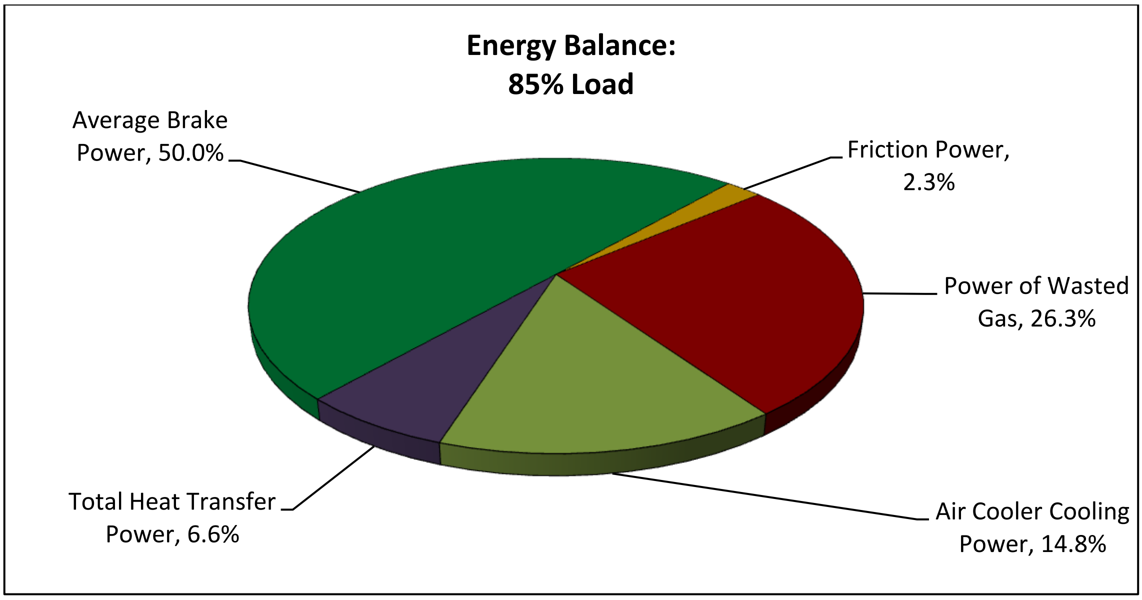


Fig. 6. 44 Energy Balance for 85% Load

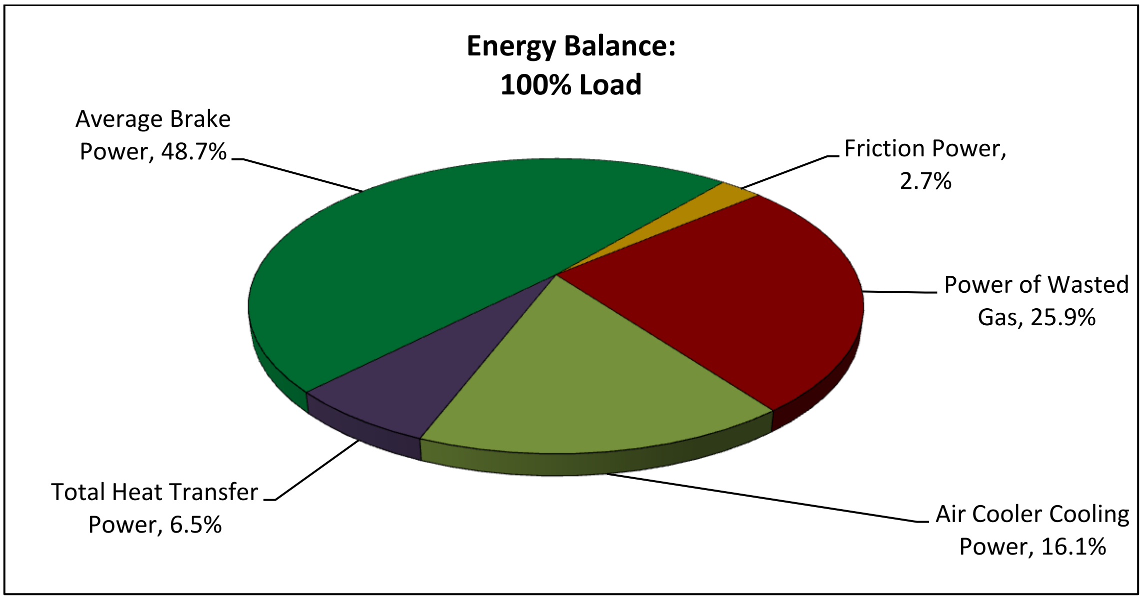


Fig. 6. 45 Energy Balance for 100% Load

The power percentages for all loads, maintain their magnitude, whereas the absolute values are raising due to the raising input power for higher loads; the input power per cycle is determined by the net energy of the fuel. The percentages showed at the above figures lie between the acceptable margins for marine diesel engines.

The Figures` values are attached at the Tables below.

Power	50% Load		75% Load	
	[kW]	[%]	[kW]	[%]
Power of Wasted Gas	2600.6	32.5%	3200.5	27.6%
Air Cooler Cooling Power	723.9	9.0%	1533.6	13.2%
Total Heat Transfer Power	592.9	7.4%	801.1	6.9%
Average Brake Power	3883.9	48.5%	5787.8	49.8%
Friction Power	211.9	2.6%	289.9	2.5%
Total Power	8013.2	100.0%	11612.9	100.0%

Table 6. 15 Energy balance for 50% & 75% Load

Power	85% Load		100% Load	
	[kW]	[%]	[kW]	[%]
Power of Wasted Gas	3462.4	26.3%	4096.6	25.9%
Air Cooler Cooling Power	1944.3	14.8%	2554.3	16.1%
Total Heat Transfer Power	870.9	6.6%	1035.9	6.5%
Average Brake Power	6569.6	50.0%	7714	48.7%
Friction Power	303.6	2.3%	427.4	2.7%
Total Power	13150.8	100.0%	15828.2	100.0%

Table 6. 16 Energy balance for 85% & 100% Load

As noticed above, the absolute value of each power fraction is raising for higher loads; figure 6.46 shows this relation.

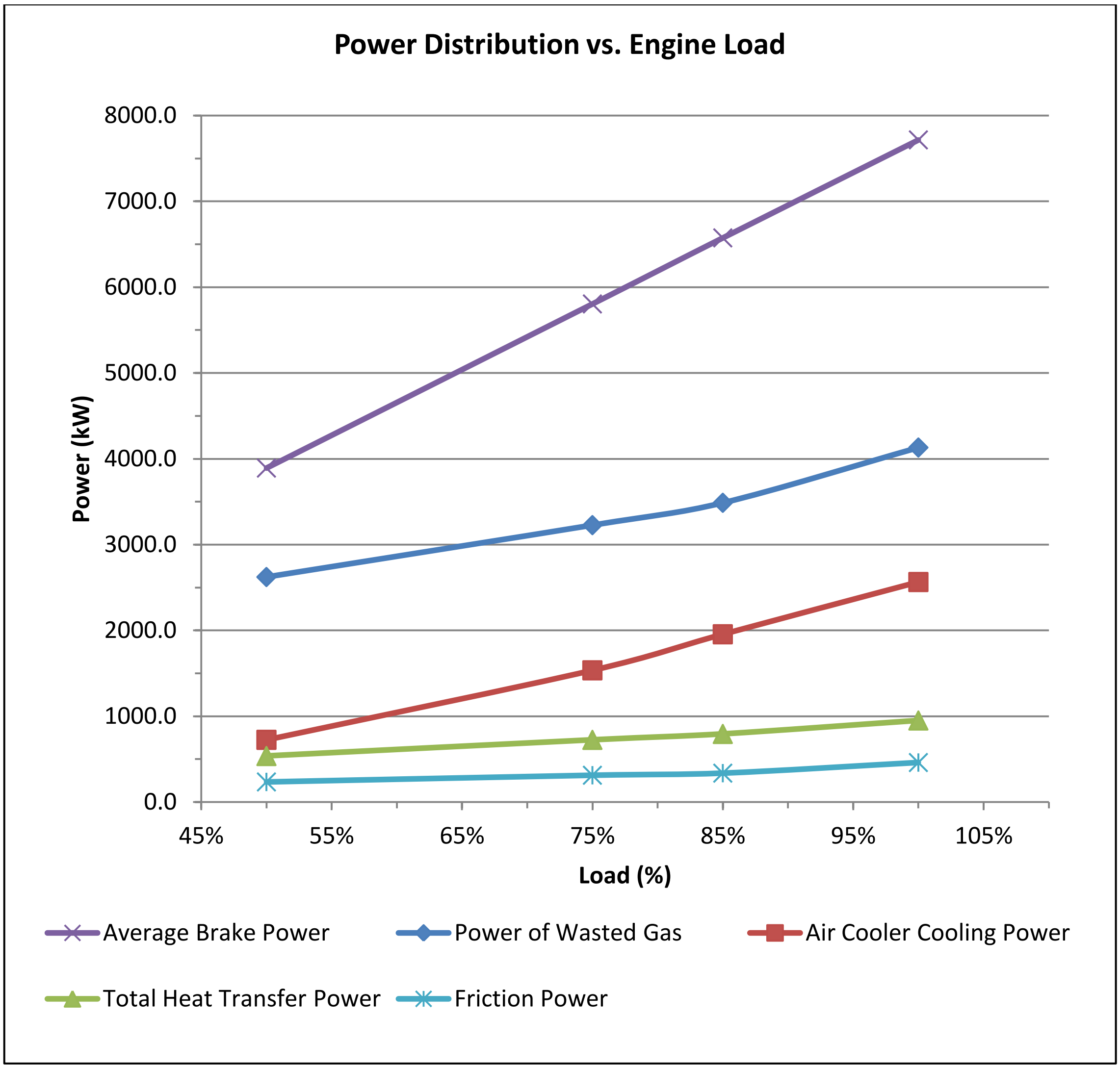


Fig. 6. 46 Power types relation with engine load

The simulation process for the validation of the 6RTA48-T`s model, continues with Sea Trials, which are discussed at Chapter 7.

CHAPTER 7

SEA TRIALS

7.1 Introduction

After the simulation of Shop Test has been completed satisfactorily, thermodynamic simulation model was evaluated in simulating the engine operation during Sea Trials. Thus, the simulation`s model input data were updated with those acquired from Sea Trials. This process targeted in succeeding convergence between the measured and calculated data, from the first ‘run’ (i.e. program execution) and with the minimum possible changes of the simulation`s model data. In other words, the model was tested on whether predicts or not, the engine operation data during Sea Trials. When the simulation of Sea Trials has been accomplished successfully, the RTA48-T simulation model is able to predict the engine`s performance of whichever operating point. This Chapter discusses how the simulation model has been tuned in order to calculate the Sea Trial data; moreover, it provides tables where the simulations` results have been attached to.

7.2 Sea Trials simulation model

Sea Trials simulation`s model input data differ from those at Shop Test. The following list contains all the input data required for the simulation model:

- Main engine rotational speed [rpm]
- Scavenging air pressure [bar]
- Barometric pressure [bar]
- Scavenging air temperature [K]
- Fuel injected per cycle [kg]
- Start of static injection [CA deg]
- M/E jacket cooling outlet (F.W.) temperature for each cylinder [K]
- M/E piston cooling outlet (L.O.) temperature for each cylinder [K]
- Turbocharger rotational speed [rpm]
- Turbine coolant (L.O.) outlet temperature [K]
- Air cooler coolant water (S.W.) temperature [K]
- Exhaust valve coolant (F.W.) temperature [K]
- Exhaust receiver`s coolant (engine room air) temperature [K]
- Inlet receiver`s coolant (engine room air) temperature [K]

A part of the Sea Trials report is attached at Appendix II, from where the input data have been acquired.

Amongst the models that were used for simulating the operation of the various engine elements, the combustion model and the friction model, require a more detailed description.

7.2.1 Combustion model

Woschni-Anisits combustion model was used, as described in Chapter 4.2.1.5.2, with the following input data:

- ***Model parameter "a "***

The estimation of constant " *a* " is based on the function showed at Fig. 6.3 (or 7.1), which is attached below. Thus, the product of "LIxRPM" at Sea Trials, is used for the estimation of the constant, according to figure 7.1.

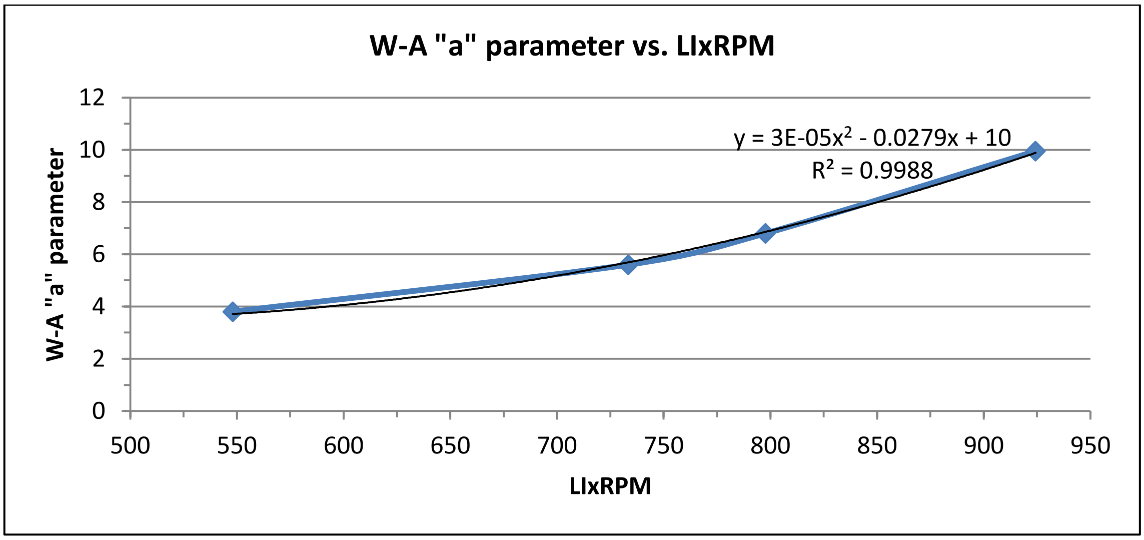


Fig. 7. 1 W-A constant "a" calculation for Sea Trials

- ***Start of static injection in CA degrees***

The injection timing was calculated by the VIT setting of the engine and by considering the FQS position, as well. The VIT and FQS functions have not been included due to the confidentiality of these data. The influence of the VIT and FQS on the injection timing, is described in Chapter 3.10.

- ***Mass of fuel injected per cycle per cylinder in kg***

The methodology for the estimation of fuel injected per cycle per cylinder, is the same as described in Appendix IV at Chapter IV.1 (Table IV.3). The calculated fuel quantity injected per cycle was corrected with the ISO fuel Net Calorific Value (NCV), because MOTHER considers that the engine is operated with the ISO fuel. Moreover, several consecutive runs of the model, showed that the calculated fuel mass should be changed by several grams, in order to achieve better convergence between the measured BHP and the calculated BHP. For this reason and in the beginning of this paragragh, the term "*estimation*" is used instead of "*calculation*". Finally, it should be stated that the precise mass of fuel injected per cycle, per cylinder (in grams), is not known, even though the consumption rate per hour is reported in tons per 24 hours. According to this fact, the aforementioned change in fuel quantity by several grams, makes sense and at the same time does not affect seriously the engine performance.

Table 7.1, below, shows the input data to the combustion model.

Sea Trial combustion model input	W-A Constant "*a*" [-]	Start of static injection [deg]	Mass of fuel inj. per cycle per cylinder [kg]
NOR. 1	6.94	-5.440	0.0272
NOR. 2	6.81	-5.523	0.0270

Table 7. 1 Combustion model input for Sea Trial

7.2.2 Friction model

The friction model coefficients k_1, k_2 were calculated as described in Chapter 6.3. The engine brake power output and rotational crankshaft speed, were required to estimate these coefficients, according to Fig. 6.5, 6.6 and 6.7.

Table 7.2 shows the friction coefficients` values.

Sea Trial 85% Load	M/E RPM	k1	k3	k2
NOR. 1	115.94	12976.0	620.0	0.00024
NOR. 2	116.55	12955.7	623.5	0.00020

Table 7. 2 Friction model coefficients

7.2.3 General remarks

- The Sea Trials simulation model is based on the Shop Test simulation model.
- The Sea Trials report did not mention the barometric pressure at the time of tests. Consequently, an assumption was made, that the barometric pressure was the same value with those at Shop Test, i.e.:

$$P_{baro} = 1.025 bar \tag{7.1}$$

- The engine was operated with heavy fuel oil during Sea Trial, instead of diesel oil which was used during Shop Tests.

- Combustion and Friction model data were changed with those attached at Tables 7.1 and 7.2.

- The initial cylinder pressure and temperature values and equivalence ratio ("General Data" tab), were revised in order the simulation model to achieve convergence between 2 or 3 calculation cycles. The less the calculation cycles are, the more accurate the simulation model`s calculations will be. Consequently, the initial values were revised with a trial-and-error procedure, according to which the trial values were evaluated with the model output; the loop ended at the time when the model was converged between 2 or 3 calculation cycles,.

- The input values were inserted at the relevant tabs in MOTHER interface.

- The A/C efficiency was calculated by taking into account the air mass flow rate, as the equation 6.12 shows. The constants` values ($C_1 = 0.806,\ C_2 = 0.0029,\ C_3 = 0$) were maintained the same with those calculated at Shop Test (Fig. 6.10). The A/C was tested for first time onboard during Sea Trials. Consequently, it is considered that it is not fouled, thus the efficiency from Sea Trial should be close to that at Shop Test at the same load. However, the A/C coolant temperature and the air mass flow rate, which affect the efficiency, differ from the two engine tests. Thus, MOTHER calculated the A/C efficiency by using the calculated air mass flow rate as input in a function which is based on Shop Test data (Eq. 6.12).

- A distinct value for the pressure loss coefficient of the Air Cooler, was selected in order to achieve convergence between the calculated and measured value. The pressure loss in the air cooler is connected mainly with the air flow rate, scavenging air temperature and pressure (Eq. 6.13). However, the RTA48-T engine`s scavenging air temperature, could be potentially affected by changing the air cooler`s coolant (i.e. sea water) flow rate inside the air cooler. Consequently, the pressure loss and the air cooler`s efficiency are affected too. The air cooler coolant`s (i.e. S.W.) flow rate was not reported in Sea Trials report nor in Performance reports; instead, an advice to reduce the scavenging air temperature, from the office`s Technical staff to the ship`s Chief Engineer, exists.

- According to the official Sea Trials report, the engine was tested at 85% of load during the "Endurance Test" and the various measurements of the engine performance data, were taken with two hours time interval. For that reason, the first measurement set is referred as "NOR. 1" and the other as "NOR. 2". The simulation results are shown in Tables in Chapter 7.3.

7.3 Sea Trials Simulation Results

The simulation results consist of three parts. The first part includes the cylinders` pressures whereas, the detailed report of MOTHER calculations is included in the second part. The energy balance calculation is attached at the end of this Chapter.

Note: *positive* error percentages means that the calculated value is greater than the measured at Sea Trial and *negative* the vise versa.

Regarding the Sea Trials measured data (Appendix II), the following could be pointed out:

- There are two measurements for the "Scavenging air pressure": the one from a local manometer and the other from a digital manometer at the Engine Control Room (ECR). For that reason, the two measurements have quite different values. As input in MOTHER simulation model, the local measurement was taken into account. Table 7.3 is a part of the Sea Trial report, regarding the scavenging air pressure.

	NOR.1	NOR.2
GROUP 6: Scavenging Air Pressure [MPa]	0.20	0.21
Scavenging Air [MPa]	0.22	0.22

Table 7. 3 Scavenging Air

- Temperature deviations are observed in the scavenging air temperature. The one measurement is referred as "Group 6: M/E Air Cooler air out" and the other as "Scavenging air temp.". The reason of these temperature differences of the same value, depends on the way the thermometers were taken the measurement; in other words, it depends on the air flow (turbulent flow) and the thermometer position. Thus, the reported deviations are reasonable. In MOTHER interface, the "M/E Air Cooler Out" temperature (Table 7.4) was used as input value at the "General Data" tab of the 'Inlet Receiver' element (inlet receiver is situated after the air cooler, Fig. 6.1). The air temperature after the air cooler was calculated by MOTHER. A part of the Sea Trial report is attached in Table 7.4, regarding the scavenging air temperature.

	NOR.1	NOR.2
GROUP 6: M/E Air Cooler Air Out [^oC]	47	48
Scavenging Air temperature [^oC]	46	47

Table 7. 4 Scavenging Air Temperature

- The Endurance Test`s engine load target is the 85% of MCR. However, the measured values during Sea Trials and the calculated Brake Horse Power, were slightly exceeded this percentage (Table 7.7, 7.8).

The first set of results is attached in Tables 7.5 and 7.6. Although, the error range between the calculated and measured values for combustion and compression pressure, is minimal, the evaluation of the model, is accomplished by considering the other performance data.

The simulation`s results of various performance data are shown in Tables 7.7 and 7.8.

7.3.1 Cylinder Pressure Results

Endurance Test NOR.1 at 115.94RPM		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error	Calculated	Measured	Error
Aver.	bar	*139.7*	*139.5*	*0.2%*	*101.2*	*101.7*	*-0.5%*
1		139.9	139	0.6%	101.3	103	-1.7%
2		139.9	140	-0.1%	101.2	101	0.2%
3		139.8	139	0.6%	101.2	102	-0.8%
4		139.5	140	-0.4%	101.0	101	0.0%
5		139.5	140	-0.4%	101.2	101	0.2%
6		139.8	139	0.6%	101.1	102	-0.9%

Table 7. 5 Pmax and Pcomp for Endurance Test NOR.1

Endurance Test NOR.2 at 116.62RPM		Maximum pressure in cylinder			Compression pressure in cylinder		
		Calculated	Measured	Error	Calculated	Measured	Error
Aver.	bar	*139.7*	*138.5*	*0.8%*	*100.9*	*101.3*	*-0.4%*
1		139.6	138	1.2%	109.0	100	0.9%
2		139.8	139	0.6%	101.0	102	-1.0%
3		139.6	138	1.2%	101.0	102	-1.0%
4		139.4	140	-0.4%	100.8	101	-0.2%
5		139.8	138	1.3%	101.0	101	0.0%
6		139.7	138	1.2%	100.9	102	-1.1%

Table 7. 6 Pmax and Pcomp for Endurance Test NOR.2

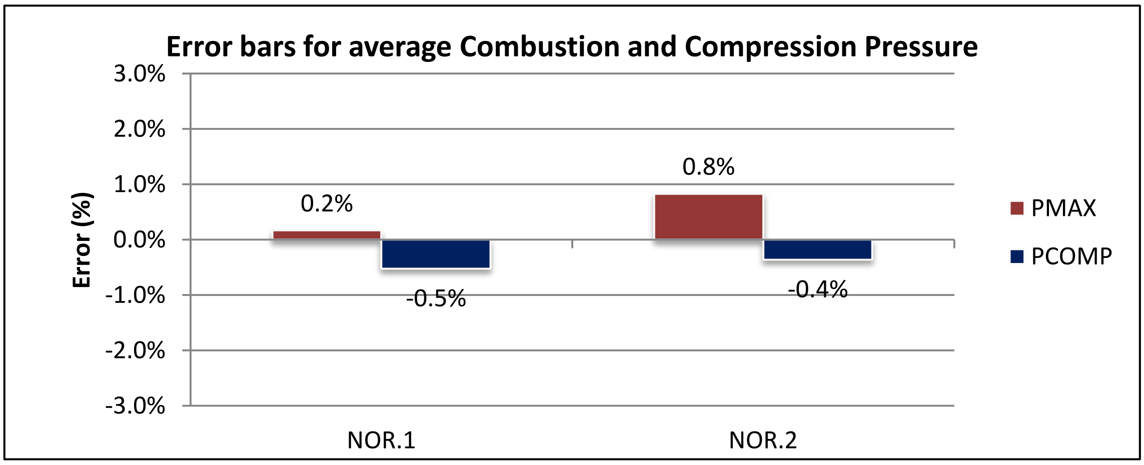


Fig. 7. 2 Error bars for average Pmax and Pcomp

7.3.2 Various Results

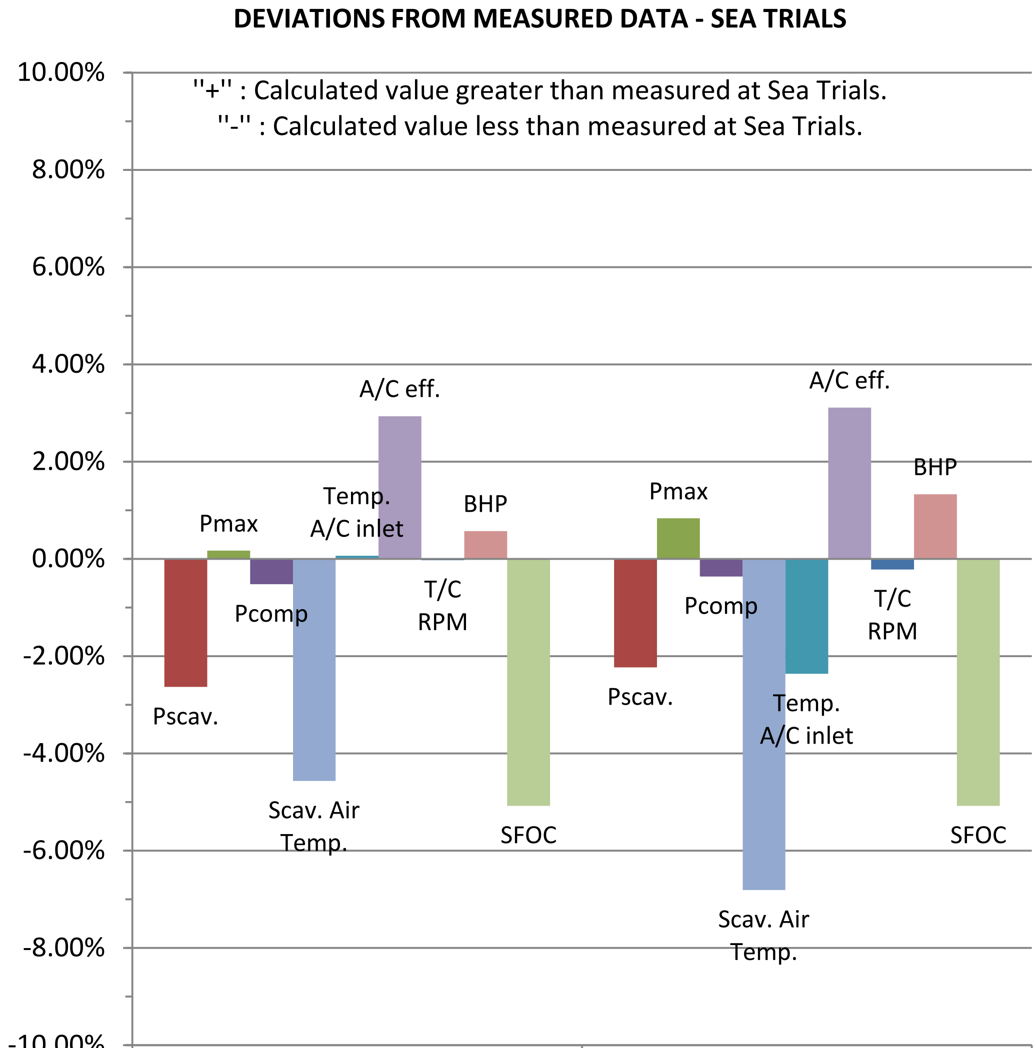
	Endurance Test NOR.1 at 115.94 RPM		MEASURED	CALCULATED	ERROR (%)
	M/E Load (% MCR)	-	86.20%	86.69%	*0.57%*
	T/C Rev.	RPM	17000	16996	*-0.03%*
PRESSURES	Scavenge air press. (incl. P_{baro})	bar	3.225	3.14	*-2.64%*
	Barometric Pressure		-	1.025	-
	Pmax (aver. value)		139.5	139.7	*0.17%*
	Pcomp (aver. value)		101.7	101.2	*-0.52%*
	Pme		-	15.91	-
	FMEP per cylinder (aver. value)		-	0.86	-
	Pres. Drop In Air Cooler		0.00882574	0.00881111	*-0.17%*
	Pres. in exh. gas receiver		-	2.80	-
TEMPERATURES	Air temp. before A/C	^oC	160	160.1	*0.06%*
	Scavenge air temp		46	43.9	*-4.57%*
	A/C coolant water		24	24	-
	Exh. Gas temp. aft. cyl. (aver. value)		339	343.9	*1.45%*
	Exh. Gas temp. aft. Turbine		240	-	-
MISC.	Compressor pressure ratio	-	-	3.072	-
	Compressor corrected air flow rate	m^3/s	-	14.15	-
	Compressor efficiency	-	-	0.82	-
	Turbine efficiency	-	-	0.83	-
	Air Cooler efficiency	-	0.831	0.855	*2.93%*
	Indicated Horse Power (IHP)	kW	-	7076.3	-
	Brake Horse Power (BHP)	kW	6637	6674.8	*0.57%*
	Engine Mech. Efficiency=BHP/IHP	-	-	0.943	-
	SFOC	g/kW-h	177.2	168.2	*-5.08%*

Table 7. 7 Detailed simulation results for NOR.1

Endurance Test NOR.2 at 116.62 RPM			MEASURED	CALCULATED	ERROR (%)
	M/E Load (% MCR)	-	86.10%	87.21%	*1.33%*
	T/C Rev.	RPM	17100	17063	*-0.22%*
PRESSURES	Scavenge air press. (incl. Pbaro)	bar	3.225	3.14	*-2.64%*
	Barometric Pressure		-	1.025	-
	Pmax (aver. value)		138.5	139.7	*0.83%*
	Pcomp (aver. value)		101.3	100.9	*-0.36%*
	Pme		-	15.91	-
	FMEP per cylinder(aver. value)		-	0.848	-
	Pres. Drop In Air Cooler		0.00882574	0.00890773	*0.93%*
	Pres. in exh. gas receiver		-	2.82	-
TEMPERATURES	Air temp. before A/C	°C	165	161.1	*-2.36%*
	Scavenge air temp.		47	43.8	*-6.81%*
	A/C coolant water		24	24	*0.00%*
	Exh. Gas temp. aft. cyl. (aver. value)		343	341.6	*-0.41%*
	Exh. Gas temp. aft. Turbine		240	-	-
MISC.	Compressor pressure ratio	-	-	3.095	-
	Compressor corrected air flow rate	m^3/s	-	14.24	-
	Compressor efficiency	-	-	0.82	-
	Turbine efficiency	-	-	0.83	-
	Air Cooler efficiency	-	0.831	0.856	*3.11%*
	Indicated Horse Power (IHP)	kW	-	7072.3	-
	Brake Horse Power (BHP)	kW	6627	6715.0	*1.33%*
	Engine Mech. Efficiency=BHP/IHP	-	-	0.949	-
	SFOC	g/kW-h	177.2	168.2	*-5.08%*

Table 7. 8 Detailed simulation results for NOR.2

Apart from the results in tables, the calculation errors are been attached for selected values in Figure 7.3.



	NOR.1: 85%	NOR.2: 85%
Pscav.	-2.64%	-2.23%
Pmax	0.17%	0.83%
Pcomp	-0.52%	-0.36%
Scav. Air Temp.	-4.57%	-6.81%
Temp. A/C inlet	0.06%	-2.36%
A/C eff.	2.93%	3.11%
T/C RPM	-0.03%	-0.22%
BHP	0.57%	1.33%
SFOC	-5.08%	-5.08%

Fig. 7. 3 Deviations from measured data

7.3.3 Energy Balance

The energy balance was calculated by MOTHER and illustrated at Figures 7.4 and 7.5, whereas, the arithmetic values are attached in Table 7.9.

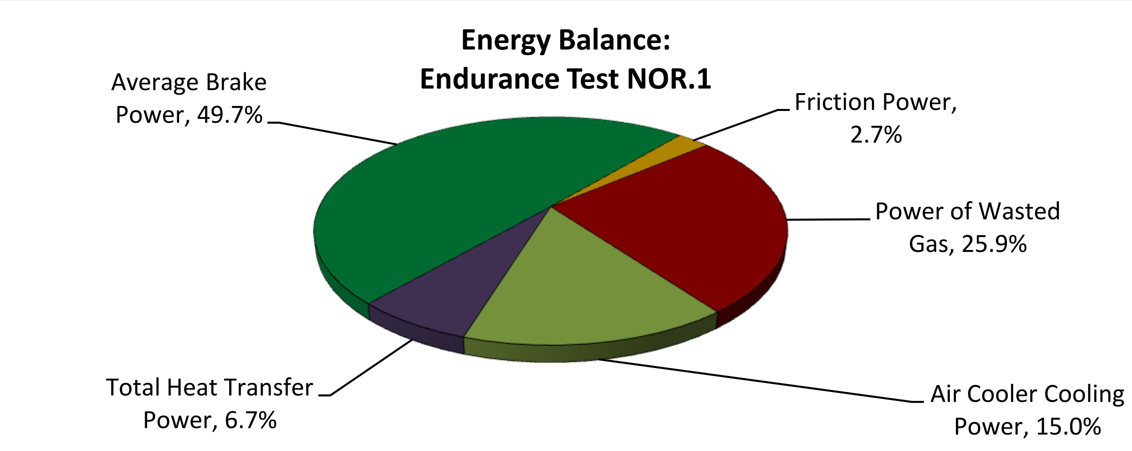


Fig. 7. 4 Energy Balance for NOR.1

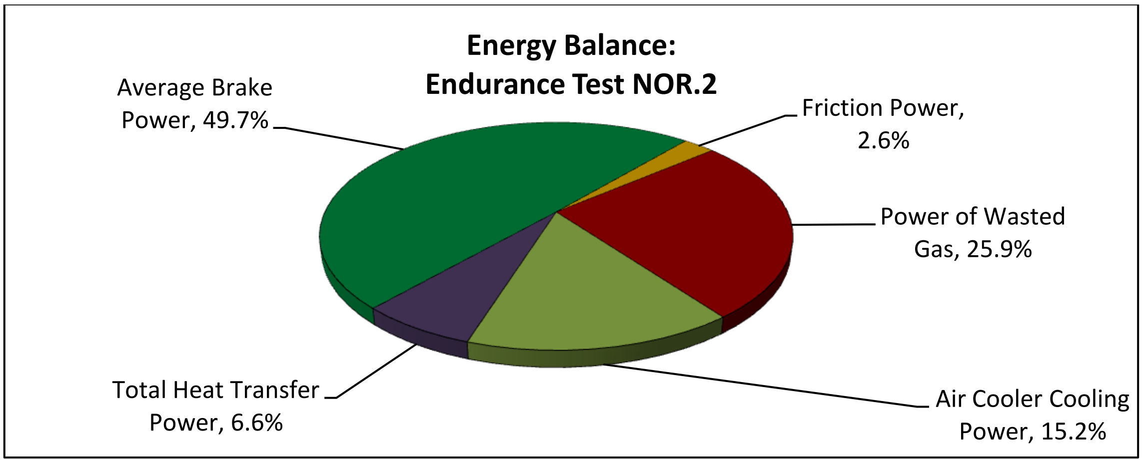


Fig. 7. 5 Energy Balance for NOR.2

Sea Trial Endurance Test	NOR.1		NOR.2	
	[kW]	[%]	[kW]	[%]
Power of Wasted Gas	3480.7	25.9%	3502.9	25.9%
Air Cooler Cooling Power	2019.8	15.0%	2049.7	15.2%
Total Heat Transfer Power	894.7	6.7%	897.7	6.6%
Average Brake Power	6674.8	49.7%	6715	49.7%
Friction Power	357.9	2.7%	354.1	2.6%
Total Power	13427.9	100.0%	13519.4	100.0%

Table 7. 9 Energy Balance values

7.4 Sea Trials results` evaluation

According to the simulation`s results shown in Chapter 7.3, the following remarks could be made:

- Combustion and compression pressure were calculated with error less than 1%. The correct calculation of compression pressure, indicated that the ports` (scavenging and exhaust) timing and effective area data, have been set correctly in MOTHER interface. In addition, the correct calculation of maximum pressure, indicated that the combustion model has been adequately calibrated. Considering also, the satisfying results from the Shop Test simulation, it is indicated that the engine model is able to predict these pressures for any load.

- The scavenging air pressure was simulated by MOTHER, according to turbocharger data. As explained above (Chapter 7.3), there are two arithmetic values for the measured scavenging air pressure (Table 7.3); amongst them, the local measurement was used for calculating the simulation error. Even though the scavenge air pressure was underestimated by MOTHER, as Tables 7.7 and 7.8, show, the calculation is satisfying because the simulations` error was kept lower than 3%.

- Regarding the air temperature before the air cooler, a deviation of 5 degrees was noticed between the two (NOR.1: i.e. 160^{o}C and NOR.2: i.e. 165 ^{o}C) measurements during Sea Trials. The time span between the measurements (2 hours) and the engine load which was maintained stable at almost 85%, could not justify this difference in air temperature. A measurement error of ±5 degrees of the thermometer, could be a possible cause for this deviation. On the other hand, the calculated by MOTHER temperatures had a deviation of +1.0 degrees and the simulation`s error for NOR.1 (160.1^{o}C) was 0.06% and for NOR.2 (161.1^{o}C) was -2.36% respectively.

- The calculated values of the air temperature after the air cooler, deviated considerably from the measured values (Table 7.7, 7.8). The air temperature, after the air cooler, was calculated through the air cooler efficiency equation (Eq. 6.12) which is based on the Shop Test data. However, the air mass flow rate, the air cooler`s coolant temperature and the air temperature before the air cooler, had different values from those of Shop Test. Furthermore, as mentioned in Chapter 7.2.3, the air cooler`s coolant flow rate is adjustable, thus the air outlet temperature and the efficiency are affected. Consequently, by considering the aforementioned remarks, the calculated errors regarding the scavenge air temperature (NOR.1: -4.57%, NOR.2: -6.81%) and the air cooler efficiency (NOR.1: 2.93%, NOR.2: 3.11%), were reasonable and therefore have been accepted.

- The simulation model converged to the measured turbocharger rotational speed with error less than 0.5%. The compression pressure is directly related with the turbocharger rotational speed. The simulation results indicated that both the compression pressure and the turbocharger speed, deviated slightly from the measured values during Sea Trial.

- Finally, the accurate calculation of the Brake Horse Power has been liable to the definition of the friction model coefficients. These coefficients have been depicted by the adjusted friction model at Shop Test (Fig. 6.5, 6.6 & 6.7) and resulted in predicting the BHP with error less than 2% for both the NOR.1 and NOR.2 trials. On the other hand, the specific fuel oil consumption was underestimated by 5.08%. MOTHER calculated the fuel consumption by considering that the engine burns the ISO fuel and also, the calculation was based on the calibration made at Shop Test simulation. In practice, the engine`s fuel consumption is higher than those measured at Shop Test, due to multiple reasons, such as different overall engine condition, different loading, fuel quality and injection system`s condition. Sea Trials` reported consumption (i.e. 177.2g/kW/h) is higher than the reported at Shop Test (i.e. 167.2g/kW/h), thus, the aforementioned statement is confirmed.

- In conclusion, the simulation model has been adjusted satisfactorily and it is able to perform predictions of the engine operation in any operating point between the 50% and 100% of MCR, provided that the appropriate input data are available.

CHAPTER 8

PERFORMANCE EVALUATION

8.1. Introduction

The scope of this thesis is the evaluation of the engine operation through the simulation model and it is provided in this Chapter. In practice, a 'M/E Full Performance Report' is sent via FAX or TELEX or e-mail by the Ship`s Chief Engineer. Readings are taken twice per voyage (or otherwise stated), if weather conditions permit. It could sometimes happen, readings to be taken when weather is rough only in case of emergency or in case the office`s Chief Engineer has been advised for it. The 'M/E Full Performance Report' contains data (mainly temperatures and pressures) that reflect the engine condition during its operation. Furthermore, it contains various ship data, like the ship draught, and sea condition, which also affect the engine performance.

The simulation`s model input data have been mentioned in Chapter 7.2 and the methodology that had been followed does not differ from that on Sea Trials (Chapter 7).

The performance evaluation was possible by considering the simulation results from MOTHER, the sea condition, the onboard measurements and some information provided by mail exchanges between the Chief Engineer (onboard the ship) and the Headquarters` Technical staff. Finally, the engine performance for each case was also evaluated through the ISO corrections` methodology which has been provided by the engine`s manufacturer. A comparison of the results that each method provides, has been also included in this chapter.

8.2. Performance reports` variables

Several input data have been updated for each performance run of the simulation model, whereas other adjustments, like the combustion and the friction model, remained unchanged. The following list shows the required input data to the simulation model, which have been acquired from "M/E Performance Reports":

- Main engine rotational speed [rpm]
- Scavenging air pressure [bar]
- Barometric pressure [bar]
- Scavenging air temperature [K]
- Fuel injected per cycle [kg]
- Start of static injection [CA deg]
- M/E jacket cooling outlet (F.W.) temperature for each cylinder [K]
- M/E piston cooling outlet (L.O.) temperature for each cylinder [K]
- Turbocharger rotational speed [rpm]

- Turbine coolant (L.O.) outlet temperature [K]
- Air cooler coolant water (S.W.) temperature [K]
- Exhaust valve coolant (F.W.) temperature [K]
- Exhaust receiver`s coolant (engine room air) temperature [K]
- Inlet receiver`s coolant (engine room air) temperature [K]

The exact injection timing was calculated as described in Chapter 3.10; whereas, for the mass of fuel injected per cycle per cylinder refer to Chapter 8.3.

The simulation model considered the data mentioned above, except for the fuel consumption, the injection timing, the ambient data and the sea water temperature, as initial values for the simulation model calculations; thus, they were calculated again by MOTHER and the results were expected to converge to their initial values. That target was accomplished according to the following subchapters, otherwise the reason of any deviation is explained.

8.3. Simulation model adjustments

The basic model from which the other models (i.e. Sea Trials, Performance cases) derive through adjustments, was the Shop Test simulation model. Thus, the simulation of the engine operation from a M/E Performance Report, was possible after adjusting the Shop Test model by importing data mentioned in Chapter 8.2 and by updating several values of combustion model and friction model respectively. Each case that has been simulated, was based on the Shop Test`s model of 75% load. The required combustion model and the friction model adjustments are described separately below.

8.3.1. Combustion model

The combustion model`s variables, as stated in Chapter 6.2, were the following:

- Model parameter “ a ”
- Start of static injection in CA degrees
- Mass of fuel injected per cycle per cylinder in kg.

The estimation of parameter “a” for each performance case, is possible through the equation shown at Fig. 8.1 (or Fig. 6.3), where the Woschni-Anisits “a” parameter is a function of the LIxRPM, which expresses the Engine Load, as determined from Shop Test simulation.

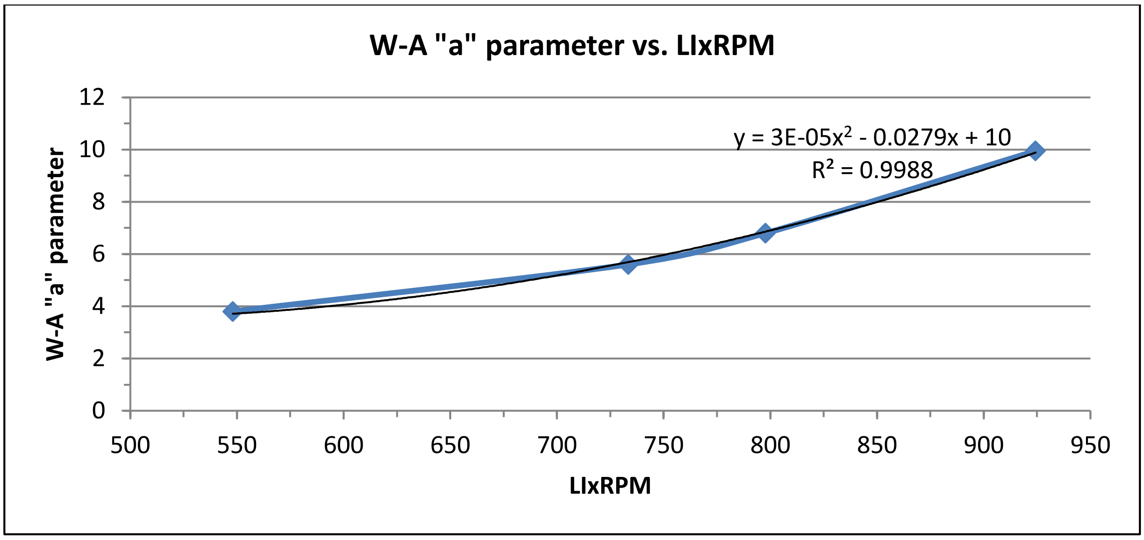


Fig. 8. 1 Woschni-Anisits 'a' parameter. vs. LIxRPM

The product of LIxRPM is the variable 'x' in Fig. 8.1 and the 'a' coefficient is calculated by the equation shown in the relevant figure.

The injection timing for each case was calculated as described in Chapter 3.10. According to VIT arrangement, the injection timing is dependent on scavenging air pressure and the engine`s rotational speed. Furthermore, the fuel quality affects the FQS setting, which is reported in M/E Engine Performance Reports. By adding the FQS value to the VIT value, the product is referred as start of static injection (SOI) and that value was inserted at the relevant tab of MOTHER interface.

The last input in the combustion model, was the fuel mass injected per cylinder per cycle. The fuel mass consumed by the engine was reported in tons per day, thus, the precise mass in milligrams injected per cylinder remained unknown; for that reason the calculated fuel mass was corrected in several cases. Due to the fact that the IHP is dependent on the injected fuel mass, its value was used for the evaluation of the injected fuel mass, as reported in Chapter 4.2.1.5.2, by checking whether the calculated by MOTHER IHP exceeded or not the measured IHP. In case MOTHER overestimated IHP, the fuel mass value was decreased by several milligrams. In cases where the IHP hadn`t been reported, the fuel mass injected per cylinder was corrected by several milligrams (decreased) according to case where the IHP have been reported.

8.3.2. Friction model

The friction model that is selected, is the "Winterbone and Tennant". The model`s coefficients have been adjusted for the Shop Test simulation (Table 6.2). As stated in Chapter 6.2, each of the friction model coefficient forms a function (see Fig. 6.5, 6.6 & 6.7). Depending the performance`s report operating point, the friction coefficients are calculated by the equations showed on the Figures 8.2, 8.3 and 8.4. The friction model coefficients figures` are attached below.

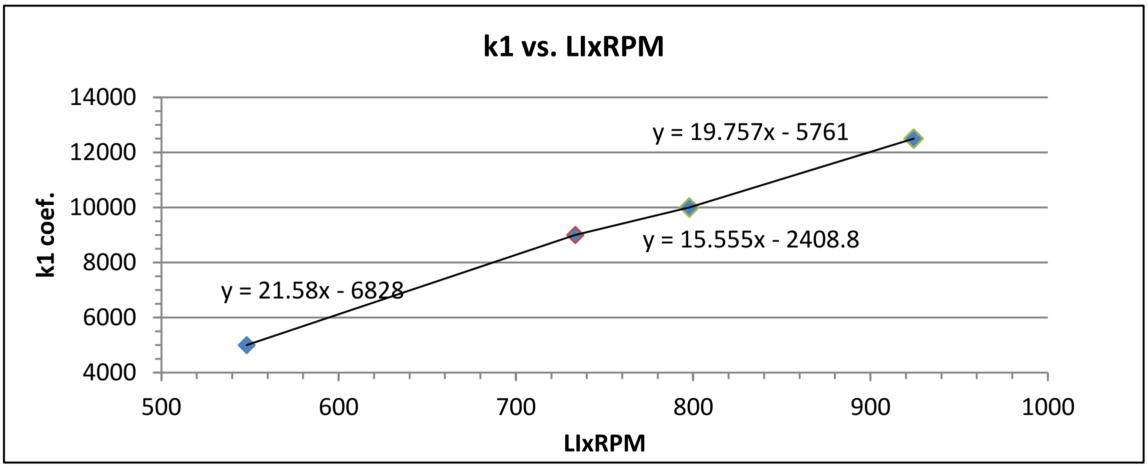


Fig. 8. 2 "k1" coef. vs. LIxRPM

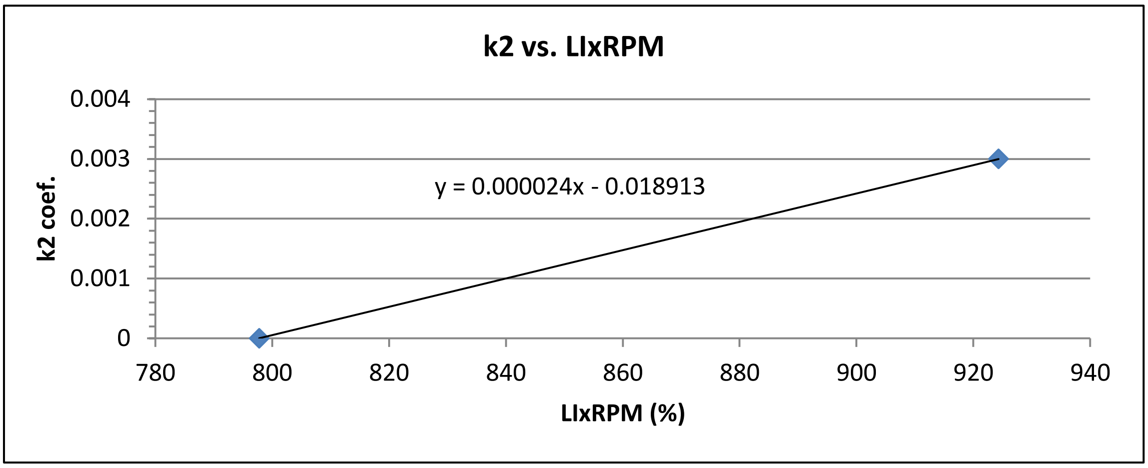


Fig. 8. 3 "k2" coef. vs. LIxRPM

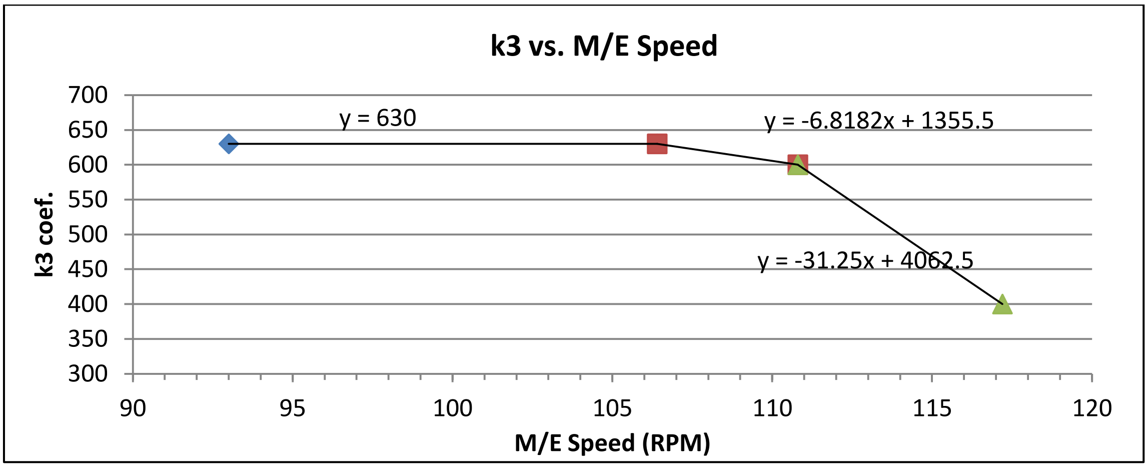


Fig. 8. 4 "k3" coef. vs. M/E Speed

The friction mean effective pressure increased in higher engine loads. Figure 8.5, illustrates the FMEP relation towards engine`s load, as calculated at Shop Test. The calculated FMEP values of Sea Trials and the various simulations of the engine operation (according to M/E Performance Reports), are also illustrated in Figure 8.5. According to this figure, the friction during the actual engine operation is higher than those during Shop Test. There are two factors that could explain the increased friction during the operation of the engine, the one is related with the calculation error of the friction coefficients and the other is connected with the wear of the engine itself. Regarding the calculation error, it should be stated that the friction model coefficients estimation was based on the reported Load Indicator position and engine speed. The calculation of the engine`s actual power output for each performance report, is challenging. The calculation methods and the difficulties in calculating the engine output are discussed in Chapter 8.4. Consequently, a clear answer towards the previous inquiry, will not be provided. Furthermore, the monitoring of wear rate of the engine parts (liner, piston rings etc) is of utmost importance and the inspection intervals are defined by the manufacturer.

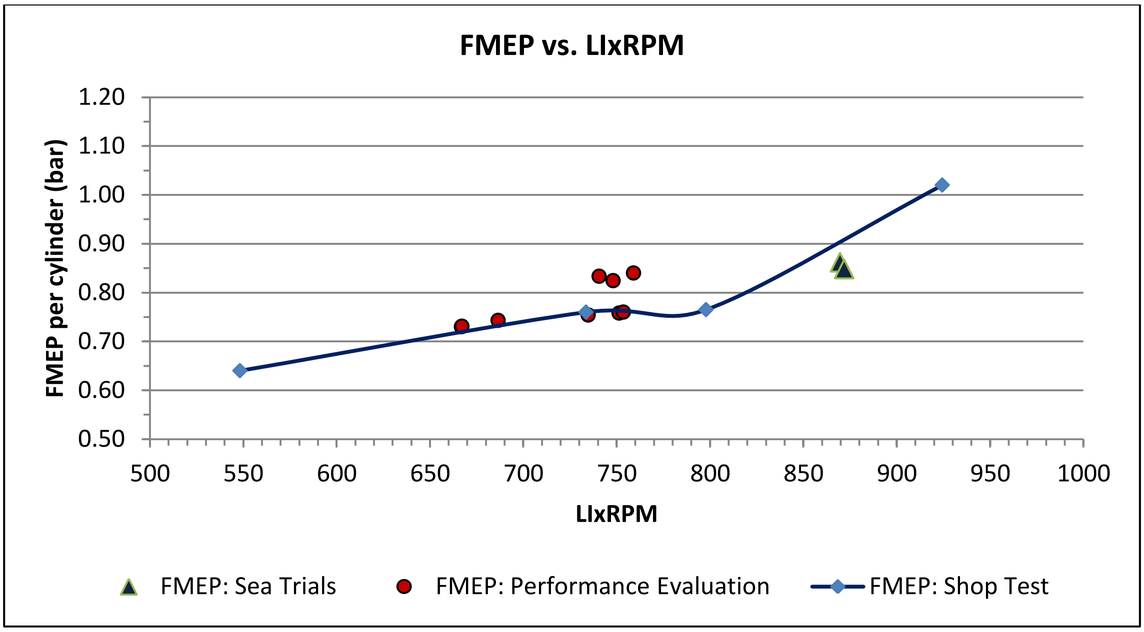


Fig. 8. 5 FMEP vs. Load

8.4. M/E Brake Power Calculation

The accurate calculation of the engine`s Brake Power, is challenging and it can be accomplished only with a torsion meter. In case a torsion meter is not installed with the other Main Engine`s equipment, manufacturer provides calculation methods in estimating the engine`s output. The calculation of the power output (BHP) of this particular 6RTA48-T main engine, was utilized by several estimation methods, which are attached below.

- Power reported in M/E Performance Report (MEPIC[3] calculation by the computer)

[3] MEPIC: Main Engine Power Index Calculation

- MEPIC calculation (Manual calculation)
- Sulzer M/E Report
- Regression calculation based on MEPIC equations
- According LI (Load Indicator) position
- According Pme (Mean effective pressure)
- According Propeller Law

The reported power in M/E Performance Report, is the calculated power output by the engine main computer situated in the Engine Control Room, which uses environmental data, performance data and fuel quality data. The accuracy of this value is dependent on updating the engine computer with the correct fuel quality data and on avoiding any mistyping.

The engine power can also be calculated by following the manufacturer\`s methodology, as described in “M/E Special Engine Manual” [12]. The relevant equations are not attached due to confidentiality reasons. According to the provided methodology, the engine output is firstly estimated through the Load Indicator position and the Mean effective pressure. Several corrections for the fuel\`s calorific value, FQS position and other factors, are made resulting in the final approximation of the engine output, which is corrected by a factor which considers the sea condition. The engine output as per MEPIC calculation, is that final value.

Regarding the regression method based on MEPIC equations, it could be noted that the corrections sequence is maintained the same except for the equations themselves, which have been calculated from the beginning. The methodology is not provided again due to copyright reasons.

Load Indicator position expresses the amount of fuel the governor supplies the engine; thus, the power output is directly related to its position. The 6RTA48-T engine, is equipped with electronic governor which working principle is to control the fuel supply to engine in order to succeed continuous operation without load and speed fluctuations. According to the calculation procedure, the relation between the engine output and LI position multiplied with the engine\`s RPM (i.e. L.I.xRPM) at Shop Test, is used for estimating the engine output. Table 8.1 shows the aforementioned calculation and Figure 8.6 illustrates its result. After calculating the LIxRPM for each case and substituting at the equation illustrated at Fig. 8.6, the product of the calculation is the power\`s output estimation.

Load (%)	Power Output (actual)	Engine Speed (planned)	Load Indicator Position	LIxRPM
50%	3892	92.9	5.9	548.1
75%	5807	106.3	6.9	733.5
85%	6577	110.8	7.2	797.8
100%	7749	117.0	7.9	924.3

Table 8. 1 LIxRPM calculation (Shop Test data)

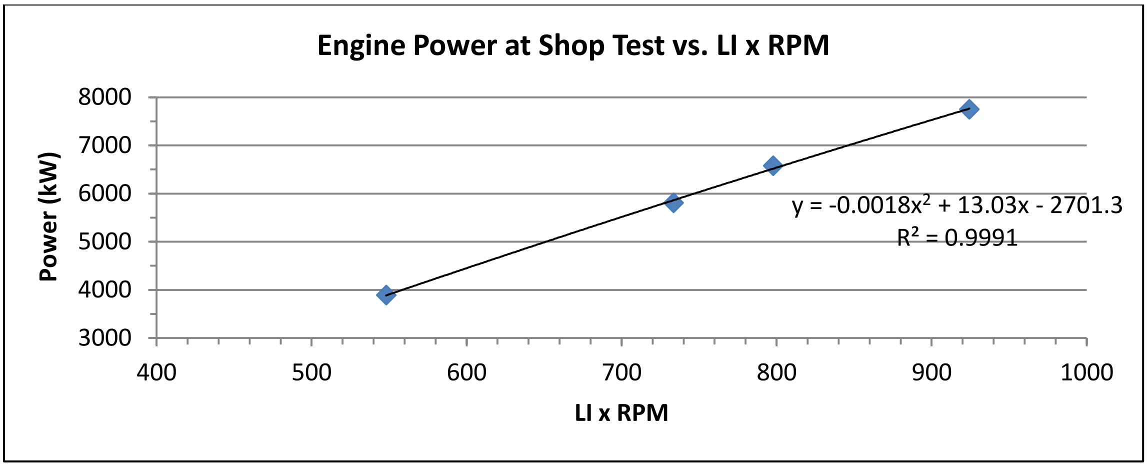


Fig. 8. 6 Power vs. LIxRPM (Shop Test data)

The power estimation procedure according to the mean effective pressure, is similar to those described above, regarding the Load Indicator position. In other words, the relation between Mean Effective Pressure and Load Indicator position at Shop Test, is approached by the linear curve shown in Fig. 8.7.

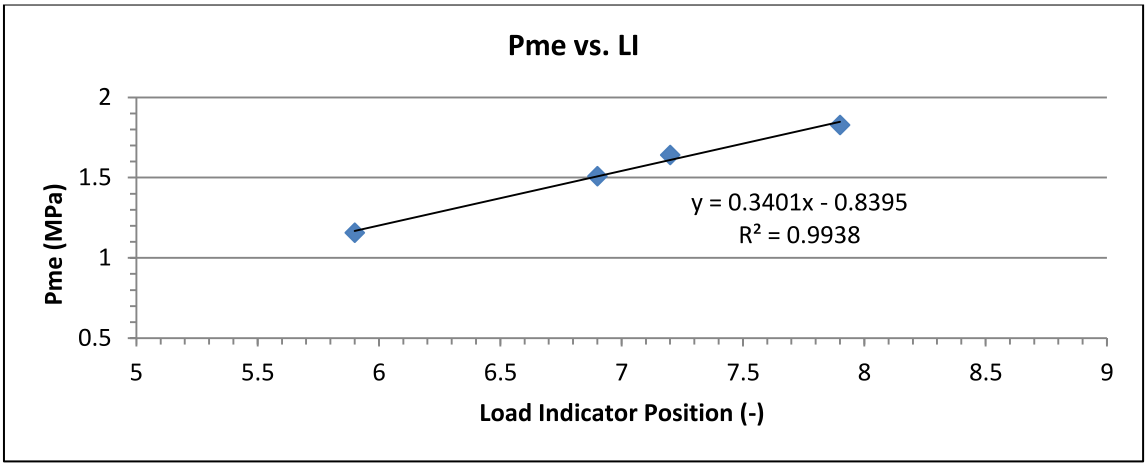


Fig. 8. 7 Pme vs. LI Pos. (Shop Test data)

When the Mean Effective Pressure has been defined (Fig. 8.7) the engine power output could be estimated by the following equation:

$$P_0 = 36.1876 \times P_{me} \times n_a \qquad (8.1)$$

where:

P_0	brake power	$[kW]$
P_{me}	mean effective pressure	$[MPa]$
N_a	engine speed	$[rpm]$

Note: The compensation factor (i.e. 36.1876) in equation 8.1, is valid for this particular engine only.

Another method for the estimation of power output is based on the propeller law, according to which the engine output is in proportion to the cube of the engine speed and the mean effective pressure is in proportion to the square of the engine speed. The propeller curve through the point of CMCR (Contract Maximum Continuous Rating), i.e. nominal power at nominal engine speed (100% power at 100% speed) is called nominal propeller characteristic. As the 6RTA48-T engine is fitted with fixed pitch propeller, it was loaded on the test bed according to this propeller characteristic. However, the power requirement of the ship with smooth and clean hull should be less than the propeller law defines. With increasing resistance, changes in wake flow conditions, due to marine growth and ageing of the vessel`s hull, a rough or mechanically damaged propeller, unfavorable sea and weather conditions or operation in shallow water, the propeller will require a higher torque to maintain its speed than it did at the time of sea trial. In such case, the operating point will then be located to the left of the original propeller curve (Fig. 8.8). The farther left the operating point is situated from the nominal propeller curve in the load diagram, the poorer the air supply to the engine and the more unfavorable the engine`s operating conditions will become. Therefore taking these into consideration, propeller is designed to light propeller condition on the new ship so as to absorb the less power than the nominal propeller curve passed the point of MCR. This adequate reserve is referred as “Propeller Margine”. As an example, the propeller design takes 5% margin in propeller speed (15% in brake horse power), in this case, engine power and speed at the new ship with the 6RTA48-T installed, are as follows:

MCR Power x Nominal speed:	7700 kW at 117 RPM
At new ship,	
Engine speed to absorb 100% load:	122.8 RPM
Absorbed power at 117 RPM:	6545 kW

Thus, the operating point of the engine should lie between the nominal propeller curve and another propeller curve which has a 5% margin of the nominal engine speed (light propeller running), in order to avoid the probability of overloading the engine. In case of serious fouling and heavy weather conditions, the propeller will require even higher speed in order to absorb the engine`s power (heavy running, the operating point moves to the left). Such a condition is unfavorable, as mentioned above, however, no matter how much the engine speed is reduced. Thus, the calculation of this power provides an indication of the reserve power and the engine load. Figure 8.8 illustrates the engine`s Load Diagram. The following formula provides an approximation for the nominal propeller`s law [12].

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{8.2}$$

Whereas for the light propeller curve, the following formula was used:

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^{2.45} \tag{8.3}$$

where:

P_1	brake power at operating point 1	$[kW]$
P_2	brake power at operating point 2	$[kW]$
n_1	engine speed at operating point 1	$[rpm]$
n_2	engine speed at operating point 2	$[rpm]$

The Load Diagram contributes to the evaluation of the engine operation and generally to control the engine operation. The calculated power by the nominal propeller curve, is used for evaluating the other calculation methods and the MOTHER output as well.

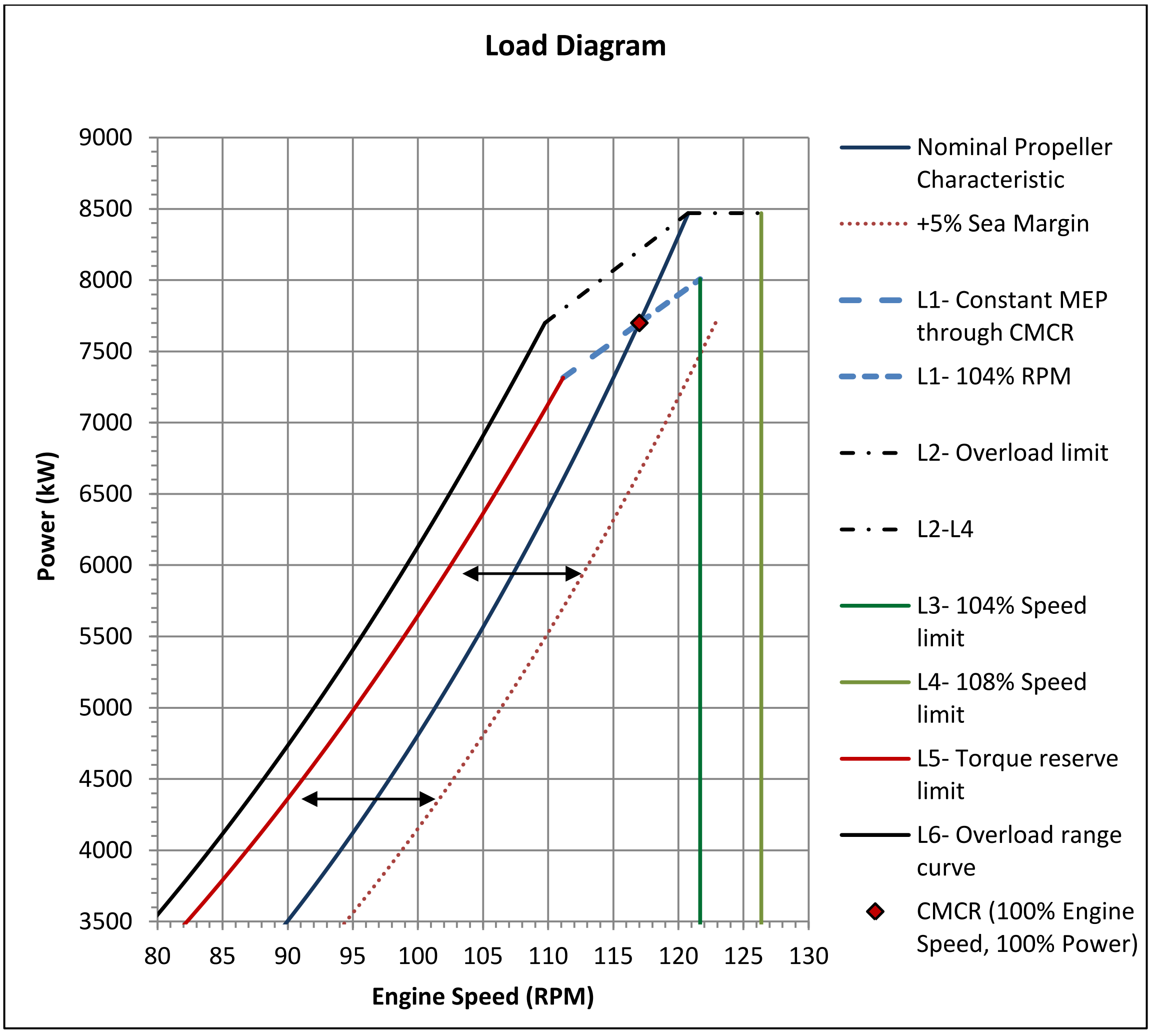


Fig. 8. 8 Load Diagram

Remarks:

- The aforementioned methods were used for the estimation of the engine output and an average value was calculated for each case.

- The outcome of each calculation method, was compared with the power output calculated by MOTHER..

- The loading diagram for each case was plotted. In fact the operation points were more than one due to the number of power estimations, but each case was described by a unique operation point (from measured onboard data and MOTHER calculation).

8.5. Simulation model runs

This Section includes the simulation results each of the selected performance reports, categorized per measurement day. The main input data are attached in a Table in each sub-section; some ship data are also included. Detailed performance data (various temperatures and pressures) are included in Appendix VII – M/E Performance Data.

The selected cases for simulation, covered the following aspects of the engine operation:

- Ship in loading and ballast condition
- Calm sea and winds
- Moderate sea state
- Normal engine operation
- Unbalanced pressures in cylinders and increased wear of major engine parts
- Operation with mixed heavy fuel
- Engine operation after overhauling and replacement of major parts (i.e. piston, piston rings, piston crown, liner, injectors)

This Section`s scope, is to provide the engine`s simulation model results and to evaluate the actual engine operation in conjunction with those results. Furthermore, each “M/E Performance Report” was evaluated according to the ISO correction methodology, in order to examine whether the two methods (i.e. simulation and ISO) coincide to the same results.

In the beginning, three cases with reported IHP values, were selected for simulation i.e. January the 28th, 2008 (Chapter 8.5.1), February the 10th, 2008 (Chapter 8.5.2) and March the 12th, 2008 (Chapter 8.5.3). The IHP measurements contributed in assessing the fuel mass injected per cycle per cylinder. Given that the recorded fuel consumption was reported in tons/day and that a measurement error existed, the precise fuel quantity in grams per cylinder per thermodynamic cycle, was not known. For that reason, the fuel quantity was estimated according to the methodology described in Charter 8.3, and afterwards it was corrected (in fact it was decreased) in such a way to achieve convergence between the calculated IHP and the reported IHP. Finally, the evaluation of consecutive performance reports (i.e. Jan. 28th, Feb. 10th, Mar. 12th) contributes in assessing the engine performance in time and in different weather, loading conditions. However, the IHP had not been reported in any other available “M/E Performance Report”.

The selected cases covered the engine operation in early 2008 and 2009, and late 2009 as well.

8.5.1. M/E Performance Report of 28 Jan. 2008

8.5.1.1. Performance data – results

Table 8.2 shows major ship data.

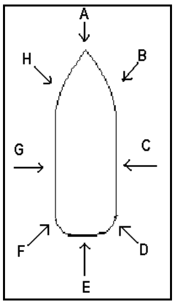
Ship Data			
Ship Condition	LADEN		
Ship Speed	14.60 kts		
Prop. Aparent Slip	1.60%		
Wind Force / Direction	3	D	
Swell Height / Direction	0.5m	D	
Current speed/ Direction	0.5kts	FE	
Draught FOR/AFT	10.10m	10.30m	
Displacement	49751 MT		

Table 8. 2 Major Ship Data

Table 8.3 shows Main Engine related data that used for the simulation model.

M/E Data		
Engine Speed	109.99 RPM	
Brake Power Reported	5917 kW	
Indicated Power Reported	5886 kW	
L.I. Pos.	6.9	
Fuel Consumption	29.05 MT/24h	
T/C Speed	15900 RPM	
Scavenge Air Pressure (relative)	1.80 bar	
Scavenge Air Pressure (total)	2.81 bar	
Air temp.: E/R / Bef. Blower	46°C	38°C
Air cooler temp. Bef. / After	155°C	50°C
Press. Drop Air Cooler	130 mmWG	0.0127483 bar
Sea Water Temp.	30°C	
Air Cooler Efficiency	84.00%	

Table 8. 3 M/E data

The simulation results for Pmax, Pcomp and IHP are illustrated below.

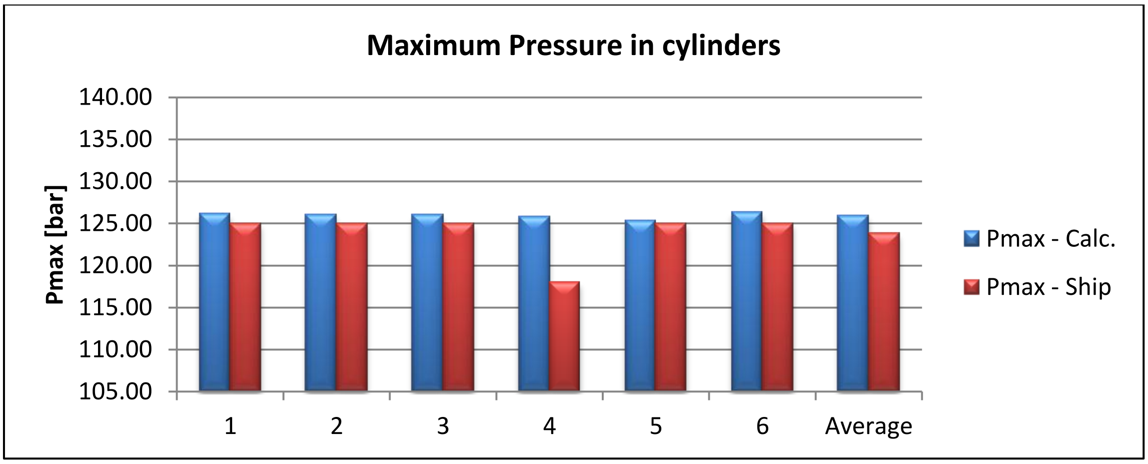


Fig. 8. 9 Pmax, Ship reported vs. “MOTHER” calculated

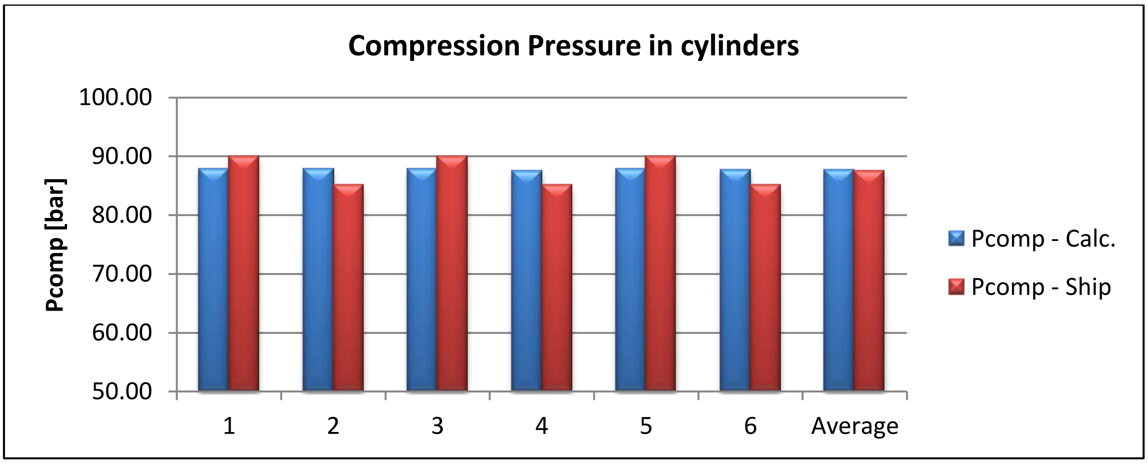


Fig. 8. 10 Pcomp, Ship reported vs. “MOTHER” calculated

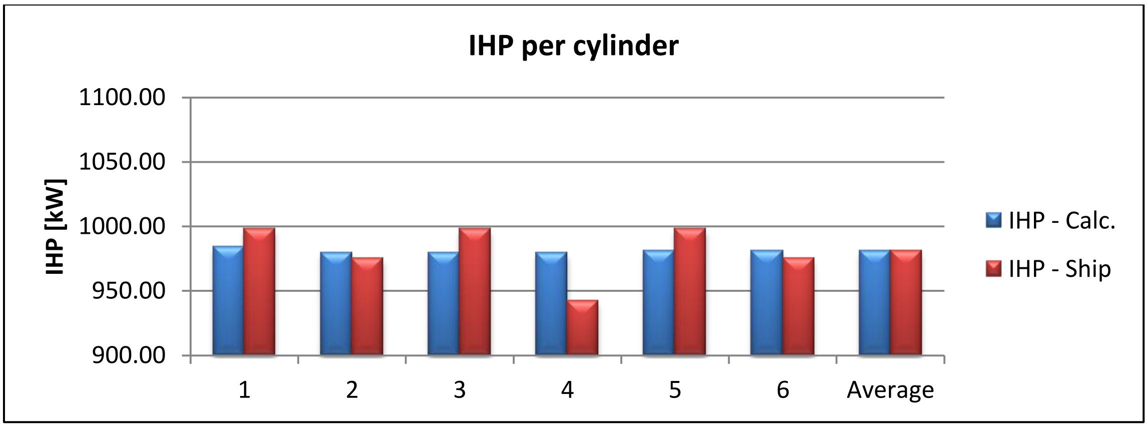


Fig. 8. 11 IHP, Ship reported vs. “MOTHER” calculated

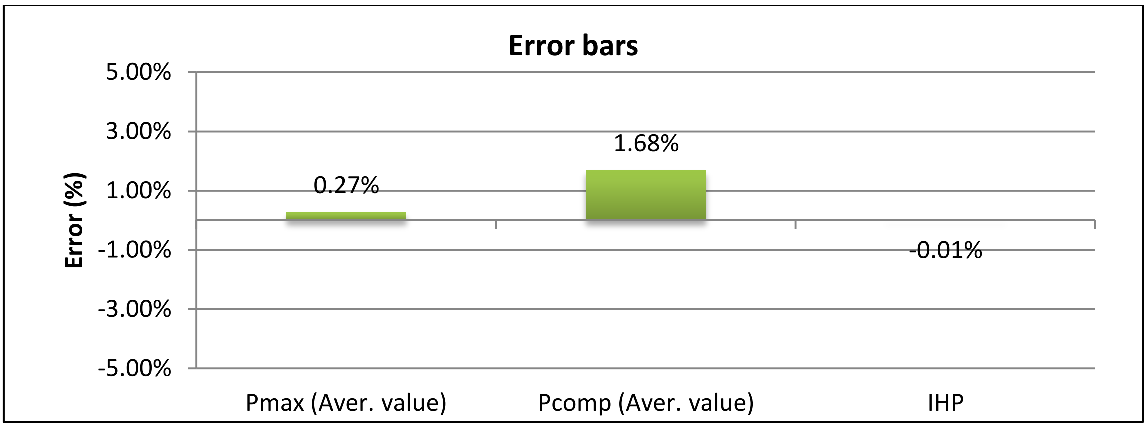


Fig. 8. 12 Error Bars for average values

Table 8.4 below shows the figures` 8.9, 8.10 & 8.11 data.

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	87.80	87.80	87.80	87.50	87.80	87.70	87.73
Pcomp – Ship		90.00	85.00	90.00	85.00	90.00	85.00	87.50
Error	%	*-2.44%*	*3.29%*	*-2.44%*	*2.94%*	*-2.44%*	*3.18%*	0.27%
Pmax – Calc.	bar	126.10	126.00	126.00	125.80	125.30	126.30	125.92
Pmax – Ship		125.00	125.00	125.00	118.00	125.00	125.00	123.83
Error	%	*0.88%*	*0.80%*	*0.80%*	*6.61%*	*0.24%*	*1.04%*	1.68%
IHP – Calc.	bar	984.30	979.90	979.90	979.20	980.90	981.00	980.87
IHP – Ship		998.00	975.00	998.00	942.00	998.00	975.00	981.00
Error	%	*-1.37%*	*0.50%*	*-1.81%*	*3.95%*	*-1.71%*	*0.62%*	-0.01%

Table 8. 4 Compression, maximum pressure and Indicated Horse Power per cylinder

The following comments could be made:

- Ship reported compression pressures had a considerable deviation of 5bar between cylinders, whereas “MOTHER” calculated pressures had a difference of 0.5bar. Unbalanced compression pressure among the cylinders could be owed to measurement error or could be a potential malfunction of the engine. High compression pressures for the correspondent engine load, increase the thermal load of various parts, such as piston rings (and especially the compression ring), piston crown, liner, exhaust valve seat, resulting in potential failure before the expected lifetime (expressed in operating hours). According to this individual report without any other information, or physical inspection of the engine, it is not possible to find the exact reason for the unbalanced compression pressure.

- Calculated average compression pressure and combustion pressure had an error less than 3%.
- Cylinder no. 4 had the minimum reported combustion pressure, i.e. 118bar, whereas MOTHER calculated 125.8bar. The lower combustion pressure could be an indication of malfunctioning injector for this cylinder.
- Mean IHP is calculated with -0.01% error, although there are deviations between individual cylinders.

Table 8.5 summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 109.99, LI POS.:6.9			**SHIP**	**CALC.**	***ERROR (%)***
PRESSURES	SCAVENGING AIR PR.	bar	2.81	2.68	*-4.63%*
	Pmax (mean value)		123.83	125.92	*1.68%*
	Pcomp (mean value)		87.50	87.73	*0.27%*
	IMEP		-	14.78	-
	BMEP		-	14.03	-
	PRESSURE DROP IN AIR COOLER		0.0127483	0.0127714	*0.18%*
	PRESSURE IN EXH. RECEIVER		-	2.40	-
TEMPERATURES	M/E AIR COOLER IN	^{o}C	155.0	156.2	*0.77%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		50.0	49.3	*-1.40%*
	AIR COOLER COOLANT WATER		30.0	30.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		366.0	333.7	*-8.83%*
	EXH. GAS AFT. TURBINE		320.0	-	-
MISC.	T/C speed	RPM	15900	15934	*0.21%*
	COMPRESSOR PRESSURE RATIO	-	-	2.67	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	12.31	-
	COMPRESSOR EFFICIENCY	-	-	0.830	-
	TURBINE EFFICIENCY	-	-	0.828	-
	AIR COOLER EFFICIENCY	-	0.8400	0.8471	*0.81%*
	IHP	kW	5886.00	5885.20	*-0.01%*
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.949	-
	BHP	kW	-	5583	-
	SFOC	g/kW-h	-	169.1	-

Table 8. 5 Simulation results for 28th Jan. 2008

The Brake Horse Power was calculated as per Chapter 8.4. The reported values and estimations` results, are attached below. Moreover, the errors from the calculated values from “MOTHER” were calculated for each power estimation.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	5917.0	76.8%	5583	72.5%	-5.64%
MEPIC Calculated	5544.0	72.0%			0.70%
SULZER Reported	5516.0	71.6%			1.21%
MEPIC Calculation (Regression)	5955.0	77.3%			-6.25%
Acc. L.I.xRPM	6150.6	79.9%			-9.23%
Acc. Pme	5999.9	77.9%			-6.95%
Acc. Propeller Law	6397.2	83.1%			-12.73%
Average	5925.7	77.0%	5583	72.5%	-5.78%

Table 8. 6 BHP

According to the table 8.4, the following figures are attached.

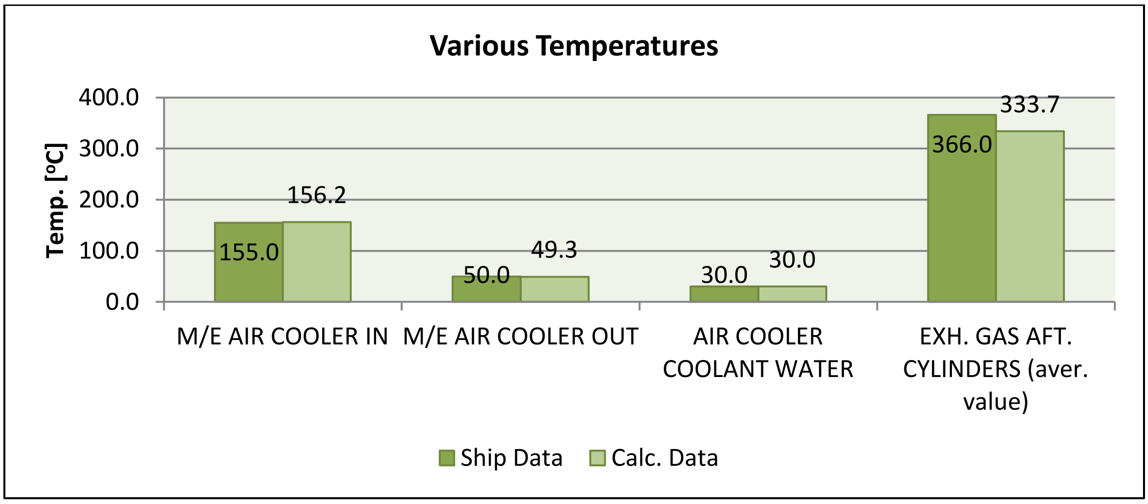


Fig. 8. 13 Various Temperatures

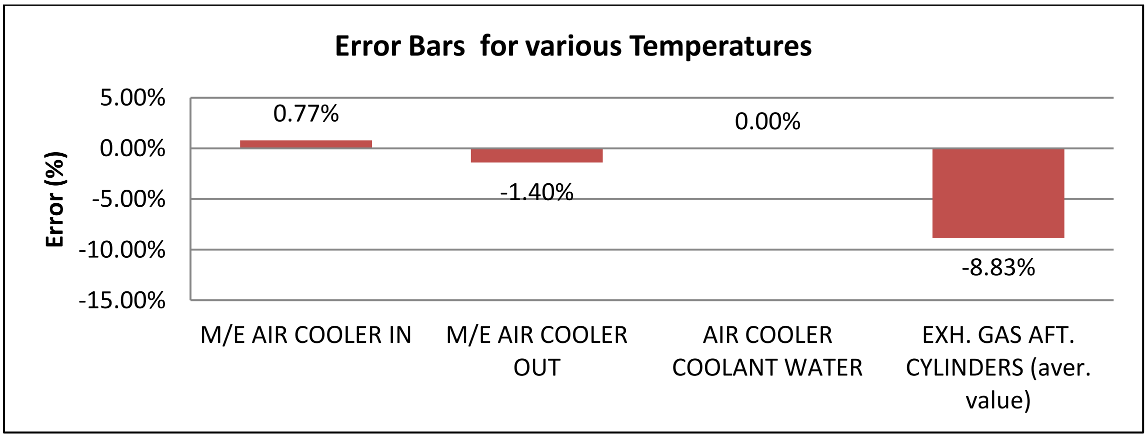


Fig. 8. 14 Error bars for various calculated temperatures

<u>Comments</u>:

- The reason of simulating the engine operation of 28th January of 2008, is the evaluation of the calculated indicated horse power with the reported IHP, and the overall evaluation of the engine performance.
- Sea conditions were favorable (apparent propeller slip equal to 1.6%), which means that the reliability of the measured data, is higher than in case of heavy sea state.
- IHP was calculated by the simulation model with minimum error, i.e. -0.01%.
- Regarding the BHP calculations, the following could be pointed out:
    - MEPIC Reported power and SULZER reported power values, deviated considerably. Possible reasons could have been either mistyping, or wrong calculation of the engine computer. Regarding the latter, it should be noted that the MEPIC calculator is updated by a person with the fuel quality data (reflected in FQS), consequently, in case of wrong calculation the human element, could be a considerable factor, as well.
    - MEPIC calculation is based on the L.I. position and on several consecutive corrections. For that reason the MEPIC calculation`s power is always lower than those calculated according to the L.I.xRPM method.
    - The L.I.xRPM value was used for the determination of the Woschni-Anisits “a” parameter and the friction model coefficients k_1, k_2 respectively.
    - MOTHER calculated BHP had the minimum deviation with the ‘MEPIC calculated’ and ‘SULZER reported’ power, whereas it had considerable deviations with the other power estimations.
    - The power estimation according the propeller law, contributed in assessing the kind of load (light or heavy) of the engine operating point.
- The scavenge air pressure was calculated with -4.63% error. Scavenge air pressure was affected by the barometric pressure, the compressor efficiency and the compression ratio too. Although MOTHER calculated scavenge air pressure was by 0.13bar lower than the measured pressure, the deviation was not an issue because the other performance parameters (compression pressure, combustion pressure and IHP) were simulated successfully.
- The mean compression pressure was estimated with 0.27% error.
- The air temperatures before and after the A/C, were predicted with error less than 1.5%.
- The pressure loss across the air cooler was calculated by MOTHER with 0.18% error. In fact, the pressure loss coefficient was set manually in MOTHER interface. The reason of that choice, was the considerable difference between the pressure loss coefficient at Shop Test and Performance reports. Specifically, the maximum pressure drop across the air cooler during Shop Test was 85mmWG, whereas in service the measured value for this particular report, was 130mmWG. As it could be noticed from the other simulations of the engine operation, the pressure drop across the air cooler was constantly maintained over 100mmWG. The pressure drop magnitude is related to the fouling of the air cooler. Although, the air cooler was cleaned at frequent intervals, the pressure drop never dropped to the first measured value during Shop Test. Engine manufacturer has set an alarm in case the pressure drop reaches 500mmWG.

- The calculated average exhaust gas temperature was by 8.83% less that the measured at the ship. The exhaust gas temperature reflects potential malfunctions in the air supply, combustion and gas systems. According to the available data and the lack of additional information, that deviation did not undermine a potential malfunction of the engine. Yet, what has particular interest, is the exhaust gas temperature in cylinder no.1, i.e. 390°C (Appendix V, Chapter V.1), which deviated considerably from both the other cylinders measurements and the mean calculated temperature by MOTHER (i.e. 333.7 °C). The increased exhaust gas temperature in cylinder no.1, could have been related with potential exhaust valve malfunction or with the increased (by 2.44%) compression pressure, or both. A clear answer to the question will not be provided here, but until the completion of evaluating the selected cases, cylinder no.1 had a history of increased pressures and temperatures.
- The engine operating point, as calculated by MOTHER, is illustrated at the engine load diagram in Fig. 8.15. The calculated operating point shows that the engine was light loaded (light propeller curve), thus the engine is operated in the safe region of load diagram and it has a 5% propeller margin, which means that for the same engine speed propeller is able to absorb extra power.
- The engine`s power distribution, as calculated by MOTHER (Fig. 8.16), is within the acceptable percentages.

The Load Diagram is illustrated at Fig. 8.15 and the Energy balance at Fig. 8.16.

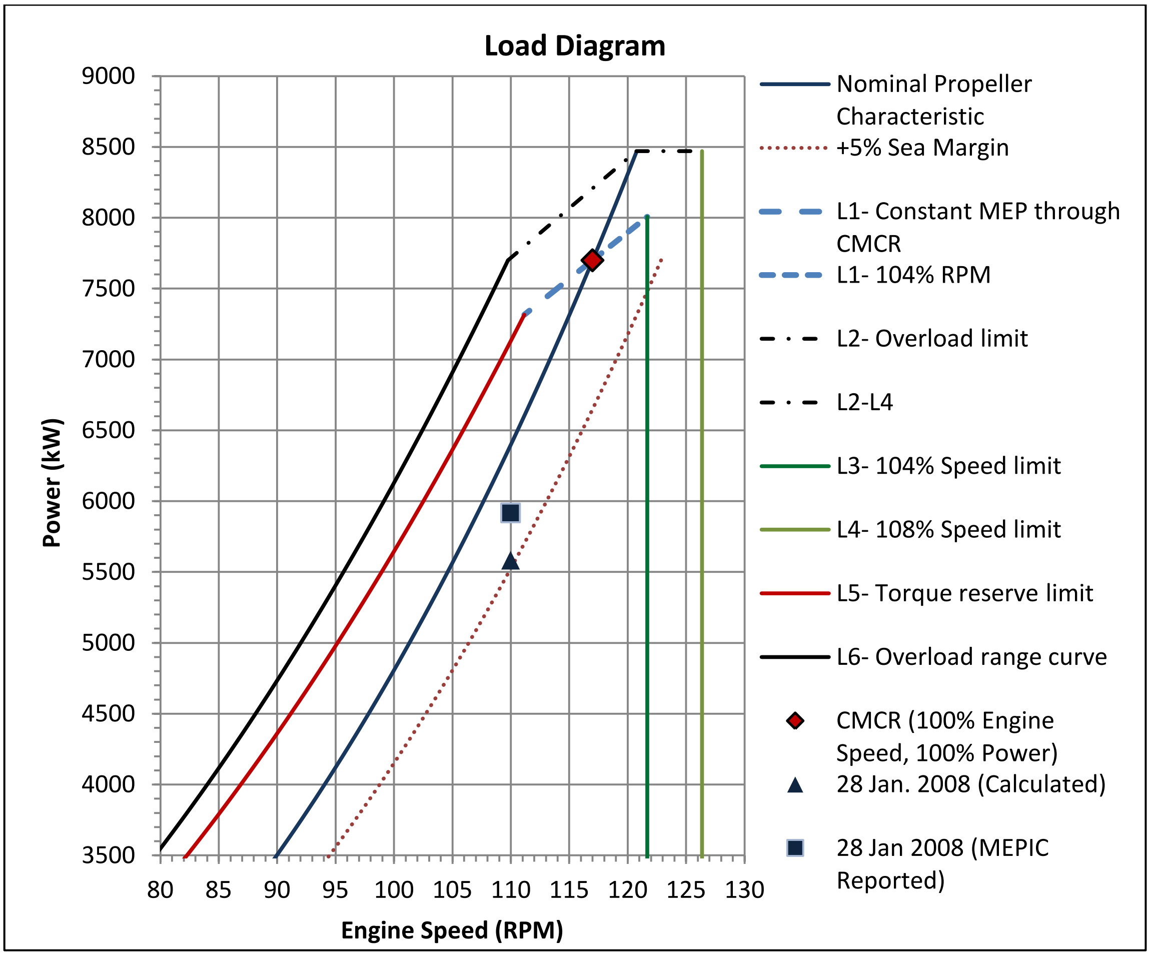


Fig. 8. 15 Load Diagram

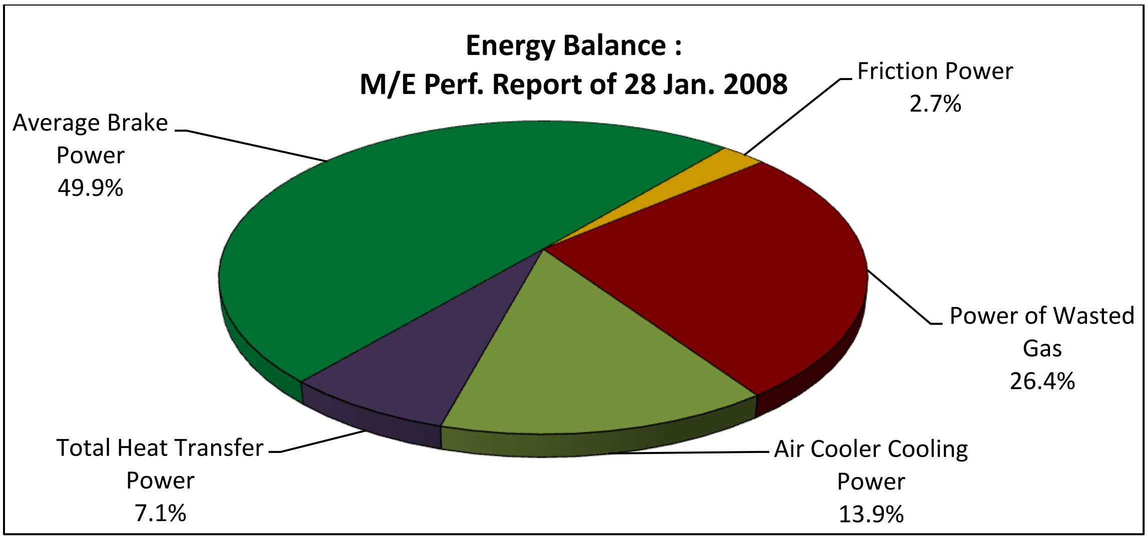


Fig. 8. 16 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	2961.2	26.4%
Air Cooler Cooling Power	1560.3	13.9%
Total Heat Transfer Power	792.0	7.1%
Average Brake Power	5583.0	49.9%
Friction Power	299.1	2.7%
Total Power	11195.6	100.0%

Table 8. 7 Energy Balance values

8.5.1.2. ISO corrections` results

This sub-section provides the results of the ISO correction methodology (which is described in Appendix VI) where the "MOTHER" output are compared with the ISO related considerations. According to the ISO correction methodology, the measured data were transformed to Shop Test conditions in order to make them comparable with Shop Test data, because both refer to the same ambient conditions. The table below shows both the corrected cylinder (maximum and compression) pressures and those depicted from Shop Test.

	(Pmax–Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	35.0	**125.0**	127.8	130.7	137.9	131.3
Pmax 2	40.0	**125.0**	127.8	125.3	137.9	131.3
Pmax 3	35.0	**125.0**	127.8	130.7	137.9	131.3
Pmax 4	33.0	**118.0**	120.8	125.3	137.9	131.3
Pmax 5	35.0	**125.0**	127.8	130.7	137.9	131.3
Pmax 6	40.0	**125.0**	127.8	125.3	137.9	131.3
Pmax Average	**36.3**	**123.8**	**126.6**	**128.0**	**137.9**	**131.3**
Pmax / $Pmax_{MCR}$	-	89%	91%	92%	99%	94%

Table 8. 8 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	1.2	**90.0**	92.8	91.2	97.9	91.0
Pcomp 2	1.2	**85.0**	87.8	91.2	97.9	91.0
Pcomp 3	1.2	**90.0**	92.8	91.2	97.9	91.0
Pcomp 4	-5.8	**85.0**	87.8	91.2	97.9	91.0
Pcomp 5	1.2	**90.0**	92.8	91.2	97.9	91.0
Pcomp 6	1.2	**85.0**	87.8	91.2	97.9	91.0
Pcomp Average	**-**	**87.5**	**90.3**	**91.2**	**97.9**	**91.0**

Table 8. 9 Corrected Pcomp

<u>Comments</u>:

- Four methods were used to calculate the maximum and compression pressures from Shop Test data (i.e. according scavenge pressure, according main engine speed, according brake power, and according compression pressure); the output of each one was compared with the corrected maximum and compression pressures, separately.
- Pressure differences below 5bars, between corrected values and reference values, are not an issue but it should be monitored in connection with past recorded data, future data and engine load.
- Average combustion pressure seems to be normal according to Pcomp calculation, even though the average corrected Pmax is 1.4bar lower than the average calculated (acc. Pcomp.).
- The engine has relatively high rotational speed, i.e. 109.9rpm, for the correspondent load (Load Diagram Fig. 8.15). According to Shop Test, the calculated combustion pressure is equal to 137.9bar for the correspondent M/E rpm, but the corrected mean combustion pressure is 126.6bar; the pressure difference expresses the potential power

increase for the same speed. In such a case, the more fuel injected in cylinders the more the combustion pressure will be. Consequently, this difference is not a issue.

- The reference combustion pressure according Power, was based on Shop Test figure, i.e. Pmax vs. Power, where the calculated Power output (according to manufacturer recommendations) was used for the estimation of combustion pressure (Pmax). The calculation output, together with calculated Pmax according M/E rpm, contributed in assessing the upper margin of combustion pressure for the correspondent power output and speed. Consequently, the mean higher combustion pressure (i.e. 131.3bar) than the mean corrected combustion pressure (i.e. 126.6bar) is reasonable and it is not an issue.
- ISO correction methodology shows that cylinder no 4, had lower than expected maximum pressure, whereas the other cylinders had the same maximum pressure.
- Mean corrected compression pressure is considered normal because the deviation from the reference compression pressure according to Pscav and Power calculation methods, was less than 5bars. On the other hand, calculated compression pressure according to M/E rpm was overestimated, due to the increased engine speed.
- The ratio $P_{MAX}/P_{MAX,MCR}$ contributes in assessing the maximum pressure magnitude in conjunction with calculated maximum pressure from Shop Test.
- The difference between maximum and compression pressure ($P_{MAX} - P_{COMP}$), shows whether or not pressure difference was equalized among the cylinders and contributed in assessing the compression pressure that the compression rings were undertaken. In this particular case, it is shown that cylinder no. 4 had the minimum pressure difference. A possible malfunctioning injector is inferred from the lower maximum pressure.
- MOTHER simulation output and ISO correction methodology, resulted in the same consideration:
    - Measured combustion pressure and measured compression pressure, were normal according to MOTHER simulation and ISO corrections.

8.5.2. M/E Performance Report of 10th Feb. 2008

Main Engine Performance Report of February the 10th, 2008, is the second performance report where the Indicated Horse Power has been reported to.

8.5.2.1. Performance Data – results

Table 8.10 shows some major ship data.

Ship Data		
Ship Condition	LADEN	
Ship Speed	11.70 kts	
Prop. Apparent Slip	15.60%	
Wind Force / Direction	5	AB
Swell Height / Direction	1.5m	D
Current speed/ Direction	2.8kts	H
Draught FOR/AFT	10.10m	10.30m
Displacement	49751 MT	

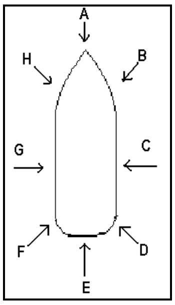


Table 8. 10 Major Ship data

Table 8.11 Main Engine related data that used in the simulation model.

M/E Data		
Engine Speed	108.87 RPM	
Brake Power Reported	5546 kW	
Indicated Power Reported	5921 kW	
L.I. Pos.	6.9	
Fuel Consumption	28.98 MT/24h	
T/C Speed	15900 RPM	
Scavenge Air Pressure (relative)	1.800 bar	
Scavenge Air Pressure (total)	2.824 bar	
Air temp.: E/R / Bef. Blower	46°C	38°C
Air cooler temp. Bef. / After	155°C	50°C
Press. Drop Air Cooler	130 mmWG	0.0127483 bar
Sea Water Temp.	29°C	
Air Cooler Efficiency	83.33%	

Table 8. 11 M/E data

As reported in Table 8.10, the ship faces adverse weather, and this can be explained by the propeller slip, the wind, current and the swell direction.

The calculated cylinder pressures are illustrated at the following figures.

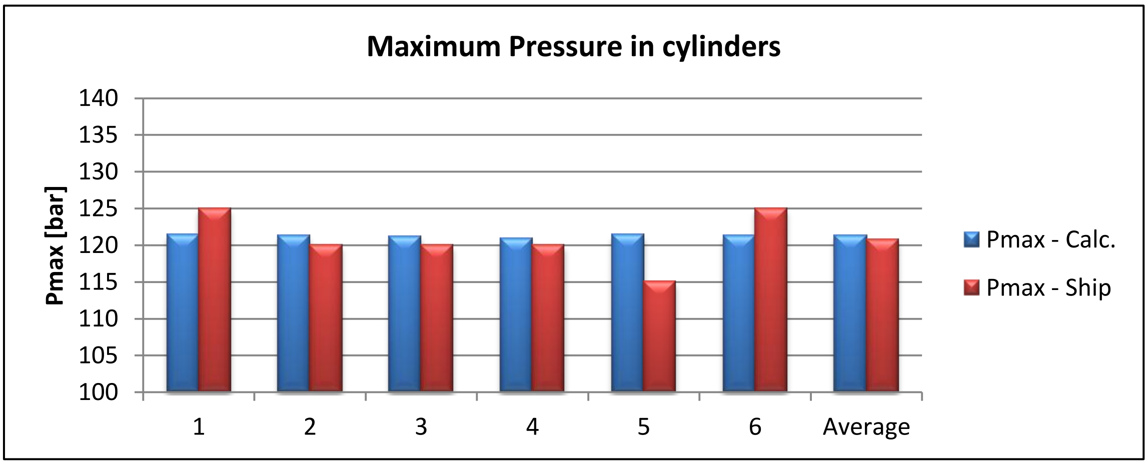


Fig. 8. 17 Pmax, Ship reported vs. "MOTHER" calculated

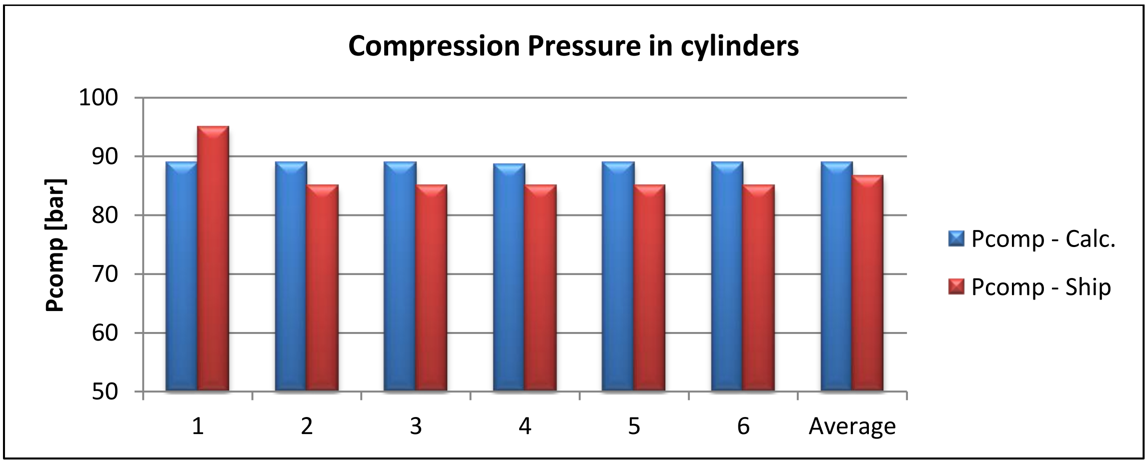


Fig. 8. 18 Pcomp, Ship reported vs. "MOTHER" calculated

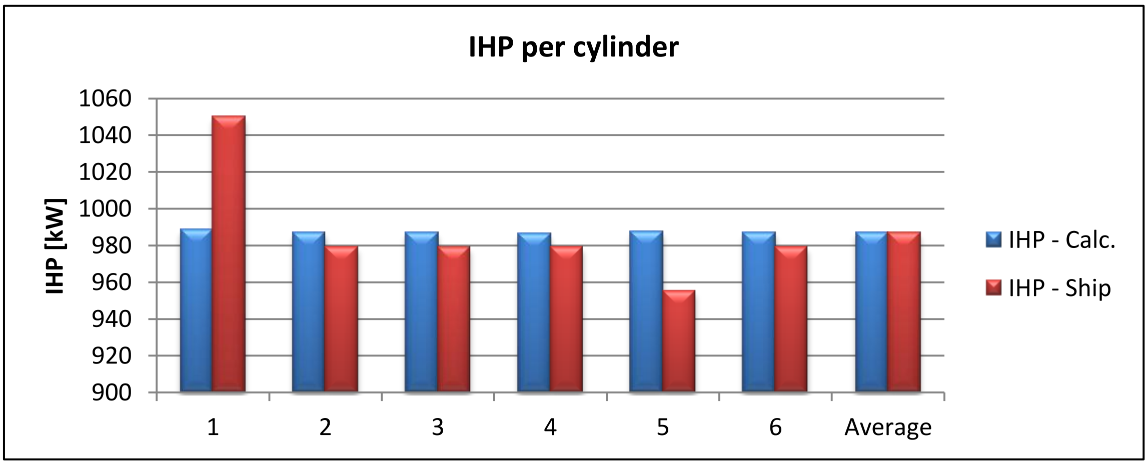


Fig. 8. 19 IHP per cylinder, Ship reported vs. "MOTHER" calculated

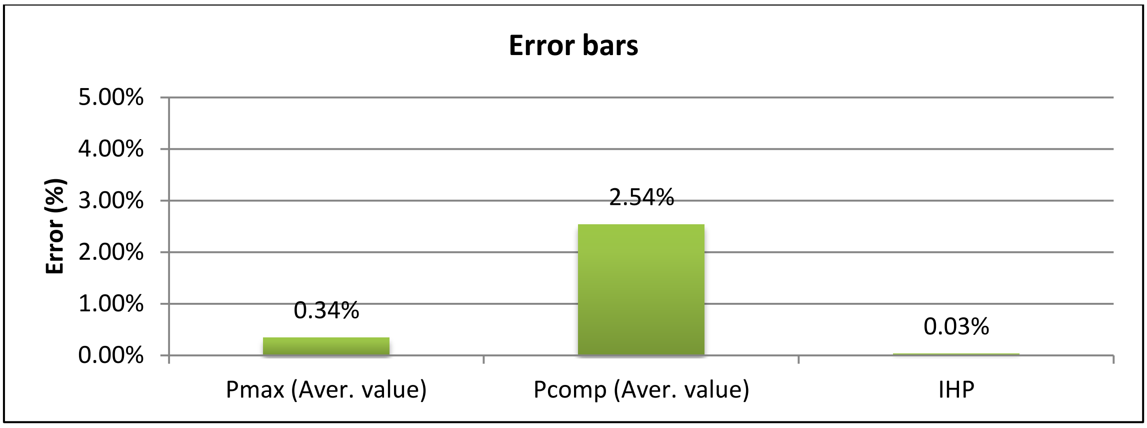


Fig. 8. 20 Error Bars for average values

The table below shows the figures` 8.17, 8.18 & 8.19 data.

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	88.9	88.9	88.9	88.6	89	88.9	88.87
Pcomp – Ship		95.0	85.0	85.0	85.0	85.0	85.0	86.67
Error	%	*-6.42%*	*4.59%*	*4.59%*	*4.24%*	*4.71%*	*4.59%*	*2.54%*
Pmax – Calc.	bar	121.4	121.3	121.2	120.9	121.4	121.3	121.25
Pmax – Ship		125.0	120.0	120.0	120.0	115.0	125.0	120.83
Error	%	*-2.88%*	*1.08%*	*1.00%*	*0.75%*	*5.57%*	*-2.96%*	*0.34%*
IHP – Calc.	bar	988.3	987	986.9	986.4	987.4	986.9	987.15
IHP – Ship		1050.0	979.0	979.0	979.0	955.0	979.0	986.83
Error	%	*-5.88%*	*0.82%*	*0.81%*	*0.76%*	*3.39%*	*0.81%*	0.03%

Table 8. 12 Compression, maximum pressure and Indicated Horse Power per cylinder

The following comments could be made:

- The mean indicated power was predicted satisfactorily with an error of 0.03%, even though reported IHP for cylinders no. 1 and no. 5 deviated considerably from the mean value.
- Average combustion and average compression pressure were predicted with an error less than 3%.
- Cylinders no 2, 3, 4 and 5 had lower measured combustion pressure than those calculated by MOTHER. Lower maximum pressure could have been caused by malfunctioning injectors (particularly in cylinder no 5). The Chief Engineer informed that the injectors for cylinders no 4, 5 and 6 would be replaced in next port. According to mail exchanges, the injectors were replaced and the simulation is continued with the next performance report in Chapter 8.5.3, sent on 12 Mar. 2008.
- Compression pressure in cylinder no. 1 (i.e. 95bar), was by 10bars higher compared to the other measured pressures. Furthermore, the combustion pressure was higher

(i.e.125bar) than the mean combustion value. By considering the high exhaust gas temperatures of 28th Jan 2008 (Chapter 8.5.1) and by combining both high compression pressure and combustion pressure in unit no 1, it is concluded that this irregularity could hide a potential flaw of the engine; for that reason the company`s technical staff had requested the crew to advise about the condition of the cylinders by inspecting them from scavenge space. In general, high (compared to normal pressures for the correspondent load) compression and combustion pressures in an individual cylinder, increases the wear rate of the various parts inside the cylinder (compression ring, piston crown, exhaust valve seat, liner) and the thermal load of them, as well. Consequently, operation under higher pressure loads than normal, could result in failure of engine parts before the accomplishment of the provided by the manufacturer, life expectancy (expressed in running hours).

- On the other hand, "MOTHER" calculated the compression pressures equalized between cylinders and by 2.54% higher than the average measured value. Lower compression pressure compared to its normal values for a specific load, could mean poor scavenging, leaks at the various valves in the cylinder head or the piston rings do not seal the combustion chamber properly. The exact cause of low compression pressure could not be found by examining only the pressure`s values.

Table 8.13, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 108.87, LI POS.: 6.9			**SHIP**	**CALC.**	***ERROR (%)***
PRESSURES	SCAVENGING AIR PR.	bar	2.82	2.70	*-4.39%*
	Pmax (mean value)		120.83	121.25	*0.34%*
	Pcomp (mean value)		86.67	88.87	*2.54%*
	IMEP		-	15.04	-
	BMEP		-	14.24	-
	PRESSURE DROP IN AIR COOLER		0.012748	0.0127567	*0.07%*
	PRESSURE IN EXH. RECEIVER		-	2.42	-
TEMPERATURES	M/E AIR COOLER IN	^oC	155.0	155.5	*0.32%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		50.0	46.6	*-6.80%*
	AIR COOLER COOLANT WATER		27.0	27.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		367.0	339.0	*-7.63%*
	EXH. GAS AFT. TURBINE		320.0	-	-
MISC.	T/C speed	RPM	15900.0	15889.0	-0.07%
	COMPRESSOR PRESSURE RATIO	-	-	2.65	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	12.260	-
	COMPRESSOR EFFICIENCY	-	-	0.829	-
	TURBINE EFFICIENCY	-	-	0.828	-
	AIR COOLER EFFICIENCY	-	0.8203	0.8475	*3.31%*
	IHP	kW	5921.0	5923.0	*0.03%*

M/E RPM: 108.87, LI POS.: 6.9			SHIP	CALC.	*ERROR (%)*
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.947	-
	BHP (mean value)	kW	-	5624	-
	SFOC	g/kW-h	-	170.0	-

Table 8. 13 Simulation results for 10th Feb. 2008

The Brake Horse Power was calculated as per 8.4. The reported values and estimations` results, are attached below.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	5921.0	76.9%	5624.0	73.0%	-5.02%
MEPIC Calculated	5474.0	71.1%			2.74%
SULZER Reported	5546.0	72.0%			1.41%
MEPIC Calculation (Regression)	5494.0	71.4%			2.37%
Acc. L.I.xRPM	6070.9	78.8%			-7.36%
Acc. Pme	5938.8	77.1%			-5.30%
Acc. Propeller Law	6203.8	80.6%			-9.35%
Average	5806.9	75.4%	5624.0	73.0%	-3.15%

Table 8. 14 BHP

According to the table 8.4, the following figures are attached.

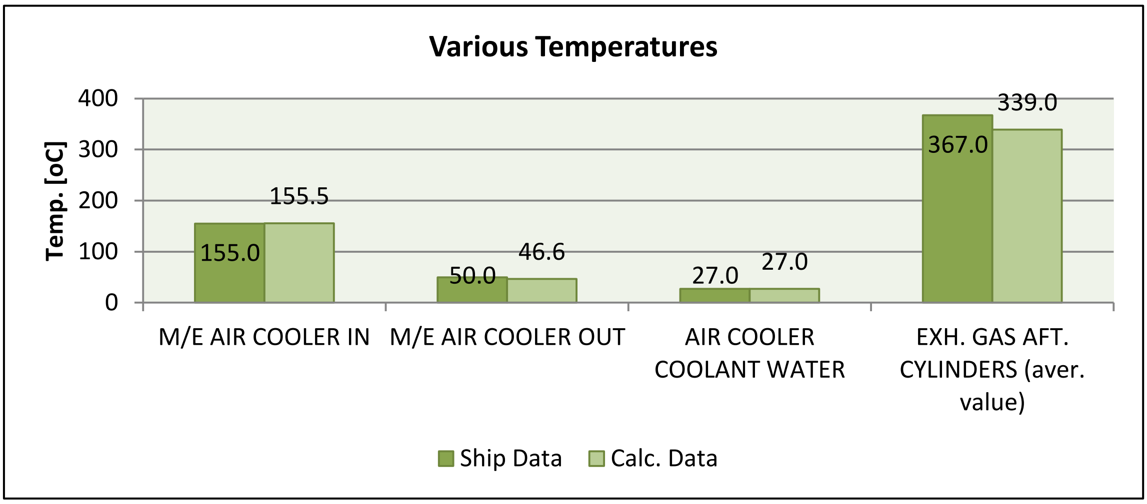


Fig. 8. 21 Various temperatures

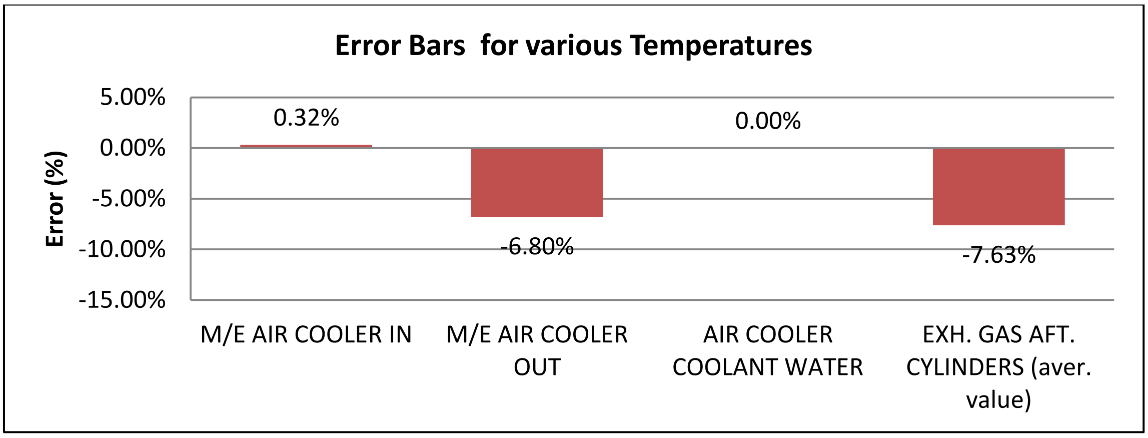


Fig. 8. 22 Error bars for various calculated temperatures

Comments:

- The reason of simulating the engine performance of 10th Feb 2008, is the existence of Indicated Horse Power measurements.
- Reported MEPIC power deviated from calculated MEPIC power by 6.8%. Moreover, a considerable deviation between the MEPIC reported and SULZER reported BHP, was noticed (Table 8.14). Apart from these deviations, MOTHER power calculation was closer to MEPIC calculation and to SULZER reported power with error less than 3%. MOTHER calculation is considered reliable because the IHP has been calculated correctly and because it was based on Shop Test figures. Thus, the engine load was at the range of 73% of MCR (Fig. 8.23).
- Regarding the air cooler efficiency, the following remarks could be made:
    - The air cooler efficiency calculation was based on Shop Test efficiency figures. For that reason MOTHER calculated higher efficiency (i.e. 0.8475) than the actual one (i.e. 0.8203).
    - Sea temperature affects the air cooler efficiency.
    - In practice, the scavenge air temperature is maintained between 48^{o}C and 50^{o}C, as per Wartsila instructions. Engine room crew is able to increase or decrease this temperature by altering the coolant water flow rate inside the air cooler through a valve. However, there were cases where the scavenge air temperature exceeds 50^{o}C (Chapter 8.5.4).
    - The combination of high scavenge air temperature and high pressure drop across the air cooler, indicates possible fouling of the air cooler. At this case, the pressure drop (i.e. 130mmWG) was not that high.
- A number of factors which were explained in Chapter 8.5.1, affects the exhaust gas temperature. Particularly, cylinder no.1 (Appendix V, Chapter V.2) continued to have high exhaust gas temperature (i.e. 393^{o}C), compared with the other cylinders and MOTHER`s output (i.e. 339$^{o}$C). Consequently, MOTHER calculations revealed a malfunction in cylinder no. 1. The question at this point is what had caused that problem. The answer is not easy to be given, because according to the engine`s history that malfunction was responsible for liner, piston rings, piston crown, failures during 2009 (Chapter 8.5.5 and 8.5.6). Manufacturer was also involved in providing its expertise

to find the failures` causes. Finally, the exact cause was never found, but the most possible cause was the increased wear inside the cylinders.

- According to Load Diagram (Fig. 8.23), the engine operating point is between the nominal propeller curve and the light propeller curve. Adverse weather conditions were responsible for shifting the operating point of January, 28th, 2008 (Fig. 8.15) from the light propeller curve, to the region between the nominal propeller curve and the light propeller curve. Furthermore, increased power demand could have been caused by increased hull and propeller fouling; but this possibility was eliminated because the load points for both performance reports (28-Jan-08 and 10-Feb-08) were situated below the nominal propeller curve.
- Figure 8.24 illustrates the calculated by MOTHER energy balance.
- The simulation model predicts satisfactorily the majority of the performance parameters. As far as, the compression pressure is concerned, the simulation model calculates a higher mean pressure which should have probably been the engine`s normal compression pressure.

The load diagram and energy balance of this case are illustrated at the Figures 8.23 and 8.24 respectively.

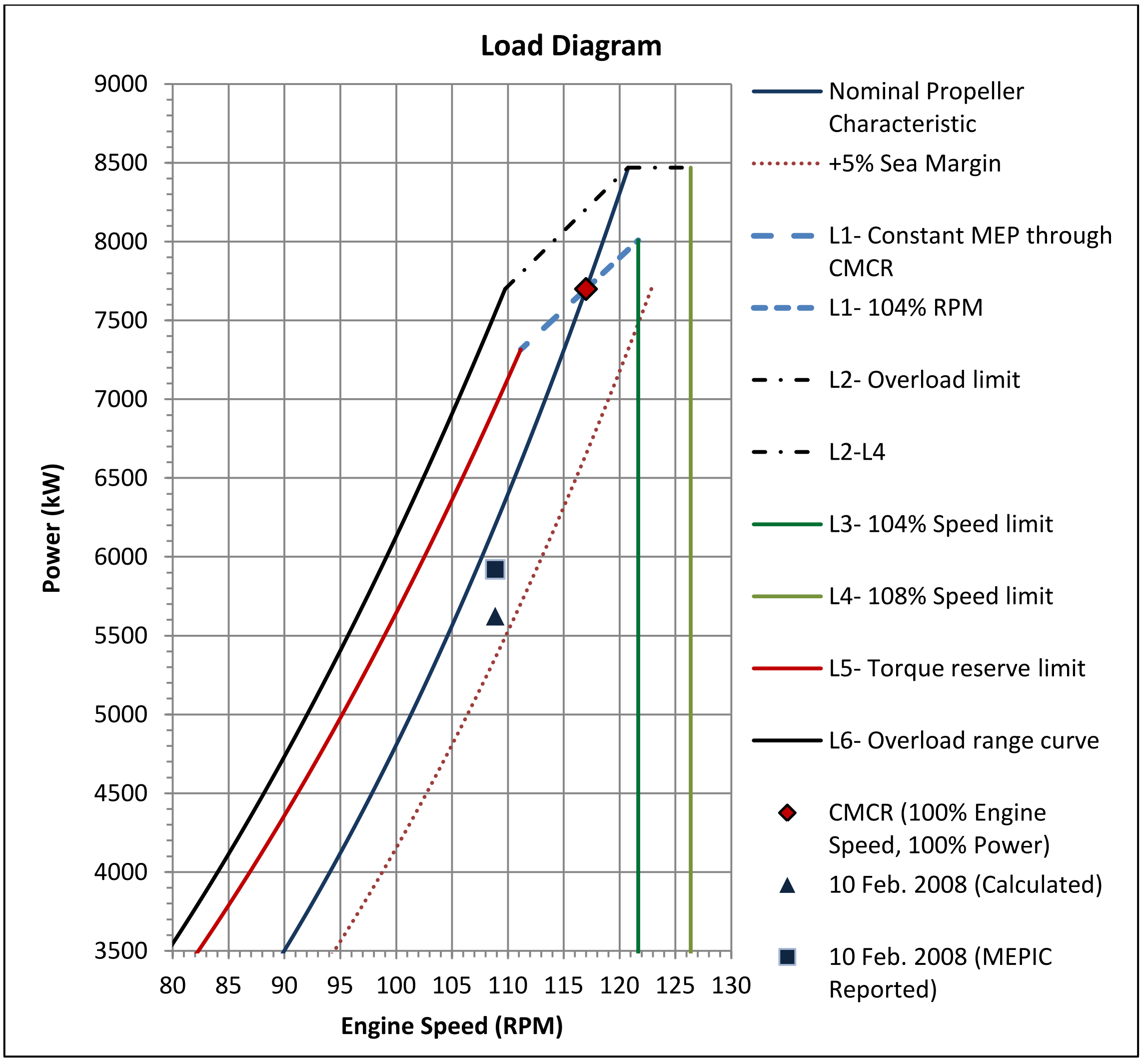


Fig. 8. 23 Load Diagram

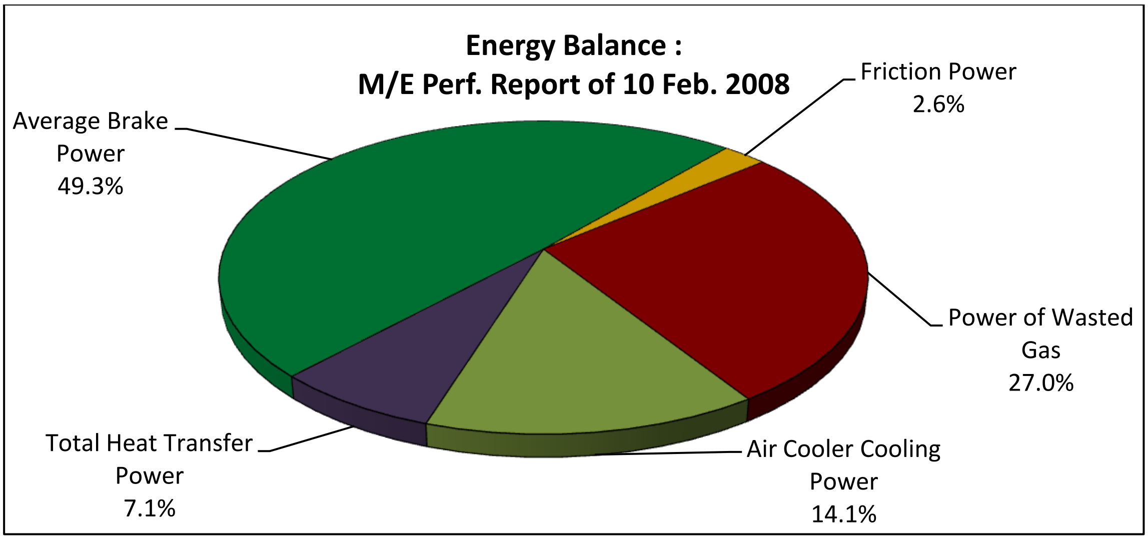


Fig. 8. 24 Energy Balance

Power	Energy Balance [kW]	Energy Balance [%]
Power of Wasted Gas	3074.0	27.0%
Air Cooler Cooling Power	1605.8	14.1%
Total Heat Transfer Power	805.4	7.1%
Average Brake Power	5624.0	49.3%
Friction Power	295.7	2.6%
Total Power	11404.8	100.0%

Table 8. 15 Energy Balance values

8.5.2.2. ISO corrections` results

This sub-section provides the results of the ISO correction methodology.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	30.0	**125.0**	127.7	135.6	135.4	131.3
Pmax 2	35.0	**120.0**	122.7	125.3	135.4	131.3
Pmax 3	35.0	**120.0**	122.7	125.3	135.4	131.3
Pmax 4	35.0	**120.0**	122.7	125.3	135.4	131.3
Pmax 5	30.0	**115.0**	117.7	125.3	135.4	131.3

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 6	40.0	**125.0**	127.7	125.3	135.4	131.3
Pmax Average	**34.2**	**120.8**	**123.6**	**127.1**	**135.4**	**131.3**
Pmax / $Pmax_{MCR}$	-	86%	88%	91%	97%	94%

Table 8. 16 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	4.2	**95.0**	97.7	91.2	95.2	91.0
Pcomp 2	-0.8	**85.0**	87.7	91.2	95.2	91.0
Pcomp 3	-0.8	**85.0**	87.7	91.2	95.2	91.0
Pcomp 4	-0.8	**85.0**	87.7	91.2	95.2	91.0
Pcomp 5	-5.8	**85.0**	87.7	91.2	95.2	91.0
Pcomp 6	4.2	**85.0**	87.7	91.2	95.2	91.0
Pcomp Average	**-**	**86.7**	**89.4**	**91.2**	**95.2**	**91.0**

Table 8. 17 Corrected Pcomp

Figures 8.16 & 8.17, correct the maximum and the compression pressure in order to be at the same conditions as at Shop Test.

The following comments could be made:

- The cylinder measured combustion pressures range between 115bar and 125bar.
- Combustion corrected cylinder pressures were lower that the reference values from Shop Test.
- Amongst the corrected combustion pressures, cylinder 5 had the lower value and the maximum deviation from the calculated values according to the following calculation methods: Pcomp, M/E speed and Power. Lower maximum pressure could indicate malfunctioning injector.
- Reported compression pressure for cylinder no 1 deviated 10bar from the other cylinders` compression pressure. Such a deviation is not normal and several factors may contribute to raise the compression pressure of individual cylinder; some of these factors could be the following:
    - The respective cylinder receives more fuel than the others.
    - The fuel pump timing has changed.
- According to the providing calculation methods from Shop Test figures, the corrected compression pressures were quite low, except for cylinder no 1.
- Calculated compression and maximum pressures according to main engine speed method, were overestimated, compared to the corrected mean values.
- These tables provided a safe comparison between the normal values during Shop Test and the values that were measured onboard. The engine operation and the fouling of

various parts was monitored in order to have a smooth and reliable operation, as per Technical Office`s recommendations.

- MOTHER calculations and ISO corrections conclude at the same results:
    - The engine`s mean compression pressure was lower than those simulated by MOTHER and those calculated by ISO methodology.
    - Measured combustion pressure should have been higher.

8.5.3. M/E Performance Report of 12th Mar. 2008

The performance report of 12/03/2008 is the third consecutive report where the Indicated Horse Power (IHP) has been reported. The current engine performance evaluation is associated with other two performance reports, i.e. 10 Feb 2008 and 17 May 2008, the latter due to the increased compression pressure.

8.5.3.1. Performance Data – results

Table 8.18 shows some major ship data.

Ship Data		
Ship Condition	LADEN	
Ship Speed	14.10 kts	
Prop. Apparent Slip	3.80%	
Wind Force / Direction	4	D
Swell Height / Direction	0.5m	H
Current speed/ Direction	1.0kts	H
Draught FOR/AFT	11.55m	11.60m
Displacement	56510 MT	

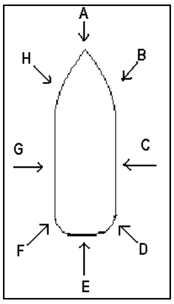


Table 8. 18 Major Ship data

Table 8.19 shows Main Engine related data that used in the simulation model.

M/E Data	
Engine Speed	109.2 RPM
Brake Power Reported	5997 kW
Indicated Power Reported	6004 kW
L.I. Pos.	6.9
Fuel Consumption	27.86 MT/24h

M/E Data		
T/C Speed	16000 RPM	
Scavenge Air Pressure (relative)	1.800 bar	
Scavenge Air Pressure (total)	2.812 bar	
Air temp.: E/R / Bef. Blower	42°C	42°C
Air cooler temp. Bef. / After	152°C	50°C
Press. Drop Air Cooler	110 mmWG	0.010787 bar
Sea Water Temp.	29°C	
Air Cooler Efficiency	82.93%	

Table 8. 19 M/E data

The calculated cylinder pressures are illustrated at the following figures.

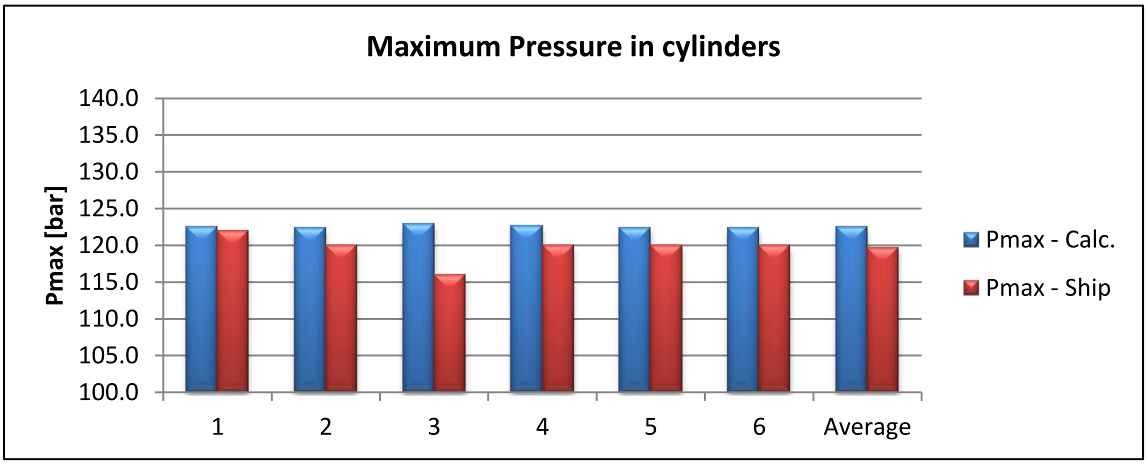


Fig. 8. 25 Pmax, Ship reported vs. "MOTHER" calculated

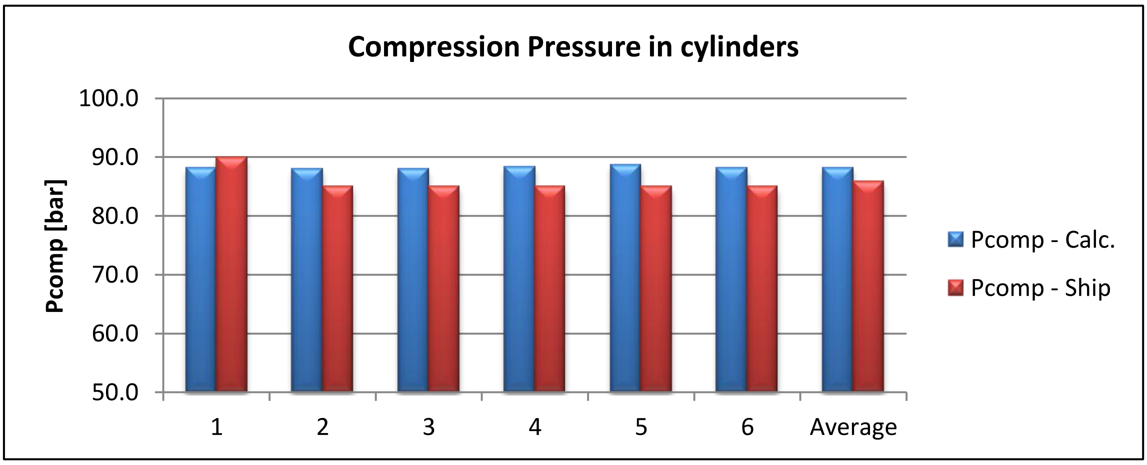


Fig. 8. 26 Pcomp, Ship reported vs. "MOTHER" calculated

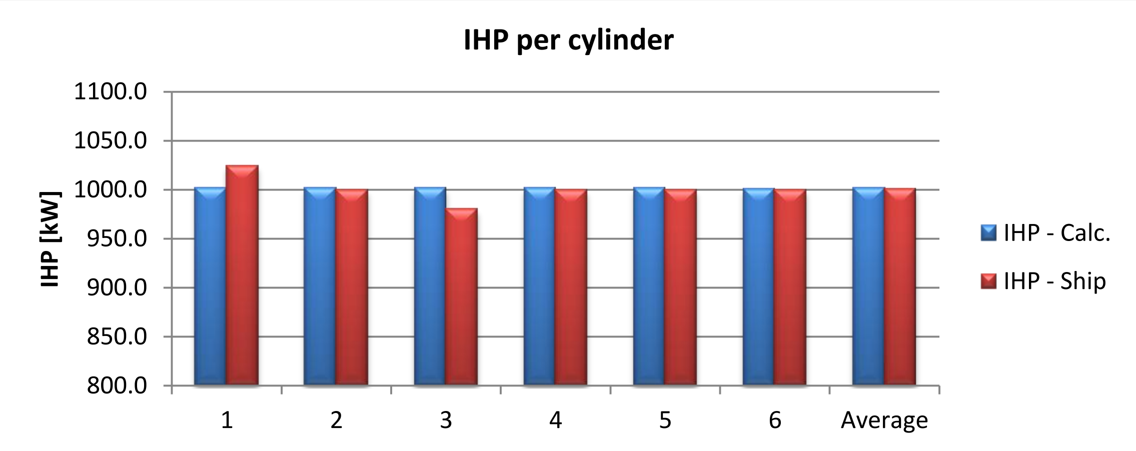


Fig. 8. 27 IHP per cylinder, Ship reported vs. "MOTHER" calculated

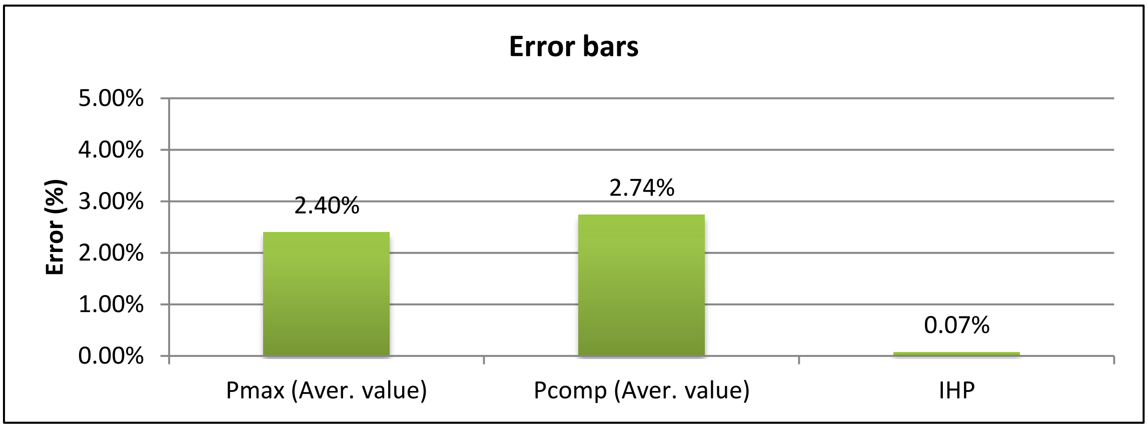


Fig. 8. 28 Error Bars for average values

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	88.1	88.0	87.9	88.3	88.7	88.1	88.18
Pcomp – Ship		90.0	85.0	85.0	85.0	85.0	85.0	85.83
Error	%	*-2.00%*	*3.76%*	*3.65%*	*3.41%*	*3.65%*	*3.76%*	*2.66%*
Pmax – Calc.	bar	122.5	122.4	122.9	122.6	122.4	122.4	122.53
Pmax – Ship		122.0	120.0	116.0	120.0	120.0	120.0	119.67
Error	%	*0.41%*	*2.00%*	*5.95%*	*2.17%*	*2.00%*	*2.00%*	*2.40%*
IHP – Calc.	bar	1001.9	1001.2	1001.2	1001.5	1001.6	1001.0	1001.40
IHP – Ship		1024.0	1000.0	980.0	1000.0	1000.0	1000.0	1000.67
Error	%	*-2.16%*	*0.12%*	*2.16%*	*0.15%*	*0.16%*	*0.10%*	0.07%

Table 8. 20 Compression, maximum pressure and Indicated Horse Power per cylinder

The following comments could be made:

- The Indicated power was predicted satisfactorily with an error of 0.07%.
- Weather conditions were better than in previous performance report.
- Although combustion and compression pressures were predicted with an error less than 3%, their values were higher than the ship`s reported values.
- Cylinder`s no 1 reported compression pressure was higher than the others by 5bar. This particular cylinder had increased compression pressure compared with the mean compression pressure, as reported in 10 Feb 2008 (Chapter 8.5.2) and in 28 Jan 2008 (Chapter 8.5.1). High compression pressure in individual cylinders could be owed to the following factors:
    - The compression volume has changed after maintenance work by chance. This could happen in case wrong compression shim has been fitted to the cylinder after the maintenance work finished.
    - A new piston is fitted to the engine. In case the new piston does not comply with the manufacturers` specifications (dimensions, geometry), it will affect the compression pressure. Ship operators sometimes prefer to supply their main engine spare parts (i.e. piston, piston rings, liner, exhaust valve, injectors etc.) instead of the original manufacturer, from other manufacturer, in a more competitive cost but that decision sometimes jeopardizes the future engine operation. Regarding the 6RTA48-T, it is clear from the ship`s managing company that the original spare parts are used in M/E overhauls under the manufacturer recommendations. Moreover, the possibility of a faulty original spare part should not be neglected in case of a serious problem.
    - Weather could potentially affect the compression pressure. In case of rough weather conditions, the measurements of performance report are not very accurate. At the time when the ship faces a wave crest in front of it, the engine`s load raises in order to maintain the speed. A measurement during this time (the ship “ascending” the wave`s crest), will give higher compression pressures compared with the case when the ship faces a valley. Regarding the report of 12th Mar 2008, the sea conditions were favorable for performing measurements of the various main engine performance data.

  Consequently, the information contained in mail exchanges between Headquarters and Ship, and also the available performance data, cannot justify the high compression pressure in cylinder no 1. Anyway, the influence of weather conditions is excluded.
- Cylinder no 3 had the lower combustion pressure (i.e. 116bar), as reported in M/E Performance Report. The lower combustion pressure, as mentioned above, could indicate injector malfunction.
- Fuel quality might have affected the measured combustion pressures. According to fuel analysis, the fuel burnt in the engine had CCAI value equal to 852. Generally, high CCAI values reflect low ignitability of the fuel; consequently the injection should be made earlier in crank angle degrees in order to reach normal combustion pressures. Depending on the fuel quality, the injection timing is adjusted through the FQS position [13]. The FQS position on March 12th, 2008, had been set at +0.5deg, resulting in lower

combustion pressures than those simulated by MOTHER. Thus, the improper FQS position, could have caused the low combustion pressure in cylinders no 2, 4, 5 and 6.

- Office technical staff had advised the ship`s Chief engineer to arrange overhauling of cylinder no 5.
- The injectors of cylinders 4, 5 and 6 had been replaced before the 12th March, 2008, according to mail exchanges (Chapter 8.5.2.1). Recorded pressures for these cylinders showed that the combustion pressures were improved compared to those pressures reported in 10 February, 2008. The impact of injectors` replacement was reflected by the equalized measurements for these particular pressures for cylinders 4, 5 and 6.
- It is noted that the measured compression pressures of individual cylinders for performance reports of 10 February and 12 March 2008, varied between 85bar and 90bar (except for cylinder no 1 with measured compression pressure at 95bar on February 10th, 2008), whereas on May 17th, 2008 (Chapter 8.5.4), the measured compression pressures exceeded the 95bar. The engine`s load for the aforementioned three cases was at the range of 72.5-76%, according to MOTHER calculations. On the other hand, MOTHER simulations resulted in consistent compression pressures at the range of 86bar for each case.

Table 8.21, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 109.2, LI POS.: 6.9			**SHIP**	**CALC.**	***ERROR (%)***
PRESSURES	SCAVENGING AIR PR.	bar	2.81	2.68	*-4.69%*
	Pmax (mean value)	bar	119.67	122.53	*2.40%*
	Pcomp (mean value)	bar	85.83	88.18	*2.74%*
	IMEP	bar	-	15.20	-
	BMEP	bar	-	14.43	-
	PRESSURE DROP IN AIR COOLER	bar	0.01079	0.01087	*0.74%*
	PRESSURE IN EXH. RECEIVER	bar	-	2.40	-
TEMPERATURES	M/E AIR COOLER IN	^oC	152.0	161.1	*5.99%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)	^oC	50.0	49.2	*-1.60%*
	AIR COOLER COOLANT WATER	^oC	29.0	29.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)	^oC	368.0	350.1	*-4.86%*
	EXH. GAS AFT. TURBINE	^oC	333.0	-	-
MISC.	T/C speed	RPM	16000.0	16015.0	*0.09%*
	COMPRESSOR PRESSURE RATIO	-	-	2.65	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	12.27	-
	COMPRESSOR EFFICIENCY	-	-	0.830	-
	TURBINE EFFICIENCY	-	-	0.829	-
	AIR COOLER EFFICIENCY	-	0.8293	0.8471	*2.15%*

M/E RPM: 109.2, LI POS.: 6.9			SHIP	CALC.	*ERROR (%)*
	IHP	kW	6004.0	6008.3	*0.70%*
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.949	-
	BHP (mean value)	kW	-	5707.8	-
	SFOC	g/kW-h	-	170.1	-

Table 8. 21 Simulation results for 12th Mar. 2008

The Brake Horse Power was calculated as per Chapter 8.4. The reported values and estimations` results, are attached below. The engine load (% of MCR) is also attached for each power estimation.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	5997.0	77.9%	5707.8	74.1%	-4.82%
MEPIC Calculated	5504.0	71.5%			3.70%
SULZER Reported	5624.0	73.0%			1.49%
MEPIC Calculation (Regression)	5524.0	71.7%			3.33%
Acc. L.I.xRPM	6094.4	79.1%			-6.34%
Acc. Pme	5956.8	77.4%			-4.18%
Acc. Propeller Law	6260.4	81.3%			-8.83%
Average	5851.5	76.0%	5707.8	74.1%	-2.46%

Table 8. 22 BHP

According to table 8.23, the following figures are attached.

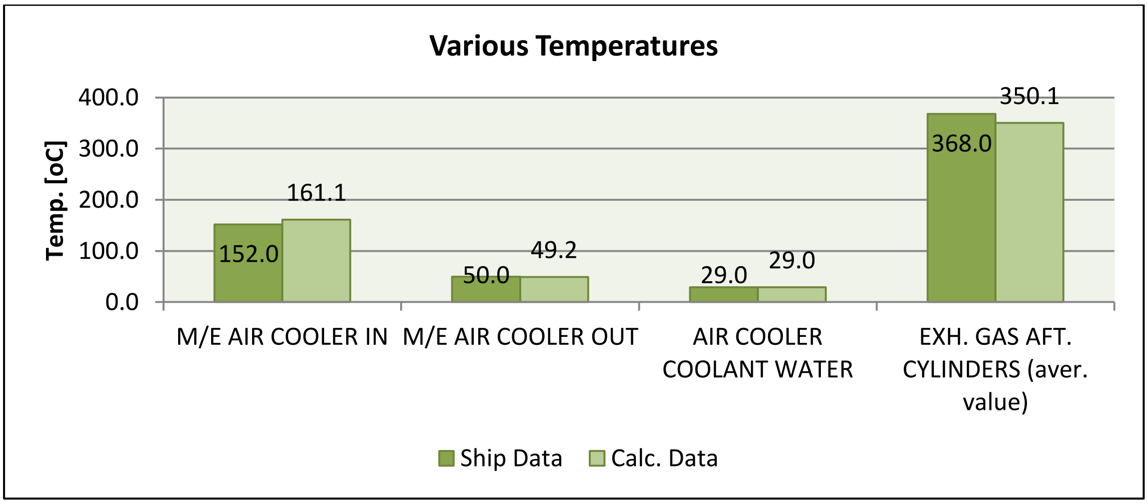


Fig. 8. 29 Various temperatures

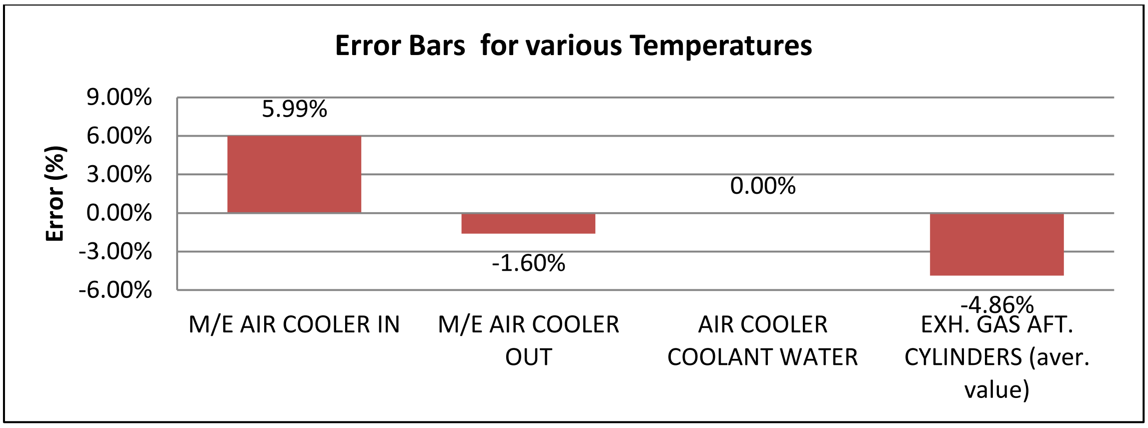


Fig. 8. 30 Error bars for various calculated temperatures

Comments:

- The reason of performing the simulation of 12th Mar 2008, is the existence of IHP data and the assessment of the engine performance in connection with the previous reports of 28th Jan and 10th Feb 2008.
- Deviations for scavenge air pressure, pressure loss coefficient, air temperature at the air cooler inlet, are shown in Table 8.21.
- The air cooler efficiency calculation error is less than 3%.
- Exhaust gas temperature is difficult to be predicted accurately as there are many factors that affect it. However, the calculation error is 4.86%. Some factors that affect the exhaust gas temperature of individual cylinders, are the following:
    - Engine load
    - Injection nozzles` condition
    - Charge air receiver`s condition (dirty or damaged)
    - Scavenge ports` condition
    - Exhaust valve`s condition
    - Fuel pump timing
    - Fuel cams arrangement
- Cylinder`s no. 1 reported exhaust gas temperature (i.e. 390$^{o}$C, Appendix V, V.3), was higher not only than the other cylinders` temperatures but the mean temperature calculated by MOTHER. Furthermore, the compression pressure remained higher than those predicted by the simulation model (Table 8.20). Consequently, by considering also the history of high compression pressure and high exhaust gas temperature (Chapter 8.5.1 7 & 8.5.2), it is concluded that MOTHER confirmed the already mentioned problem with cylinder no. 1 and moreover, the high exhaust gas temperature was related with the relatively high compression pressure in the cylinder.
- Regarding the engine brake power output calculation refer to Table 8.22.
- MOTHER predicted power converged better with the SULZER reported power (1.41% error) and MEPIC calculated power with regression (2.37% error).
- MEPIC reported and SULZER reported power, deviated considerably the one from the other.

- The calculated by MOTHER engine load is 74.1%; the load diagram is illustrated at Fig. 8.31.
- According to Fig. 8.31, the operating point is located between the nominal propeller curve and the light propeller curve. Consequently, the engine had reserve power to be used in case the propeller needed more torque for the same speed.
- According to MOTHER calculated energy balance, the power percentages are within the normal range for diesel engines.

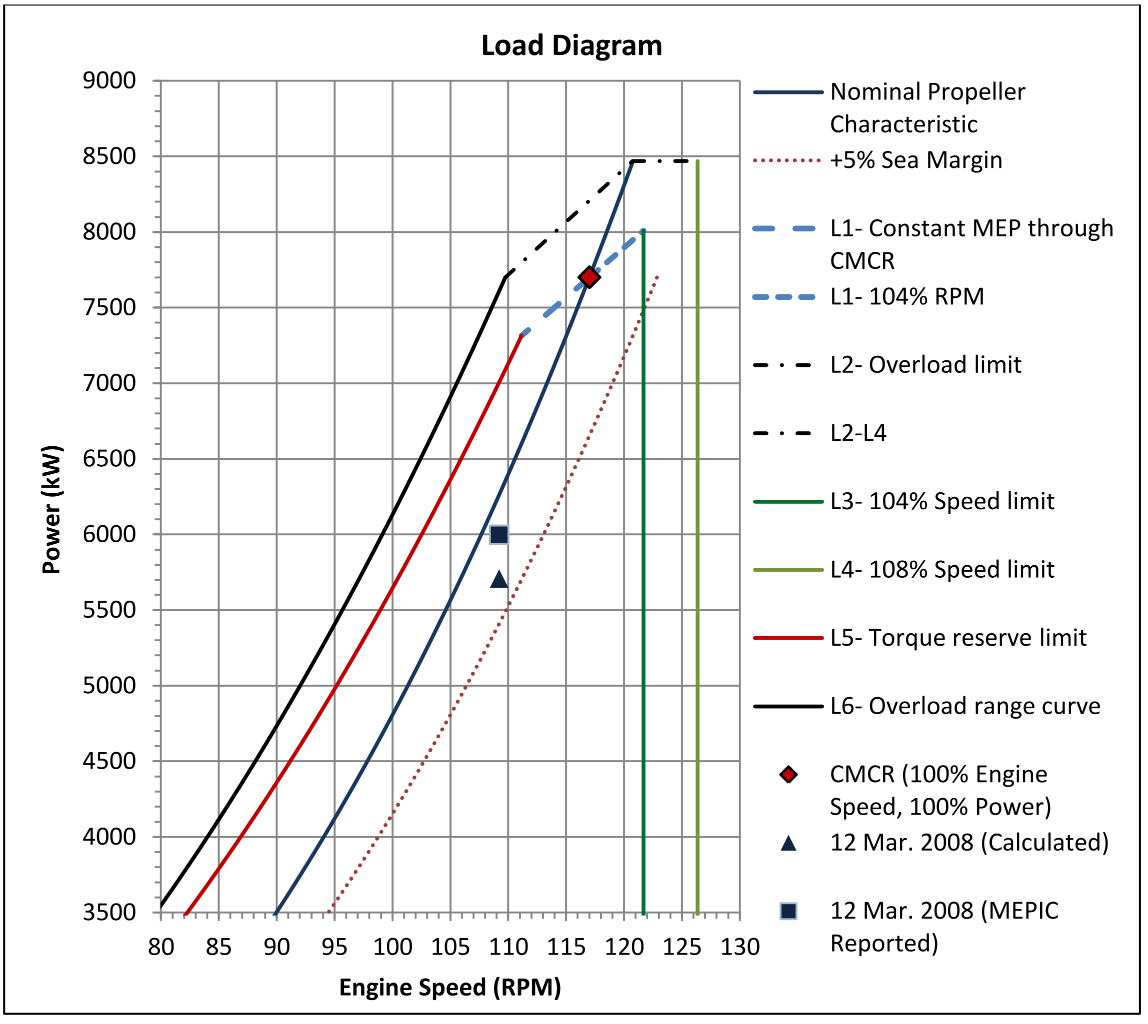


Fig. 8. 31 Load Diagram

The energy balance of this case is illustrated at the figure below.

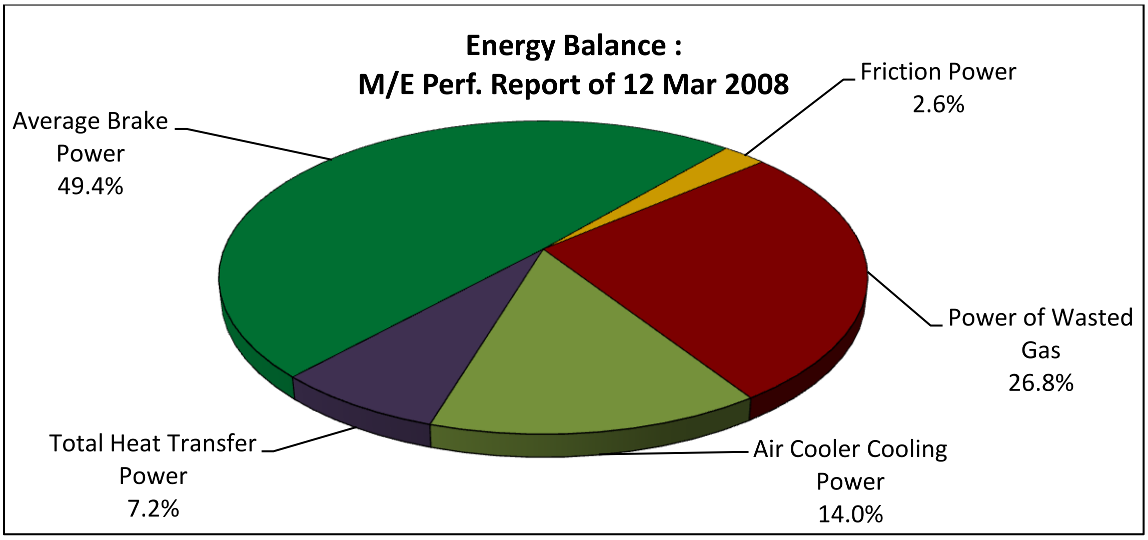


Fig. 8. 32 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	3097.1	26.8%
Air Cooler Cooling Power	1622.6	14.0%
Total Heat Transfer Power	829.4	7.2%
Average Brake Power	5707.8	49.4%
Friction Power	297.4	2.6%
Total Power	11554.3	100.0%

Table 8. 23 Energy Balance values

8.5.3.2. ISO corrections` results

Tables 8.24 and 8.24, below, include the ISO corrections results.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	32.0	**122.0**	125.9	130.7	136.2	132.2
Pmax 2	35.0	**120.0**	123.9	125.3	136.2	132.2
Pmax 3	31.0	**116.0**	119.9	125.3	136.2	132.2
Pmax 4	35.0	**120.0**	123.9	125.3	136.2	132.2

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 5	35.0	**120.0**	123.9	125.3	136.2	132.2
Pmax 6	35.0	**120.0**	123.9	125.3	136.2	132.2
Pmax Average	**33.8**	**119.7**	**123.6**	**126.2**	**136.2**	**132.2**
Pmax / $Pmax_{MCR}$	-	86%	88%	90%	97%	95%

Table 8. 24 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	2.3	**90.0**	93.9	91.2	96.0	91.9
Pcomp 2	0.3	**85.0**	88.9	91.2	96.0	91.9
Pcomp 3	-3.7	**85.0**	88.9	91.2	96.0	91.9
Pcomp 4	0.3	**85.0**	88.9	91.2	96.0	91.9
Pcomp 5	0.3	**85.0**	88.9	91.2	96.0	91.9
Pcomp 6	0.3	**85.0**	88.9	91.2	96.0	91.9
Pcomp Average	**-**	**85.8**	**89.8**	**91.2**	**96.0**	**91.9**

Table 8. 25 Corrected Pcomp

Comments:

- Corrected average combustion pressure was lower than the average reference values calculated from Shop Test (Table 8.24). However, according to Pcomp calculation, it seems that the lower corrected mean Pmax, lied within the acceptable pressure difference of 3bar.
- Cylinder no. 3 had the lower Pmax, which could indicate possible fault injectors.
- According Pscav calculation, average corrected compression pressure was normal. On the other hand, the lower by 1.4bar mean corrected compression pressure from the ISO calculated pressure, shows that the normal compression pressure might have been somewhat higher than measured.
- Although corrected compression pressure was slightly lower than the calculated pressures (Table 8.25), the engine`s operation seems to be balanced, compared to the Perf. Report of 10th Feb. 2008 (see 8.5.2).
- Cylinder no 1 corrected compression pressure exceeds the reference compression pressure according Pscav and Power (Table 8.25) calculation.
- According to ISO corrections, there is one conclusion:
    - Actual combustion pressure and compression pressure were slightly higher than the measured values.
- Consequently, MOTHER and ISO method coincide to the same result regarding the cylinder pressures:

- M/E was operated with slightly lower cylinder combustion pressure and compression pressure.
- Cylinder`s no. 1, compression pressure was high.

8.5.4. M/E Performance Report of 17 May 2008

This “M/E Performance Report” has been selected for evaluation due to the high compression pressures. It is remarkable that there are similarities with the report of 12th Mar. 2008 (Chapter 8.5.3), such as the barometric pressure, the ship`s draught and the correspondent displacement. Yet, the reported average compression pressure between the 12th Mar. 2008 and 17th May 2008, yields about 10bar. The analysis of the various performance parameters follows below.

8.5.4.1. Performance Data – results

Ship Data		
Ship Condition	LADEN	
Ship Speed	13.60 kts	
Prop. Apparent Slip	8.43%	
Wind Force / Direction	5	F
Swell Height / Direction	2m	F
Current speed/ Direction	-0.1kts	F
Draught FOR/AFT	11.70m	11.93m
Displacement	56590 MT	

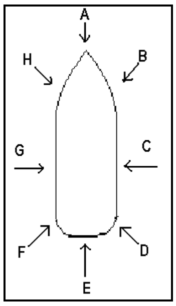


Table 8. 26 Major Ship data

M/E Data	
Engine Speed	109.98 RPM
Power Reported	5408 kW
L.I. Pos.	6.9
Fuel Consumption	28.03 MT/24h
T/C Speed	15900 RPM
Scavenge Air Pressure (relative)	1.80 bar
Scavenge Air Pressure (total)	2.81 bar

M/E Data		
Air temp.: E/R / Bef. Blower	45 °C	43 °C
Air cooler temp. Bef. / After	158 °C	56 °C
Press. Drop Air Cooler	111 mmWG	0.0108851 bar
Sea Water Temp.	28 °C	

Table 8. 27 M/E data

The calculated cylinder pressures are illustrated at the following figures.

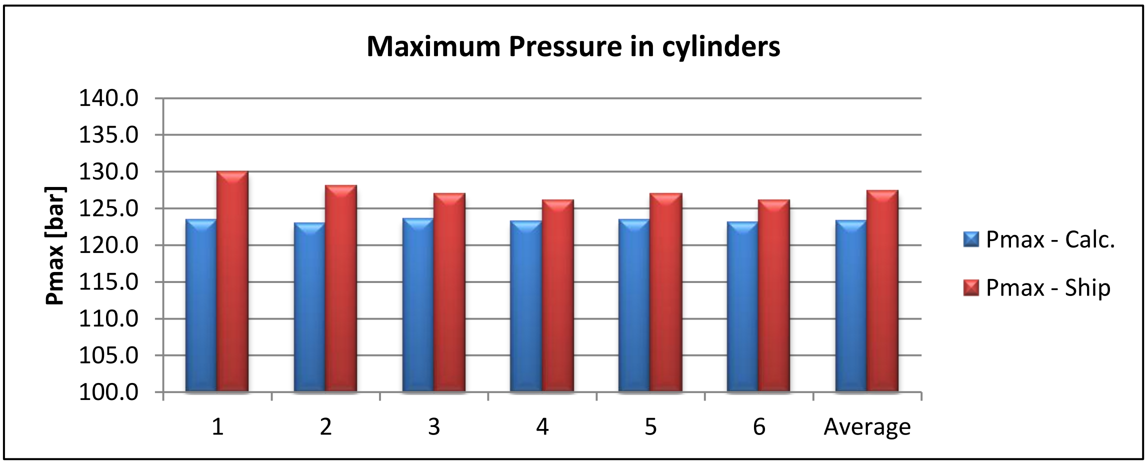


Fig. 8. 33 Pmax, Ship reported vs. "MOTHER" calculated

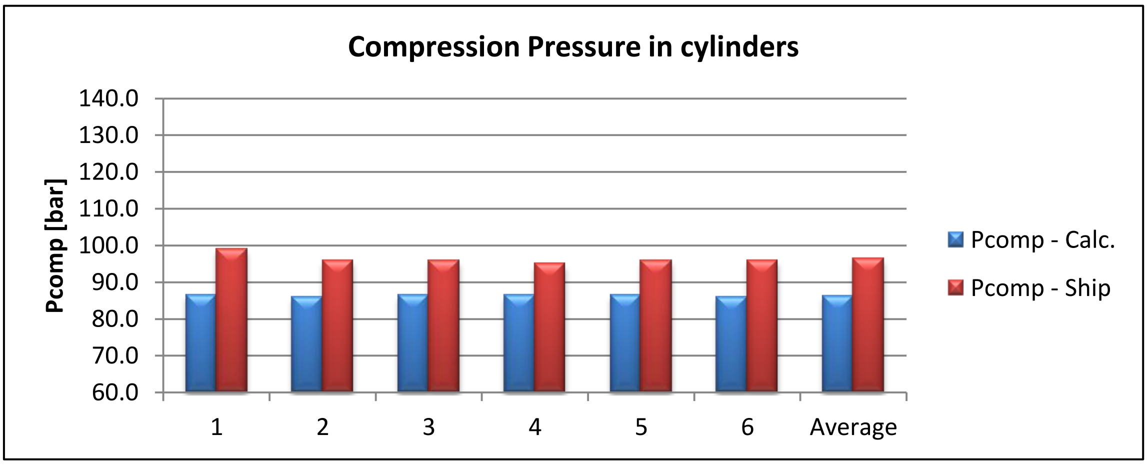


Fig. 8. 34 Pcomp, Ship reported vs. "MOTHER" calculated

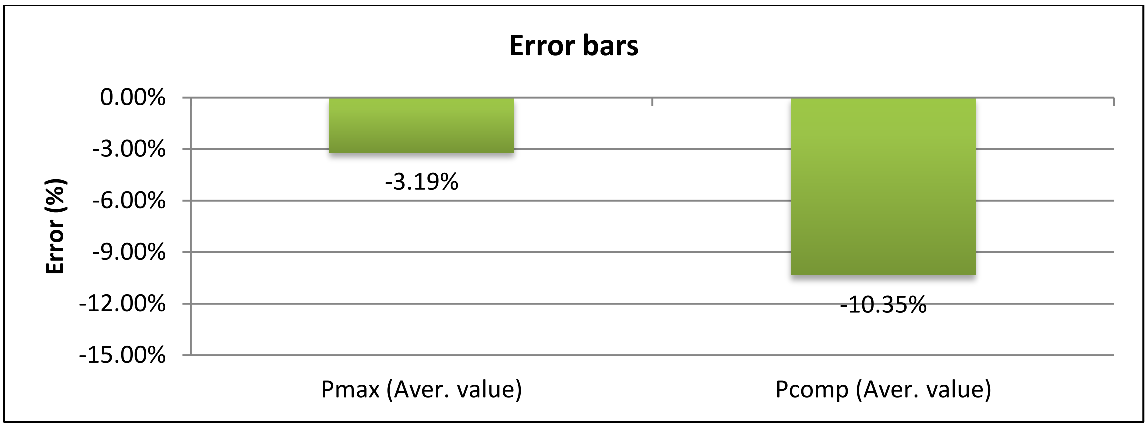


Fig. 8. 35 Error Bars for average values

The table below shows the figures` 8.33 & 8.34 data.

Value / Cyl.		1	2	3	4	5	6	Aver.
Pcomp – Calc.	bar	86.5	86.1	86.6	86.4	86.5	86.1	86.37
Pcomp – Ship		99.0	96.0	96.0	95.0	96.0	96.0	96.33
Error	%	*-12.63%*	*-10.31%*	*-9.79%*	*-9.05%*	*-9.90%*	*-10.31%*	*-10.35%*
Pmax – Calc.	bar	123.5	122.9	123.6	123.2	123.4	123.0	123.27
Pmax – Ship		130.0	128.0	127.0	126.0	127.0	126.0	127.33
Error	%	*-5.00%*	*-3.98%*	*-2.68%*	*-2.22%*	*-2.83%*	*-2.38%*	*-3.19%*

Table 8. 28 Compression and maximum pressure per cylinder

The following comments could be made:

- Average combustion pressure and compression pressure have not been predicted, as expected, with an error less than 3%.
- According to MOTHER simulation results, all cylinders had considerably higher than normal compression pressures. That high compression pressures indicates that the cylinders were overloaded for the correspondent engine speed.
- The engine performance of May 17th, 2008, was completely different from those on 12th March 2008. The common values amongst the aforementioned reports, were the barometric pressure (i.e. 1012mmWG), the ship`s draught and the correspondent displacement. The engine speed and turbocharger`s speed, as long as the scavenge air pressure, deviated less than 1% from the relevant values of March 12th, 2008.
- Apart from the mentioned causes of increased compression pressure (compression volume change, weather contribution) in Chapter 8.5.3, the following could be added (in this case, all cylinders had increased Pcomp, compared to case of Chapter 8.5.3 where only cylinder no 1 had increased Pcomp):
    - Increased scavenge air pressure could have been caused by increased compressor speed. Such a possibility is excluded because both turbocharger speed and scavenge air pressure were normal.

  - Increased scavenge air temperature. Such a possibility is applicable to this case and it could be a potential cause for the increased Pcomp, because the measured scavenging temperature was 56°C, when the acceptable range lies between 48°C and 50°C.
- Given that the combustion pressure is affected by the injection angle of the fuel and the temperature inside the cylinder, it is considered that either the FQS position advanced the injection, either the high scavenge air pressure affected the combustion pressure, or both had happened. According to WARTSILA, if FQS input valve is advanced by 1deg, Pmax rises by about 4 to 5bar.
- Cylinder no 1, had the most increased values of combustion and compression pressures. This particular cylinder had constantly high compression pressure, as reported from performance reports from 28 Jan. 2008 up to 17 May, 2008. Technical staff from Headquarters requested again the Chief Engineer to evaluate the condition of cylinder no 1 by superficial inspection (17 May 2008).

Table 8.29, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 109.2, LI POS.: 6.9			**SHIP**	**CALC.**	***ERROR (%)***
PRESSURES	SCAVENGING AIR PR.	bar	2.812	2.645	*-5.94%*
	Pmax (mean value)		127.33	123.27	*-3.19%*
	Pcomp (mean value)		96.33	86.37	*-10.35%*
	IMEP		-	15.11	-
	BMEP		-	14.35	-
	PRESSURE DROP IN AIR COOLER		0.01089	0.01094	*0.52%*
	PRESSURE IN EXH. RECEIVER		-	2.37	-
TEMPERATURES	M/E AIR COOLER IN	°C	158.0	161.0	*1.90%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		56.0	48.4	*-13.57%*
	AIR COOLER COOLANT WATER		28.0	28.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		384.2	355	*-7.59%*
	EXH. GAS AFT. TURBINE		335.0	-	-
MISC.	T/C speed	RPM	15900	15932	*0.20%*
	COMPRESSOR PRESSURE RATIO	-	-	2.63	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	12.16	-
	COMPRESSOR EFFICIENCY	-	-	0.829	-
	TURBINE EFFICIENCY	-	-	0.829	-
	AIR COOLER EFFICIENCY	-	0.7846	0.8466	*7.90%*
	IHP	kW	-	6014	-
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.95	-
	BHP (mean value)	kW	-	5712.9	-
	SFOC	g/kW-h	-	169.8	-

Table 8. 29 Simulation results for 17th May. 2008

The Brake Horse Power calculations results are included in Table 8.30.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	5402.0	70.2%	5712.9	74.2%	5.76%
MEPIC Calculated	5426.0	70.5%			5.29%
SULZER Reported	5402.0	70.2%			5.76%
MEPIC Calculation (Regression)	5437.0	70.6%			5.07%
Acc. L.I.xRPM	6037.8	78.4%			-5.38%
Acc. Pme	5865.0	76.2%			-2.59%
Acc. Propeller Law	6399.0	83.1%			-10.72%
Average	5709.8	74.2%	5712.9	74.2%	0.05%

Table 8. 30 BHP

According to Table 8.29, the following figures are attached.

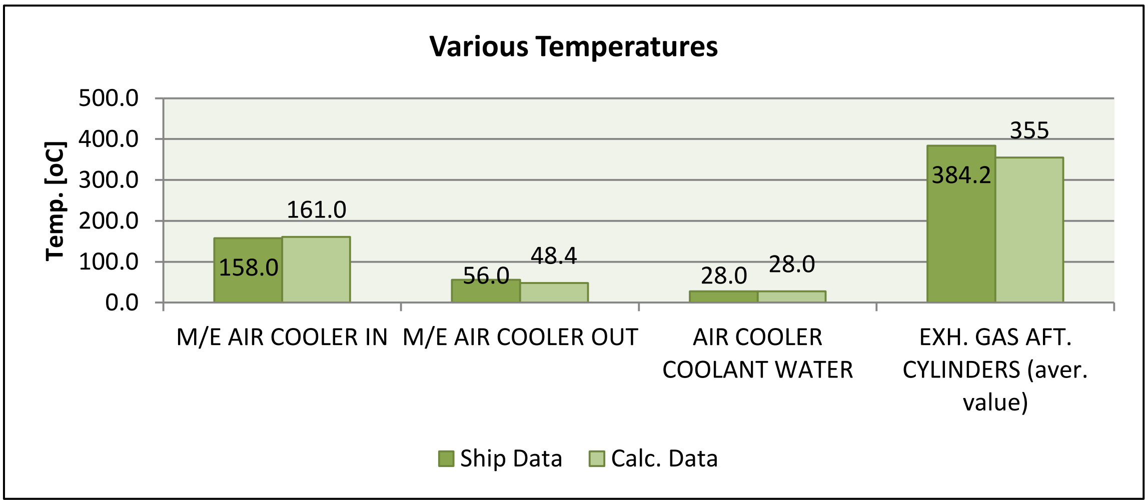


Fig. 8. 36 Various temperatures

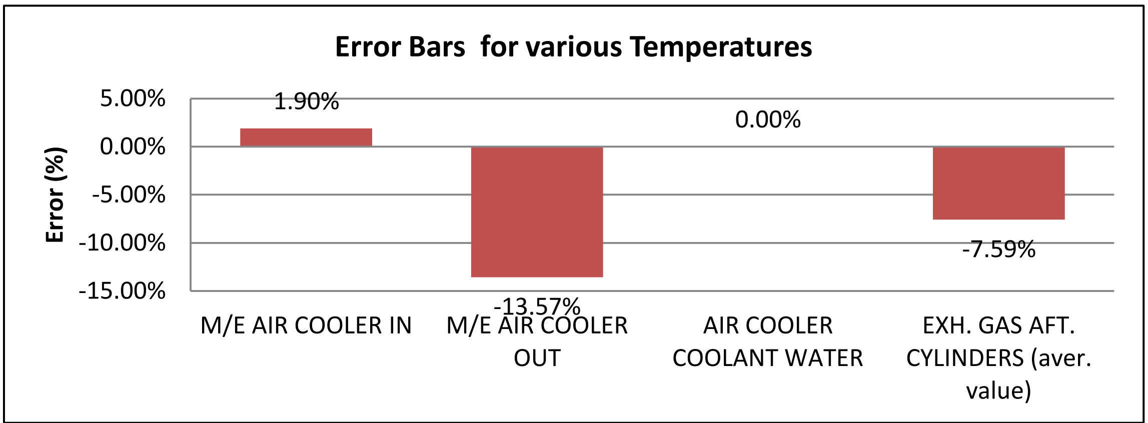


Fig. 8. 37 Error bars for various calculated temperatures

Comments:

- The reason of simulating the engine operation at 17th May 2008, is the evaluation of its performance which varied considerably from those examined at the previous performance reports (i.e. Chapter 8.5.1, 8.5.2 & 8.5.3). The barometric pressures of 17th May and 12th Mar 2008, were the same and equal to 1.012bar.
- According to the available mail exchanges between Headquarters and Ship, the following works had been accomplished before the 17th May 2008:
  - Overhauling of cylinder no 5 and inspection through scavenge ports (March 2008).
  - Replacement of cylinder`s no 5 lubrication quill (March 2008).
  - Replacement of injectors (two injectors for each cylinder) of cylinders no 4, 5 and 6 (March 2008).
- As mentioned above, measured compression pressures were higher than in the calculated by MOTHER.
- Another issue that arose from the performance data, was the high air temperature after the air cooler (i.e. 56^oC). WARTSILA recommends to maintain this temperature between 48^oC and 50 ^oC. The high temperature could have caused by the following factors:
  - Air cooler fouling from the sea side
  - Cooling water valve not fully opened
- According to Chief Engineer`s message to Headquarters (18/05/2008), air cooler running hours -sea side- since last cleaning, were 3800 (hours), whereas it should have been cleaned in 3000hours intervals, as per manufacturer advice. Furthermore, he stated that they had adjusted the scavenge air temperature by turning the sea water valve at fully opened position. Consequently, the aforementioned factors were both contributed to the high air temperature and the MOTHER calculation error is justified.
- MOTHER calculated the scavenge air temperature that should have been measured in order to achieve the reference air cooler efficiency as measured at Shop Test. In fact, the actual air cooler efficiency was 0.7846, whereas MOTHER calculated 0.8466.
- Pressure loss coefficient was selected in order to achieve convergence between calculated and measured pressure drop across the air cooler.
- Even though the calculated turbocharger speed was 0.20% higher than the measured speed, scavenging air pressure was calculated with error -5.94%.
- According to Appendix V, Chapter V.4, the exhaust gas temperatures was increased in cylinders no. 1, 5, and 6. Cylinder no. 1 continued to face high compression pressures, accompanied with high exhaust gas temperatures, as well. For first time other units faced that high temperatures. According to the engine`s history, these phenomenally unrelated incidents, had triggered a series of failures during 2009, which are explained in Chapters 8.5.5 and 8.5.6.
- Besides the high exhaust gas temperature, cylinder no. 1 faced increased piston underside temperature (i.e. 72^oC, normal values lie at the range of 65^oC). The aforementioned irregularities had been noticed by the Headquarters` Technical staff, who had requested the Chief Engineer to inspect the engine`s condition by checking the piston rings, the liner, the piston crown and piston skirt.

- Even though IHP has not been reported, it was considered that it will be close to those reported in 12 Mar. 2008 (i.e. 6004kW), because the engine`s speed and load in both reports were adjacent.
- Table 8.30, shows the reported and calculated Brake Power as described in Chapter 8.4. MOTHER calculated engine load was 74.2% of MCR, whereas the Ship reported was 70.2%. Power calculations according to LIxRPM and Pme, provided an estimation about the range of the engine output, because they have not been corrected for the fuel quality, the exhaust gas temperature etc. From the other hand, the power calculation according to propeller law, contributes in assessing the engine and propeller load.
- Fig. 8.38, illustrates the load diagram and the operating point of 17 May 2008, as calculated by MOTHER. Furthermore, the reported (MEPIC) operating point is illustrated as well.
- Although the engine faced high cylinder compression pressures, the engine load was lower than the nominal propeller curve`s load, which means that the engine had reserve power which could have been potentially used in case the propeller required to absorb more power at the same speed. From the other hand, MEPIC reported operating point was below the "5% propeller margin" curve.
- Fig. 8.39, illustrates the energy balance. The power percentages lie between the acceptable limits for diesel engines.

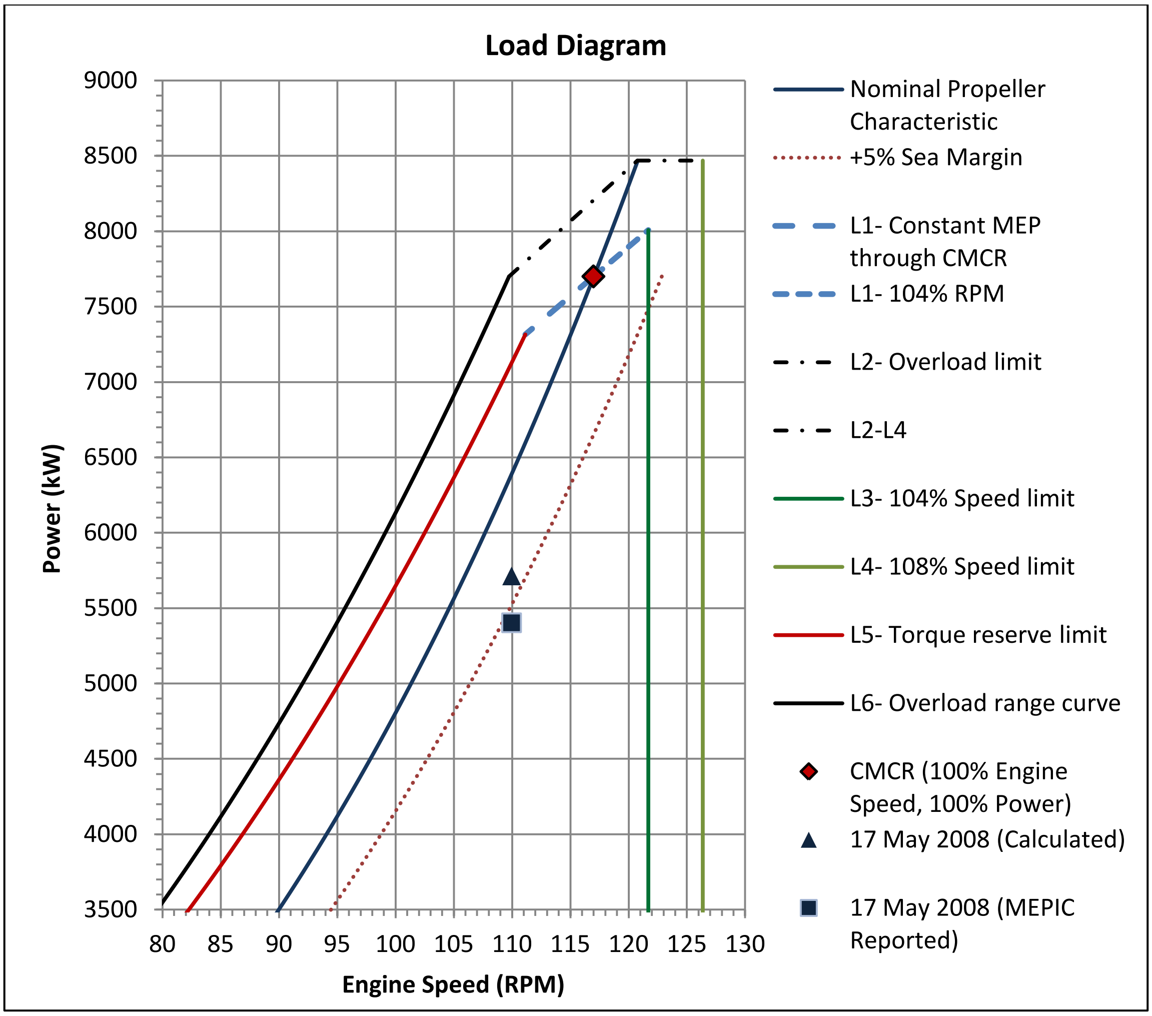


Fig. 8. 38 Load Diagram

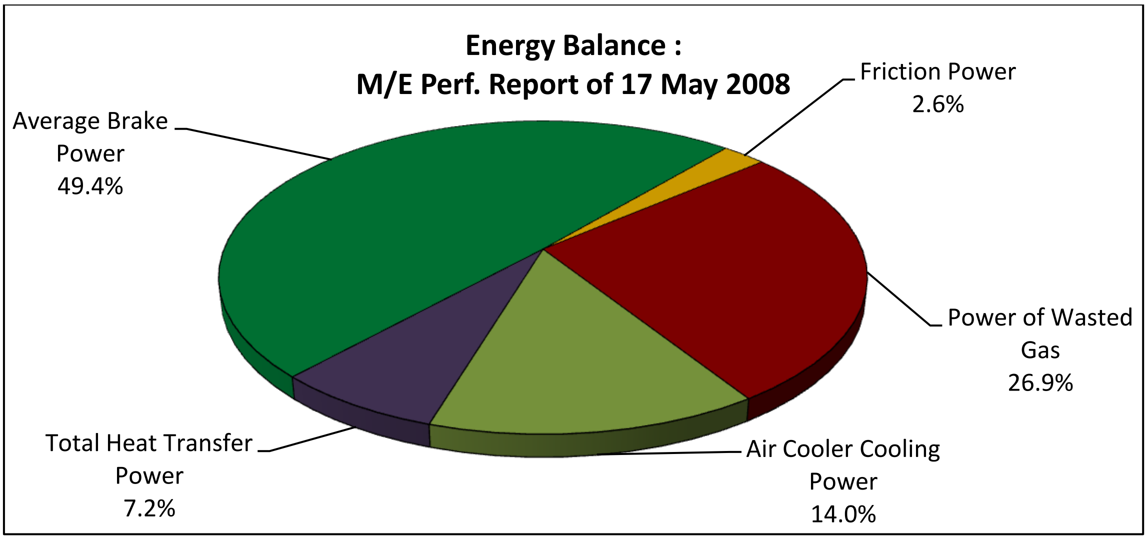


Fig. 8. 39 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	3112.5	26.9%
Air Cooler Cooling Power	1616.0	14.0%
Total Heat Transfer Power	828.0	7.2%
Average Brake Power	5712.9	49.4%
Friction Power	298.0	2.6%
Total Power	11567.4	100.0%

Table 8. 31 Energy Balance values

8.5.4.2. ISO corrections` results

The first two tables show the corrected combustion and compression pressures and they are compared with reference Pmax and Pcomp respectively.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	31.0	**130.0**	133.7	139.3	137.9	130.9
Pmax 2	32.0	**128.0**	131.7	136.6	137.9	130.9
Pmax 3	31.0	**127.0**	130.7	136.6	137.9	130.9
Pmax 4	31.0	**126.0**	129.7	135.6	137.9	130.9
Pmax 5	31.0	**127.0**	130.7	136.6	137.9	130.9
Pmax 6	30.0	**126.0**	129.7	136.6	137.9	130.9
Pmax Average	**31.0**	**127.3**	**131.0**	**136.9**	**137.9**	**130.9**
Pmax / $Pmax_{MCR}$	-	91%	94%	98%	99%	94%

Table 8. 32 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	2.7	**99.0**	102.7	91.2	97.9	90.6
Pcomp 2	0.7	**96.0**	99.7	91.2	97.9	90.6
Pcomp 3	-0.3	**96.0**	99.7	91.2	97.9	90.6
Pcomp 4	-1.3	**95.0**	98.7	91.2	97.9	90.6
Pcomp 5	-0.3	**96.0**	99.7	91.2	97.9	90.6
Pcomp 6	-1.3	**96.0**	99.7	91.2	97.9	90.6
Pcomp Average	**-**	**96.3**	**100.0**	**91.2**	**97.9**	**90.6**

Table 8. 33 Corrected Pcomp

<u>Comments</u>:

- Corrected combustion pressure is lower than the calculated combustion pressure according Pcomp and M/E rpm. From the other hand, according to power output the combustion pressure is normal (Table 8.32).
- Corrected compression pressure is considerably higher than the calculated pressures (Table 8.33).
- ISO correction methodology indicates a possible engine malfunction especially in cylinder no 1 due to the high compression pressure. According to the calculation methodology, the high compression pressures (corrected Pcomp) resulted in high combustion pressures (Acc Pcomp).
- The sea water temperature was not that high (i.e. 28^{o}C$^{)}$ to explain the scavenge air temperature (i.e. 56 ^{o}C$^{)}$.
- These pressures cannot be justified by the weather conditions.
- The above corrections showed a tendency only. The exact causes of operational disturbances could be found in conjunction with the data acquired onboard and the inspections` results.
- Reference average values for combustion pressure (according to Pcomp and M/E rpm calculation), indicated that the corrected combustion pressure was low. On the other hand, according to MOTHER, the Ship`s reported mean combustion pressure is higher than the calculated Pmax.
- Finally, both ISO correction methodology and MOTHER calculations, showed that compression pressures were higher than expected.

8.5.5. M/E Performance Report of 8 Mar. 2009

Main engine faced high compression pressure in cylinder no 1 from early 2008, as reported in Chapters 8.5.1, 8.5.2, 8.5.3 and 8.5.4. The other cylinders also started to face high pressures and temperatures from May 2008 (Chapter 8.5.4), as well. Although the engine was inspected many times and overhauls were made according the manufacturer`s recommendations, the exact causes of the malfunction weren`t found. WARTSILA stated that the problem was probably caused by poor lubrication of the cylinder liner. During, March and April 2009, the majority of pistons, piston rings, piston skirts and liners were replaced, due to increased wear. Maintenance works had been accomplished before the 8th March 2009 and others had been scheduled to be finished before 28th April 2009. The simulation of the engine operation on 8 March and 28 April 2009, intended to investigate how the maintenance works had affected the engine performance in conjunction with the simulation model`s calculated reference data.

8.5.5.1. Performance Data – results

Table 8.34 below shows some major ship data and Table 8.35 shows some engine performance data.

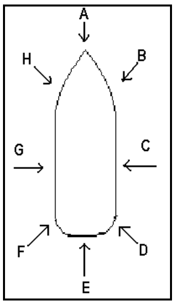
Ship Data			
Ship Condition	BALLAST		
Ship Speed	13.80 kts		
Prop. Apparent Slip	3.10%		
Wind Force / Direction	5	E	
Swell Height / Direction	1.5m	E	
Current speed/ Direction	0kts	E	
Draught FOR/AFT	5.90m	7.98m	
Displacement	35187 MT		

Table 8. 34 Major Ship data

M/E Data	
Engine Speed	105.6 RPM
Power Reported	4619 kW
L.I. Pos.	6.5
Fuel Consumption	26.83 MT/24h
T/C Speed	14900 RPM

M/E Data		
Scavenge Air Pressure (relative)	1.30 bar	
Scavenge Air Pressure (total)	2.80 bar	
Air temp.: E/R / Bef. Blower	41 ^{o}C	41 ^{o}C
Air cooler temp. Bef. / After	143 ^{o}C	47 ^{o}C
Press. Drop Air Cooler	102 mmWG	0.010003 bar
Sea Water Temp.	28 ^{o}C	

Table 8. 35 M/E data

The calculated cylinder pressures are illustrated at the following figures.

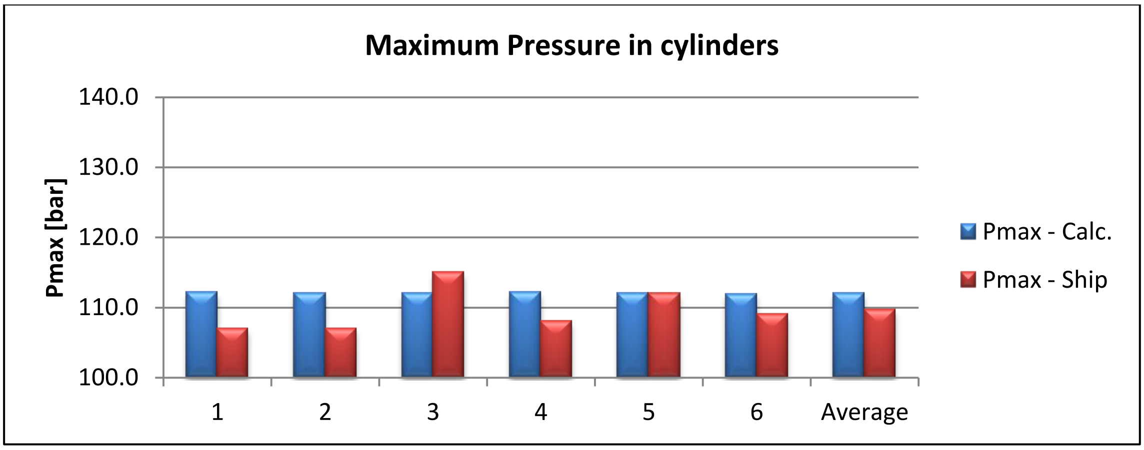


Fig. 8. 40 Pmax, Ship reported vs. "MOTHER" calculated

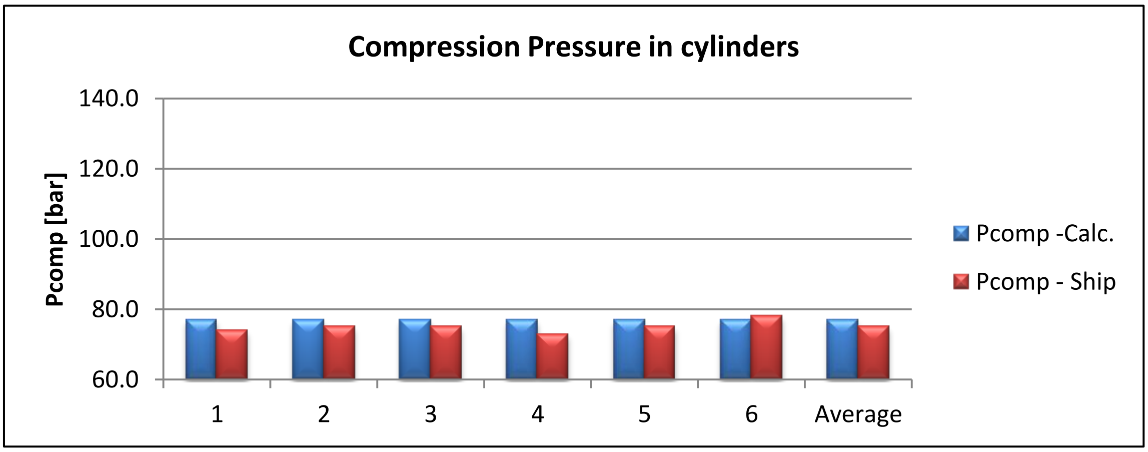


Fig. 8. 41 Pcomp, Ship reported vs. "MOTHER" calculated

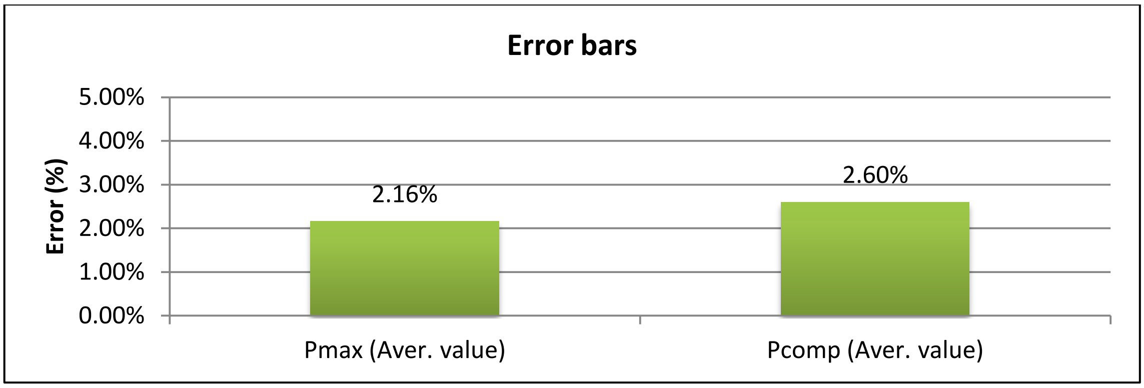


Fig. 8. 42 Error Bars for average values

The table below shows the figures` 8.40 & 8.41 data.

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	77.0	77.0	77.0	76.9	76.9	76.9	76.95
Pcomp – Ship		74.0	75.0	75.0	73.0	75.0	78.0	75.00
Error	%	*4.05%*	*2.67%*	*2.67%*	*5.34%*	*2.53%*	*-1.41%*	*2.60%*
Pmax – Calc.	bar	112.1	112.0	112.0	112.2	112.0	111.9	112.03
Pmax – Ship		107.0	107.0	115.0	108.0	112.0	109.0	109.67
Error	%	*4.77%*	*4.67%*	*-2.61%*	*3.89%*	*0.00%*	*2.66%*	*2.16%*

Table 8. 36 Compression and maximum pressure per cylinder

The following comments could be made:

- MOTHER calculated average combustion and compression pressures were higher than the measured pressures, but less than the 3% error margin.
- Up to 8 March 2009, the following works had been accomplished:
    - New piston was installed in unit no 1.
    - Spare piston crown was installed in unit no 5.
- Compression pressure of cylinder no 6, had the higher value (i.e. 78bar), whereas lower pressure was measured in cylinder no 4 (i.e. 73bar).
- Low compression and combustion pressures could have been caused by the effective low load of the engine; the ship was in ballast condition and weather conditions were good. On the other hand, the excess wear rates that have been reported, could have provoked the high pressures and temperature in individual cylinders.
- The higher combustion pressure was measured in cylinder no. 3 (i.e. 115bar). Assuming that the normal average maximum pressure was close to 110bar (according to MOTHER), the high combustion pressure indicates that the injection was too early for the fuel burnt, thus, the FQS position was probably not set properly. According to manufacturer, in case the FQS input value is advanced (+) by 1^{o}, Pmax rises about 4 to 5bar.

- Lower combustion pressure indicates poor combustion, thus either the FQS value or the injection nozzles needed to have been checked.
- Both injectors of unit no 6, were replaced on 14 March 2009.

The table that follows, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 105.6, LI POS.: 6.5			SHIP	CALC.	*ERROR (%)*
PRESSURES	SCAVENGING AIR PR.	bar	2.312	2.327	*0.65%*
	Pmax (mean value)		109.67	112.03	*2.16%*
	Pcomp (mean value)		75.00	76.95	*2.60%*
	IMEP		-	13.03	-
	BMEP		-	12.28	-
	PRESSURE DROP IN AIR COOLER		0.01000	0.010086	*0.83%*
	PRESSURE IN EXH. RECEIVER		-	2.33	-
TEMPERATURES	M/E AIR COOLER IN	^oC	143.0	141.8	*-0.84%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		47	46.0	*-2.13%*
	AIR COOLER COOLANT WATER		28.0	28.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		370.2	326.0	*-11.93%*
	EXH. GAS AFT. TURBINE		340.0	-	-
MISC.	T/C speed	RPM	14900	14720	*0.20%*
	COMPRESSOR PRESSURE RATIO	-	-	2.31	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	10.78	-
	COMPRESSOR EFFICIENCY	-	-	0.823	-
	TURBINE EFFICIENCY	-	-	0.823	-
	AIR COOLER EFFICIENCY	-	0.8348	0.8418	*0.84%*
	IHP	kW	-	6014	-
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.943	-
	BHP (mean value)	kW	-	4694.8	-
	SFOC	g/kW-h	-	170.0	-

Table 8. 37 Simulation results for 8 Mar. 2009

The Brake Horse Power calculations results are included in Table 8.38.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	4619.0	60.0%	4694.8	60.97%	5.76%
MEPIC Calculated	4763.0	61.9%			5.29%
SULZER Reported	4619.0	60.0%			5.76%
MEPIC Calculation (Regression)	4756.0	61.8%			5.07%
Acc. L.I.xRPM	5394.2	70.1%			-5.38%
Acc. Pme	5240.5	68.1%			-2.59%
Acc. Propeller Law	5661.4	73.5%			-10.72%
Average	5007.6	65.0%	4694.8	60.97%	0.05%

Table 8. 38 BHP

According to Table 8.37, the following figures are attached.

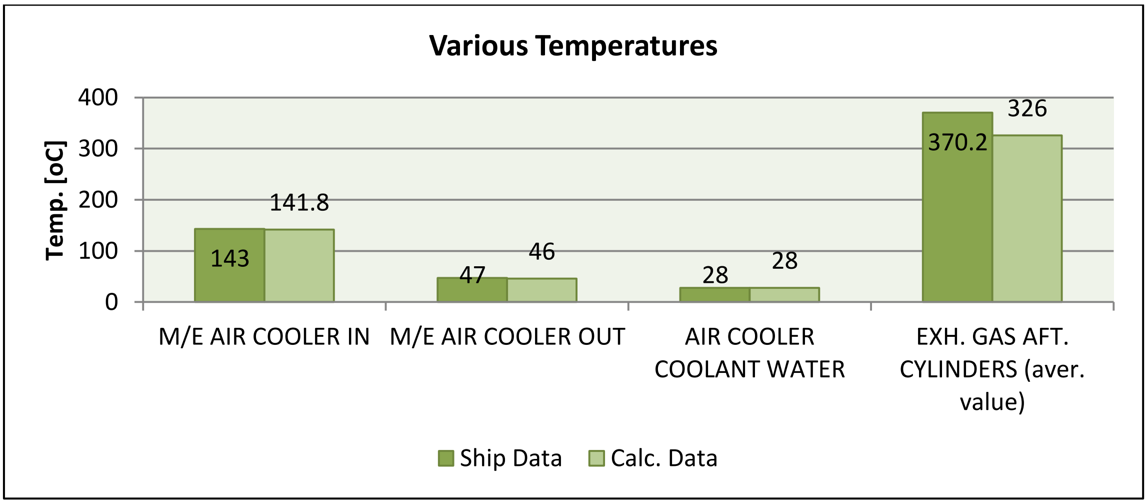


Fig. 8. 43 Various temperatures

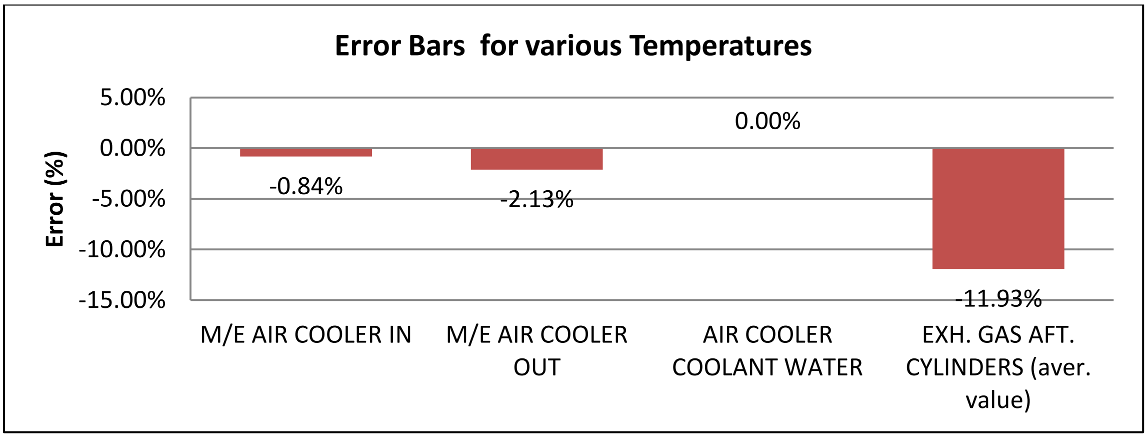


Fig. 8. 44 Error bars for various calculated temperatures

Comments:

- The reason of simulating the engine operation of 8th March 2009, is the evaluation of its performance given that the engine faced pressure disturbances particularly in cylinder no 1 (early 2008) and several parts were replaced before 8 March 2009 and others had been scheduled for later.
- Scavenge air pressure and temperature have been calculated with error less than 1%.
- According to reported data, the air cooler efficiency was high -0.8348- even though the sea temperature (i.e. 28^{o}C) is quite high. MOTHER calculated air cooler efficiency converges with the actual efficiency with 0.84% error.
- The air cooler pressure drop was 102mmWG. This measurement is considerably lower compared to the reported pressure drop in 28th Jan. (i.e. 130mmWG), 10th Feb. (i.e. 130mmWG), and 12th Mar. 2008 (i.e. 110mmWG). Such a difference could be justified by a cleaner air cooler and the lower engine load (resulting in reduced air mass flow rate through the a/c).
- The temperature difference between the air after the cooler (i.e. 47^{o}C) and the sea water inlet (i.e. 28^{o}C) is 19 ^{o}C. Low temperature difference indicates clean air cooler and high temperature fouled a/c. In M/E Performance report of 28 Apr. 2009 (Chapter 8.5.6), this temperature difference increases to 20^{o}C.
- MOTHER calculated average exhaust gas temperature was lower by 11.93% than the temperature measured onboard. According to simulation results and measured data, possible factors that could justify an increased exhaust gas temperature, could be the following:
    - High air inlet temperature (i.e. 41^{o}C)
    - Quite high sea water temperature (i.e. 28^{o}C)
    - Increased exhaust gas temperature (compared to the other cylinders) of cylinder no 1.
    - The burnt heavy fuel oil.
- Brake power output is shown in Table 8.38. MOTHER calculated load (i.e. 60.97% of MCR) converged to the ship reported power and MEPIC calculated power. Calculation methods according Pme and LIxRPM, estimated higher power values because they were based on Shop Test figures and they have not been corrected with factors such as: the fuel burnt, the exhaust gas temperature etc.
- The operating point of 8th March 2009, is illustrated in Fig. 8.45. It can be pointed out that the engine was low loaded and the propeller margin was more than 5%. Consequently, the engine had reserve power in case the propeller required more power to absorb in order to maintain its speed.
- The energy balance is illustrated in Fig. 8.46. The power percentages are rational and within the expected ones.
- Summarizing the aforementioned remarks, the following could be pointed out:
    - The engine was operated in low load.
    - Compression and combustion pressures could be justified by the relatively low load.

- MOTHER calculated higher compression and combustion pressures` values which might be the normal values.
- Unit no 6 faced high compression pressure. For that reason, it was inspected on 14 March 2008 (after the report of 8th March 2009). The upper positioned lubrication quills were blocked and the liner, piston crown and piston rings were found with excessive wear. Consequently, the high compression pressure was justified and MOTHER calculation has been proven correct because lower compression pressure was calculated (predicted) for cylinder no 6.
- Several maintenance works affected the following performance report of 28th April 2008 (Chapter 8.5.6)

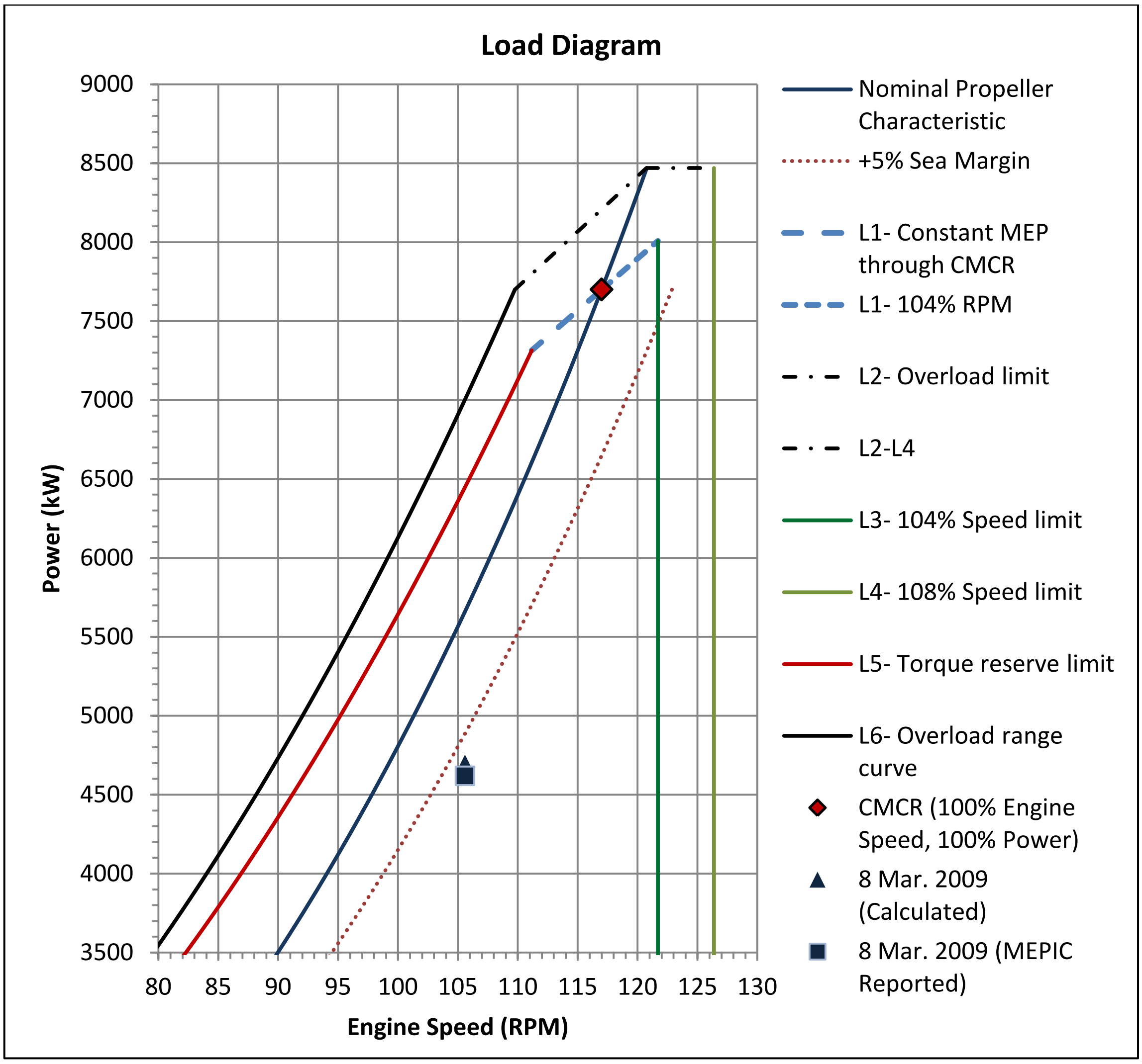


Fig. 8. 45 Load Diagram

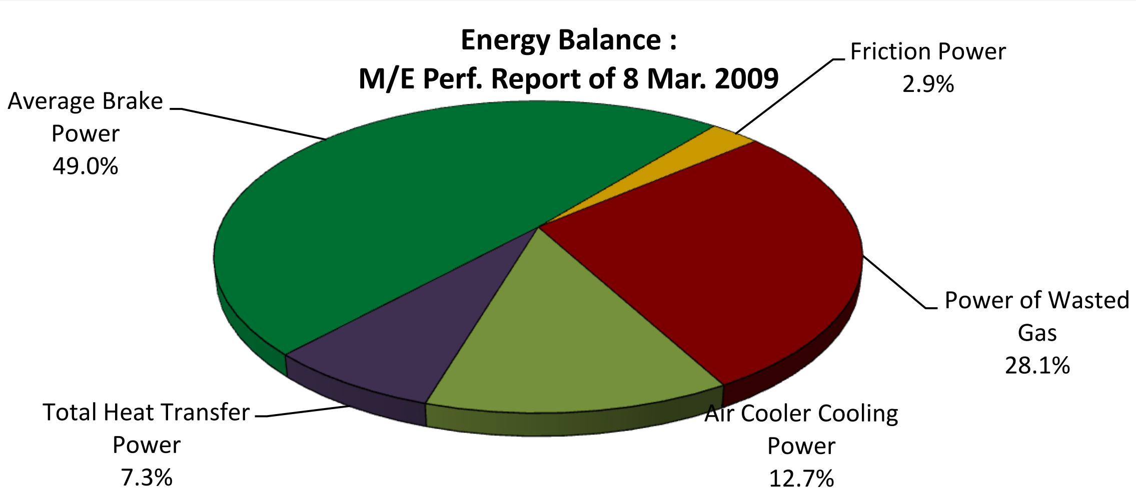


Fig. 8. 46 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	2688.4	28.1%
Air Cooler Cooling Power	1219.7	12.7%
Total Heat Transfer Power	697.9	7.3%
Average Brake Power	4694.8	49.0%
Friction Power	281.2	2.9%
Total Power	9582.0	100.0%

Table 8. 39 Energy Balance values

8.5.5.2. ISO corrections` results

The first two tables show the corrected maximum and compression pressures and they are compared with calculated Pmax and Pcomp respectively.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	33.0	**107.0**	110.8	112.3	128.1	118.2
Pmax 2	32.0	**107.0**	110.8	113.5	128.1	118.2
Pmax 3	40.0	**115.0**	118.8	113.5	128.1	118.2
Pmax 4	35.0	**108.0**	111.8	111.0	128.1	118.2
Pmax 5	37.0	**112.0**	115.8	113.5	128.1	118.2
Pmax 6	31.0	**109.0**	112.8	117.2	128.1	118.2
Pmax Average	**34.7**	**109.7**	**113.4**	**113.5**	**128.1**	**118.2**
Pmax / $Pmax_{MCR}$	-	79%	81%	81%	92%	85%

Table 8. 40 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	-2.7	**74.0**	77.8	75.0	88.4	78.6
Pcomp 2	-2.7	**75.0**	78.8	75.0	88.4	78.6
Pcomp 3	5.3	**75.0**	78.8	75.0	88.4	78.6
Pcomp 4	-1.7	**73.0**	76.8	75.0	88.4	78.6
Pcomp 5	2.3	**75.0**	78.8	75.0	88.4	78.6
Pcomp 6	-0.7	**78.0**	81.8	75.0	88.4	78.6
Pcomp Average	**-**	**75.0**	**78.8**	**75.0**	**88.4**	**78.6**

Table 8. 41 Corrected Pcomp

Comments:

- Average corrected combustion pressure, seems to be normal according Pcomp calculation, but according to the other two calculation methods it seems to be lower than the normal one.
- Average corrected compression pressure was high according to the Pscav calculation and normal according to the Power calculation. Even though, these results are contradicting, calculation according Pscav is more reliable because the other methods (i.e. acc. M/E rpm & Power) tend to calculate higher that the corrected compression pressures.
- Cylinder`s no 3 corrected combustion pressure was 118.8bar, whereas the calculated pressure according Pcomp, was 113.5bar. As stated above, high combustion pressure in individual cylinder has been probably caused by faulty adjustment of FQS position, or higher fuel quantity was transferred by the fuel pump. If neither of these had provoked the high pressure, the cylinder should have been inspected.
- Cylinder`s no 6 corrected compression pressure was 81.8bar, whereas the calculated according Pscav compression pressure (from Shop Test) was 75bar. This difference shows that this particular cylinder was overloaded.
- ISO corrections and MOTHER calculations, resulted in the same consideration regarding the combustion pressure, which is the following:
    - Measured combustion pressures in individual cylinders (except for unit no 3 and 5) were lower that the reference values.
- On the other hand, ISO corrections and MOTHER calculations, resulted in a contradicting result regarding the compression pressure:
    - According to MOTHER calculations, the Ship had lower reference mean compression pressure, whereas, according to ISO corrections, the mean corrected Pcomp was higher than the ISO –reference value (based on Pscav. Calc.).

8.5.6. M/E Performance Report of 28 Apr. 2009

The engine performance had been affected due to the problem with the increased wear in liners and piston rings, from March 2009 (Chapter 8.5.5). By the end of April 2009, several major parts of the engine were replaced due to increased wear. The simulation of the engine performance of 28th April 2009, intends to investigate how the new engine parts had affected the performance of individual cylinders and the overall performance accordingly.

8.5.6.1. Performance Data – results

Table 8.42 below shows some major ship data and Table 8.43 shows some engine performance data.

Ship Data		
Ship Condition	BALLAST	
Ship Speed	14.80 kts	
Prop. Apparent Slip	-0.06%	
Wind Force / Direction	3	A
Swell Height / Direction	2m	A
Current speed/ Direction	0.2kts	E
Draught FOR/AFT	4.18m	6.13m
Displacement	23540 MT	

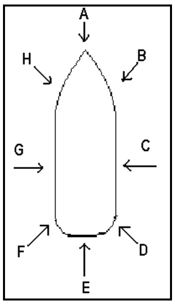


Table 8. 42 Major Ship data

M/E Data		
Engine Speed	109.65 RPM	
Power Reported	5188 kW	
L.I. Pos.	6.7	
Fuel Consumption	26.58 MT/24h	
T/C Speed	16100 RPM	
Scavenge Air Pressure (relative)	1.65 bar	
Scavenge Air Pressure (total)	2.663 bar	
Air temp.: E/R / Bef. Blower	44 °C	43 °C
Air cooler temp. Bef. / After	156 °C	52 °C
Press. Drop Air Cooler	150 mmWG	0.01471 bar
Sea Water Temp.	32 °C	

Table 8. 43 M/E data

The calculated cylinder pressures are illustrated at the following figures.

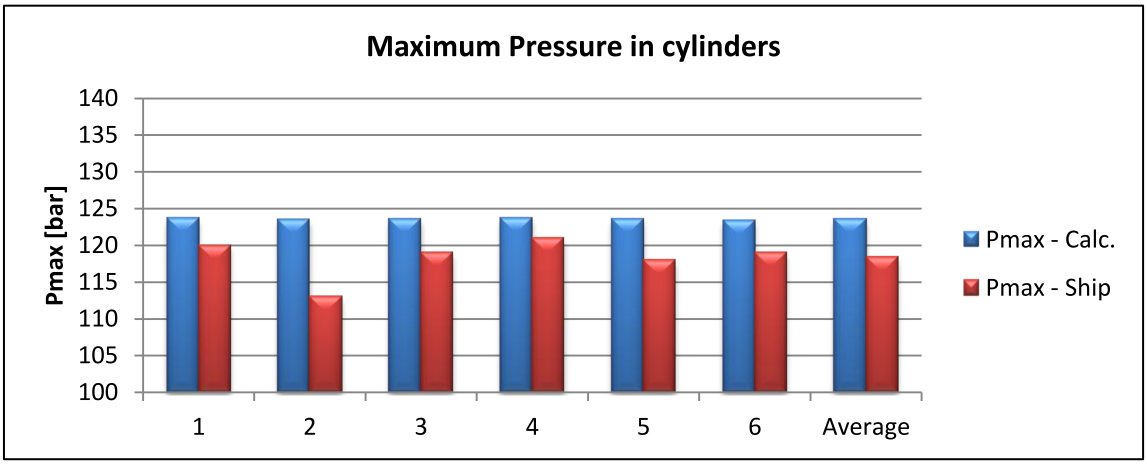


Fig. 8. 47 Pmax, Ship reported vs. "MOTHER" calculated

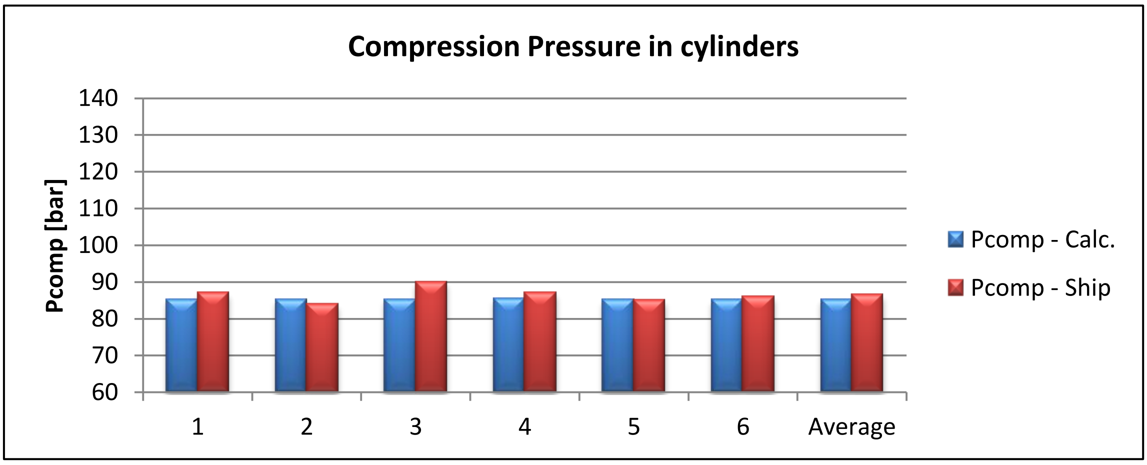


Fig. 8. 48 Pcomp, Ship reported vs. "MOTHER" calculated

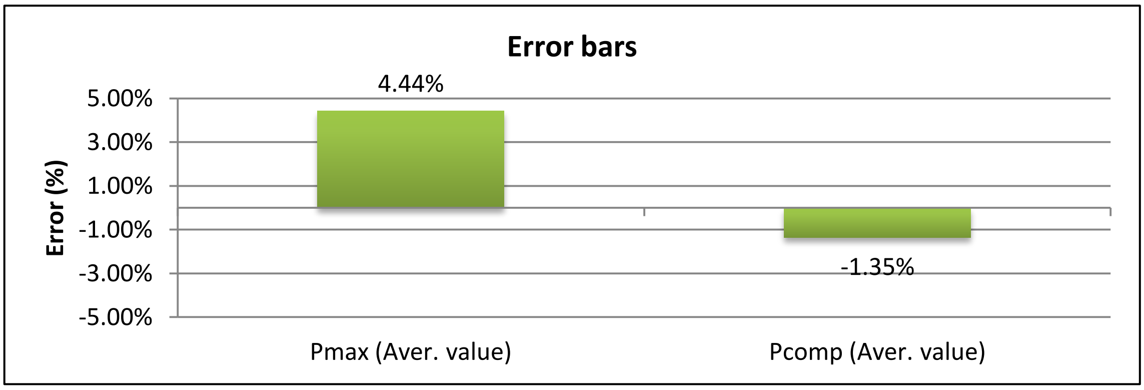


Fig. 8. 49 Error Bars for average values

The table below shows the figures` 8.40 & 8.41 data.

Value / Cyl.		1	2	3	4	5	6	Aver.
Pcomp – Calc.	bar	85.4	85.3	85.4	85.5	85.2	85.2	85.33
Pcomp – Ship		87	84	90	87	85	86	86.50
Error	%	*-1.84%*	*1.55%*	*-5.11%*	*-1.72%*	*0.24%*	*-0.93%*	*-1.35%*
Pmax – Calc.	bar	123.7	123.5	123.6	123.7	123.6	123.4	123.58
Pmax – Ship		120	113	119	121	118	119	118.33
Error	%	*3.08%*	*9.29%*	*3.87%*	*2.23%*	*4.75%*	*3.70%*	*4.44%*

Table 8. 44 Compression and maximum pressure per cylinder

The following comments could be made:

- The following works had been completed before the 28th April 2009:
    - Liners of units no 4 and 6, were replaced.
    - Pistons skirts of units no 1,2,3,4 and 6, were replaced.
    - Piston crowns of units no 1,2,5 and 6, were replaced.
    - Injectors of unit no 6, had already replaced before 28th March 2009.
- Unit`s no 2 liner was replaced after the 28th April 2009, according to mail exchanges.
- Measured compression pressures were not equalized among cylinders. The higher pressure was measure in unit no 3 (i.e. 90bar) and the lower in unit no 2 (i.e. 84bar).
- Onboard measurements of 8th March 2009 (Table 8.36) and 28th April 2009 (Table 8.44), compared to MOTHER calculations, revealed that the compression pressure of cylinder no 4 was improved. The liner and piston skirt replacement contributed in improving and increasing the compression pressure (i.e. on March the 8th 2009, measured Pcomp was almost 4bars lower than the calculated by MOTHER whereas, on April the 28th 2009, measured Pcomp was 1.5bar higher than the calculated value by MOTHER).
- The liner, piston crown and piston skirt replacement of unit no 6, affected the compression pressure (i.e. on March the 8th 2009, it was higher than the normal mean Pcomp by 1.1bar, whereas, on April the 28th 2009, it was by 0.8bar higher).
- Regarding compression pressures of units no 1, 2 and 3, it is not possible to claim that the piston skirts` replacement improved them. Although, on 8th March 2009 measured Pcomp (for units no 1, 2 and 3) were lower than expected, on 28th April 2009 compression pressures varied between these particular cylinders.
    - The lower pressure was measured for unit no 2, i.e. 84bar. According to mail exchanges, the office`s engineer claimed that the forthcoming liner replacement could not explain such low pressures (compression and combustion) and most probably the measurement was taken before the indicator had been cool enough. He also advised to check the injectors at next port of call, in case they were leaking.
    - The higher measured Pcomp for unit no 3, indicates that the cylinder was probably overloaded, in case the measurement is reliable. Furthermore, there is a possibility the combustion space to have been slightly chanced after the maintenance works had been accomplished. Even a minor change of combustion

space, affects the compression ratio and thus, the compression pressure. Such a possibility can be justified if the compression pressure of cylinder no 3 is constantly higher than the normal mean value for M/E Performance Reports after the 28th April 2009.

- The combustion pressures were lower than it is expected according to MOTHER calculations. The lower combustion pressure was measured at unit no 2 (i.e. 113bar) and the higher at unit no 4 (i.e. 121bar). The lower combustion pressure could potentially indicate faulty injectors (for that reason Chief Engineer had been advised by Headquarters` technical staff, to check the injectors) or could have been caused by wrong setting of the FQS position for the fuel burnt. Even wrong injection timing was possible to happen, in case the VIT settings had been modified by mistake.
- Even though both injectors of unit no 6 were replaced on 14th March 2009, combustion pressure (i.e. 119bar) seemed to remain lower than the expected according to MOTHER calculation (i.e.123.4bar) – Table 8.44.
- Apart from comparing the simulation results with the measured data, the ship`s load condition and weather conditions, should be taken into account too. The ship was in ballast condition and the weather conditions were favorable, which can be justified by the propeller apparent slip`s value, i.e. -0.06%. Negative propeller slip value means that the current increases the ship`s speed, reducing the engine load respectively. Consequently, it is expected that the engine operating point is located near to the ‘light propeller curve’ (Fig. 8.45).
- Practice has proven that low engine loading results in lower pressures and temperatures because the fuel consumption is lower and the ship`s speed can be maintained with the minimum required power. According to the current M/E Perf. Report, the engine`s low load can be justified by the load indicator`s position which is 6.7. Although the mean combustion pressure was low (i.e. 118.33bar), it cannot be assigned to the relative engine low load due to the problems history and the unbalanced measured values.

Table 8.45, summarizes the calculated data and provides a comparison with the measured data.

	M/E RPM: 109.65, LI POS.: 6.7		SHIP	CALC.	*ERROR (%)*
PRESSURES	SCAVENGING AIR PR.	bar	2.663	2.620	*-1.61%*
	Pmax (mean value)	bar	118.33	123.58	*4.44%*
	Pcomp (mean value)	bar	86.50	85.33	*-1.35%*
	IMEP	bar	-	14.06	-
	BMEP	bar	-	13.41	-
	PRESSURE DROP IN AIR COOLER	bar	0.0147096	0.0147468	*0.25%*
	PRESSURE IN EXH. RECEIVER	bar	-	2.34	-
TEMPERATURES	M/E AIR COOLER IN	^oC	156.0	160.7	*3.01%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)	^oC	52.0	51.7	*-0.58%*
	AIR COOLER COOLANT WATER	^oC	32.0	32	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)	^oC	375.0	326.4	*-12.96%*
	EXH. GAS AFT. TURBINE	^oC	325.0	-	-
MISC.	T/C speed	RPM	16100	15930	*-1.06%*
	COMPRESSOR PRESSURE RATIO	-	-	2.61	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	12.220	-
	COMPRESSOR EFFICIENCY	-	-	0.827	-
	TURBINE EFFICIENCY	-	-	0.828	-
	AIR COOLER EFFICIENCY	-	0.8387	0.8469	*0.98%*
	IHP	kW	-	-	-
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.954	-
	BHP (mean value)	kW	-	5281.2	-
	SFOC	g/kW-h	-	168.2	-

Table 8. 45 Simulation results for 28 Apr. 2009

The Brake Horse Power calculations results are included in Table 8.46.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	5188.0	67.4%	5281.2	68.59%	1.80%
MEPIC Calculated	5258.0	68.3%			0.44%
SULZER Reported	5188.0	67.4%			1.80%
MEPIC Calculation (Regression)	5200.0	67.5%			1.56%
Acc. L.I.xRPM	5899.5	76.6%			-10.48%
Acc. Pme	5711.4	74.2%			-7.53%
Acc. Propeller Law	6338.1	82.3%			-16.68%
Average	5540.4	72.0%	5281.2	68.59%	-4.68%

Table 8. 46 BHP

According to Table 8.37, the following figures are attached.

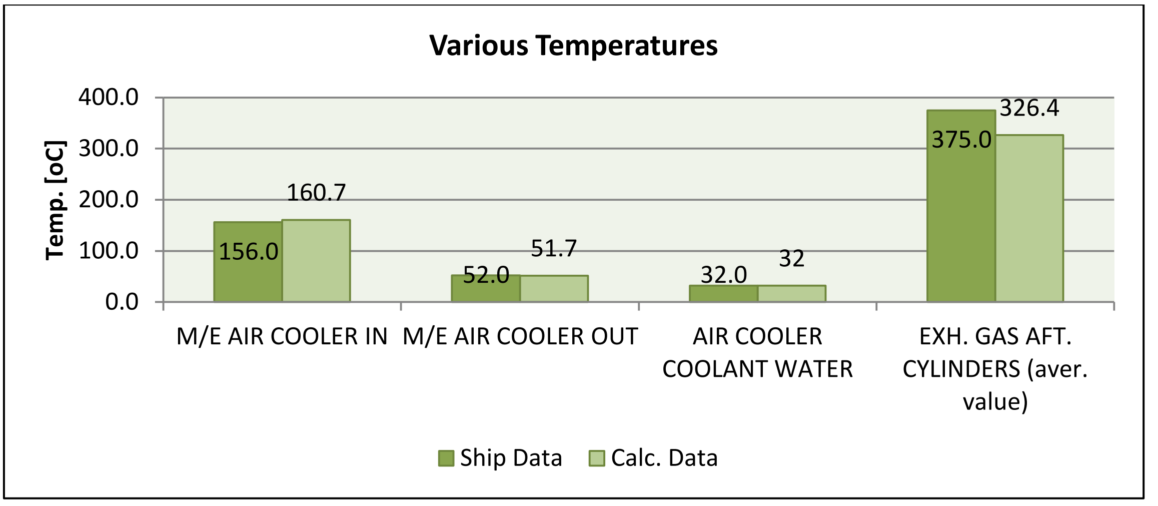


Fig. 8. 50 Various temperatures

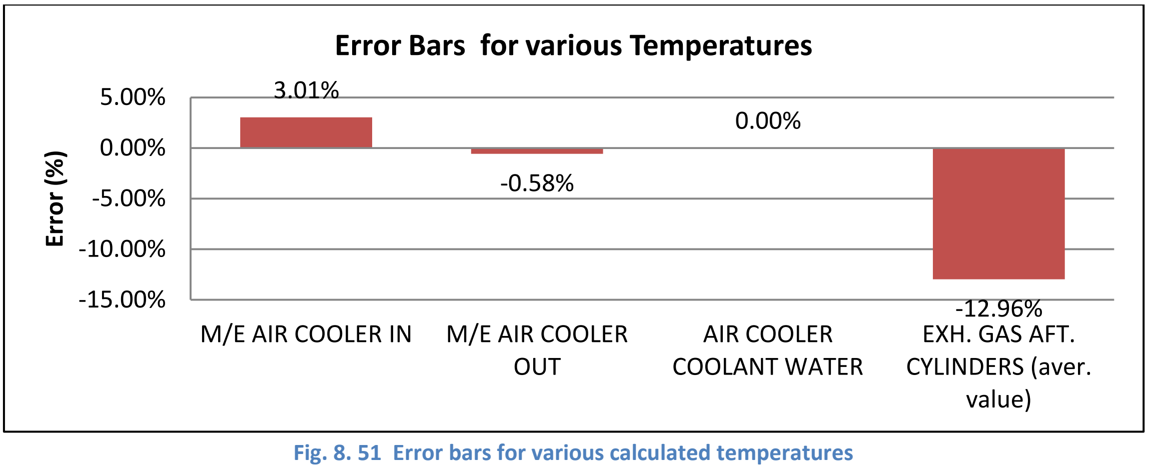


Fig. 8. 51 Error bars for various calculated temperatures

Comments:

- The reason of simulating the engine operation at 28th April 2009, is the evaluation of its performance in conjunction with the maintenance works that were made.
- Scavenging pressure is calculated with error less than 2%.
- The measured pressure drop across the air cooler was high, i.e. 150mmWG. It was increased from 102mmWG (Table 8.35) at previous performance report, to 150mmWG (Table 8.43) the current report. The high pressure loss indicates that the air cooler`s air side was fouled; without exceeding the alarm value (i.e. 500mmWG). The pressure loss coefficient (in MOTHER interface) was manually selected in order to achieve convergence between the calculated pressure drop and measured pressure drop. For that reason the calculation error is 0.25%.
- The measured scavenge air temperature –52°C– exceeded the manufacturer`s recommendation to maintain it between 48°C and 50°C. Increased sea water

temperature (i.e. 32°C) and air cooler fouling were major factors which contributed in raising the scavenge air temperature and decreasing the air cooler`s efficiency. Furthermore, the scavenge air temperature can be adjusted by the sea water valve opening percentage.

- MOTHER simulates the scavenging air temperature with -1.35% error (Fig. 8.43, 8.44).
- The temperature difference between the air after cooler (i.e. 52°C) and the coolant water inlet (32°C) is 20 °C. Increasing temperature difference indicates fouled air cooler. In 8th March 2009, this temperature difference was 19°C. Combining the aforementioned increased temperature difference and the increased pressure drop (i.e. Δp), it is deducted that the air cooler was fouled and was needed cleaning.
- The air temperature before the air cooler is affected by the air temperature at compressor`s inlet, the compression ratio and by compressor efficiency. MOTHER calculated this temperature with 3% error. The lower calculated scavenging pressure compared to the measured onboard, can be justified by the lower calculated turbocharger rotational speed compared to the measured one.
- Calculated turbocharger speed is 1.06% lower than the reported speed in M/E Perf. Report.
- The air cooler efficiency (as calculated from the measured values) remained high despite the high scavenge air and sea water temperature.
- Exhaust gas temperature was underestimated by 12.96%. The measured exhaust gas temperature was affected directly by the following factors:
    - The fouled air cooler
    - The high sea temperature (i.e. 32°C)
    - The high engine room temperature (i.e. 44°C)
    - The heavy fuel oil. Compared with diesel oil, the use of heavy fuel oil can normally be expected to give an exhaust temperature increase of approx. 5°C.
- Calculated power deviated less than 2% from reported MEPIC and calculated MEPIC power (Table 8.46). The calculated power according LIxRPM, Pme and Propeller Law, were considerably higher than the calculated by MOTHER power and the reported power too. Such a difference, expresses the reserve power of the engine for the current speed.
- The engine operating point is illustrated in Load Diagram in Fig. 8.45. The engine load is low because the operating point is located on the right of light (+5% margin) propeller curve. Such a performance it can also be justified by the negative propeller slip, good weather conditions and ballast condition of the ship.
- The energy balance, as calculated by MOTHER, is illustrated in Fig. 8.46.

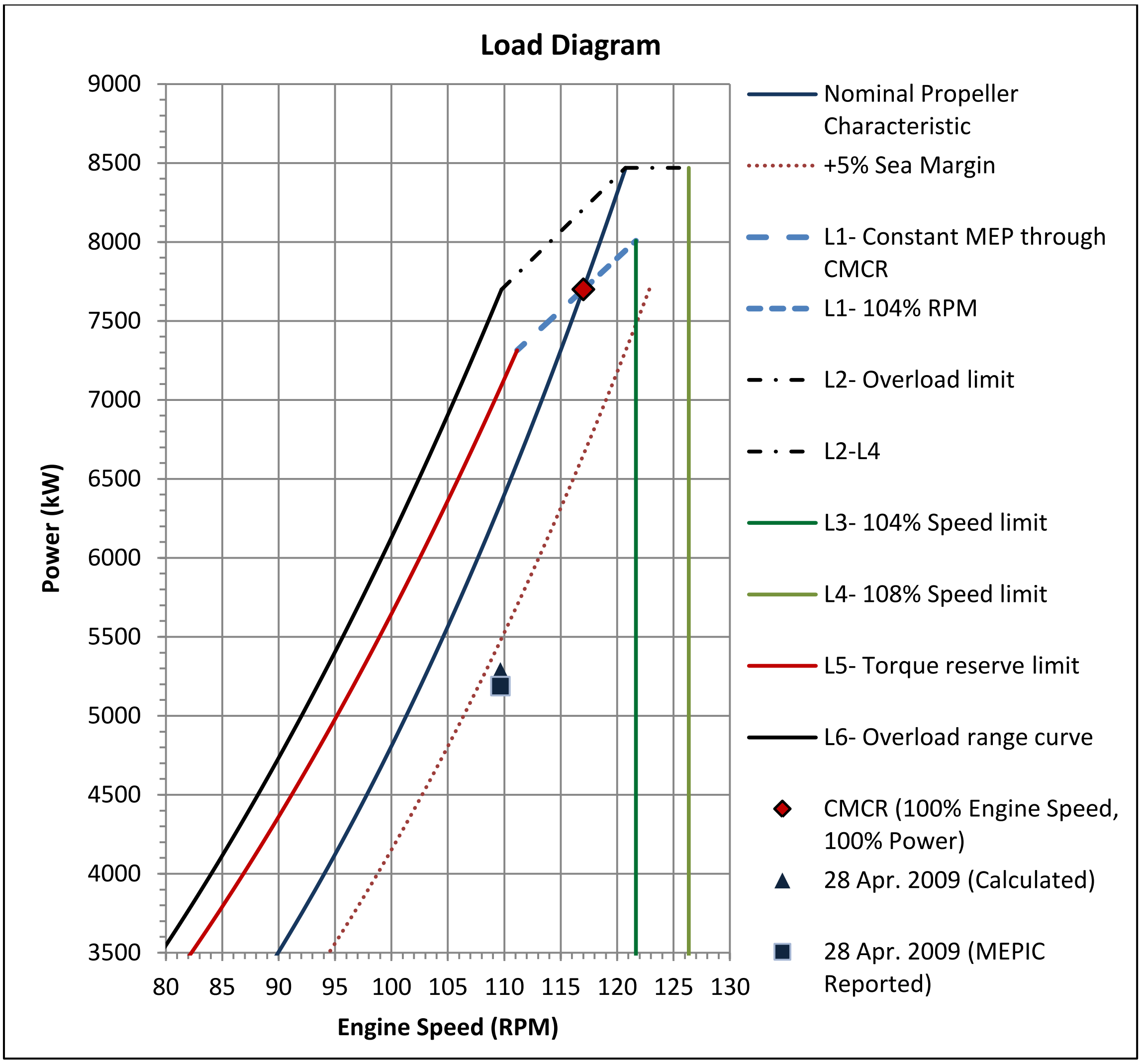


Fig. 8. 52 Load Diagram

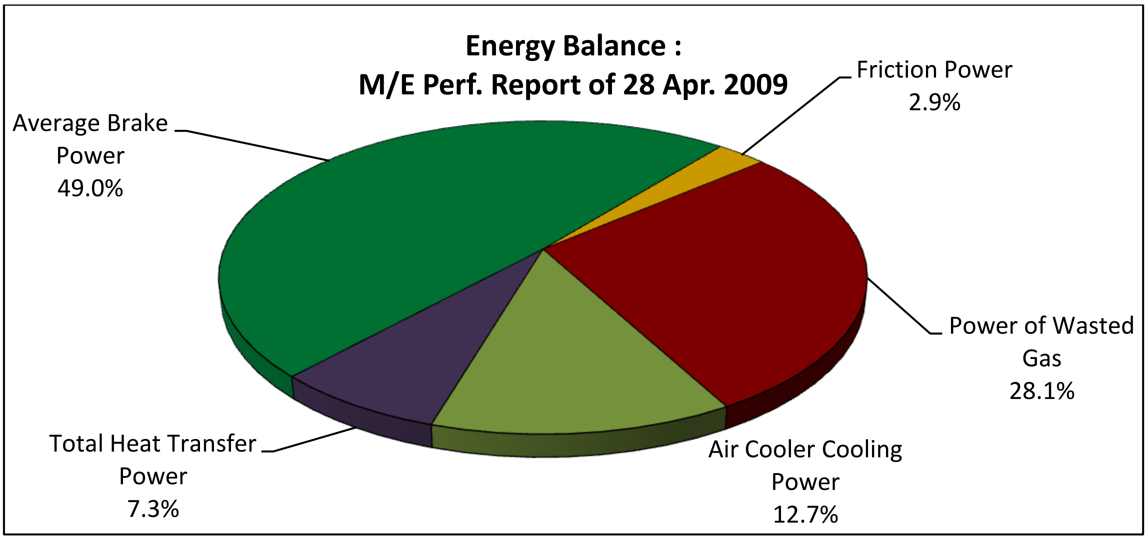


Fig. 8. 53 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	2791.2	26.1%
Air Cooler Cooling Power	1573.1	14.7%
Total Heat Transfer Power	760.0	7.1%
Average Brake Power	5281.2	49.3%
Friction Power	296.3	2.8%
Total Power	10701.8	100.0%

Table 8. 47 Energy Balance values

8.5.6.2. ISO corrections` results

This sub-section provides the results of the ISO correction methodology.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	33.0	**120.0**	124.0	127.5	137.2	132.4
Pmax 2	29.0	**113.0**	117.0	124.2	137.2	132.4
Pmax 3	29.0	**119.0**	123.0	130.7	137.2	132.4
Pmax 4	34.0	**121.0**	125.0	127.5	137.2	132.4
Pmax 5	33.0	**118.0**	122.0	125.3	137.2	132.4
Pmax 6	33.0	**119.0**	123.0	126.4	137.2	132.4
Pmax Average	**31.8**	**118.3**	**122.4**	**127.0**	**137.2**	**132.4**
Pmax / $Pmax_{MCR}$		85%	88%	91%	98%	95%

Table 8. 48 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	1.7	**87.0**	91.0	91.2	97.1	92.2
Pcomp 2	-5.3	**84.0**	88.0	91.2	97.1	92.2
Pcomp 3	0.7	**90.0**	94.0	91.2	97.1	92.2
Pcomp 4	2.7	**87.0**	91.0	91.2	97.1	92.2
Pcomp 5	-0.3	**85.0**	89.0	91.2	97.1	92.2
Pcomp 6	0.7	**86.0**	90.0	91.2	97.1	92.2
Pcomp Average	**-**	**86.5**	**90.5**	**91.2**	**97.1**	**92.2**

Table 8. 49 Corrected Pcomp

The following comments can be made:

- Corrected average combustion pressure was lower than the calculated values according to the three providing calculation methods in Table 8.48 (i.e. according Pcomp, M/E rpm and Power).
- The first calculation method is based on the measured compression pressure and gives an estimation about the combustion pressure according to Shop Test figures.
- Combustion pressures according M/E rpm and power are usually overestimated, but in general, they provide the upper limit of the combustion pressure that could be measured in the cylinders.
- Cylinder no 3 had the lower corrected combustion pressure, whereas cylinder no 4 the higher corrected combustion pressure.
- The pressure difference between combustion and compression pressure, is related with the stress that the piston rings are undertaking due to the combustion. High pressures could result in piston rings collapsing. For that reason, this difference is monitored constantly and it is defined by the reference Shop Test values. Table 8.48, shows that cylinder no 4 faced the higher Pmax-Pcomp among the other cylinders.
- Corrected compression pressure seems to be normal according to Pscav calculation and Power calculation, even though it is lower by 2bar.
- Calculated compression pressure according to M/E rpm, was higher than the other two calculations (i.e. acc. Pcomp and Power). The calculated compression pressure according to m/e rpm is based on Shop Test figures and it shows the upper safe margin for the compression pressure for the given engine speed. In practice, the engine was operated at lower power output according to its speed and compared to Test bed output, in order to have power reserve in case of adverse weather, increased hull fouling etc. Thus, the higher calculated compression pressure (similarly combustion pressure) according to m/e rpm, expresses the power reserve of the engine. Furthermore, it should be stated that high power output provoke high pressures in the engine.
- Summarizing the ISO corrections` results and the MOTHER simulation results, the following comments can be made:
    - First, both MOTHER and ISO methodology resulted in the same consideration: that the attained mean combustion pressure was lower than it should have been.
    - Second, MOTHER and ISO calculations showed than the mean compression pressure was normal, even though MOTHER calculated the compression pressure by 1.35% lower than the measured.
    - Cylinder no 2 (which liner was replaced after April the 28th, 2009) faced too low compression and combustion pressures.

8.5.7. M/E Performance Report of 29 Oct. 2009

The performance parameters` measurements on 29th October 2009 was held when the engine was burning Heavy Fuel Oil (HFO). The purpose of simulating the current engine`s performance is the investigation whether or not the reported fuel quality had affected the engine`s performance. Furthermore, the results of this simulation will be compared with those depicted from the simulation of the engine operation on November the 1st, 2009, when the engine was burning a mixture of Heavy Fuel Oil and LSFO with low ignitability properties (Chapter 8.5.8).

8.5.7.1. Performance Data – results

Table 8.42 below shows some major ship data and Table 8.43 shows some engine performance data.

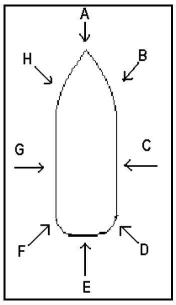
Ship Data			
Ship Condition	LOADED		
Ship Speed	12.90 kts		
Prop. Aparent Slip	8.20%		
Wind Force / Direction	3	C	
Swell Height / Direction	2m	C	
Current speed/ Direction	0.5kts	C	
Draught FOR/AFT	10.19m	10.61m	
Displacement	50536 MT		

Table 8. 50 Major Ship data

M/E Data		
Engine Speed	104.18 RPM	
Power Reported	4423 kW	
L.I. Pos.	6.5	
Fuel Consumption	23.018 MT/24h	
T/C Speed	15000 RPM	
Scavenge Air Pressure (relative)	1.40 bar	
Scavenge Air Pressure (total)	2.413 bar	
Air temp.: E/R / Bef. Blower	43 ^{o}C	43 ^{o}C
Air cooler temp. Bef. / After	140 ^{o}C	46 ^{o}C
Press. Drop Air Cooler	115 mmWG	0.011277 bar
Sea Water Temp.	27 ^{o}C	

Table 8. 51 M/E data

Fuel Oil Analysis		
ISO-F GRADE (2005)		RMF 180
Viscosity at 50°C	cSt	170.2
Density at 15°C	kg/l	0.985
Sulphur Content	% (v/v)	2.62
CCAI	-	855
Net Specific Energy	MJ/kg	40.39

Table 8. 52 Fuel Oil Analysis

The calculated cylinder pressures are illustrated at the following figures.

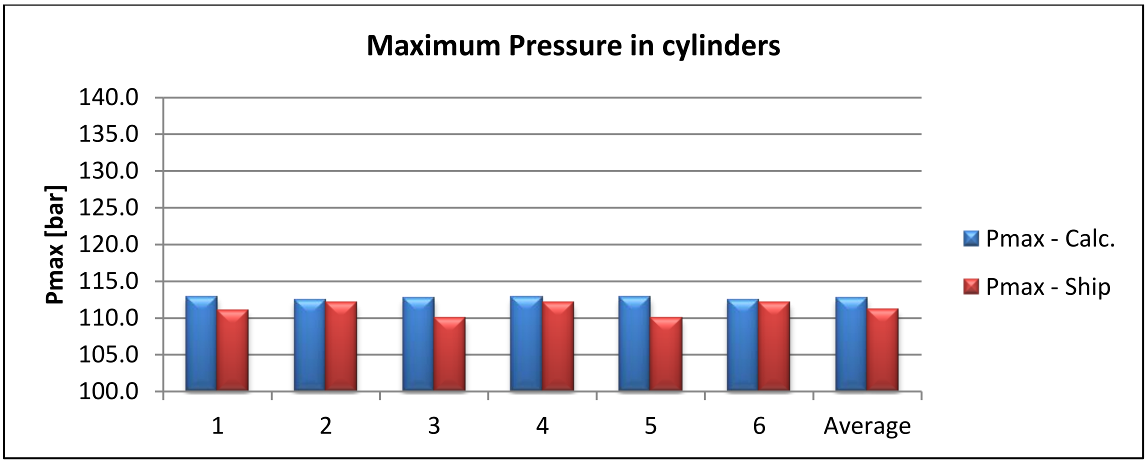


Fig. 8. 54 Pmax, Ship reported vs. "MOTHER" calculated

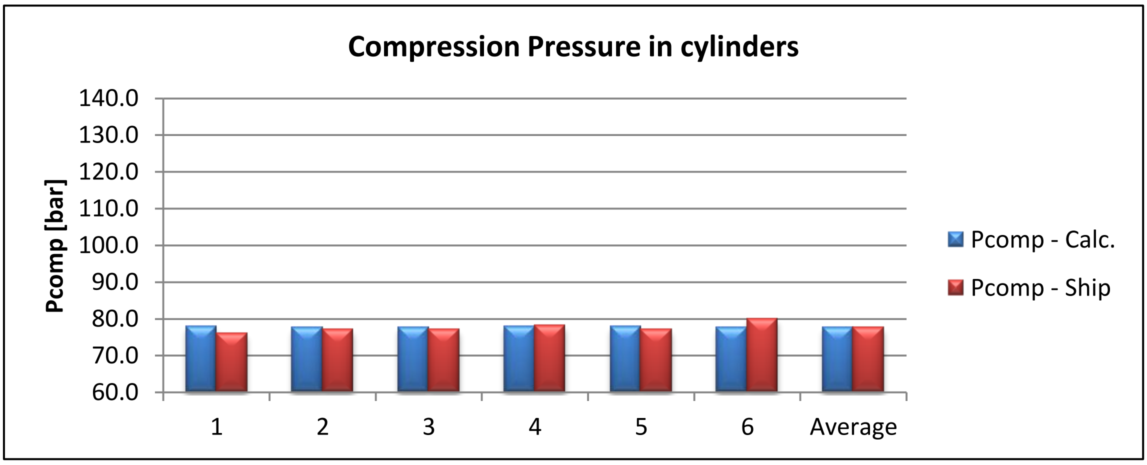


Fig. 8. 55 Pcomp, Ship reported vs. "MOTHER" calculated

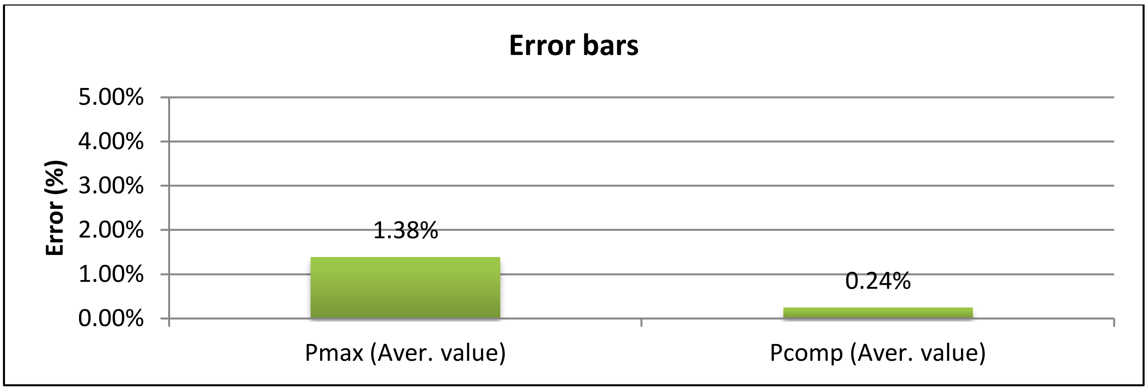


Fig. 8. 56 Error Bars for average values

Table 8.53 shows the figures` 8.54 & 8.55 data.

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	77.8	77.5	77.7	77.8	77.8	77.5	77.68
Pcomp – Ship		76.0	77.0	77.0	78.0	77.0	80.0	77.50
Error	%	*2.37%*	*0.65%*	*0.91%*	*-0.26%*	*1.04%*	*-3.13%*	*0.24%*
Pmax – Calc.	bar	112.9	112.5	112.7	112.8	112.8	112.5	112.70
Pmax – Ship		111.0	112.0	110.0	112.0	110.0	112.0	111.17
Error	%	*1.71%*	*0.45%*	*2.45%*	*0.71%*	*2.55%*	*0.45%*	*1.38%*

Table 8. 53 Compression and maximum pressure per cylinder

The following comments could be made:

- Average combustion pressure has been simulated with 1.38% error. According to MOTHER calculations the ship was operated under quite low combustion pressures.
- Measured combustion pressure was balanced among cylinders and the pressure difference between the maximum (i.e. 112bar) and the minimum (i.e. 110bar) measured pressure was 2bar.
- Low combustion pressures indicated possible injector malfunctioning or late injection in cylinders which may have been caused by wrong FQS position adjustment.
- According to fuel analysis report (Table 8.52), Calculated Carbon Aromaticity Index (CCAI) was 855 which means that the fuel ignitability was lower compared to other heavy fuel oil with CCAI equal to 830. The relative low ignitability can be avoided by advancing the fuel injection through the FQS position. The FQS position on October the 29th, 2009, was at +0.8deg position. The FQS position was calculated according a methodology provided by the manufacturer (i.e. WARTSILA). The methodology is not showed due to copyright reasons. The calculation resulted in FQS position of +1.1deg, which is by +0.3deg greater than the Ship`s reported value (i.e. +0.8deg). If the FQS position had been set at +1.1deg, the combustion pressure would have been greater than the reported. Consequently, the low combustion pressure could have been avoided by retarding more the injection timing.

- Average calculated compression pressure was 0.24% higher than the mean measured onboard, consequently it can be considered that the ship had normal compression pressures.
- Cylinder no 6 had the higher measured compression pressure (i.e. 80bar), compared to the other cylinders` pressures. However, cylinder`s no 6 exhaust gas temperature (i.e. 370°C), and also, the piston underside piston temperature (i.e. 59°C), were normal according to M/E Performance Report. Finally, the high measured Pcomp does not indicate directly a possible malfunction. Pressure differences at the range of 5bar among the cylinders, are admissible but it should be monitored.
- Cylinder no 1 had the lower compression pressure (i.e. 76bar). As mentioned above, such pressure does not indicate a malfunction, because this unit`s temperatures (exhaust gas temp.: 360 °C, piston underside temp.: 61°C) were normal and balanced with the other cylinders.
- In order to conclude whether the measured pressures of cylinders no 1 and 6 are normal, the precedent and future performance data should be taken into account and be connected with the engine load and weather conditions.

Table 8.54, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 104.18, LI POS.: 6.5			**SHIP**	**CALC.**	***ERROR (%)***
PRESSURES	SCAVENGING AIR PR.	bar	2.413	2.3665	*-1.99%*
	Pmax (mean value)		111.17	112.70	*1.38%*
	Pcomp (mean value)		77.50	77.68	*0.24%*
	IMEP		-	13.14	-
	BMEP		-	12.30	-
	PRESSURE DROP IN AIR COOLER		0.01128	0.0113319	*0.48%*
	PRESSURE IN EXH. RECEIVER		-	2.101	-
TEMPERATURES	M/E AIR COOLER IN	°C	140.0	146.3	*4.50%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		46.0	47.5	*3.26%*
	AIR COOLER COOLANT WATER		27.0	27	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		361.2	314.3	*-12.98%*
	EXH. GAS AFT. TURBINE		335.0	-	-
MISC.	T/C speed	RPM	15000	14900	*-0.67%*
	COMPRESSOR PRESSURE RATIO	-	-	2.33	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	10.840	-
	COMPRESSOR EFFICIENCY	-	-	0.825	-
	TURBINE EFFICIENCY	-	-	0.823	-
	AIR COOLER EFFICIENCY	-	0.8319	0.8399	*0.97%*
	IHP	kW	-	4939.10	-
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.936	-
	BHP (mean value)	kW	4533.64	4639.1	*2.33%*
	SFOC	g/kW-h	-	169.8	-

Table 8. 54 Simulation results for 29th Oct. 2009

The Brake Horse Power calculations results are included in Table 8.55.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	4423.0	57.4%	4639.1	60.25%	-4.82%
MEPIC Calculated	4597.0	59.7%			3.70%
SULZER Reported	4423.0	57.4%			1.49%
MEPIC Calculation (Regression)	4581.0	59.5%			3.33%
Acc. L.I.xRPM	5186.1	67.4%			-6.34%
Acc. Pme	5041.8	65.5%			-4.18%
Acc. Propeller Law	5436.1	70.6%			-8.83%
Average	4812.6	62.5%	4639.1	60.25%	-3.89%

Table 8. 55 BHP

According to Table 8.54, the following figures are attached.

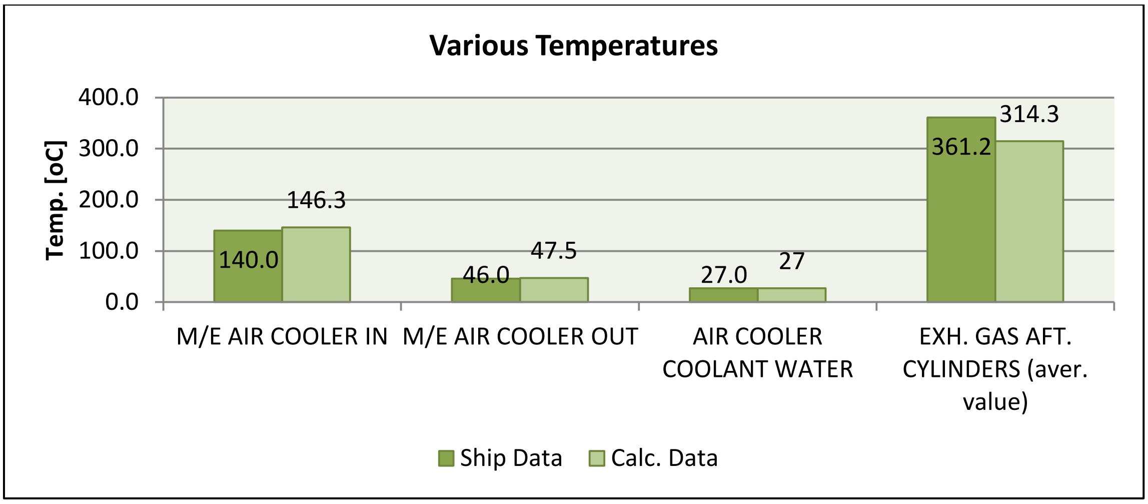


Fig. 8. 57 Various temperatures

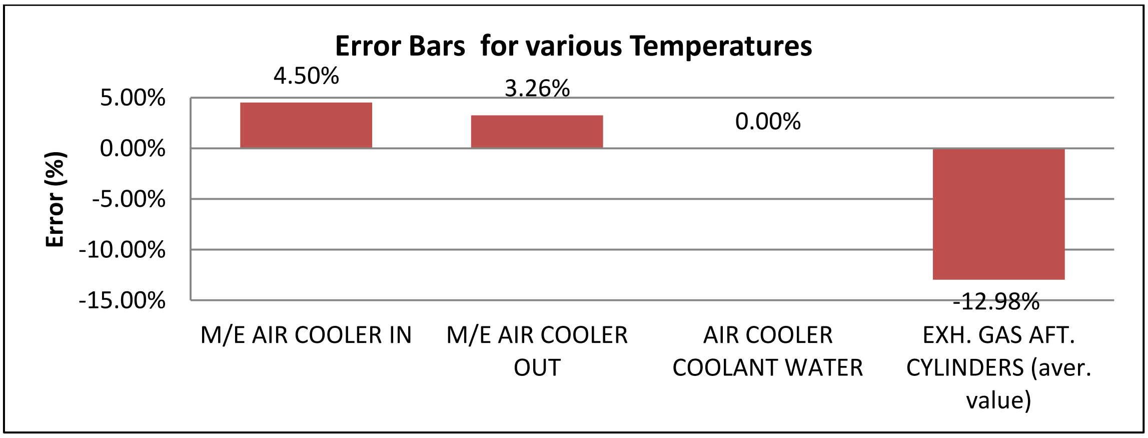


Fig. 8. 58 Error bars for various calculated temperatures

Comments:

- The engine operation on 29th October, 2009, with HFO, is evaluated in order to be compared with the operation of November the 1st, 2009, when the engine was operated with mixed fuel (Chapter 8.5.8).
- Air temperature before the air cooler was calculated with 3.86% error. The compressor efficiency, compressor pressure ratio and ambient air temperature, are the main parameters that affect the air temperature before the air cooler.
- Lower measured scavenging air temperature, than MOTHER calculated, can be justified by MOTHER calculation error and the opening percentage of the coolant water valve.
- Exhaust gas temperature was underestimated by -12.98%.
- The engine operation point is illustrated at the Load Diagram (Fig. 8.59). Although the Ship was loaded, the engine operating point was adjacent to the light propeller curve.
- The energy balance calculation is illustrated in Fig. 8.60. The power percentages complies with the reference percentages for diesel engines.

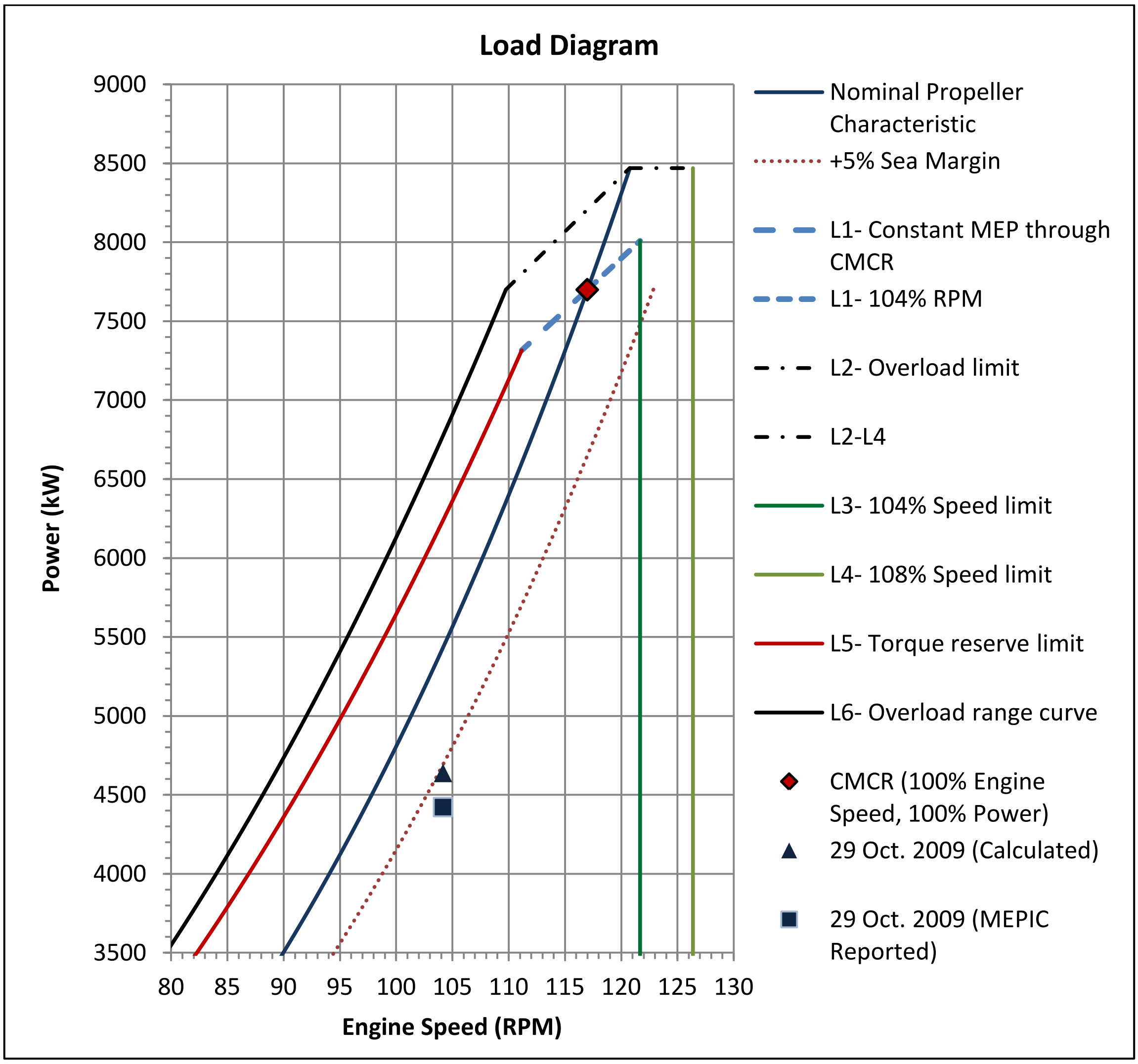


Fig. 8. 59 Load Diagram

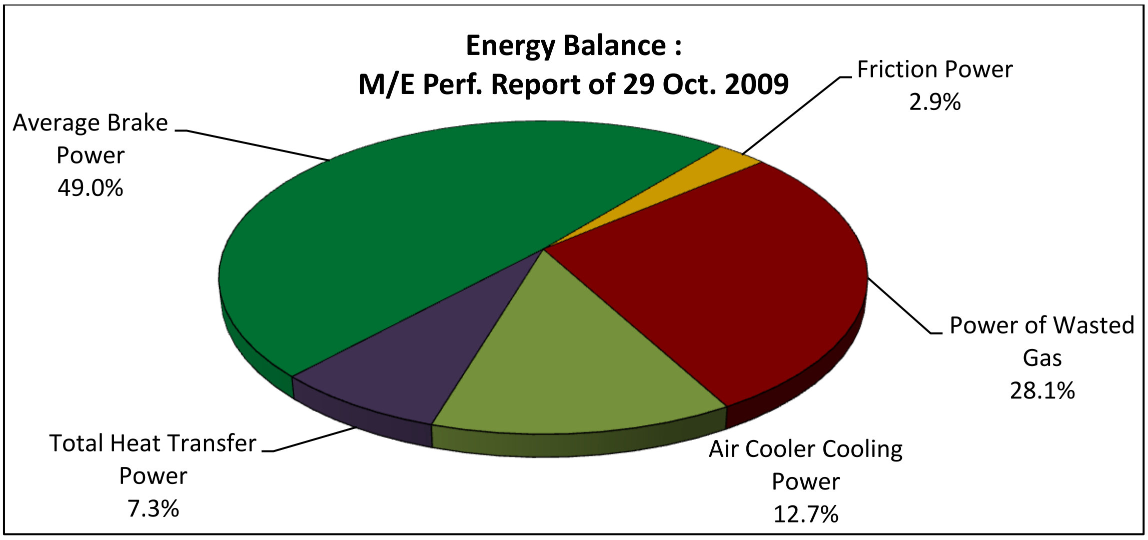


Fig. 8. 60 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	2579.9	27.3%
Air Cooler Cooling Power	1252.4	13.3%
Total Heat Transfer Power	690.7	7.3%
Average Brake Power	4639.1	49.2%
Friction Power	272.7	2.9%
Total Power	9434.8	100.0%

Table 8. 56 Energy Balance values

8.5.7.2. ISO corrections` results

This sub-section provides the results of the ISO correction methodology.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	35.0	**111.0**	115.4	114.8	125.0	120.8
Pmax 2	35.0	**112.0**	116.4	116.0	125.0	120.8
Pmax 3	33.0	**110.0**	114.4	116.0	125.0	120.8
Pmax 4	34.0	**112.0**	116.4	117.2	125.0	120.8
Pmax 5	33.0	**110.0**	114.4	116.0	125.0	120.8
Pmax 6	32.0	**112.0**	116.4	119.6	125.0	120.8
Pmax Average	**33.7**	**111.2**	**115.5**	**116.6**	**125.0**	**120.8**
Pmax / $Pmax_{MCR}$	-	80%	83%	83%	89%	86%

Table 8. 57 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	-0.2	**76.0**	80.4	78.2	85.8	80.9
Pcomp 2	0.8	**77.0**	81.4	78.2	85.8	80.9
Pcomp 3	-1.2	**77.0**	81.4	78.2	85.8	80.9
Pcomp 4	0.8	**78.0**	82.4	78.2	85.8	80.9
Pcomp 5	-1.2	**77.0**	81.4	78.2	85.8	80.9
Pcomp 6	0.8	**80.0**	84.4	78.2	85.8	80.9
Pcomp Average	**-**	**77.5**	**81.9**	**78.2**	**85.8**	**80.9**

Table 8. 58 Corrected Pcomp

Comments:

- Average corrected combustion pressure (i.e. 115.5bar) was slightly lower than the reference combustion pressure according to Pcomp calculation (i.e. 116.6bar) and significantly lower according to M/E rpm (i.e. 125bar) and Power calculation (i.e. 120.8bar) respectively (Table 8.57).
- Average corrected compression pressure (i.e. 81.9bar) was higher than the reference pressure according Pscav (i.e. 78.2bar) and Power (i.e. 80.9bar) and lower than the compression pressure calculated according to M/E rpm (i.e. 85.8bar, Table 8.58).
- In practice, pressure deviations within two bar (2bar) between ISO corrected and calculated values according to Shop Test figures are not an issue.
- Calculations according to M/E rpm, express the maximum combustion and compression pressures than the engine can develop at the correspondent rotational speed, according to Shop Test data.
- The engine operation was balanced due to minor deviations among the measured combustion and compression pressures; expect for cylinder`s no 6 measured compression pressure, i.e. 80bar.
- High compression pressure and low combustion pressure indicates that the combustion in the engine was poor or the ignition was late (in terms of crank angle degrees) for the fuel type used. In case of poor combustion, the injection nozzles should have been checked. In case the ignition was delayed, the FQS position should have been checked in conjunction with the HFO used.
- By comparing the MOTHER`s based results and ISO corrections, the following could be stated:
    - Both MOTHER and ISO methodology conclude that the measured combustion pressure was lower than the calculated pressure.
    - Regarding the compression pressure, MOTHER simulation concluded that the engine had normal compression pressure because the calculation error was 0.24% (Table 8.53). From the other hand, according to ISO correction methodology, compression pressure seemed to be high according Pscav calculation and normal (deviation within 2bar range) according to Power calculation. Finally, by combining the aforementioned results, the compression pressure could be considered rather normal.

8.5.8. M/E Performance Report of 1 Nov. 2009

The engine performance parameters` measurements have been carried out in order to assess and control the engine performance, because a mixture of Low Sulfur Heavy Fuel Oil and Heavy Fuel Oil with low ignitability properties was burnt by the engine. The office`s technical staff advised the Chief Engineer to proceed in mixing the two fuels in a ratio of 1/3 HFO and 2/3 LSFO. According to fuel analysis report sent from FOBAS, the fuel required attention regarding the engine operating parameters and the combustion profile, while using this fuel. The performance parameters of November the 1st, 2009, are compared with the previous measured parameter of October the 29th, 2009 (Chapter 8.5.7), in order to assess the fuel impact to the performance.

8.5.8.1. Performance Data – results

Table 8.59 below shows some major ship data and Table 8.60 shows some engine performance data.

Ship Data		
Ship Condition	LOADED	
Ship Speed	11.70 kts	
Prop. Apparent Slip	16.70%	
Wind Force / Direction	5	B
Swell Height / Direction	2m	A
Current speed/ Direction	0.5kts	B
Draught FOR/AFT	10.18m	10.55m
Displacement	50467 MT	

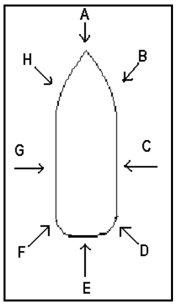


Table 8. 59 Major Ship data

M/E Data		
Engine Speed	104.22 RPM	
Power Reported	4474 kW	
L.I. Pos.	6.5	
Fuel Consumption	23.437 MT/24h	
T/C Speed	15100 RPM	
Scavenge Air Pressure (relative)	1.50 bar	
Scavenge Air Pressure (total)	2.512 bar	
Air temp.: E/R / Bef. Blower	44 °C	44 °C
Air cooler temp. Bef. / After	150 °C	47 °C
Press. Drop Air Cooler	115 mmWG	0.011277 bar
Sea Water Temp.	29 °C	

Table 8. 60 M/E data

Fuel Oil Analysis		
ISO-F GRADE (2005)		RMG 380
Viscosity at 50°C	cSt	356.6
Density at 15°C	kg/l	0.984
Sulphur Content	% (v/v)	0.70
CCAI	-	846
Net Specific Energy	MJ/kg	41.04

Table 8. 61 Fuel Oil Analysis

The calculated cylinder pressures are illustrated at the following figures.

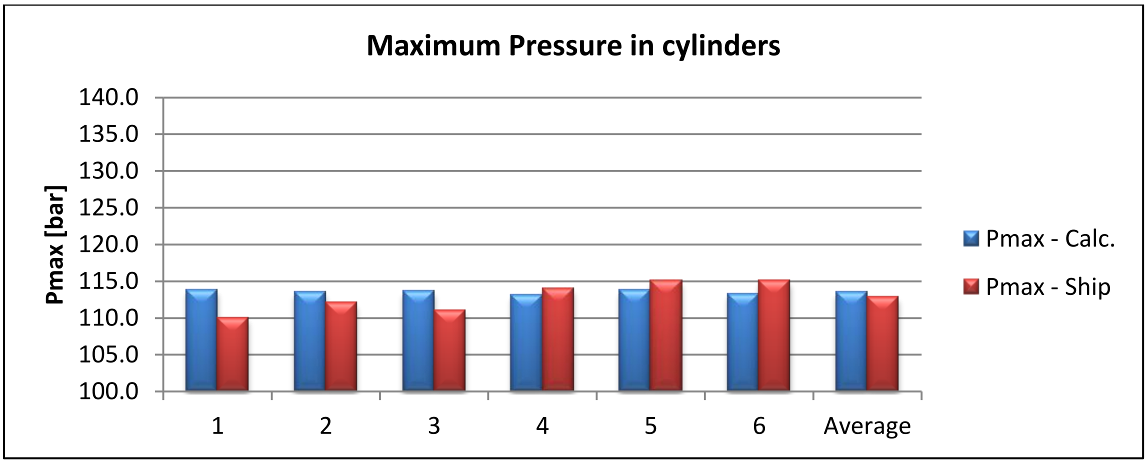


Fig. 8. 61 Pmax, Ship reported vs. "MOTHER" calculated

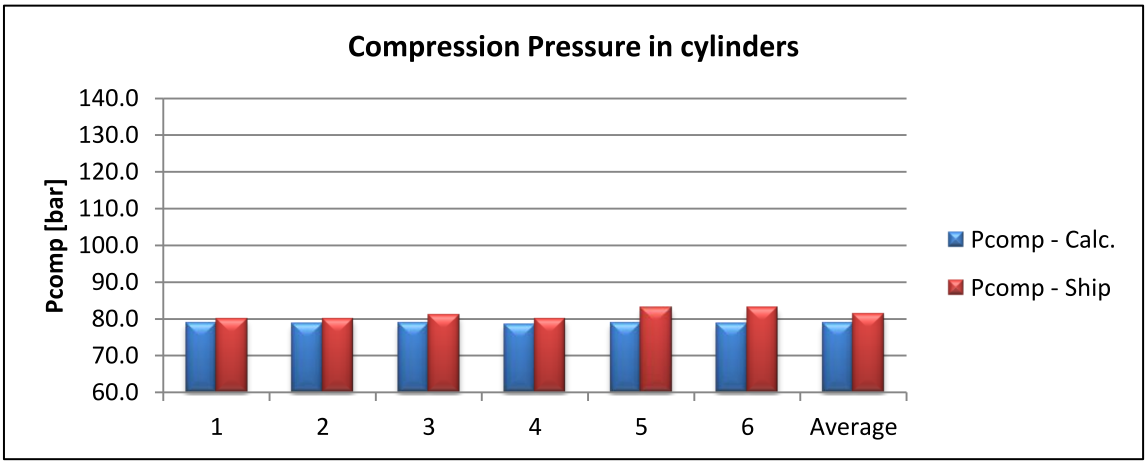


Fig. 8. 62 Pcomp, Ship reported vs. "MOTHER" calculated

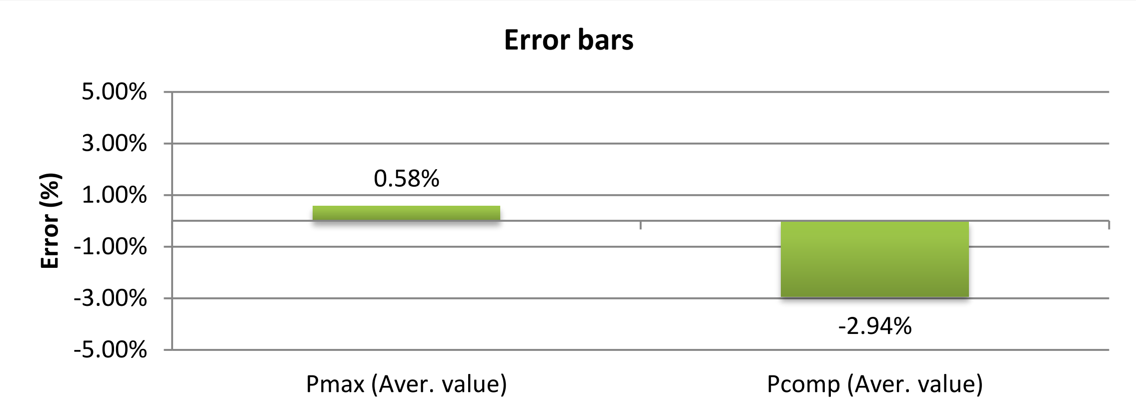


Fig. 8. 63 Error Bars for average values

Table 8.62 shows the figures` 8.54 & 8.55 data.

Value / Cyl.		**1**	**2**	**3**	**4**	**5**	**6**	**Aver.**
Pcomp – Calc.	bar	79.0	78.7	78.9	78.5	79.0	78.6	78.78
Pcomp – Ship	bar	80.0	80.0	81.0	80.0	83.0	83.0	81.17
Error	%	*-1.25%*	*-1.63%*	*-2.59%*	*-1.88%*	*-4.82%*	*-5.30%*	*-2.94%*
Pmax – Calc.	bar	113.8	113.5	113.6	113.1	113.7	113.3	113.49
Pmax – Ship	bar	110.0	112.0	111.0	114.0	115.0	115.0	112.83
Error	%	*3.45%*	*1.29%*	*2.34%*	*-0.79%*	*-1.13%*	*-1.48%*	*0.58%*

Table 8. 62 Compression and maximum pressure per cylinder

The following comments could be made:

- Average compression pressure has been simulated with -2.94% error. The higher measured compression pressure compared to the calculated, can be justified by the adverse weather that faces the ship (Table 8.59). The ship is loaded and the apparent propeller slip ratio is 16.70%. However, the calculation error does not exceed the admissible error range of ±3%.
- Average combustion pressure has been simulated with the minimum possible error, i.e. 0.58%. However, MOTHER calculations show that the measured combustion pressures should have been quite higher than the attained pressures. Fuel quality is the prime suspect for quite lower combustion pressures.
- According to instructions provided by the office to the ship, the fuel used in the engine is a mixture of two fuels, i.e. ratio of 1/3 LSFO and 2/3 of HFO with low ignitability properties.
- The low ignitability of the HFO had been controlled by mixing the fuels and by advancing the ignition in the cylinders. The FQS position was in +0.8deg position, compared to other performance reports (refer to Chapters 8.5.1 to 8.5.7) were its position was +0.5deg.

- Cylinder`s no 1 measured combustion pressure, i.e. 110bar, was the lower measured pressure among the cylinders. The low combustion pressure could be justified by the following factors:
    - Fuel quality.
    - Possibly poor fuel atomizers. Given that the combustion pressure on October the 29^{th}, 2009 (Chapter 8.5.7) was lower than the calculated by MOTHER value, and there was not available any information regarding the most recent fuel injectors replacement, it is concluded that the low measured combustion pressures (29^{th} Oct. and 1^{st} Nov. 2009) could potentially have been caused by poor fuel atomizers.
- Cylinders no 5 and 6 had the higher measured compression pressures (i.e. 83bar) amongst the other cylinders. Although the compression pressure deviation from the mean pressure (i.e. 81.17bar) was less than 2bar, a possible cause of this deviation (given that the other units were operated with lower Pcomp) could have been a malfunction of the exhaust valve system. Such a statement could not stand, unless other factors, such as the exhaust gas temperature, the exhaust gas smoke color and various pressures of the exhaust valve system, result to the same conclusion, i.e. the exhaust valve system is defective and needs physical inspection. Returning back in the reported measured values on November the 1^{st}, 2009, it was noted that the exhaust gas temperatures of cylinders no 5 and 6, were 370°C and 365°C respectively (Appendix V, Chapter V.8), whereas the average exhaust gas temperature was 365.2°C. Thus, the exhaust gas temperature seems to be normal (deviation less than 5°C). Furthermore, other information regarding the exhaust gas smoke and pressures of the exhaust valve system were not available. Consequently, it is not possible to give a clear answer to that question.The aforementioned facts were mentioned in order to prove that the engine performance is affected by various factors. In case of operational disturbance every possible cause should be evaluated and checked in conjunction with physical inspection and measured performance data, in order to improve the engine`s performance, prevent from future damages and most importantly to work cost effectively.

Table 8.63, summarizes the calculated data and provides a comparison with the measured data.

M/E RPM: 104.22, LI POS.: 6.5			SHIP	CALC.	*ERROR (%)*
PRESSURES	SCAVENGING AIR PR.	bar	2.512	2.390	*-4.86%*
	Pmax (mean value)		112.83	113.49	*0.58%*
	Pcomp (mean value)		81.17	78.78	*-2.94%*
	IMEP		-	13.16	-
	BMEP		-	12.43	-
	PRESSURE DROP IN AIR COOLER		0.01128	0.0113316	*0.48%*
	PRESSURE IN EXH. RECEIVER		-	2.13	-
TEMPERATURES	M/E AIR COOLER IN	^{o}C	150.0	148.9	*-0.73%*
	M/E AIR COOLER OUT (SCAV. AIR TEMP.)		47.0	47.9	*1.91%*
	AIR COOLER COOLANT WATER		29.0	29.0	*0.00%*
	EXH. GAS AFT. CYLINDERS (mean value)		365.2	313.3	*-14.20%*
	EXH. GAS AFT. TURBINE		335	-	-
MISC.	T/C speed	RPM	15100	15014	-0.57%
	COMPRESSOR PRESSURE RATIO	-	-	2.368	-
	COMPRESSOR CORRECTED AIR FLOW	m^3/s	-	11.040	-
	COMPRESSOR EFFICIENCY	-	-	0.824	-
	TURBINE EFFICIENCY	-	-	0.823	-
	AIR COOLER EFFICIENCY	-	0.8512	0.8532	*0.23%*
	IHP	kW	-	13.16	-
	ENGINE MECH. EFFICIENCY=BMEP/IMEP	-	-	0.944	-
	BHP (mean value)	kW	4552.67	4688.1	*2.97%*
	SFOC	g/kW-h	-	169.7	-

Table 8. 63 Simulation results for 1st Nov. 2009

The Brake Horse Power calculations` results are included in Table 8.64.

BHP [kW]	Power	Engine Load (% MCR)	MOTHER Calculation	Engine Load (% MCR)	Error (%) from 'MOTHER' calc. power
MEPIC Reported	4474.0	58.1%			4.79%
MEPIC Calculated	4632.0	60.2%			1.21%
SULZER Reported	4474.0	58.1%			4.79%
MEPIC Calculation (Regression)	4692.0	60.9%	4688.1	60.88%	-0.08%
Acc. L.I.xRPM	5188.8	67.4%			-9.65%
Acc. Pme	5043.7	65.5%			-7.05%
Acc. Propeller Law	5442.3	70.7%			-13.86%
Average	4849.5	63.0%	4688.1	60.88%	-3.33%

Table 8. 64 BHP

According to Table 8.63, the following figures are attached.

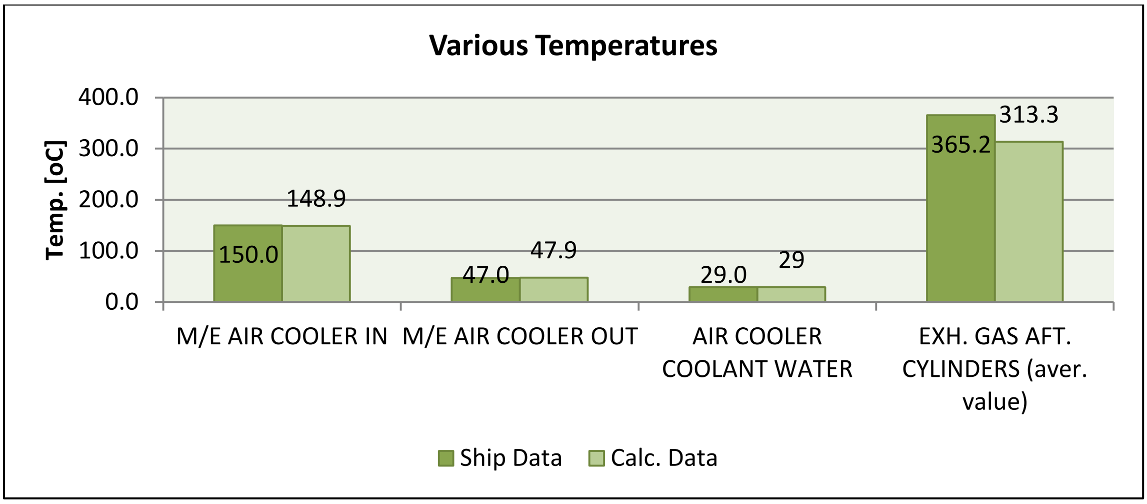


Fig. 8. 64 Various temperatures

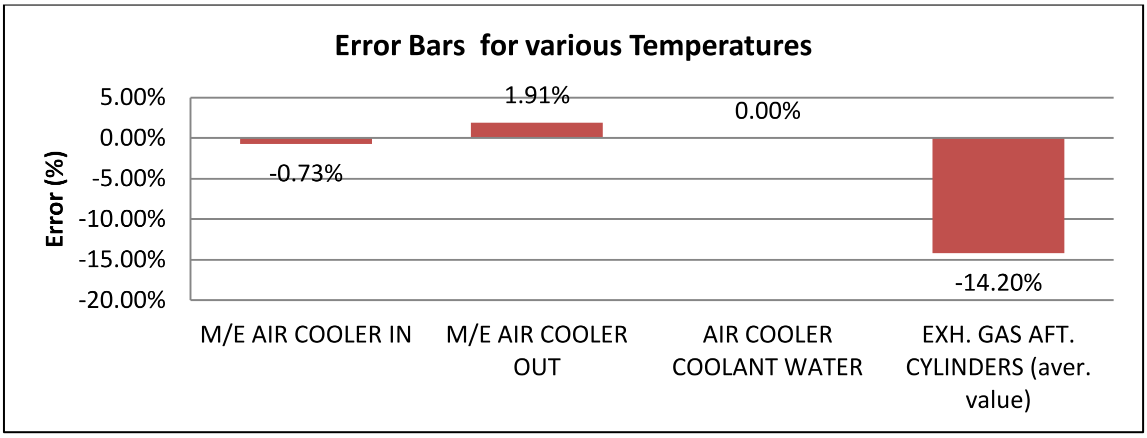


Fig. 8. 65 Error bars for various calculated temperatures

Comments:

- The engine operation at 1th November, 2009, had a particular interest because it was burning a mixture of two fuels.
- Calculated scavenge air pressure (i.e. 2.39bar) was considerably lower (-4.86%) than the measured scavenging pressure (i.e. 2.512bar). Increased scavenge air pressure was caused by the turbocharger and the prime suspect for such pressure, is the turbocharger`s nozzle ring which may be fouled or choked. It is noted that the turbocharger had been operating for 17400 hours after the last overhauling. According to manufacturer, the turbocharger is cleaned at frequent intervals before the major overhauling has taken place.
- According to MOTHER calculations, both scavenge air pressure and compression pressure should have been lower than the measured values.

- It was reported that turbocharger rotational speed had been fluctuated from 15000rpm to 15200rpm and load indicator from 6.4 to 6.5. The average rotational speed, i.e. 15100rpm, has been inserted in Table 8.63. Fluctuations of T/C`s rotational speed could have been caused by the fluctuations of the load indicator which is responsible for the fuel injected per cycle per cylinder. Thus, the fuel mass injected per cycle was not constant; when an increased fuel mass was injected, the engine load was raised producing more exhaust gas resulting in increased exhaust gas flow rate to the turbine. The increased exhaust gas flow rate caused the increased turbocharger rotational speed. Consequently, the increased measured scavenge air pressure is justified by the increase in the T/C`s speed. The question at this stage of evaluation is what had caused the load indicator fluctuation.
- According to Table 8.59, the loaded ship (according to reported displacement, Table 8.59) faced adverse weather (propeller slip 16.70%), which means the ship`s resistance was increased. The load change is expressed by the load indicator position, which was fluctuated from 6.4 to 6.5. Given that the speed setting in the Engine Control Room was set at 105rpm and that the engine load was changing due to weather conditions, load indicator position was changing in order to maintain the engine speed constant. In case the speed was lower than 105rpm, the load indicator position was increased (automatically by the electronic governor) in order to supply the engine with adequately more fuel, resulting in increasing the engine speed. Exactly the opposite was happened in case the engine speed was higher than 105rpm. It was reported that the engine speed had been fluctuated from 103.7rpm to 105.5rpm. Consequently, the load indicator fluctuation reason has been identified and justified.
- The average engine speed, i.e. 104.22rpm, was used for simulating the engine operation in MOTHER, because this simulation model is not able to simulate transient engine operation.
- Given that the engine load was changing in time, it is unknown at which exactly engine speed and T/C speed the measurements were carried out. For that reason, the average engine speed and average T/C speed was reported in M/E Full Performance Report. These average values were used as input in the thermodynamic model in MOTHER interface.
- Air cooler related temperatures (air inlet and scavenging air temperature) were simulated with error less than 2%.
- Air cooler efficiency, as calculated from the measured values, is high (i.e. 0.8512) and 0.23% lower than those calculated by MOTHER (i.e. 0.8532). The air cooler efficiency has been calculated according to Shop Test measured a/c efficiency.
- The power output calculations results` and MOTHER calculated BHP, are showed in Table 8.63. MOTHER calculated brake output converges with “MEPIC calculated” power and “Regression calculation”, with calculation error less than 2%.
- The brake output was low, as long as the engine load. Even though the ship faced adverse weather, the low load indicator setting and engine speed setting, resulted in maintained the engine load low and eliminating the chances of reaching the torque rich region in Loading Diagram. The engine operating point is illustrated at the Load Diagram in Fig. 8.59.

- The energy balance results, as calculated by MOTHER, are illustrated in Fig. 8.60.

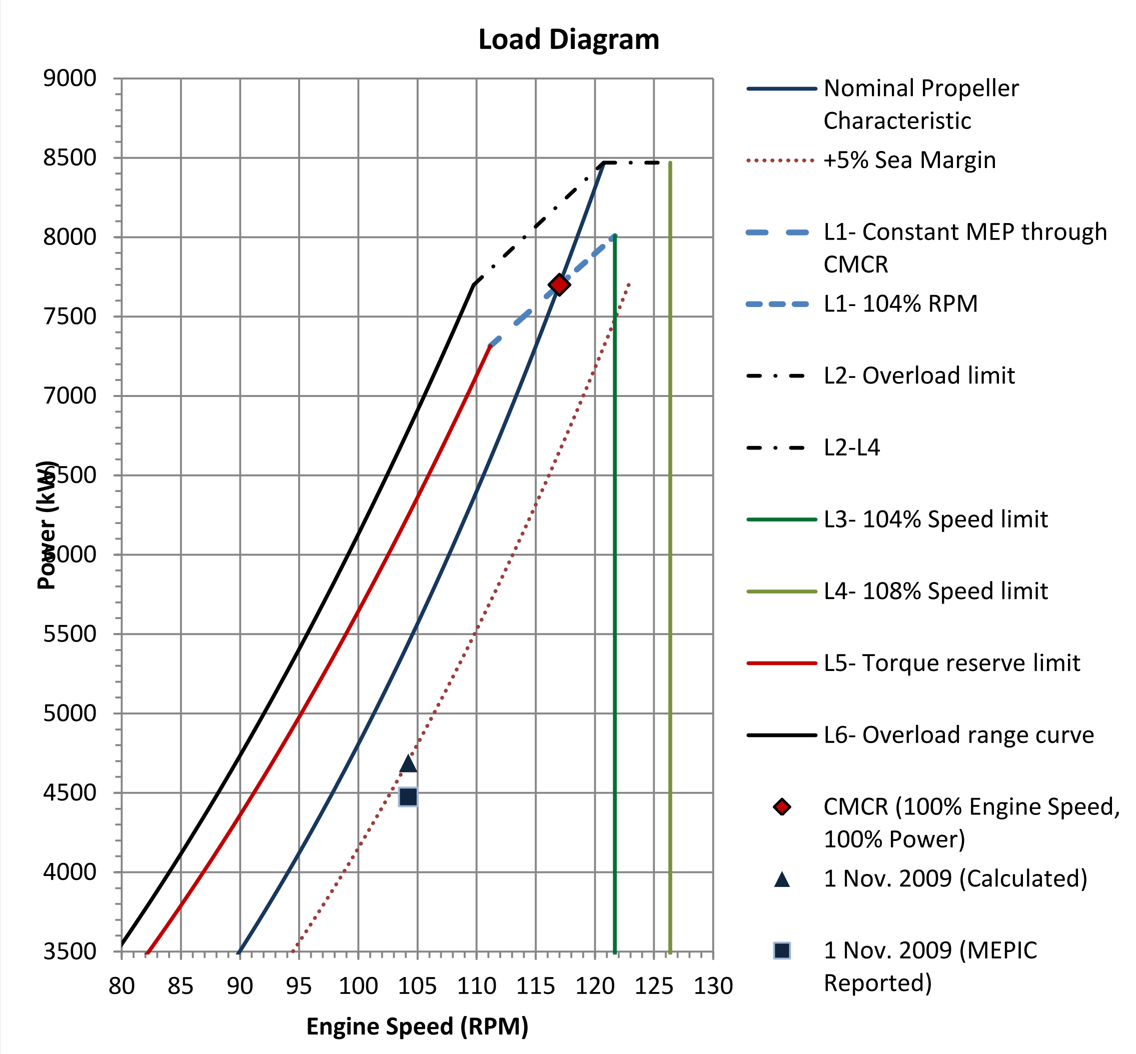


Fig. 8. 66 Load Diagram

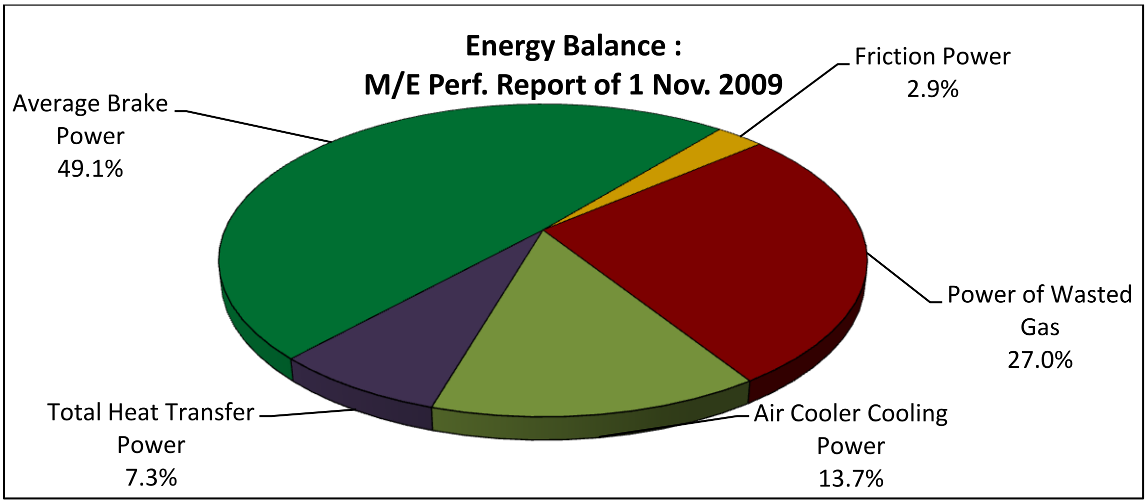


Fig. 8. 67 Energy Balance

Power	Energy Balance	
	[kW]	[%]
Power of Wasted Gas	2571.7	27.0%
Air Cooler Cooling Power	1311.5	13.7%
Total Heat Transfer Power	694.6	7.3%
Average Brake Power	4688.1	49.1%
Friction Power	272.9	2.9%
Total Power	9538.8	100.0%

Table 8. 65 Energy Balance values

8.5.8.2. ISO corrections` results

This sub-section provides the results of the ISO correction methodology.

	(Pmax – Pcomp)	Actual	Corrected	Shop Test		
				Acc Pcomp	Acc M/E RPM	Acc Power
Pmax 1	30.0	**110.0**	114.6	119.6	125.0	123.1
Pmax 2	32.0	**112.0**	116.6	119.6	125.0	123.1
Pmax 3	30.0	**111.0**	115.6	120.8	125.0	123.1
Pmax 4	34.0	**114.0**	118.6	119.6	125.0	123.1
Pmax 5	32.0	**115.0**	119.6	123.1	125.0	123.1
Pmax 6	32.0	**115.0**	119.6	123.1	125.0	123.1
Pmax Average	**31.7**	**112.8**	**117.4**	**121.0**	**125.0**	**123.1**
Pmax / $Pmax_{MCR}$	-	81%	84%	87%	89%	88%

Table 8. 66 Corrected Pmax

	(Pmax – Pmax av.)	Actual	Corrected	Shop Test		
				Acc Pscav	Acc M/E RPM	Acc Power
Pcomp 1	-2.8	**80.0**	84.6	81.3	85.8	83.0
Pcomp 2	-0.8	**80.0**	84.6	81.3	85.8	83.0
Pcomp 3	-1.8	**81.0**	85.6	81.3	85.8	83.0
Pcomp 4	1.2	**80.0**	84.6	81.3	85.8	83.0
Pcomp 5	2.2	**83.0**	87.6	81.3	85.8	83.0
Pcomp 6	2.2	**83.0**	87.6	81.3	85.8	83.0
Pcomp Average	**-**	**81.2**	**85.7**	**81.3**	**85.8**	**83.0**

Table 8. 67 Corrected Pcomp

<u>Comments</u>:

- Average corrected combustion pressure (i.e. 117.4bar) was lower than those calculated according to Shop Test figures. ISO correction methodology shows that the measured combustion pressure should have been higher than those in Table 8.66.
- The deviation between combustion and compression pressure, was balanced among the cylinders.
- Average corrected compression pressure (i.e. 85.7bar) was considerably higher than those pressures calculated according to scavenge air pressure and power figures from Shop Test (Table 8.67). Although it seems that the calculated compression pressure according to M/E rpm (i.e. 85.8bar), is normal, it should be stated that this calculation contributes in understanding the upper limit of compression pressure.
- Cylinders no 5 and 6 had considerably high compression pressure compared with the other measure, according to both three calculations based on Shop Test figures.
- By comparing the results from MOTHER simulation and from ISO correction methodology, the following could be stated:
    - Both MOTHER simulation and ISO corrections resulted in lower compression pressures for all the cylinders.
    - Both MOTHER simulation and ISO corrections resulted in higher combustion pressures, even though the calculated by MOTHER combustion pressure was slightly higher than the measured onboard.

CHAPTER 9

CONCLUSIONS

9.1 Introduction

The creation of a simulation model is a demanding procedure which depends on available geometric and operational data. The simulation of 6RTA48-T main engine was completed successfully by using the MOtor THERmodynamics software. This Chapter intends to provide the final remarks regarding the simulation procedure and to summarize the results of the engine operation simulation.

The concluding remarks are categorized in three distinctively representative sections, i.e. the achieved targets, the comparison between "MOTHER" and ISO method, and the simulation of the engine operation at service .

Finally, several ideas for further work to be done in order to continue this work, are also provided.

9.2 Achieved Targets

The work presented in this Thesis, is summarized to the following points:

i. The 6RTA48-T simulation model was set-up and calibrated according to Shop Test records (Chapters 4, 5 and 6).
ii. Shop Test and Sea Trials simulations were successful according to the results provided in Chapters 6 and 7.
iii. The validated engine simulation model was used for creating reference conditions at any operating point between 50% and 100% of MCR, to compare with recorded service data (Chapter 8).

9.3 "MOTHER" versus ISO correction methodology

"MOTHER"`s thermodynamic simulation model is capable of predicting the engine operation in any operating point, provided that the required input data are available. On the other hand, ISO correction methodology transforms the recorded service data to the same ambient conditions with Shop Test, in order for the two sets of data to be compared.

The main difference between these methods lies in the idea that Shop Test data comprises the reference engine performance, consequently, they serve for the assessment and evaluation during the engine`s lifetime. In contradiction, MOTHER does not require Shop Test records for evaluating the engine`s service data, because the MOTHER-calculated performance parameters comprise the reference data.

The utilization of the ISO methodology, combined with MOTHER simulation results`, resulted in common considerations regarding the engine performance, except for two cases (i.e. Chapters 8.5.4 (17 May 2008) and 8.5.5 (8 Mar 2009)), where the results were contradicting.

Although ISO correction methodology is effective (despite its vulnerabilities), and highly appreciated by Ship Operators, this Thesis demonstrated and justified (without underestimating the ISO methodology), that there is another method for fulfilling the same purpose of evaluating M/E performance. Finally, "MOTHER" goes one step further because of its capability to predict the engine performance in various loadings and conditions.

No matter what evaluation methodology an operator follows, unless the engine`s condition is evaluated by in-site inspection, it is not possible to make certain and safe conclusions about its condition. Thus, the evaluation procedure gives a valuable hint, but it cannot stand alone.

9.4 Simulating the engine operation at service

Amongst the available M/E Performance Reports, it was decided to simulate specific cases where the actual engine operation had particular interest. The engine operational simulation covered the following cases:

- Ship in loading and ballast condition
- Calm sea and winds
- Moderate sea state
- Normal engine operation
- Unbalanced pressures in cylinders and increased wear of major engine parts
- Operation with mixed heavy fuel
- Engine operation after overhauling and replacement of major parts (i.e. piston, piston rings, piston crown, liner, injectors)

Headquarters' technical staff instructions to the Chief Engineer intended to eliminate the chance of operating the engine in high loads; thus, the possibility of the engine`s operating point reaching the "torque rich" region in loading diagram was reduced, thereby reducing potential engine damages through overloading. For that reason, they had advised the engine`s speed setting not to exceed the 110.8rpm and the turbocharger speed to be lower than 16800rpm. In case of adverse and severe weather, the engine`s speed was decreased in order to prevent overloading and to protect the shafting system.

MOTHER confirmed that the engine was operated in loads close to the 'light' propeller curve, as per Technical staff instructions. The loading diagram, in Fig. 9.1, shows the operating points for the selected cases, as calculated by MOTHER.

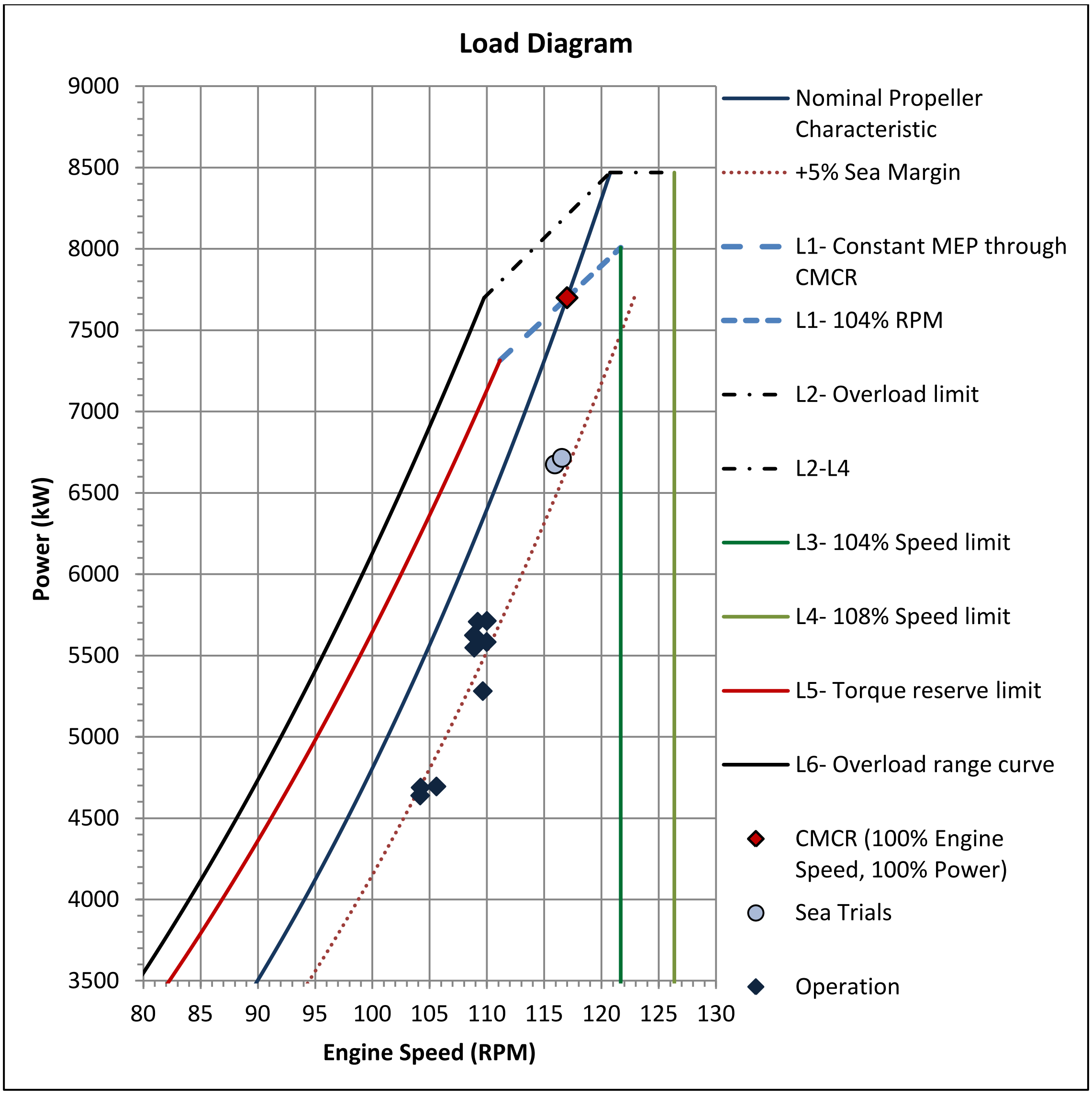


Fig. 9. 1 **Loading Diagram (MOTHER-calculated data)**

All parameters resulted in reasonable residuals, except for exhaust gas temperature, which indicated an issue. Either the factors reported in Chapter 8 were responsible for that deviation, or the impact of some of them (such as HFO) on the exhaust gas temperature was beyond MOTHER`s computational capability. Moreover, although the fuel quantity had been corrected for the usage of the ISO fuel (NCV=42700kJ/kg), the actual impact of various parameters on the exhaust gas temperature could not be simulated by MOTHER exactly; but they can be inferred based on the residuals of all performance parameters. Figure 9.2, illustrates the residuals of the exhaust gas temperature.

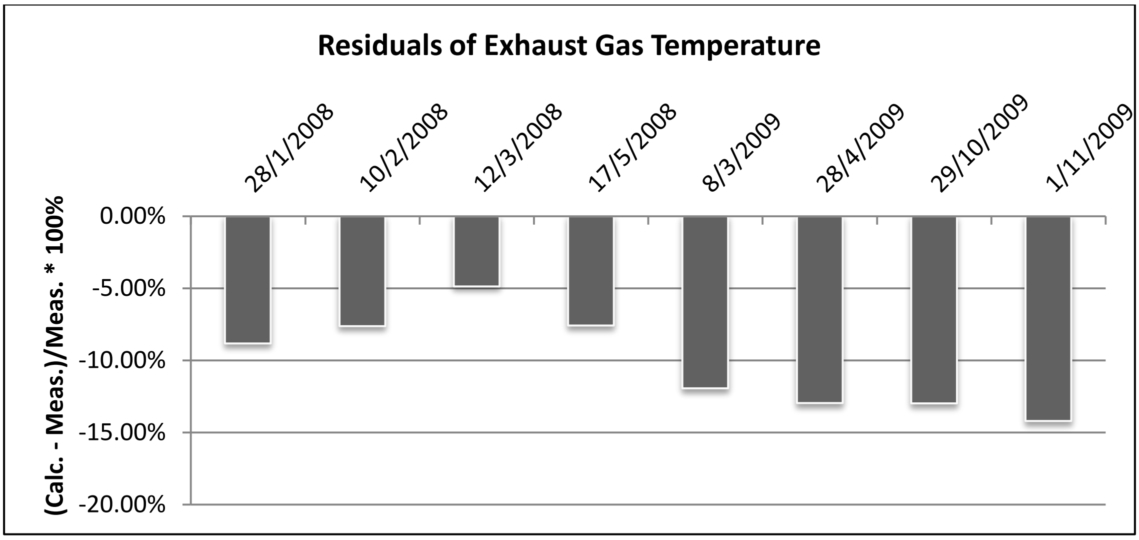


Fig. 9. 2 Residuals of Exhaust Gas Temperatures

The simulations` results have been discussed in Chapter 8. The Air Cooler and the cylinder block are two major parts of the engine, where condition and performance affects the overall main engine performance. Main concluding remarks regarding these parts and the HFO`s influence on the engine performance, are provided below.

9.4.1 Air Cooler

MOTHER has proved that the A/C was operated with decreased efficiency compared to those attained at Shop Test and Sea Trials (Fig. 9.3). Such a behavior was rational because fouling of both air and sea side and changes in sea water flow rate, had affected its efficiency, and the scavenge air temperature, as well. Figure 9.3, illustrates the Air Cooler Efficiency with respect to Brake Horse Power, as calculated by MOTHER.

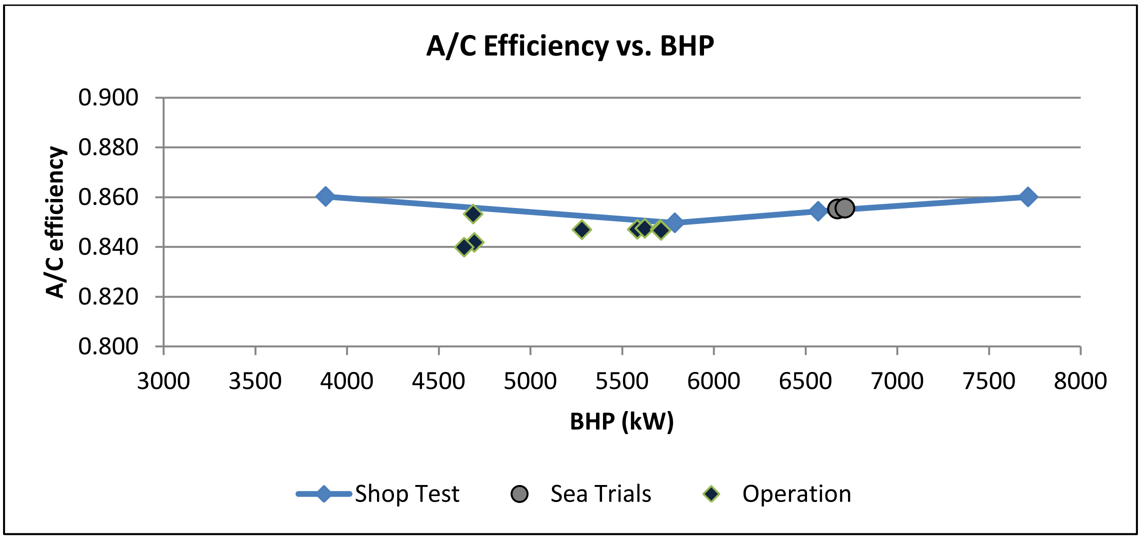


Fig. 9. 3 A/C Efficiency vs. BHP

The most representative case was discussed in Chapter 8.5.4.1 (17 May 2008), where the residual of scavenge air temperature (Fig. 9.5) and A/C efficiency (Fig. 9.4) indicated a malfunction. Scavenge air temperature had been improved to normal value by the time the crew increased sea water flow rate to the A/C, according to mail exchanges. Figures 9.4 and 9.5, illustrate the residuals of Air Cooler Efficiency and Scavenge Air Temperature.

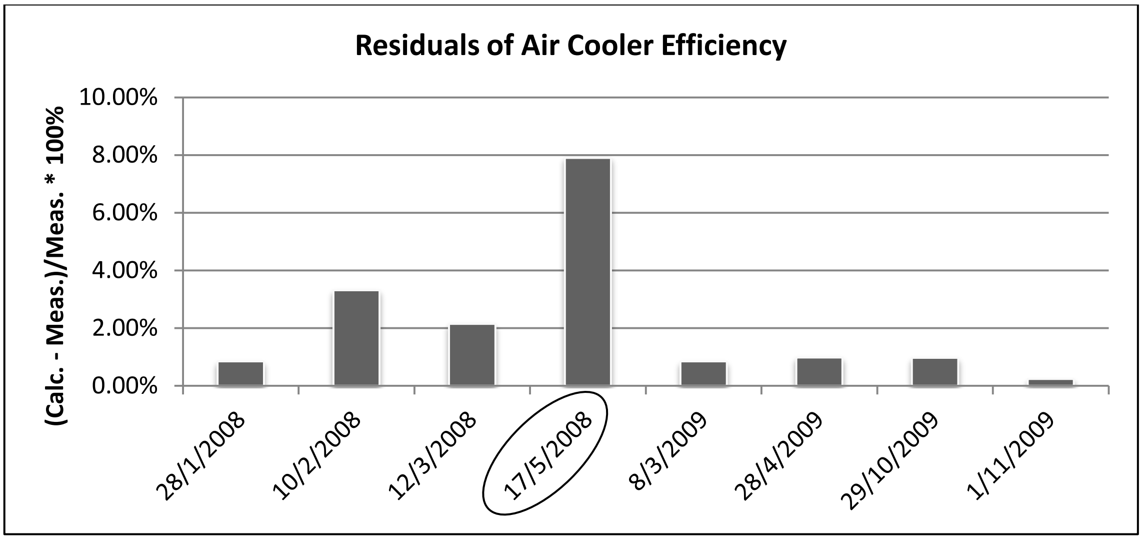


Fig. 9. 4 Residuals of A/C Efficiency

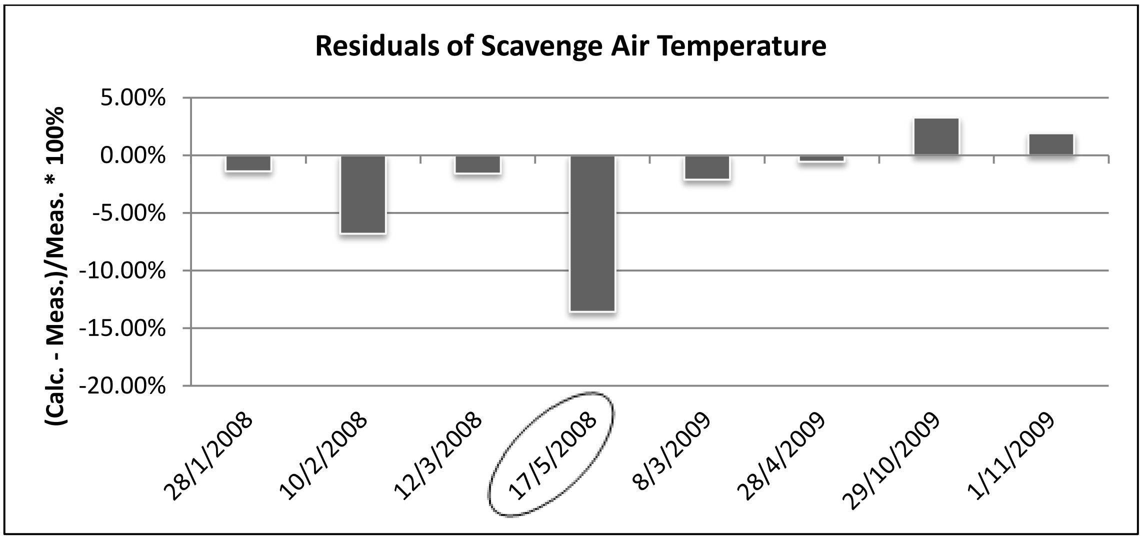


Fig. 9. 5 Residuals of Scavenge Air Temperature

9.4.2 Cylinder(s)

Pressure residuals (either Pcomp, or Pmax) in most cases indicated an issue which have been justified according to the available information (i.e. mail exchanges). Specifically, simulation model results, showed that cylinder no. 1, faced high compression pressure since early 2008. Consequently, the model`s calculations confirmed an actual problem of the engine, which was developed and resulted in serious damages (due to excessive wear) to cylinder liners, piston skirts and piston rings. Although, MOTHER is unable to predict directly such failures, the investigation of the possible causes and the available information, resulted in confirming some of them indirectly.

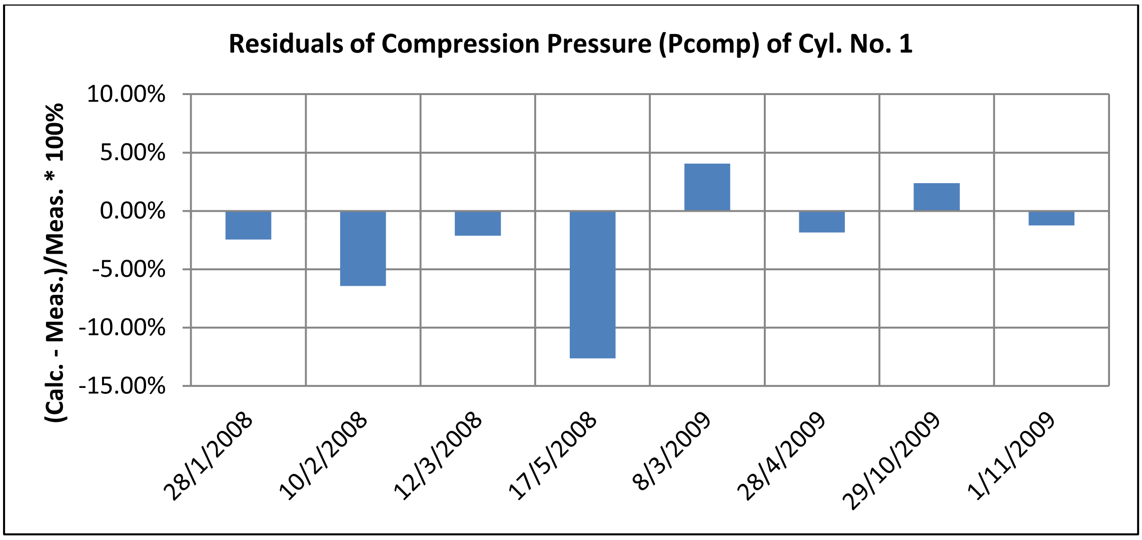


Fig. 9. 6 Residuals` history of cylinder`s no. 1 Compression Pressure

Besides cylinder`s no. 1 problem, the following cases confirmed considerations owed to MOTHER calculations:

- Low combustion pressure (Pmax) in cylinders no. 5 and 6 has been caused by poor injectors (Chapters 8.5.3.1) according to simulation of "M/E Performance of 10 Feb. 2008" and available mail exchanges. Figure 9.7, illustrates the combustion pressure`s residuals before and after the injectors` replacement.

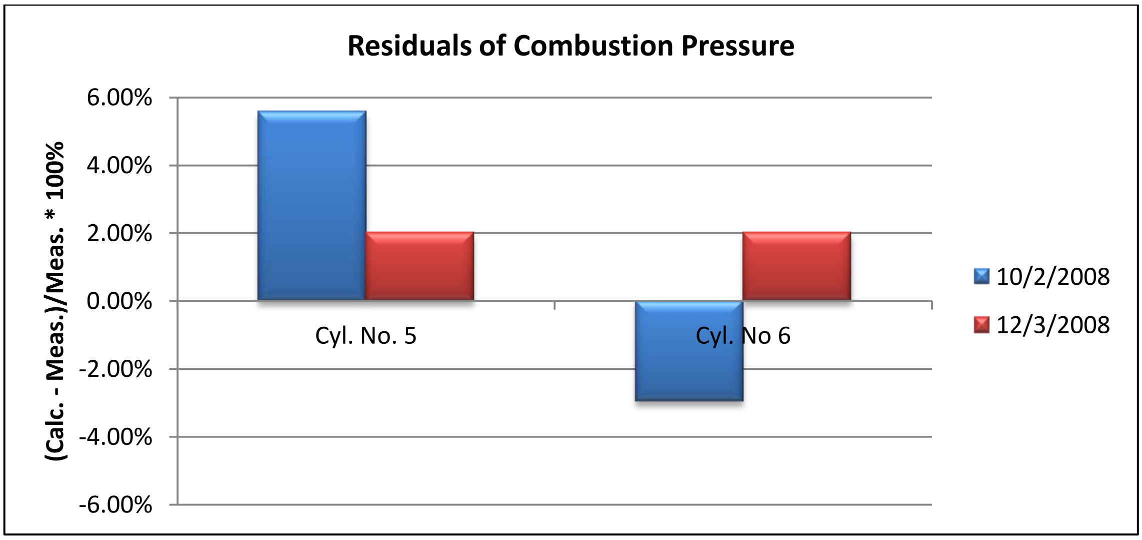


Fig. 9. 7 Residuals of Pmax

- Overhauling of unit no. 4 (liner and piston skirt were replaced), before the 28 Apr. 2009, had improved cylinder`s compression pressure (Pcomp, Chapter 8.5.6.1). Figure 9.8, illustrates residuals of compression pressure from three consecutive simulations.

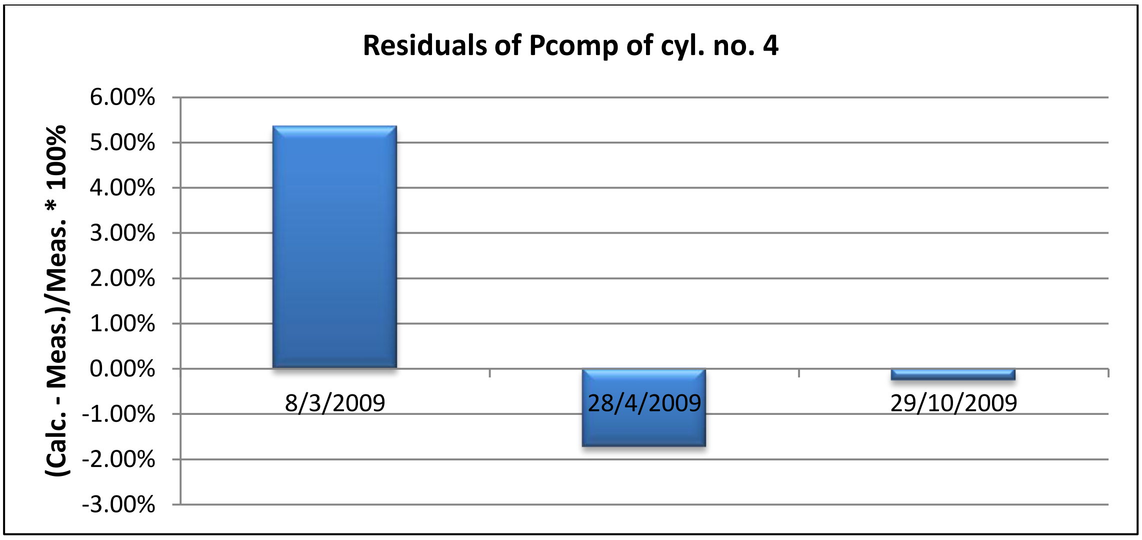


Fig. 9. 8 Residuals of Pcomp for cyl. no. 4

- Overhauling of unit no. 6 (liner, piston crown and piston skirt were replaced) after the 8th Mar 2009, had slightly improved the cylinder`s compression pressure (Pcomp, Chapter 8.5.6.1). Figure 9.9, illustrates the residuals of compression pressure.

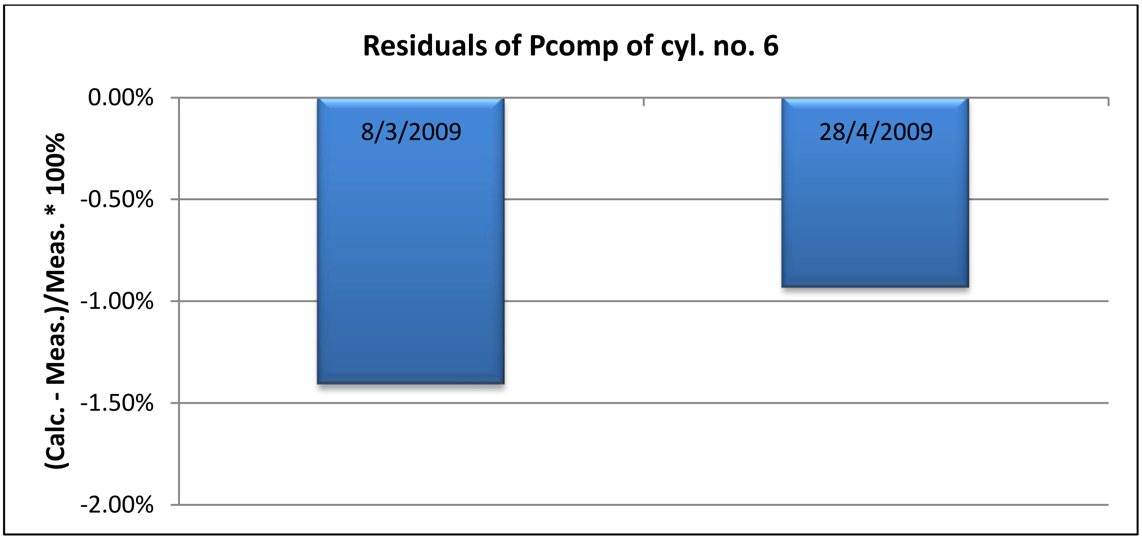


Fig. 9. 9 Residuals of Pcomp of cyl. no. 6

Finally, throughout the evaluation procedure possible causes were provided, in case of deviation between recorded and calculated data; yet, some of them were confirmed according to the available information and some others were not confirmed due to the lack of information.

9.4.3 Fuel Oil

Fuel Oil`s quality affects the combustion profile and the energy produced during thermodynamic cycle. Furthermore, heavy fuel oil requires specialized treatment, which depends on its quality, in order the engine to be operated safe and without potential damages. Fuel oil analysis is of utmost importance and determines the onboard treatment.

Regarding the MOTHER-simulated cases, one of them, had particular interest, i.e. 1 Nov. 2009, because the engine was burning a mixture of two fuels (i.e. one third of HFO and two thirds of LSFO with low ignitability properties, Chapter 8.5.8). The minimal residual of 0.58% (Fig. 9.10) of average combustion pressure shows that the ship`s reported pressures were normal. Consequently, the impact of the 'faulty' fuel had been altered onboard, and the main factor that contributed to this (besides mixing the fuels), was the FQS position, which had been taken into account by MOTHER.

On the other hand, the evaluation of M/E Performance Report of 29 Oct. 2009, showed that if the FQS position had been advanced by 0.3deg, the combustion pressures would have been higher. Thus, the residual of 1.38% was justified. Figure 9.10, illustrates the residuals of average combustion pressure.

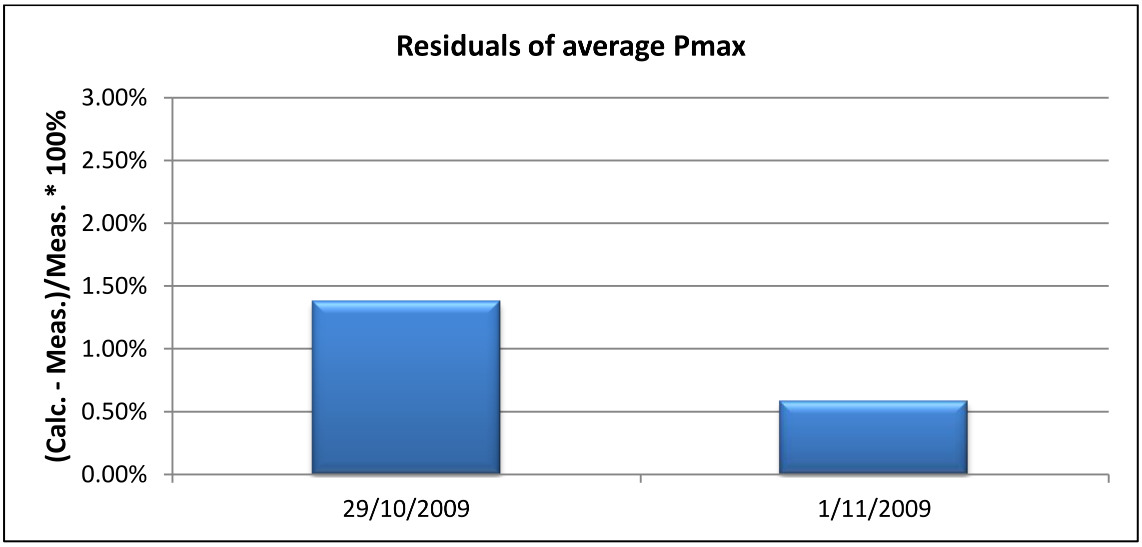


Fig. 9. 10 Residuals of average Pmax

9.5 Recommendations for future work

Herein are provided several ideas for continuation of work in this field.

- Accurate and reliable onboard brake power measurements.
- Governor simulation in MOTHER interface.
- VIT and FQS simulation in MOTHER interface.
- Simulation of the engine operation under low loads (below 50% of MCR), where the Blower is activated due to low scavenge air pressure.
- Simulation of the engine operation in the torque rich region of the Load Diagram.

APPENDIX I:
SHOP TEST

ENGINE PARTICULARS

Model: 2-stroke, single acting, airless injection, direct reversible, crosshead type, exhaust gas turbocharged marine diesel engine, DIESEL UNITED – SULZER 6RTA48-T

Manufacturer	:	DIESEL UNITED, LTD.
Classification Society	:	NK
Nos. of cylinder	:	6
Cylinder Bore	:	480 mm
Piston Stroke	:	2000 mm
Maximum Continuous Rating	:	7700kW x 117 r/min
Maximum Continuous Pressure	:	142 bar
Mean Effective Pressure	:	18.2 bar
Mean Piston Speed at Rated Output	:	7.80 m/sec
Direction of Rotation (Ahead)	:	Clockwise seen from after side
Firing Order (Ahead)	:	1-5-3-4-2-6-(1)
Crank Shaft	:	Semi-built up type
Piston Cooling	:	Lubrication oil
Cylinder Cooling	:	Fresh water
V.I.T. (Variable Injection Timing)	:	Equipped
Exhaust Gas Turbocharger	:	ABB-Turbo System United Ltd. Type TPL 73-B12 x 1 set With plain bearing & force lub. oil

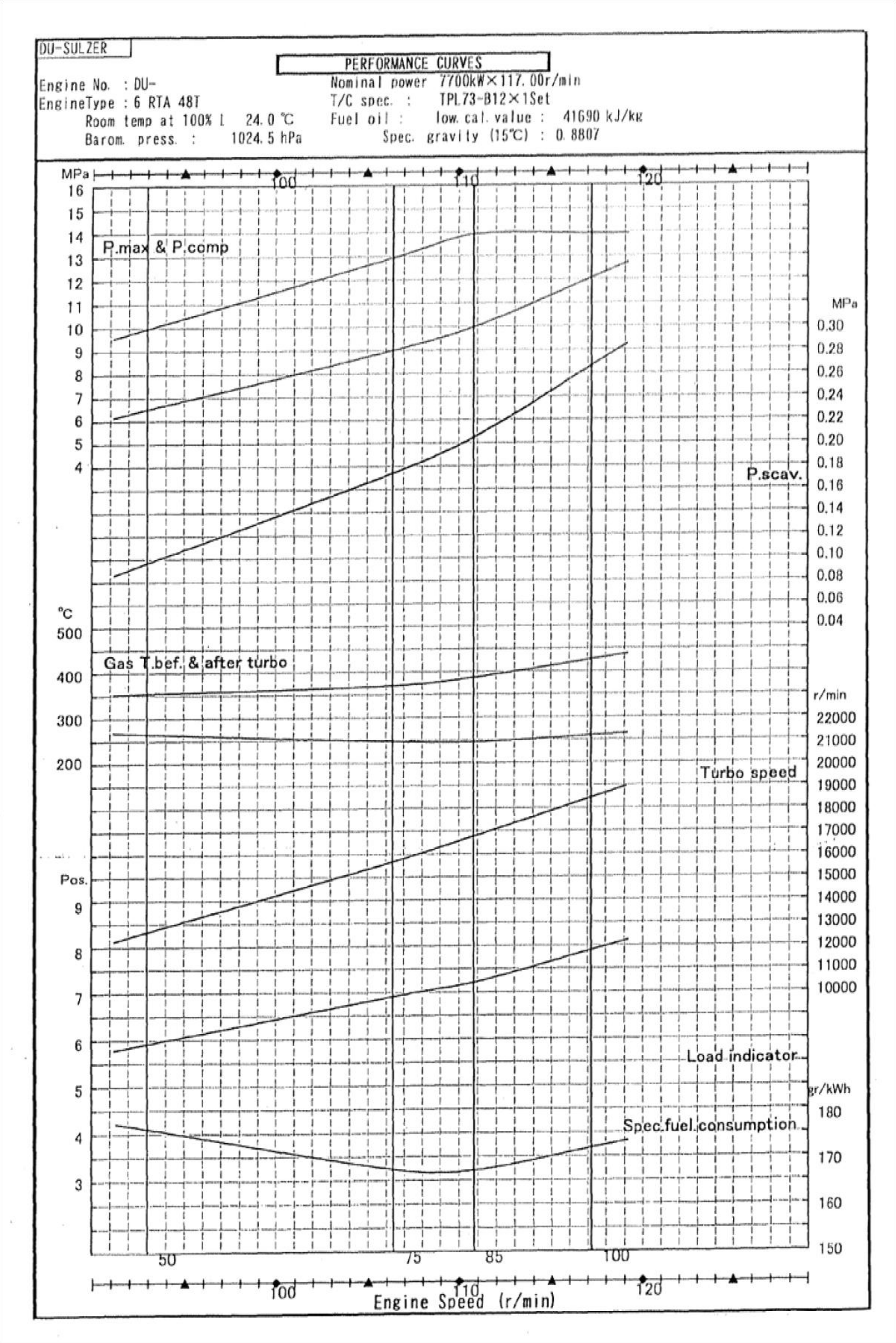


Fig. I. 1 Performance curves

SHOP TRIAL RESULTS

Load		**%L**	**50**		**75**		**85**		**100**	
Time		H:m	9:25		10:10		11:10		12:10	
General data										
Engine Power plan/act.		kW	3850	3892	5775	5807	6545	6577	7700	7749
Eng. Speed plan/act.		r/min	92.9	93	106.3	106.4	110.8	110.8	117	117.2
Brake force / Pme		Ton/Mpa	56.9	1.156	74.2	1.508	80.7	1.640	89.9	1.827
Gov. Terminal / Load Ind.			59	5.9	68	6.8	72	7.2	79	7.9
VIT		Pos.	-0.5		+1.1		+1.5		-1.0	
FQS			-0.5		-0.5		-0.5		-0.5	
Fuel consp. /measured time		kg / min	293	25	415	25	751	40	569	25
Fuel consp. / spec. consp.		kg/h / g/kW-h	703.2	176.4	996.0	167.5	1126.5	167.2	1365.6	172.1
ISO spec. consp.								167.2		
Cyl. Lubri-cator	ACM moter	r/min	169		235		264		303	
	Screw/Man 691	Pos.	4C	0	4C	0	4C	0	4C	0
	Total lub. oil feed	kg/h	9.28		12.91		14.50		16.64	
	Spec. cyl. oil consp.	g/kW-h	2.38		2.22		2.20		2.15	
Scavenge Air										
Auxiliary Blower			off		off		off		off	
T/C speed		r/min	12700		15700		16800		18500	
Barometric press./room temp.		hPa/°C	1025.5	22.5	1025.5	23.0	1025.0	23.0	1024.5	24.0
Blower filter loss		mmAq	25		50		67		85	
Scavenge air press. /local		MPa	0.097		0.175		0.206		0.265	
P. drop across A/C		mmAq	60		80		85		96	
Temp. bef. blower			22.7		23.5		25.8		25.8	
Temp. aft. blower		°C	100		140		155		185	
Temp. aft. A/C			32		39		40		44	
Exhaust Gas										
Exh. Back press		mmAq	50		135		165		240	
Temp. bef. Turbine - elec.		°C	358		374		387		427	
Temp. aft. Turbine - elec.			265		246		244		258	
	Average		312		316		324		354	
	1		313		318		328		356	
	2		304		308		315		342	
Temp. aft. cylinder	3	°C	300		303		313		343	
	4		305		310		318		343	
	5		323		326		335		369	
	6		325		330		337		371	
kW=F(ton)xN(r/min)x0.7355										

Table I. 1 Shop Trial Results Part 1

Load			**%L**	**50**		**75**		**85**		**100**	
Time			H:m	9:25		10:10		11:10		12:10	
Cylinder Pressures											
Pmax/Pcomp	Average		MPa	9.97	6.48	13.00	9.0	13.95	9.97	13.98	120.8
	1			9.9	6.4	13.0	9.0	13.9	9.9	14.0	11.9
	2			10.0	6.5	13.0	9.0	14.0	10.0	14.0	12.1
	3			10.1	6.6	13.0	9.0	14.0	10.0	14.0	12.2
	4			9.9	6.4	12.9	9.0	13.9	9.9	14.0	12.1
	5			9.9	6.4	13.0	8.8	13.9	10.0	13.9	12.0
	6			10.0	6.6	13.1	9.1	14.0	10.0	14.0	12.2
System											
Air spring press.			MPa	0.71		0.71		0.70		0.70	
Coolant press.	Cylinder			0.45		0.45		0.45		0.45	
	Piston/ Air Cooler			0.40	0.27	0.40	0.25	0.40	0.24	0.40	0.24
Coolant temperature	Cyl.	In	°C	75		74		74		74	
		Out 1-4 cyl		82, 83, 82, 82		83, 84, 83, 84		84, 85, 83, 83		84, 85, 84, 84	
		Out 5-6cyl		81, 82		82, 82		83, 84		83, 84	
	Piston	In		41.8		41.9		41.7		41.7	
		Out 1-4 cyl		58, 57, 58, 57		60, 58, 60, 58		61, 59, 61, 58		62, 60, 61, 59	
		Out 5-6cyl		58, 56		60, 58		61, 59		62, 60	
	T/C oil inlet			56		71		77		84	
	A/C	Inlet		21		21		21		21	
		Outlet		25		29		30		30	
Oil press.	Bearing/Crosshead		MPa	0.40	1.13	0.40	1.15	0.40	1.15	0.40	1.15
	T/C			0.20		0.20		0.18		0.19	
Oil temp.	inlet		°C	41.8		41.9		41.7		41.7	
Fuel	Press/Inlet temp.		MPa/°C	0.39	25	0.38	25	0.37	25	0.35	25

Table I. 2 Shop Trial Results Part 2

SETTING TABLE Sheet A

FUEL PUMPS: equipped with VIT			Firing order: Ahead: 1 – 5 – 3 – 4 – 2 – 6 – (1)						
Fuel pump setting indicat. Pos. 8 FQS & VIT cyl. in pos. 0	Direction of rotation		Ahead						Astern
	Cyl. No.		1	2	3	4	5	6	1
Effective Stroke begin	Suction valve closes at a plunger stroke "a"	(Plan) 7.63mm	7.62	7.65	7.61	7.64	7.62	7.61	7.67
	Suction valve closes before TDC	°	5.0	5.1	5.0	5.1	5.0	4.7	0.9
Effective Stroke end	Spill valve opens at a plunger stroke "b"	(Plan) 33.44mm	33.47	33.47	33.45	33.46	33.46	33.48	33.42
	Spill valve opens after TDC	°	12.0	11.9	11.9	11.8	11.9	12.3	16.1
Effective plunger stroke		(Plan) 25.81mm	25.85	25.82	25.84	25.82	25.84	25.87	25.75
Injection angle		°	17.0	17.0	16.9	16.9	16.9	17.0	17.0
Fuel linkage is limited at load indicator pos.: 8.1			Safety valve opening pressure: 115 MPa						

Table I. 3 Setting Table

APPENDIX II:
SEA TRIAL RESULT

ENDURANCE TEST

A part of the Sea Trials` measurements, is attached at the following tables. Data related to Diesel Generators and tests of other machinery, have not been included.

LOAD		**NOR (85%)**	**NOR (85%)**	**NOR (85%)**	**AVR.**
TIME		22:00	23:00	0:00	-
LOAD INDICATOR		7.50	7.33	7.48	7.4
T/C REV.	RPM	17000	16800	17100	16967
SCAVENGING AIR	MPa	0.20	0.20	0.21	0.203
SHAFT REV.	RPM	115.94	116.55	116.62	116.37
BHP	kW	6637	6553	6627	6605.7
SPEED	kt	-	-	-	-
SEA CONDITION		Rough	Rough	Rough	-
DIRECTION OF WIND		-	-	-	-
VELOCITY OF WIND	m/s	-	-	-	-
KIND OF FUEL		HEAVY FUEL OIL			

Table II. 1 Endurance Test

LOAD				**NOR (85%)**	**NOR (85%)**	**AVR.**
TIME				22:00	0:00	
M/E MONITOR CENTRAL CONTROL CONSOLE	MAIN AIR RESERVOIR	NO.1	MPa	2.30	2.02	2.16
		NO.2		2.50	2.60	2.55
	CONTROL AIR			0.75	0.76	0.755
	JACKET COOL FW. IN.			0.31	0.32	0.315
	COOL S.W. IN			0.19	0.17	0.18
	M/E START AIR			2.3	2.0	2.15
	L.O. IN			0.405	0.405	0.405
	CROSSHEAD L.O. IN			1.22	1.20	1.210
	SCAVENGING AIR			0.20	0.21	0.205
	SHAFT REV.		rpm	115.94	116.55	116.25
	T/C REV.			17000	17100	17050
	LOAD INDICATOR			7.50	7.33	7.42
	VIS. CON. UNIT VISCOSITY		cSt	13	13	13.00
	VIS. CON. UNIT TEMP.		^{o}C	124	123	123.5
	F.Q.S. POSITION			1.2	1.2	1.2
	D/G L.O. IN	NO.1	MPa	-	-	-
		NO.2		0.45	0.45	0.45
		NO.3		-	-	-
GROUP – 6	M/E EXH. GAS CYL. OUT. AVR		^{o}C	339	343	341
	M/E EXH. GAS OUT CYL	NO.1		345	347	346
		NO.2		326	326	326
		NO.3		325	331	328
		NO.4		325	332	329
		NO.5		351	36	356
		NO.6		362	364	363
	M/E EXH. GAS T/C IN			396	400	398
GROUP – 4	M/E L.O. IN		^{o}C	45	45	45
	M/E T/C L.O. OUT			79	80	79.5
GROUP – 2	M/E JACKET COOL F.W. IN			75	76	75.5
	M/E JACKET COOL F.W. CYL. OUT	NO.1		86	86	86
		NO.2		85	85	85
		NO.3		86	85	85.5
		NO.4		85	86	85.5
		NO.5		86	86	86
		NO.6		86	86	86
GROUP – 3	M/E PISTON COOL L.O. CYL OUT	NO.1	^{o}C	63	62	62.5
		NO.2		62	62	62
		NO.3		63	63	63
		NO.4		61	61	61
		NO.5		64	64	64
		NO.6		61	61	61
		AVR.		62.3	62.2	62.3
GROUP – 6	M/E AIR COOLER AIR OUT		^{o}C	47	48	47.5
GROUP – 2	M/E COOL S.W. IN			25	25	25
GROUP – 4	M/E THUST PAD L.O. IN			46	46	46

GROUP – 10	D/G LO. IN	NO.1	°C	45	43	44
GROUP – 11		NO.2		63	63	63
GROUP – 12		NO.3		41	40	40.5
GROUP – 10	D/G HT F.W. OUT	NO.1		69	69	69
GROUP – 11		NO.2		75	75	75
GROUP – 12		NO.3		68	67	67.5
GROUP – 14	HEAVY F.O. SETT. TANK		°C	60	57	58.5
	HEAVY F.O. SERV. TANK			85	85	85
GROUP – 15	S.W. SERV. PUMP OUT			16	16	16
GROUP – 5	M/E F.O. IN		MPa	0.87	0.87	0.87
	JACKET IN		°C	76	76	76
	JACKET OUT	NO.1		87	87	87.0
		NO.2		85	85	85.0
		NO.3		86	86	86.0
		NO.4		84	84	84.0
		NO.5		88	87	87.5
		NO.6		87	87	87.0
		AVR.		86.2	86.0	86.1
	CROSSHEAD L.O IN		MPa	1.17	1.20	1.20
	EXH. VALVE AIR SPRING			0.75	0.67	0.7
	BEARING L.O. IN			0.34	0.39	0.4
	STARTING AIR			2.2	2.1	2.2
	CONTROL AIR			0.75	0.75	0.75
	CYLINDER COOL F.W. IN			0.32	0.34	0.3
	SCAVENGING AIR			0.22	0.22	0.22
	F.O. IN PRESS			0.83	0.83	0.8
	F.O. OUT PRESS			0.8	0.8	0.8
	THRUST BEARING FORE		°C	43	43	43
	SCAVENGING AIR			46	47	46.5
	F.O. IN TEMP.			115	115	115
TURBO CHARGER	BLOWER AIR FILTER MANO.		mm	60	60	60
	BLOWER AIR IN. (TEMPORARY)		°C	26.1	25.9	26.0
	EXH. GAS OUT.			240	240	240
	L.O. OUT			75	75	75
	L.O. IN		MPa	0.16	0.16	0.16
AIR COOLER	COOL S.W. IN		°C	24	24	24
	COOL S.W. OUT			32	32	32
	AIR IN			160	165	162.5
	AIR CLR. MANO.		mm	90	90	90

Table II. 2 Sea Trials Results

LOAD				**NOR (85%)**	**NOR (85%)**	**AVR.**
TIME				22:00	0:00	
	MAXIMUM PRESSURE	NO.1	MPa	13.9	13.8	
		NO.2		14.0	13.9	
		NO.3		13.9	13.8	
		NO.4		14.0	14.0	
		NO.5		14.0	13.8	
		NO.6		13.9	13.8	
		AVR.		13.95	13.85	
	COMPRESSION PRESSURE	NO.1	MPa	10.3	10.0	
		NO.2		10.1	10.2	
		NO.3		10.2	10.2	
		NO.4		10.1	10.1	
		NO.5		10.1	10.1	
		NO.6		10.2	10.2	
		AVR.		10.17	10.13	

Table II. 3 Sea Trials cylinders` pressures

APPENDIX III:

ENGINE LAYOUT

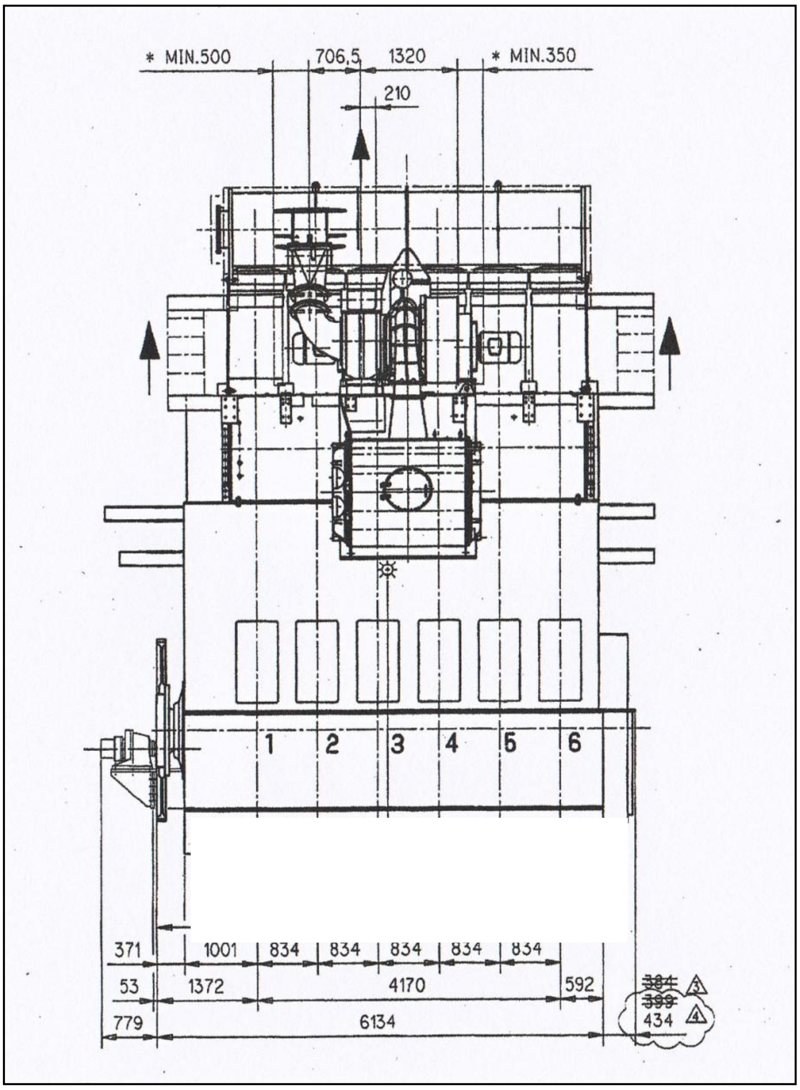


Fig. III. 1 Engine Outline

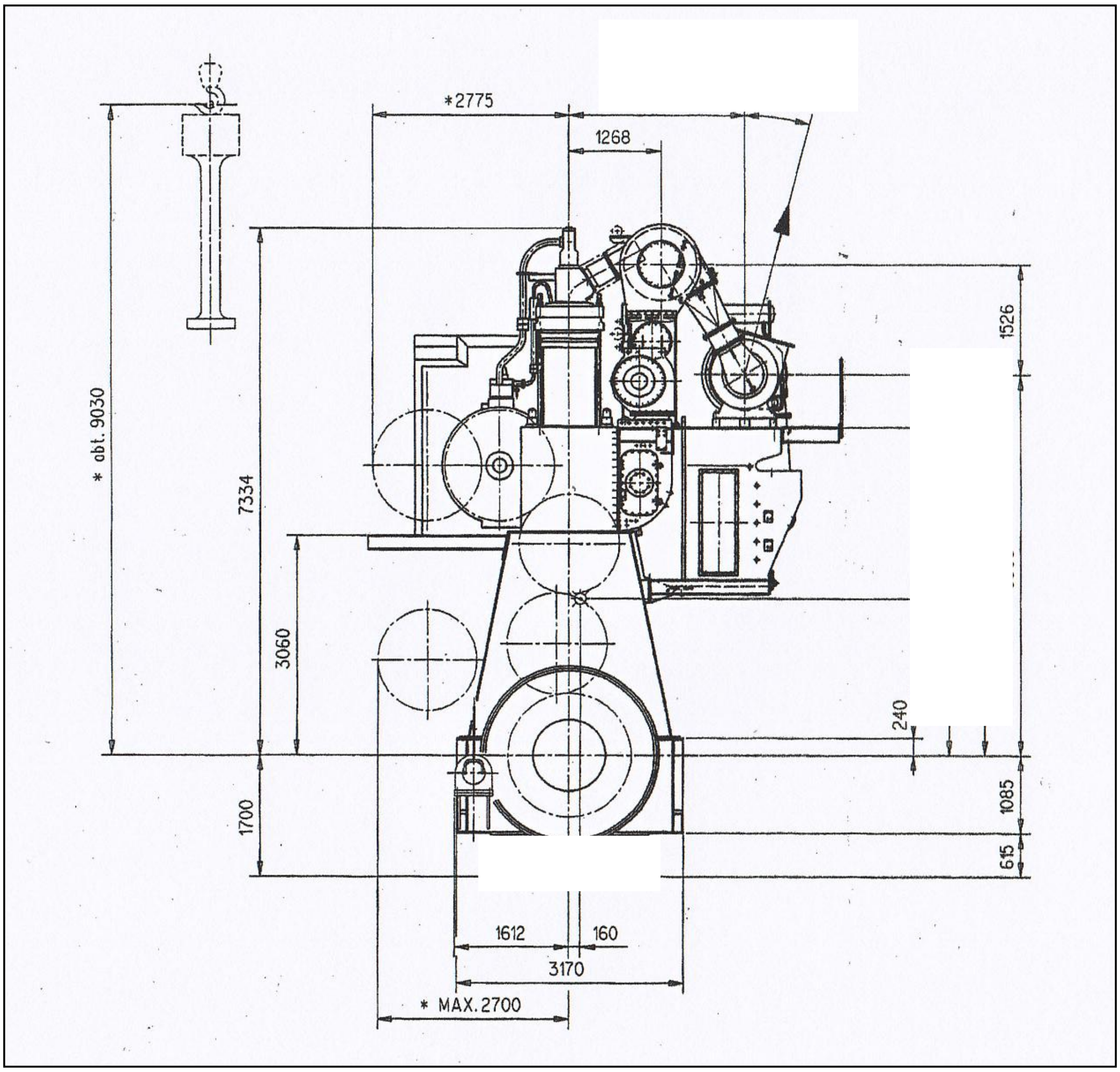


Fig. III. 2 Engine Outline (looking fore)

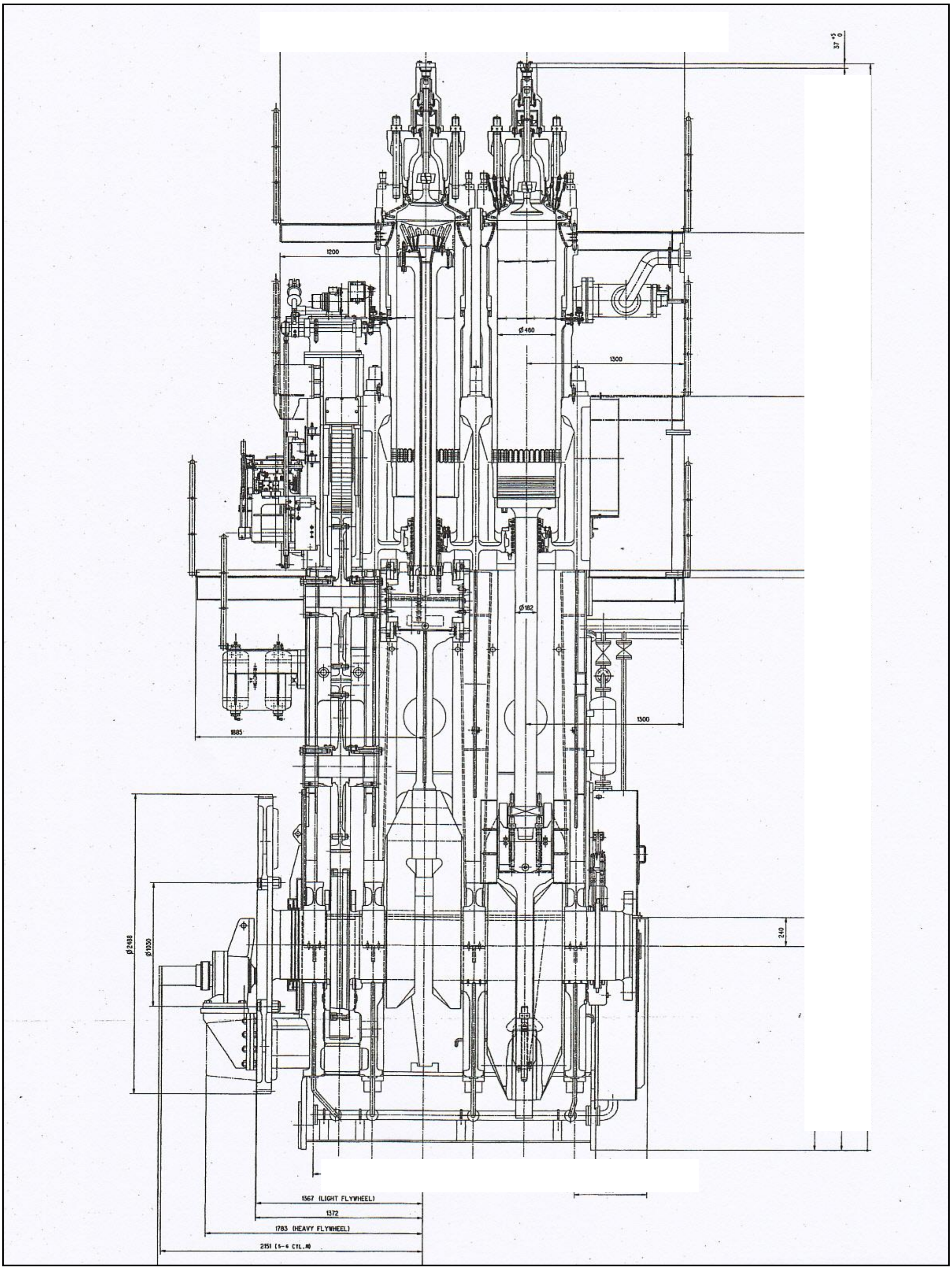


Fig. III. 3 Longitudinal Section

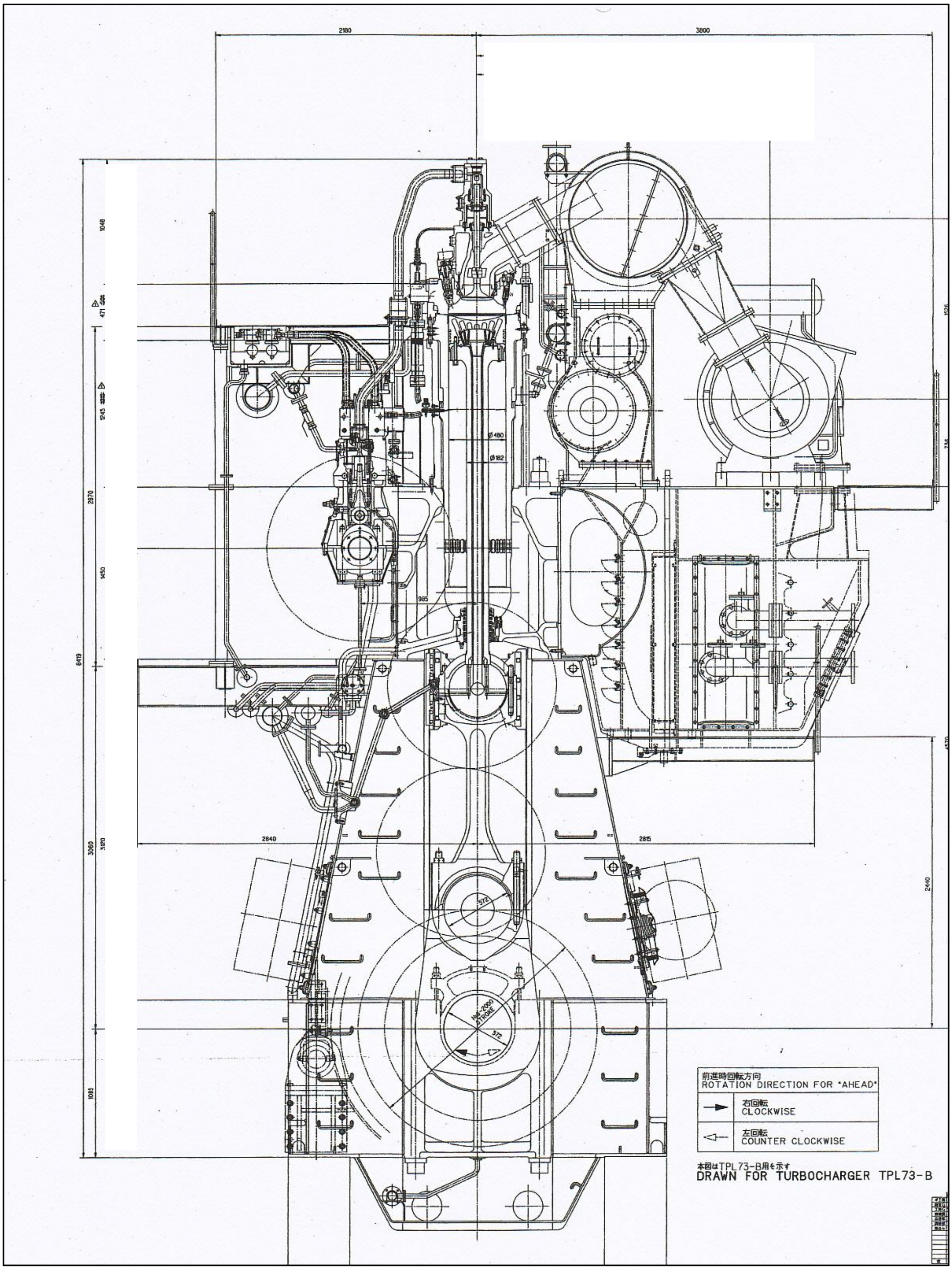


Fig. III. 4 Cross Section

Various Photos

Engine room

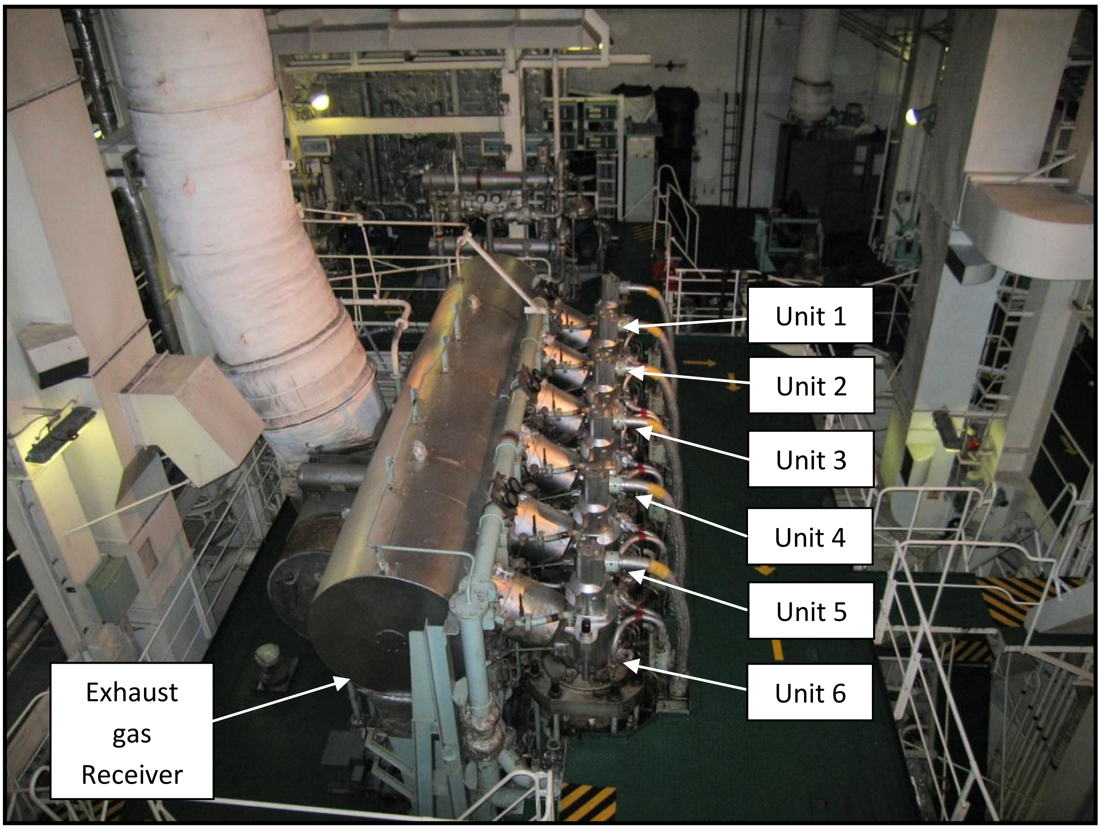


Fig. III. 5 The 6RTA48-T in the Engine Room (looking fore)

Cylinder head

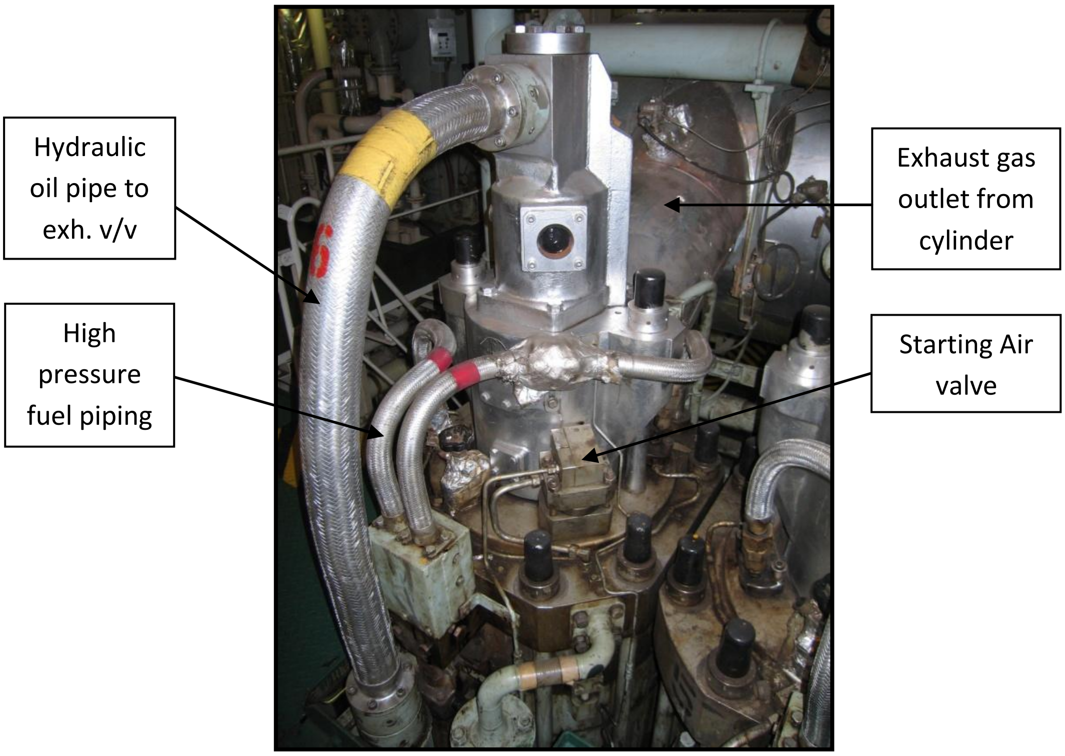


Fig. III. 6 No.6 Cylinder head and Exhaust valve

Note: Each cylinder head is similar to the one showed at Fig.6 above.

Exhaust valve

Fig. III. 7 Exhaust valve

Turbocharger

Fig. III. 8 T/C TPL73B-12

Liner lower part with scavenge ports

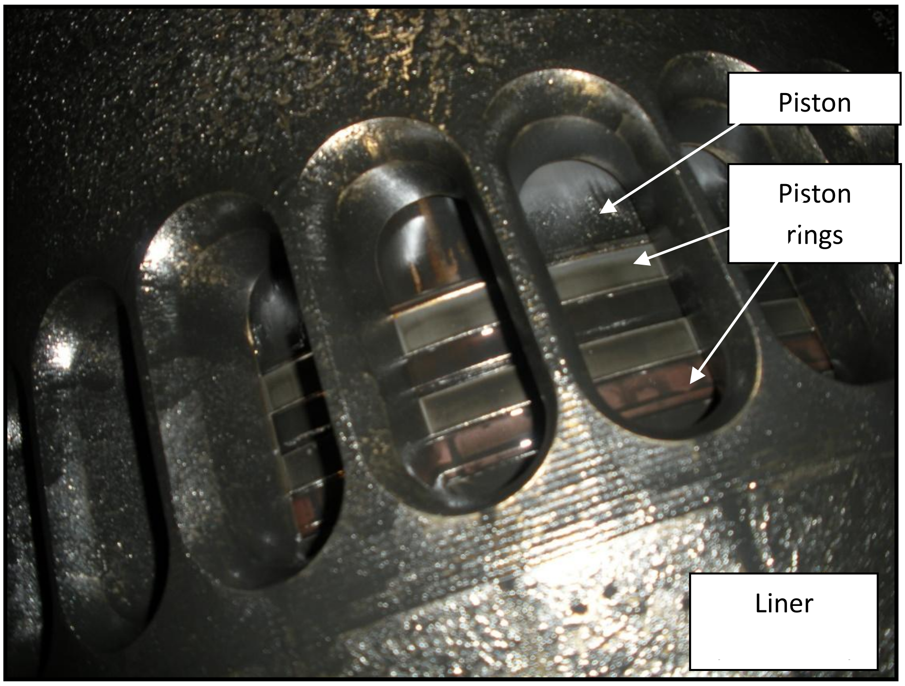


Fig. III. 9 Scavenge ports and piston rings

Scavenging space

Fig. III. 10 Scavenging space

Exhaust gas thermometers

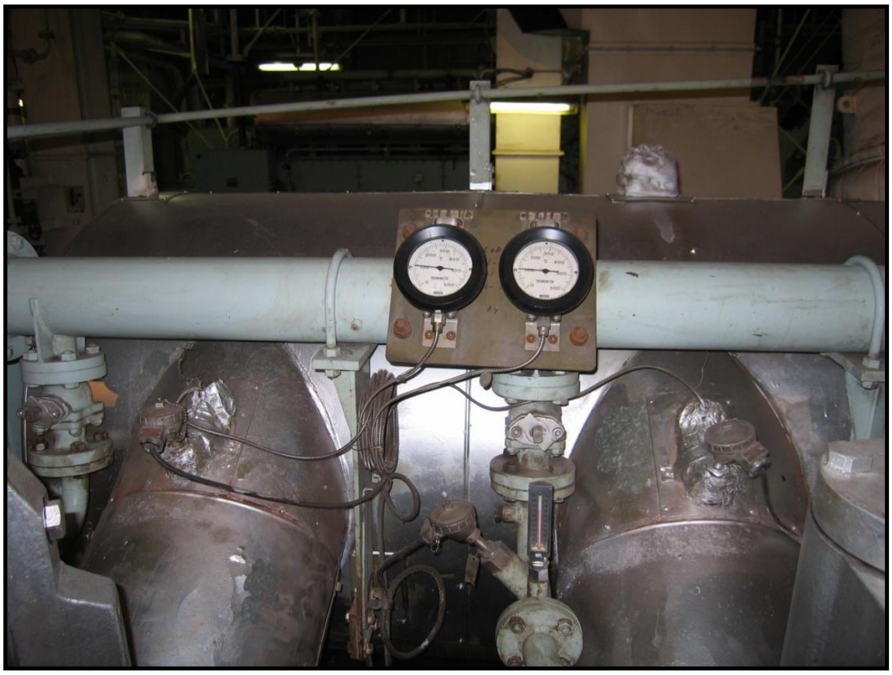


Fig. III. 11 Thermometers measuring the Exhaust gas temperature after cylinders

Note: The exhaust gas temperature is measured by analogue thermometers, like those is Fig.III.11, and by digital too. The digital thermometers` output is shown at the Engine Control Room.

Fuel injection pump with exhaust valve actuator

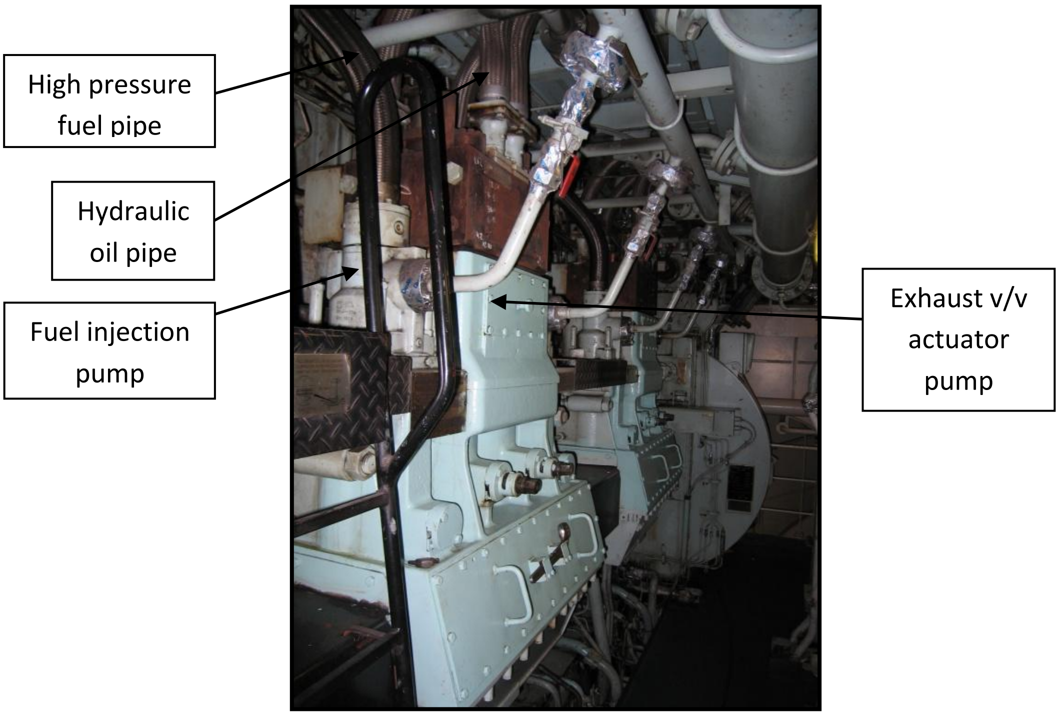


Fig. III. 12 Fuel injection pump and exhaust valve actuator pump

Engine`s aft end

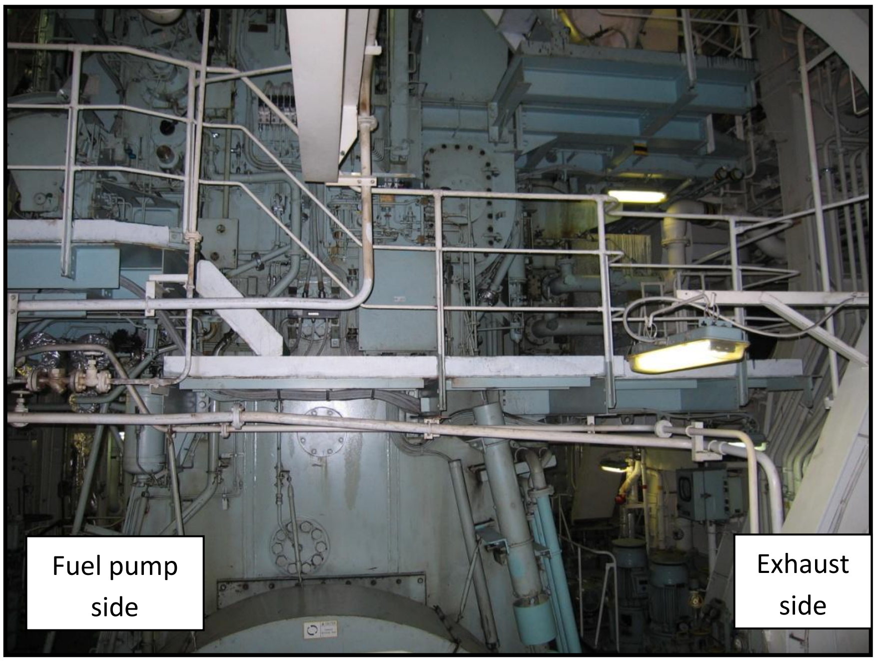


Fig. III. 13 The aft end of the engine

Engine Control Room

Fig. III. 14 Engine Control Room (ECR)

Bulk carrier`s deck view

Fig. III. 15 Ship`s deck view

APPENDIX IV:

VARIOUS CALCULATIONS

This Appendix`s purpose is to provide the details of several calculations that were made as they were essential for the engine`s simulation. These calculations are the following:

IV. 1. Fuel Injected per cyle

MOTHER requires the mass of fuel injected during one thermodynamic cycle in kilograms per cylinder. However, the consumed fuel, during Shop Tests, was measured in tones for a specified period of time (i.e. minutes). Consequently, the duration of one thermodynamic cycle is required in order to use the reported data for the calculation of the fuel injected per kilogram per cylinder.

From thermodynamics, we have: $\varphi = \omega \cdot t$, and the angular velocity is:

$$\omega = \frac{2\pi N}{60} \tag{1}$$

where,

φ	crank angle	$[\deg]$
ω	angular velocity	$[rad/s]$
t	thermodynamic cycle `s time	$[s]$
N	crank shaft rotational velocity	$[rpm]$

Thus, the crank angle equation can be transformed as follows:

$$\Delta\varphi\left({}^{o}\right) = \frac{2\pi N}{60}\Delta t\left(\frac{360}{2\pi}\right) = 6 \cdot N\left(rpm\right)\cdot \Delta t\left(\sec\right) \tag{2}$$

Finally, the time for one thermodynamic cycle, i.e. 360°, will be:

$$\Delta t\left(\sec\right) = \frac{360}{6 \cdot N} \tag{3}$$

The mass of fuel injected per cycle at Shop Test, can now be calculated as explained below:

- Firstly, the thermodynamic cycle`s time is calculated by the equation (3).
- According to the total reported consumed fuel mass, the consumption per minute is calculated by dividing the fuel mass with the duration of the measurement (refer to Table IV.1 at [5]).

- The consumption per second is then calculated (Table IV.1 at [6]).
- The fuel consumption per second per cylinder is calculated by dividing the product of [5] with the number of cylinders, i.e. 6 (Table IV.1 at [7]).
- Finally, the required from MOTHER, fuel mass injected per cycle, is calculated by multiplying the consumption per second with the duration of the cycle.

Table IV.1 shows the calculation of fuel mass injected per cycle for each engine load tested at Shop Test.

LOAD	N [1]	Δt [2]	Shop Test consum. [3]	Measured Time [4]	Cons. per min [5]=[3]/[4]	Cons. per sec [6]=[5]/60	Cons. per cylinder [7]=[6]/6	Fuel injected [8]=[2]*[7]
	[RPM]	[sec]	[kg]	[min]	[kg/min]	[kg/sec]	[kg/sec]	[kg]
100%	117.2	0.512	569	25	22.760	0.379333	0.06322	0.032365
85%	110.8	0.542	751	40	18.775	0.312917	0.05215	0.028240
75%	106.4	0.564	415	25	16.600	0.276667	0.04611	0.026002
50%	93.0	0.645	293	25	11.720	0.195333	0.03256	0.021006

Table IV. 1 Fuel injected per cycle (without correction for ISO fuel`s NCV)

The above fuel quantity needs to be corrected for the ISO fuel Low Calorific Value, because MOTHER uses the ideal fuel ("ISO fuel") for all the calculations. Table 4.4 shows the difference in calorific value between the fuel used at Shop Test and the ISO fuel. The compensation factor is calculated by the division of the low calorific value used at Shop Test, by the ISO fuel`s calorific value.

Shop Test Fuel Low Cal. Val. [1]	41690 kJ/kg
ISO Fuel Low cal. Val. [2]	42700 kJ/kg
Compensation factor [3]=[1]/[2]	0.976

Table IV. 2 Correction for ISO fuel

Table IV.3 depicts the corrected fuel mass injected per cycle, as MOTHER requires.

FINAL CALCULATION FOR FUEL INJECTED PER CYCLE			
LOAD	**Fuel injected per cycle (uncorrected) [kg]**	**Compensation factor for ISO fuel [-]**	**Fuel injected per cycle (corrected) [kg]**
	[1]	[2]	[3]=[1]*[2]
100%	0.032365	0.976	0.03160
85%	0.028240	0.976	0.02757
75%	0.026002	0.976	0.02539
50%	0.021006	0.976	0.02051

Table IV. 3 Mass of fuel injected per cycle (corrected)

IV. 2. Exhaust valve effective area and lift diagram

Exhaust valve area calculation

The physical geometry of the poppet valve and its location are shown in the figure 4.5 below, characterized by a lift L, above a seat at an angle ϕ, which has inner and outer diameters d_{is} and d_{os} respectively. The valve stem diameter d_{st}, partially obscures the aperture.

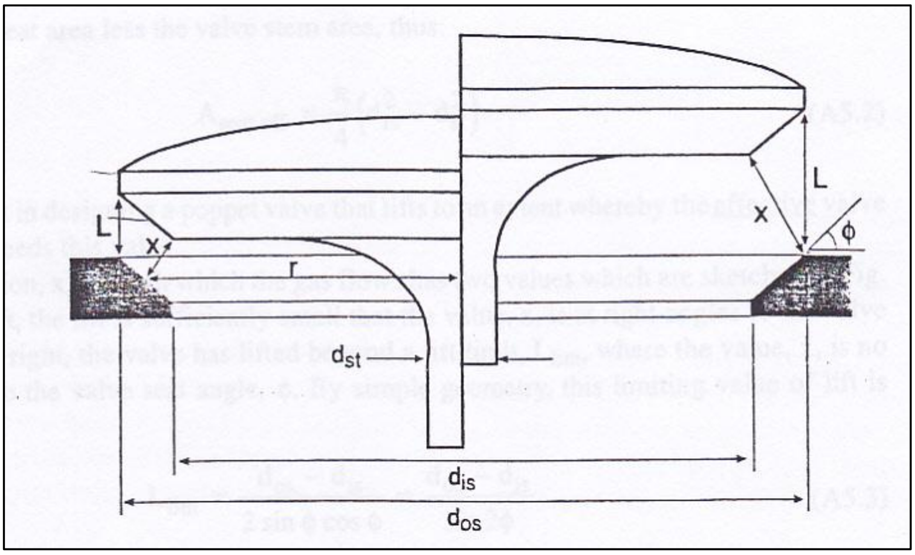


Fig. IV. 1 Valve curtain area at two lift positions

The port-to-pipe-area ratio k, for this particular geometry is expressed as:

$$Area\ ratio,\ k = \frac{valve\ curtain\ area}{pipe\ area} = \frac{A_t}{A_p} = \frac{\pi d_{is} L}{\frac{\pi}{4} d_p^2} \tag{4}$$

For accuracy of incorporation of poppet valve flow into the engine simulation, it is vital to calculate correctly the geometrical throat area of the restriction A_t. From Fig. IV.1, it can be observed that the valve curtain area at the throat, when the valve lift is L, is that which is represented by the frustum of the cone defined by the side length dimension x, the valve seat angle ϕ, and the inner or outer seat diameters, d_{is} and d_{os}; or also of dimension r, depending on the amount of the valve lift, L.

The effective area of the seat of the valve through which the gas flow to or from the port is given by the seat area less the valve stem area, thus:

$$A_{seat\ eff.} = \frac{\pi}{4}\left(d_{is}^2 - d_{st}^2\right) \tag{5}$$

The effective valve curtain area does not exceed this value.

The dimension x, through which the gas flows has two values which are sketched in Fig. IV.1 . On the left, the lift is sufficiently small that the value, x, is at right angles to the valve seat and, on the right, the valve has lifted beyond a lift limit, $L_{\lim}$, where the value x, is no longer normal to the seat angle, ϕ. By simple geometry, this limiting value of lift is given by:

$$L_{\lim} = \frac{d_{os} - d_{is}}{2\sin\varphi\cos\varphi} = \frac{d_{os} - d_{is}}{\sin 2\varphi} \tag{6}$$

For the first stage of the poppet valve lift where: $L \le L_{\lim}$, then the valve curtain area A_t, is given from the values of x and r as:

$$x = L\cos\varphi \tag{7}$$

$$r = \frac{d_{is}}{2} + x\sin\varphi \tag{8}$$

Whence,

$$A_t = \pi L\cos\varphi\left(d_{is} + L\sin\varphi\cdot\cos\varphi\right) \tag{9}$$

For the second stage of poppet valve lift where: $L > L_{\lim}$, then the valve curtain area, A_t, is given from the values of x as:

$$x = \sqrt{\left(L - \frac{d_{os} - d_{is}}{2}\tan\varphi\right)^2 + \left(\frac{d_{os} - d_{is}}{2}\right)^2} \tag{10}$$

Whence,

$$A_t = \pi\left(\frac{d_{os} + d_{is}}{2}\right)\sqrt{\left(L - \frac{d_{os} - d_{is}}{2}\tan\varphi\right)^2 + \left(\frac{d_{os} - d_{is}}{2}\right)^2} \tag{11}$$

According to the equations (9) and (11), the valve effective area is calculated. The MOTHER interface requires the valve effective area to be calculated in respect with the crank angle. Figure IV.2 shows an example for the valve lift in relation to crank angle degrees.

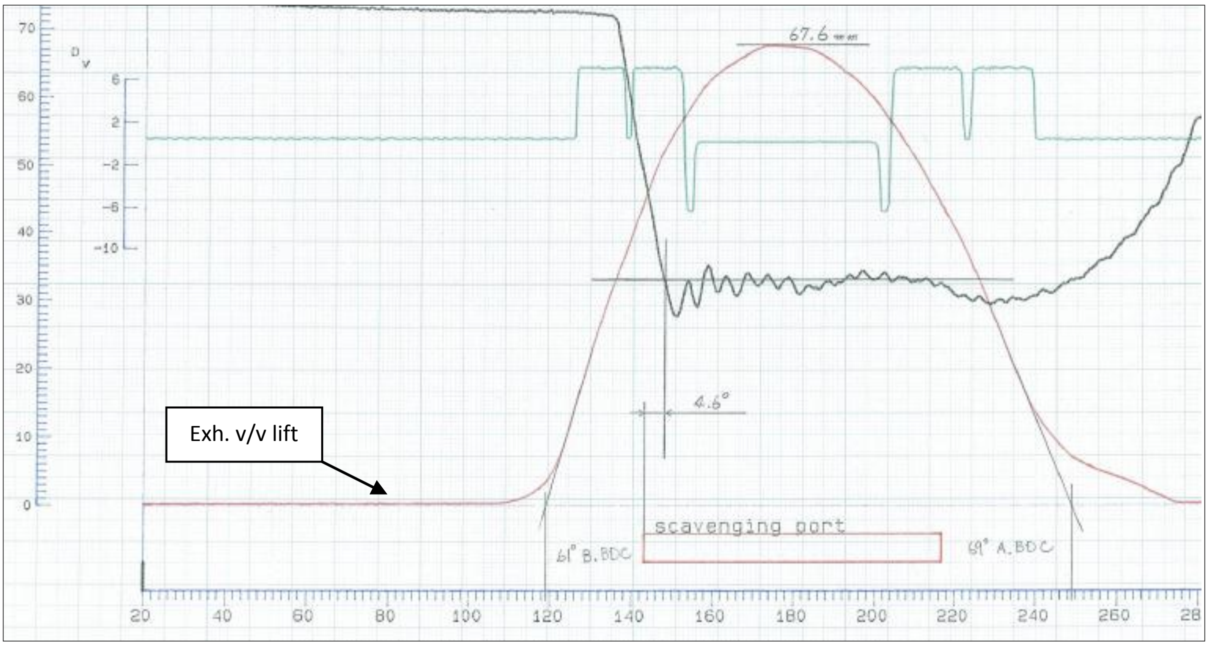


Fig. IV. 2 Example for exh. v/v lift diagram

Exhaust valve lift diagram

The exhaust valve lift diagram, attached in Figure IV.2, refers to another RTA diesel engine; the differences between the reference exhaust valve and the 6RTA48-T`s exhaust valve, can be depicted at the Table IV.4.

	Reference exh. v/v for Fig. IV.2	Exh. v/v for the 6RTA48-T
Valve Lift	67.6mm	71mm
Open valve in CA	119^o	132^o
Close valve in CA	249^o	250.8^o

Table IV. 4 Exh. v/v data

The differences in valve lift, opening and closing times, result into the consideration that the example lift diagram (Fig. IV.2), could not be used for the MOTHER inputs. In order to overcome this discrepancy, a methodology was followed, in which the form of the exhaust valve`s curve (Fig. IV.2) is maintained, whereas some of its values are altered. The steps of this methodology are described below.

- Firstly the shape of the Figure`s IV.2 lift diagram, was digitized.
- Hereupon, the digitized data were transformed by stretching the initial curve in a way that the new curve had the correct lift (i.e. 71mm) and opening/closing angles (refer to Table IV.4). At the same time, the initial`s curve shape maintained. The result of this regression is showed on the Figure IV.3.

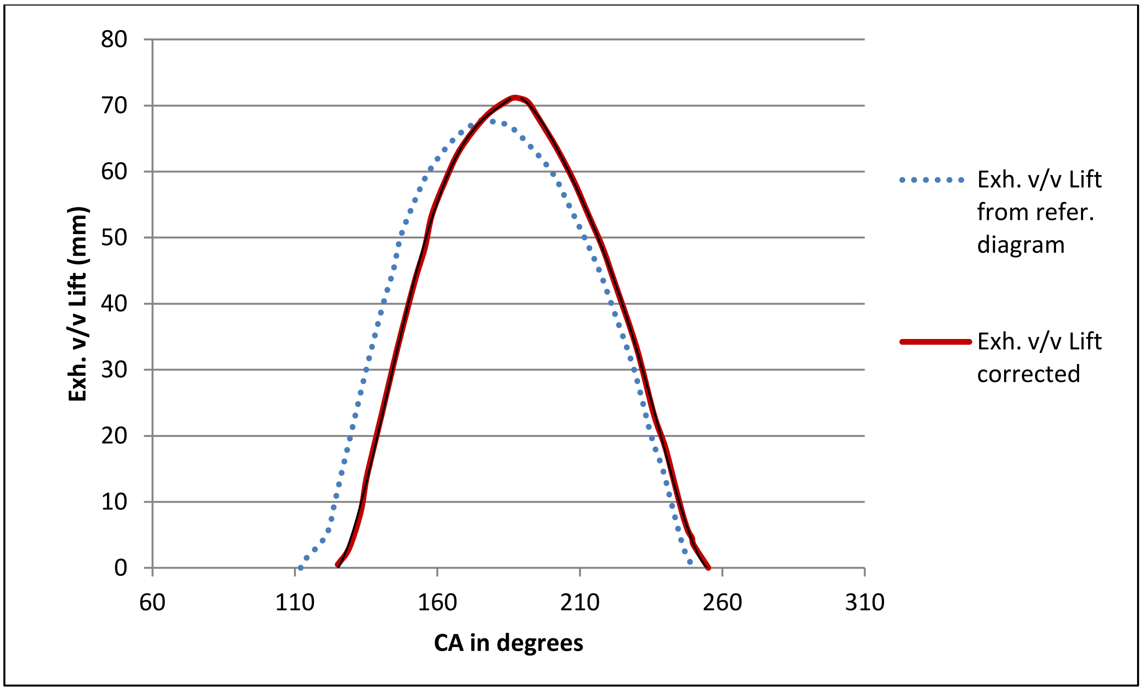


Fig. IV. 3 Lift diagram comparison

- When the valve lift is known, it is possible to calculate the valve effective area by using the equations 9 & 11. In other words, the "Valve lift figure" was transformed into the "Valve effective area figure". The following figure illustrates the exhaust valve`s effective area as a function of crank angle.

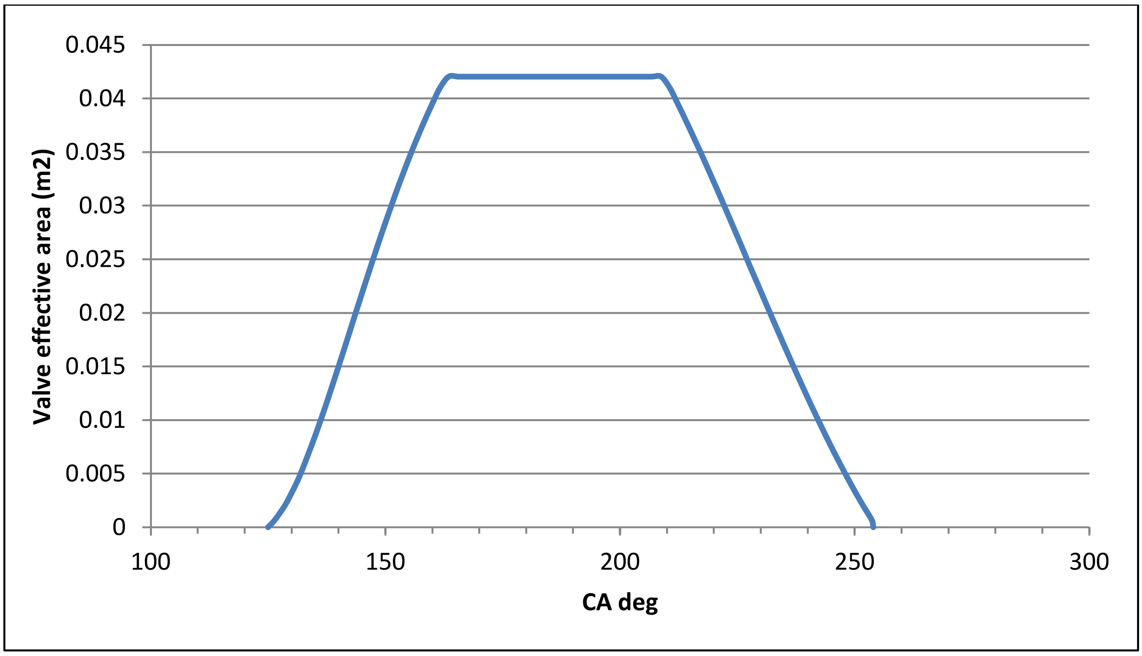


Fig. IV. 4 Exhaust valve effective area

At this point, it should be noted that the exhaust valve effective area data, as calculated above, were evaluated by running the one cylinder model with fuel quantity close to zero in order to check whether the compression pressure is calculated as measured at Shop Test for all loads. First runs showed that calculated compression pressure was by 1 to 2bar less than the measured value, for each load.

The last deficiency of the input data, was overcome by affecting slightly the exhaust valve`s effective area curve (Fig. IV.4). It was decided to alter the x-axis position (i.e. crank angle) of several points, without affecting their y-axis values (i.e. Valve eff. area). Thus, an updated exhaust valve effective area figure, has been produced. Figure IV.5 shows the initial curve for the exhaust valve`s effective area, as long as the new curve.

The aforementioned procedure was utilized because the initial exhaust valve lift data was not accurate for this engine (i.e. 6RTA48-T) and it was not possible to acquire them from at any engine manual.

The transposition of several points at x-axis direction, was selected because this action results in changing the exhaust valve`s effective area curve; thus, the relation of the exhaust valve`s lift with the crank angle is changed; consequently, the compression pressure is expected to be altered.

The new Effective area data were evaluated at each load. Finally, the calculated compression pressure converges to the measured and the results of the one cylinder model were satisfying (see Chapter 4.2.7). Figure IV.5 illustrates the first curve for the exhaust valve effective area (dashed curve) and the new one (continuous line).

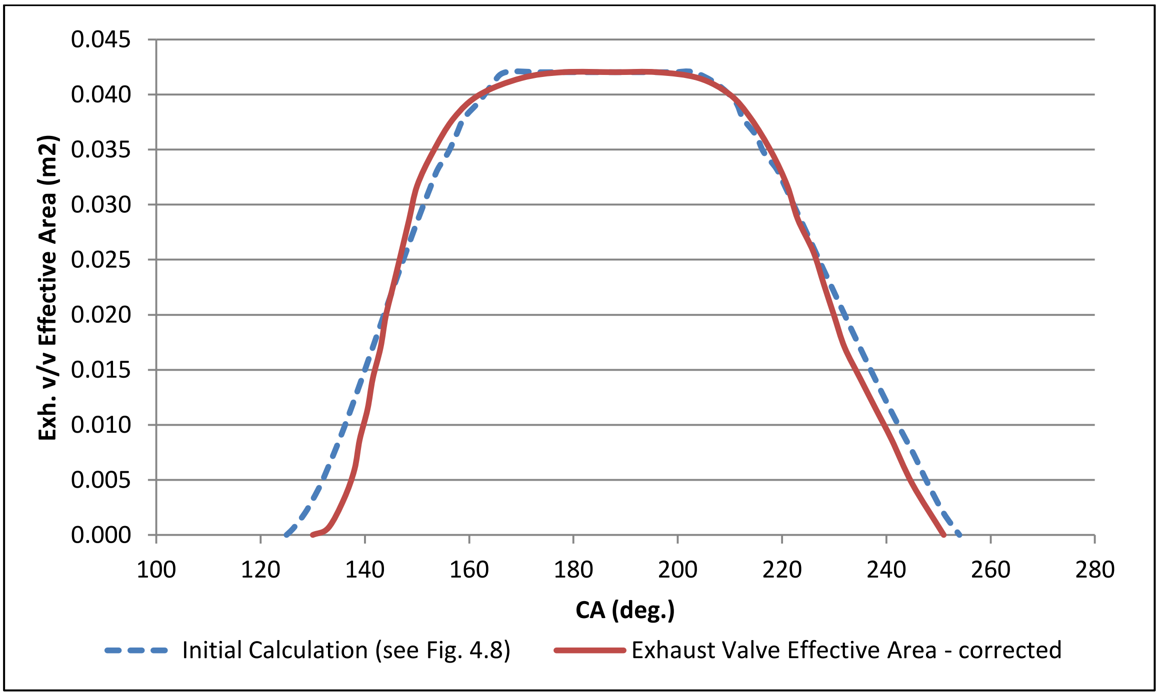


Fig. IV. 5 Corrected Exhaust Valve Effective Area

IV. 3. Scavenge ports effective area

The scavenge ports` effective area diagram is not included into any engine manual; for that reason a methodology is followed in order to calculate it. The most essential dimension for this calculation, is the distance between the BDC and the lower part of the ports (Fig. IV.6), which is not stated in any manual. This distance is calculated below from the available data. The required calculations are attached at the sub chapters below.

- **Distance between BDC and lower part of the port**

The unknown distance between BDC and the lower part of scavenge port, is illustrated at Fig. IV.6. According to Fig. IV.2, the scavenge ports open at 143^o and close at 217^o in crank angle degrees. The first attempt, of the ports configuration, was the usage of these data, which are coming from another RTA engine. Thus, the ports` effective area was calculated for the given timing. Afterwards, these input data were evaluated for each simulation load, by comparing the calculated with the measured cylinder pressures (maximum and compression) and by examining the mass flow rate.

It was noticed a backflow of gas though the scavenge ports, even at high loads. This inconsistency was corrected by altering the opening and closing crank angle of the ports without affecting the other output like the maximum pressure and compression pressure. It should be noted that Fig. IV.2 shows an example about the ports` opening and closing angle and not of this particular engine. A range of opening angles between 143^o and 155^o were examined, whether the mass backflow is eliminated or not. By completing several runs of the simulation model, the mass backflow was eliminated when the opening angle is 154^o and the closing angle is 206^o. The calculation of closing angle is related with the scavenge ports geometry and it is explained below. The BDC is at 180^o.

As the crankshaft is rotated by 180^o, the piston moves upwards from the BDC towards the TDC; this vertical movement of the piston is defined as one stroke. The 6RTA48-T`s stroke is 2000mm. Consequently, the piston displacement in relation with CA, will be:

2000mm/180^o= **11.11mm/deg. CA**.

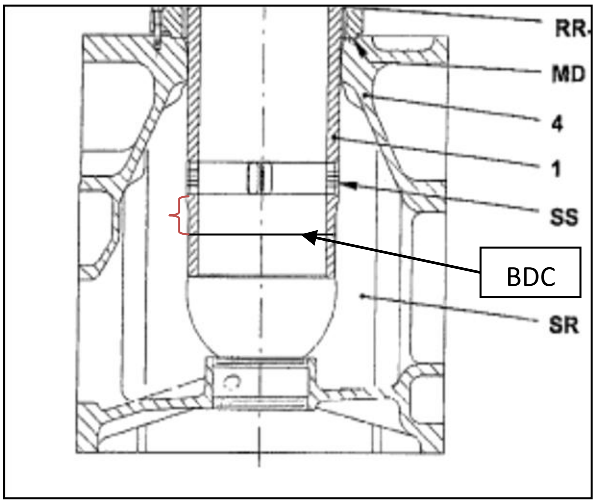


Fig. IV. 6 Lower liner part with scavenge ports

Key to illustration:

RR	Annular space
MD	Metallic sealing
SS	Scavenge ports
SR	Scavenge space (piston underside)
BDC	Bottom Dead Center
1	Cylinder liner
4	Cylinder block

According to the figure below, the port height is 120mm, so this vertical distance in terms of crank angle degrees will be as follows:

$$120mm \cdot \frac{1}{11.11 mm/CA\deg} = 10.80 CA \text{ deg} \quad (12)$$

Figure IV.7 illustrates the scavenge port with the appropriate dimensions.

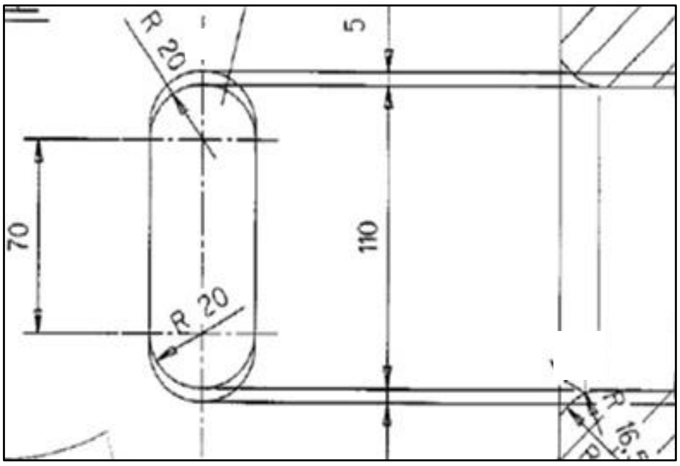


Fig. IV. 7 Scavenge Port

Thus, as the ports open at 154°, they are fully open at:

$$154^o + 10.8^o = 164.8^o \quad (13)$$

Consequently, the distance below the ports until the BDC, can be calculated, by converting the crank angle degrees in mm, as explained above. As the piston moves downwards from the lowest ports position to the BDC, the crankshaft rotates by $180^o - 164.8^o = 15.2^o$; in mm will be:

$$15.2 CA \text{ deg } \; x \; 11.11 CA \text{ deg}/mm = 168.90 mm \quad (14)$$

The scavenge ports will close after the crankshaft is rotated by 15.2^{o} from the BDC, and by another 10.8^{o} from the ports lowest part; thus, it closes at:

$$180^{\circ} + 15.2^{\circ} + 10.8^{\circ} = 206^{\circ} \tag{15}$$

The scavenge ports` area calculation is accomplished by dividing the port area into three simple geometric shapes, i.e. one hemicycle at the upper port`s part, one rectangle and a second semicycle at the lower part; refer to Fig. IV.7. The calculation is based on the piston displacement in mm and afterwards the calculated distance in mm, is transformed in crank angle degrees in order to calculate the ports area in relation to CA degrees. Table IV.5 shows the aforementioned procedure; the column named ''Piston Displacement in mm'' refers to the time when the piston moves downwards and the scavenging ports are being opening. The total ports Surface is calculated by multiplying the ''Port Surface'' by 30 which is the total ports` number.

	Piston Displacement	Port Surface	CA in relation to the piston displacement	Crank Angle from TDC	Total Ports Surface
	[mm]	[mm^2]	deg. CA	deg. CA	[m^2]
1st section - semicycle	0.00	0.00	0.00	**154.00**	0
	1.00	33.13	0.09	154.09	0.00099
	10.00	331.33	0.90	154.90	0.00994
	20.00	662.65	1.80	155.80	0.01988
	25.00	828.32	2.25	156.25	0.02485
2nd section - rectangle	26.00	868.32	2.34	145.34	0.02605
	40.00	1428.32	3.60	157.60	0.04285
	60.00	2228.32	5.40	159.40	0.06685
	80.00	3028.32	7.20	161.20	0.09085
	95.00	3628.32	8.55	162.55	0.10885
3rd section - semicycle	100.00	3793.98	9.00	163.00	0.11382
	110.00	4125.31	9.90	163.90	0.12376
	115.00	4291.07	10.35	164.35	0.1287
	120.00	4456.74	10.80	164.80	0.13370
Piston going DOWN	150.00	4456.74	13.50	167.50	0.13370
	200.00	4456.74	18.00	172.00	0.13370
	250.00	4456.74	22.50	176.50	0.13370
BDC	288.89	4456.74	26.00	**180.00**	0.13370
Piston going UP	300.00	4456.74	27.00	181.00	0.13370
	350.00	4456.74	31.50	185.50	0.13370
	400.00	4456.74	36.00	190.00	0.13370
	450.00	4456.74	40.50	194.50	0.13370
PORT starts to CLOSE	457.78	4456.74	41.20	**195.20**	0.13370
1st section - semicycle	460.00	4383.11	41.40	195.40	0.1314
	470.00	4051.79	42.30	196.30	0.1215
	480.00	3720.46	43.20	197.20	0.1116
	482.78	3628.33	43.45	197.45	0.1088
2nd section - rectangle	500.00	2939.44	45.00	199.00	0.0882
	525.00	1939.44	47.25	201.25	0.0582
	550.00	939.44	49.50	203.50	0.0282
	552.78	828.32	49.75	203.80	0.0249
3rd section - semicycle	555.00	754.70	49.95	203.95	0.0226
	565.00	423.37	50.85	**204.85**	0.0127
	570.00	257.71	51.30	205.30	0.0077
	575.00	92.04	51.75	205.75	0.0028
	577.78	0.00	52.00	**206.00**	0.0000

Table IV. 5 Scavenge Ports` Surface Calculation

Figure IV.8 below, illustrates the scavenge ports` effective area, as it appears in MOTHER interface.

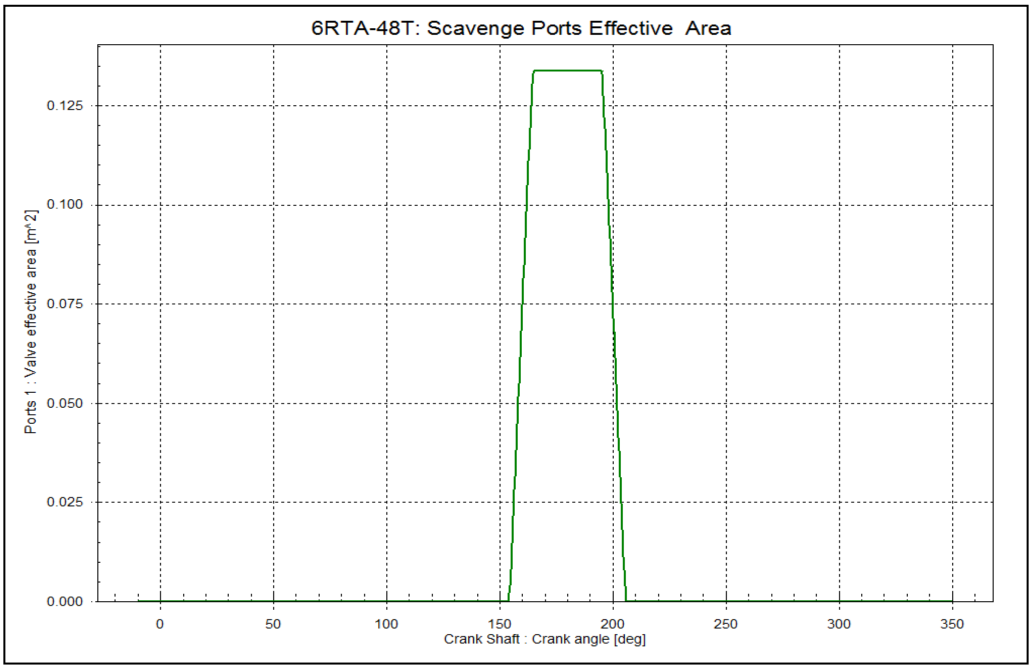


Fig. IV. 8 Scavenge ports` effective area vs. Crank Angle

IV. 4. Piston crown area

The piston crown was designed in AutoCad©, according to the manufacturer`s dimensions. The piston crown top is a curved surface, which area was calculated through the AutoCad. The area that MOTHER requires as input, is the curved surface on top of the piston crown. The three-dimensional AutoCad model is illustrated in Figure 1, below.

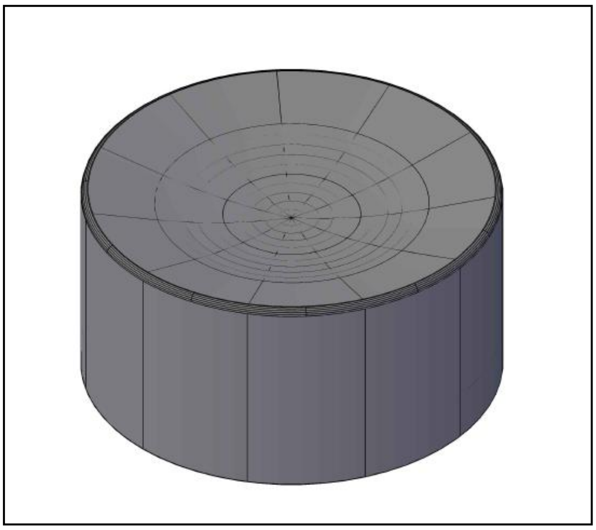


Fig. IV. 9 Piston crown

AutoCad calculates the total area of the solid (i.e. the piston crown), including the side area and the bottom area; thus, by subtracting these parts from the total area, gives the required for MOTHER piston crown area.

AutoCad© Total Area: $S_{tot.} = 714593mm^2$

Side Area (cylinder): $S_1 = 2\pi rh = 2\pi \cdot 238.1 \cdot 232 = 347078.1mm^2$

Bottom Area: $S_2 = \pi r^2 = \pi(238.1)^2 = 178101.9mm^2$

Piston Crown Area: $S_{crown} = S_{tot.} - (S_1 + S_2) = 714593 - (347078.1 + 178101.9) \Rightarrow$

$S_{crown} = 189413mm^2 \Rightarrow S_{crown} \cong 0.189m^2$

IV. 5. Cylinder head area

The cylinder head area inside the combustion chamber is a complex surface. Figure 2 illustrates the combustion chamber and required surface area (red bold line). It was considered that the required area, is described by a ductile surface with outside diameter the cylinder head`s diameter (i.e. $d_{out}$=485mm) and inside diameter the exhaust valve`s spindle bottom diameter (i.e. d_{in}=272.7mm).

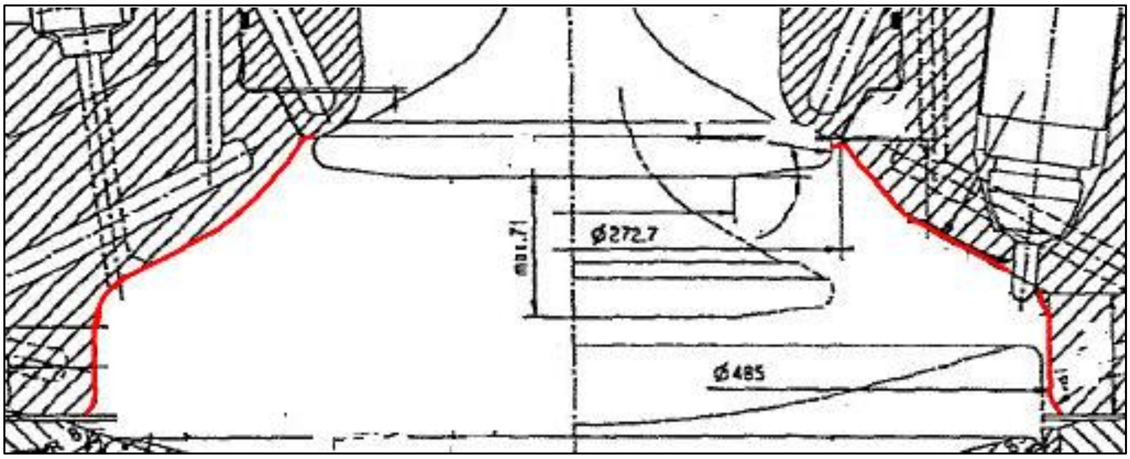


Fig. IV. 10 Combustion chamber

The cylinder head surface area estimation, follows below:

$$S_{cyl.head} = \pi\left(r_2^2 - r_1^2\right) = \pi\left((485 \cdot 0.5)^2 - (272.7 \cdot 0.5)^2\right) = 126339mm^2$$

$$S_{cyl.head} \cong 0.126m^2$$

IV. 6. Exhaust valve back area

The required exhaust valve back area is illustrated at Figure 3. The exact dimensions of the lower part of the exhaust valve spindle are not available. Thus, it was considered that the required area is described by two geometric objects; the lower part of the exhaust valve spindle is approximated by a truncated cone and the other part, by a cylinder. The total exhaust valve back area is the product of the addition of side surfaces of the two aforementioned geometric objects. The dimensions that were used for the calculations are approximate and they depicted from the Fig. 3.7 by transposing the Figure`s scale to 1:1 scale.

Cone side area: $S = \pi r\left(l + r\right)$

where,
r: cone base diameter, l: slant height

The calculation follows below.

Truncated cone side surface:

$$S_1 = \pi r_1\left(l_1 + r_1\right) - \pi r_2\left(l_2 + r_2\right) =$$
$$= \pi \cdot 128 \cdot \left(121 + 128\right) - \pi \cdot 37 \cdot \left(44 + 37\right)$$
$$= 90713 mm^2$$
$$S_1 \cong 0.09 m^2$$

Cylinder side surface:

$$S_2 = 2\pi r h = 2\pi \cdot 48 \cdot 110 = 33175 mm^2$$
$$\Rightarrow S_2 \cong 0.03 m^2$$

Total exhaust valve side area:

$$S = S_1 + S_2 = 0.09 + 0.03 \Rightarrow S = 0.12 m^2$$

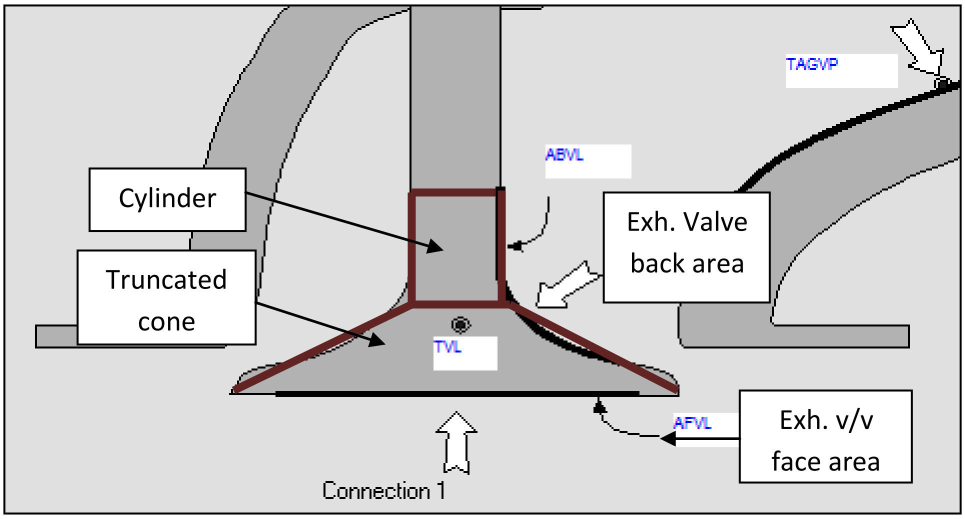


Fig. IV. 11 Exhaust valve

IV. 7. Exhaust valve face area

The exhaust valve face area is illustrated in Figure 3 and it has the shape of a cycle. Thus, the area is calculated by the following equation:

Exh. v/v face area: $S_{AFVL} = \pi r_1^2 = \pi \cdot 128^2 = 51472 mm^2$

$S_{AFVL} \cong 0.05 m^2$

IV. 8. Inlet receiver`s volume

The inlet receiver`s shape is a half cylinder (Fig. 3.8). The receiver`s dimensions have been provided by the manufacturer. The volume calculation is as follows:

Inlet receiver volume: $V_{inlet} = \dfrac{\pi r_{inlet}^2 l}{2} = \dfrac{\pi \cdot 350^2 \cdot 4792}{2} \cdot 10^{-9} \Rightarrow$

$V_{inlet} = 3.68 m^3$

where,
r: inlet receiver`s radius, $l$: inlet receiver`s length

IV. 9. Exhaust receiver`s volume

The exhaust receiver has a cylindrical shape (Fig. 3.9), with radius $r = 508mm$ and length $l = 4792mm$. The volume calculation is the following.

Exhaust receiver volume: $V_{exh.} = \pi r_{exh}^2 l = \pi \cdot 508^2 \cdot 4792 \cdot 10^{-9} \Rightarrow$

$V_{exh.} = 3.89 m^3$

IV. 10. Air cooler equivalent area

The air cooler equivalent area is the minimum flow area of the air cooler used for the calculation of the air cooler pressure loss. According to Fig. 3.10, the air cooler type is referred as shell-and-tube air cooler, where the cooling water passes inside the tubes and the air outside them. The required surface area is calculated by subtracting the total tube area from the air cooler`s side area. However, the total number of tubes is provided by manufacturer but the accurate number of tubes in the vertical direction is unknown. For that reason, the number of tubes placed vertically is calculated firstly, and afterwards the calculation of the equivalence area is made.

Figure 4, shows that horizontally there are 10 tubes; the total number of tubes is 260. Thus, vertically, they are 260/10=26 tubes, with 12mm diameter each.

$$N_{vertical} = 26 tubes, d_{tube} = 12mm$$

The distance between the first row and the last row of tubes inside the air cooler will be (refer to Figure 4):

$$h_{a/c} = (1390mm - 2x25mm) = 1340mm$$

If the total vertical height of tubes is subtracted from the air cooler height, the product will be the effective air cooler height, where only air passes through:

$$h_{eff.} = (h_{a/c} - N_{vert.} x d_{tube}) = (1340mm - 26x12mm) = 1028mm$$

The air cooler`s equivalent area can now be calculated by multiplying the effective height with the air cooler`s length:

$$S_{eff.area} = h_{eff.} \cdot l_{a/c} = 1028 \cdot 1625 \cdot 10^{-6} \Rightarrow$$

$$S_{eff.area} = 1.67m^2$$

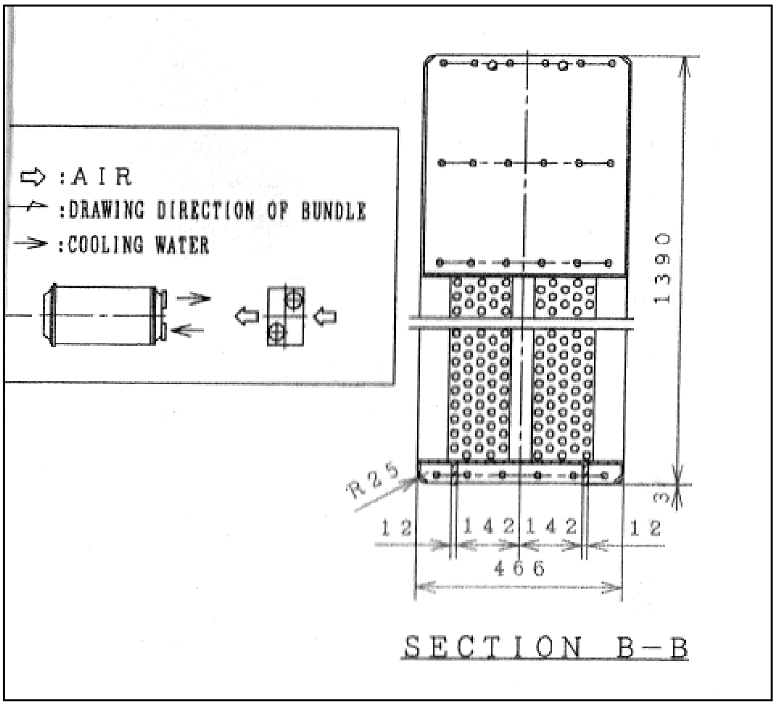


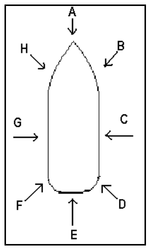
Fig. IV. 12 Air cooler section B-B

APPENDIX V:

M/E PERFORMANCE DATA

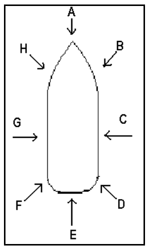
This Appendix`s purpose is to provide the detailed measurements of the engine performance parameters for each of the selected M/E Performance Reports in Chapter 8.5.

V. 1. M/E Performance of January 28th, 2008

	VOYAGE/LADEN-BALLAST:	LADEN					
VARIOUS SHIP	VESSEL SPEED	Knots		14,6			
	BAROMETRIC PRESS	mmWG		1010			
	WIND FORCE/DIRECTION			3	D		
	SWELL HEIGHT/DIRECTION	M		0,5	D		
	CURRENT SPEED/DIRECTION	Knots		0,5	FE		
	COURSE	Deg		308			
	DRAUGHT FOR/AFT	M		10.10	10,30		
	CORRESPOND DISPLACEMENT	MTs		49751			
	M/E SPEED	Kn		14.84			
	PROPELLER APARENT SLIP	%		1.60			
VARIOUS M/E	ENGINE SPEED	R/MIN		109.99			
	SPEED SETTING	POS. BAR		111	RPM		
	ECR L.I./F.O LEVER	POS:		6.9	N/A		
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:		6.9	6.9		
	POWER	KW		5917			
	FUEL CONSUMPTION	MT/24 HRS		29.05			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.2	0.5		
	T/C SPEED	RPM		15900			
	SCAVENGE PRESS (LCL/RMT)	Mpa		0.18	0.19		
	AIR TEM: E/R/BEF. BLOWER	DEG C		46	38		
	AIR COOL TEMP BEF/AFTER	DEG C		155	50		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		130			
	BLOWER SUCTION	mmWG		100			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0,7			
	OIL: BEARING/CROSSHEAD	Mpa		0.43	1.21		
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.21			
	FUEL OIL AFTER FILTER	MPa		0.78			
	PISTON/JACKET/AIR CLR	MPa	0,43	0.32	0.2		
	SEA WATER	MPa		0.22			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	9	8.5	9	8.5	9	8.5
	CYL MAX COMBUSTION PRESS (Mpa)	12.5	12.5	12.5	11.8	12.5	12.5
	EXH T BEFR/AFTER T/C (LOCAL)	432				320	
	EXH T BEFR/AFTER T/C (REMT)	422				N/A	

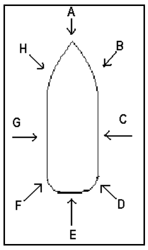
VARIOUS TMPERATURES	EXH T AFTER CYL (LOCAL)	390	375	360	355	380	370
	EXH T AFTER CYL (REMOTE)	390	359	358	344	376	369
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	43	43	43	43	43	43
	PISTON COOLING OUT(REMT):	62	60	62	59	62	59
	JACKET COOLING IN:	77	77	77	77	77	77
	JACKET COOLING OUT(LOCAL):	85	84	84	84	84	84
	JACKET COOLING OUT(REMT):	85	84	84	84	84	84
	PISTON UNDERSIDE TEMP.	N/A	N/A	N/A	N/A	N/A	N/A
	T/C COOLING IN/OUT	43/70					
	FUEL OIL BFR H.P.PUMP/F.MTR	125/130					
	SEA WATER	30					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	333.8	ALUM.+SILICON CONTENT		PPM	19
	DENSITY AT 15 °C	Kg/l	0.9829	NCV		MJ/kg	40.15
	SULPHER CONTENT	%	3.25	CCAI		-	845
	VANADIUM CONTENT	PPM	101				
REMARKS: -							

V. 2. M/E Performance of February 10th, 2008

	VOYAGE/LADEN-BALLAST:	LADEN			
VARIOUS SHIP	VESSEL SPEED	Knots	11.7		
	BAROMETRIC PRESS	mmWG	1024		
	WIND FORCE/DIRECTION		5	AB	
	SWELL HEIGHT/DIRECTION	M	1.5	D	
	CURRENT SPEED/DIRECTION	Knots	2.8	H	
	COURSE	Deg	267		
	DRAUGHT FOR/AFT	M	10.10	10.30	
	CORRESPOND DISPLACEMENT	MTs	49751		
	M/E SPEED	Kn	13.87		
	PROPELLER APARENT SLIP	%	15.6		
VARIOUS M/E	ENGINE SPEED	R/MIN	108.87		
	SPEED SETTING	POS. BAR	110	RPM	
	ECR L.I./F.O LEVER	POS:	6.9	N/A	
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:	6.9	6.9	
	POWER	KW	5921		
	FUEL CONSUMPTION	MT/24 HRS	28.90		
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE	3.2	0.5	
	T/C SPEED	RPM	15900		
	SCAVENGE PRESS (LCL/RMT)		0.18	0.19	
	AIR TEM: E/R/BEF. BLOWER	DEG C	46	38	
	AIR COOL TEMP BEF/AFTER	DEG C	155	50	
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG	130		
	BLOWER SUCTION	mmWG	100		
	AIR: VALVE AIR SPRING SUPPLY	Mpa	0.7		
	OIL: BEARING/CROSSHEAD	Mpa	0.43/1.21		
	V/V DRIVE ACTUATOR SUPPLY	Mpa	1.21		
	FUEL OIL AFTER FILTER	MPa	0.78		

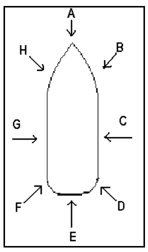
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.2		
	SEA WATER	MPa		0.22			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	9.5	8.5	8.5	8.5	8.5	8.5
	CYL MAX COMBUSTION PRESS (Mpa)	12.5	12	12	120	11.5	12.5
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	432				320	
	EXH T BEFR/AFTER T/C (REMT)	422				N/A	
	EXH T AFTER CYL (LOCAL)	393	361	359	355	375	372
	EXH T AFTER CYL (REMOTE)	390	359	358	352	374	369
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	43	43	43	43	43	43
	PISTON COOLING OUT(REMT):	61	60	62	60	62	60
	JACKET COOLING IN:	77	77	77	77	77	77
	JACKET COOLING OUT(LOCAL):	85	84	84	84	84	84
	JACKET COOLING OUT(REMT):	85	84	84	84	84	84
	PISTON UNDERSIDE TEMP.	N/A	N/A	N/A	N/A	N/A	N/A
	T/C COOLING IN/OUT	43/70					
	FUEL OIL BFR H.P.PUMP/F.MTR	125/130					
	SEA WATER	27					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	339.3	ALUM.+SILICON CONTENT		PPM	19
	DENSITY AT 15 °C	Kg/l	0.9895	NCV		MJ/kg	40.28
	SULPHER CONTENT	%	2.78	CCAI		-	852
	VANADIUM CONTENT	PPM	65				
REMARKS: -							

V. 3. M/E Performance of March 12th, 2008

	VOYAGE/LADEN-BALLAST:	LADEN			
VARIOUS SHIP	VESSEL SPEED	Knots	14.1		
	BAROMETRIC PRESS	mmWG	1012		
	WIND FORCE/DIRECTION		4	D	
	SWELL HEIGHT/DIRECTION	M	0.5	H	
	CURRENT SPEED/DIRECTION	Knots	1	H	
	COURSE	Deg	53		
	DRAUGHT FOR/AFT	M	11.55	11.60	
	CORRESPOND DISPLACEMENT	MTs	56510		
	M/E SPEED	Kn	14.65		
	PROPELLER APARENT SLIP	%	3.8		
VARIOUS M/E	ENGINE SPEED	R/MIN	109.2		
	SPEED SETTING	POS. BAR	111	RPM	
	ECR L.I./F.O LEVER	POS:	6.9	N/A	
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:	6.9	6.9	
	POWER	KW	5997		
	FUEL CONSUMPTION	MT/24 HRS	27.86		
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE	3.2	0.5	
	T/C SPEED	RPM	16000		
	SCAVENGE PRESS (LCL/RMT)		0.18	0.19	
	AIR TEM: E/R/BEF. BLOWER	DEG C	46	42	

	AIR COOL TEMP BEF/AFTER	DEG C		152	50		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		110			
	BLOWER SUCTION	mmWG		100			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.7			
	OIL: BEARING/CROSSHEAD	Mpa		0.43/1.21			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.21			
	FUEL OIL AFTER FILTER	MPa		0.78			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.2		
	SEA WATER	MPa		0.22			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	9	8.5	8.5	8.5	8.5	8.5
	CYL MAX COMBUSTION PRESS (Mpa)	12.2	12	11.6	12	12	12
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	440				333	
	EXH T BEFR/AFTER T/C (REMT)	432				N/A	
	EXH T AFTER CYL (LOCAL)	390	365	371	355	387	375
	EXH T AFTER CYL (REMOTE)	390	360	361	352	375	370
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	43	43	43	43	43	43
	PISTON COOLING OUT(REMT):	61	60	62	60	60	62
	JACKET COOLING IN:	77	77	77	77	77	77
	JACKET COOLING OUT(LOCAL):	85	85	84	84	85	84
	JACKET COOLING OUT(REMT):	85	84	84	84	84	84
	PISTON UNDERSIDE TEMP.	N/A	N/A	N/A	N/A	N/A	N/A
	T/C COOLING IN/OUT	43/70					
	FUEL OIL BFR H.P.PUMP/F.MTR	122/132					
	SEA WATER	29					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	339.3	ALUM.+SILICON CONTENT		PPM	19
	DENSITY AT 15 °C	Kg/l	0.9895	NCV		MJ/kg	40.28
	SULPHER CONTENT	%	2.78	CCAI		-	852
	VANADIUM CONTENT	PPM	65				
REMARKS: -							

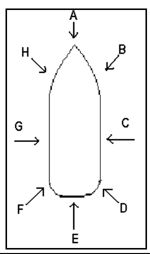
V. 4. M/E Performance of May 17th, 2008

	VOYAGE/LADEN-BALLAST:	LADEN			
VARIOUS SHIP	VESSEL SPEED	Knots	13.6		
	BAROMETRIC PRESS	mmWG	1012		
	WIND FORCE/DIRECTION		5	F	
	SWELL HEIGHT/DIRECTION	M	2	F	
	CURRENT SPEED/DIRECTION	Knots	-0.1	F	
	COURSE	Deg	219		
	DRAUGHT FOR/AFT	M	11.70	11.93	
	CORRESPOND DISPLACEMENT	MTs	56590		
	M/E SPEED	Kn	14.83		
	PROPELLER APARENT SLIP	%	8.43		
VARIOUS M/E	ENGINE SPEED	R/MIN	109.98		
	SPEED SETTING	POS. BAR	111	RPM	
	ECR L.I./F.O LEVER	POS:	6.8	N/A	
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:	6.8	6.8	
	POWER	KW	5402		

	FUEL CONSUMPTION	MT/24 HRS		28.03			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.2	0.8		
	T/C SPEED	RPM		16000			
	SCAVENGE PRESS (LCL/RMT)			0.19	0.18		
	AIR TEM: E/R/BEF. BLOWER	DEG C		45	43		
	AIR COOL TEMP BEF/AFTER	DEG C		158	56		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		111			
	BLOWER SUCTION	mmWG		105			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.7			
	OIL: BEARING/CROSSHEAD	Mpa		0.43/1.21			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.25			
	FUEL OIL AFTER FILTER	MPa		0.81			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.2		
	SEA WATER	MPa		0.24			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	9.9	9.6	9.6	9.5	9.6	9.6
	CYL MAX COMBUSTION PRESS (Mpa)	13	12.8	12.7	12.6	12.7	12.6
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	450				335	
	EXH T BEFR/AFTER T/C (REMT)	438				N/A	
	EXH T AFTER CYL (LOCAL)	390	385	375	360	390	405
	EXH T AFTER CYL (REMOTE)	395	383	373	356	388	400
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	45	45	45	45	45	45
	PISTON COOLING OUT(REMT):	63	62	63	61	63	60
	JACKET COOLING IN:	77	77	77	77	77	77
	JACKET COOLING OUT(LOCAL):	84	84	85	84	84	84
	JACKET COOLING OUT(REMT):	85	84	84	84	84	84
	PISTON UNDERSIDE TEMP.	72	68	63	62	69	68
	T/C COOLING IN/OUT	45/73					
	FUEL OIL BFR H.P.PUMP/F.MTR	126/130					
	SEA WATER	28					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	364.6	ALUM.+SILICON CONTENT		PPM	35
	DENSITY AT 15 °C	Kg/l	0.99	NCV		MJ/kg	40.43
	SULPHER CONTENT	%	2.27	CCAI		-	851
	VANADIUM CONTENT	PPM	119				
REMARKS: -							

V. 5. M/E Performance of March 8th, 2009

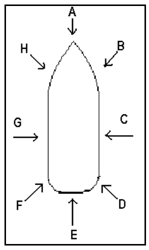
	VOYAGE/LADEN-BALLAST:	BALLAST		
VARIOUS SHIP	VESSEL SPEED	Knots	13.8	
	BAROMETRIC PRESS	mmWG	1012	
	WIND FORCE/DIRECTION		5	E
	SWELL HEIGHT/DIRECTION	M	1.5	E
	CURRENT SPEED/DIRECTION	Knots	0	E
	COURSE	Deg	123	
	DRAUGHT FOR/AFT	M	5.90	7.98
	CORRESPOND DISPLACEMENT	MTs	35187	



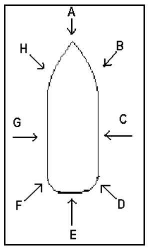
	M/E SPEED	Kn		14.24			
	PROPELLER APARENT SLIP	%		3.10%			
VARIOUS M/E	ENGINE SPEED	R/MIN		105.6			
	SPEED SETTING	POS. BAR		106	RPM		
	ECR L.I./F.O LEVER	POS:		6.3	N/A		
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:		6.5	6.5		
	POWER	KW		4619			
	FUEL CONSUMPTION	MT/24 HRS		28.83			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.2	0.5		
	T/C SPEED	RPM		14900			
	SCAVENGE PRESS (LCL/RMT)			0.13	0.12		
	AIR TEM: E/R/BEF. BLOWER	DEG C		41	41		
	AIR COOL TEMP BEF/AFTER	DEG C		143	47		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		102			
	BLOWER SUCTION	mmWG		105			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.7			
	OIL: BEARING/CROSSHEAD	Mpa		0.43/1.25			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.25			
	FUEL OIL AFTER FILTER	MPa		0.77			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.2		
	SEA WATER	MPa		0.2			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	7.4	7.5	7.5	7.3	7.5	7.8
	CYL MAX COMBUSTION PRESS (Mpa)	10.7	10.7	11.5	10.8	11.2	10.9
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	440				340	
	EXH T BEFR/AFTER T/C (REMT)	425				N/A	
	EXH T AFTER CYL (LOCAL)	390	385	375	360	377	377
	EXH T AFTER CYL (REMOTE)	383	372	372	351	371	378
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	43	43	43	43	43	43
	PISTON COOLING OUT(REMT):	61	60	65	60	61	60
	JACKET COOLING IN:	75	75	75	75	75	75
	JACKET COOLING OUT(LOCAL):	83	84	86	84	84	84
	JACKET COOLING OUT(REMT):	83	84	85	84	83	83
	PISTON UNDERSIDE TEMP.(L/R)	74/70	67/68	67/71	65/61	62/64	68/66
	T/C COOLING IN/OUT	45/65					
	FUEL OIL BFR H.P.PUMP/F.MTR	118/119					
	SEA WATER	28					
FUEL ANALYSIS	VISCOSITY AT 50^{o}C	cSt	337	ALUM.+SILICON CONTENT		PPM	27
	DENSITY AT 15 ^{o}C	Kg/l	0.9882	NCV		MJ/kg	39.99
	SULPHER CONTENT	%	3.74	CCAI		-	850
	VANADIUM CONTENT	PPM	65				
REMARKS: -							

V. 6. M/E Performance of April 28th, 2009

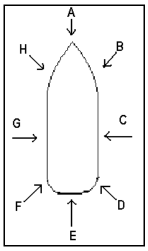
	VOYAGE/LADEN-BALLAST:	BALLAST					
VARIOUS SHIP	VESSEL SPEED	Knots		14.8			
	BAROMETRIC PRESS	mmWG		1013			
	WIND FORCE/DIRECTION			3	A		
	SWELL HEIGHT/DIRECTION	M		2	A		
	CURRENT SPEED/DIRECTION	Knots		0.2	E		
	COURSE	Deg		180			
	DRAUGHT FOR/AFT	M		4.18	6.13		
	CORRESPOND DISPLACEMENT	MTs		23540			
	M/E SPEED	Kn		14.79			
	PROPELLER APARENT SLIP	%		-0.06%			
VARIOUS M/E	ENGINE SPEED	R/MIN		109.65			
	SPEED SETTING	POS. BAR		110	RPM		
	ECR L.I./F.O LEVER	POS:		6.7	N/A		
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:		6.8	6.7		
	POWER	KW		5188			
	FUEL CONSUMPTION	MT/24 HRS		26.58			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.2	0.5		
	T/C SPEED	RPM		16100			
	SCAVENGE PRESS (LCL/RMT)			0.18	0.165		
	AIR TEM: E/R/BEF. BLOWER	DEG C		44	43		
	AIR COOL TEMP BEF/AFTER	DEG C		156	52		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		150			
	BLOWER SUCTION	mmWG		105			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.7			
	OIL: BEARING/CROSSHEAD	Mpa		0.43/1.25			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.25			
	FUEL OIL AFTER FILTER	MPa		0.77			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.2		
	SEA WATER	MPa		0.2			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	8.7	8.4	9	8.7	8.5	8.6
	CYL MAX COMBUSTION PRESS (Mpa)	12	11.3	11.9	12.1	11.8	11.9
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	435				325	
	EXH T BEFR/AFTER T/C (REMT)	420				N/A	
	EXH T AFTER CYL (LOCAL)	380	380	365	375	375	375
	EXH T AFTER CYL (REMOTE)	367	354	356	363	378	372
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	45	45	45	45	45	45
	PISTON COOLING OUT(REMT):	63	64	63	60	63	61
	JACKET COOLING IN:	74	74	74	74	74	74
	JACKET COOLING OUT(LOCAL):	82	84	85	84	84	84
	JACKET COOLING OUT(REMT):	82	83	82	83	83	82
	PISTON UNDERSIDE TEMP.(L/R)	66/67	70/73	56/58	60/59	63/64	65/64
	T/C COOLING IN/OUT	45/72					
	FUEL OIL BFR H.P.PUMP/F.MTR	124/120					
	SEA WATER	32					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	374.8	ALUM.+SILICON CONTENT		PPM	6
	DENSITY AT 15 °C	Kg/l	0.99	NCV		MJ/kg	39.87
	SULPHER CONTENT	%	3.65	CCAI			851
	VANADIUM CONTENT	PPM	83				
REMARKS: -							



V. 7. M/E Performance of October 29th, 2009

	VOYAGE/LADEN-BALLAST:	LADEN					
VARIOUS SHIP	VESSEL SPEED	Knots		12.9			
	BAROMETRIC PRESS	mmWG		1013			
	WIND FORCE/DIRECTION			2	C		
	SWELL HEIGHT/DIRECTION	M		2	C		
	CURRENT SPEED/DIRECTION	Knots		0.5	C		
	COURSE	Deg		19			
	DRAUGHT FOR/AFT	M		10.19	10.61		
	CORRESPOND DISPLACEMENT	MTs		50536			
	M/E SPEED	Kn		14.05			
	PROPELLER APARENT SLIP	%		8.20%			
VARIOUS M/E	ENGINE SPEED	R/MIN		104.18			
	SPEED SETTING	POS. BAR		105	RPM		
	ECR L.I./F.O LEVER	POS:		6.5	6.4		
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:		6.4	6.4		
	POWER	KW		4423			
	FUEL CONSUMPTION	MT/24 HRS		23.018			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.2	0.8		
	T/C SPEED	RPM		15000			
	SCAVENGE PRESS (LCL/RMT)			0.15	0.14		
	AIR TEM: E/R/BEF. BLOWER	DEG C		43	43		
	AIR COOL TEMP BEF/AFTER	DEG C		140	46		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		115			
	BLOWER SUCTION	mmWG		50			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.78			
	OIL: BEARING/CROSSHEAD	Mpa		0.44/1.23			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.23			
	FUEL OIL AFTER FILTER	MPa		0.8			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.15		
	SEA WATER	MPa		0.21			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	7.6	7.7	7.7	7.8	7.7	8
	CYL MAX COMBUSTION PRESS (Mpa)	11.1	11.2	11	11.2	11	11.2
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	430				335	
	EXH T BEFR/AFTER T/C (REMT)	419				N/A	
	EXH T AFTER CYL (LOCAL)	360	370	370	370	375	370
	EXH T AFTER CYL (REMOTE)	356	348	368	358	370	367
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	45	45	45	45	45	45
	PISTON COOLING OUT(REMT):	62	62	63	61	62	60
	JACKET COOLING IN:	76	76	76	76	76	76
	JACKET COOLING OUT(LOCAL):	81	85	84	81	84	84
	JACKET COOLING OUT(REMT):	83	83	83	83	83	83
	PISTON UNDERSIDE TEMP.(L/R)	61/60	60/61	57/55	57/58	60/58	59/61
	T/C COOLING IN/OUT	45/69					
	FUEL OIL BFR H.P.PUMP/F.MTR	126/134					
	SEA WATER	27					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	399.4	ALUM.+SILICON CONTENT		PPM	34
	DENSITY AT 15 °C	Kg/l	0.985	NCV		MJ/kg	40.39
	SULPHER CONTENT	%	2.62	CCAI		-	855
	VANADIUM CONTENT	PPM	34				
REMARKS: -							

V. 8. M/E Performance of November 1st, 2009

	VOYAGE/LADEN-BALLAST:	LADEN					
VARIOUS SHIP	VESSEL SPEED	Knots		11.7			
	BAROMETRIC PRESS	mmWG		1012			
	WIND FORCE/DIRECTION			5	B		
	SWELL HEIGHT/DIRECTION	M		2	A		
	CURRENT SPEED/DIRECTION	Knots		0.5	B		
	COURSE	Deg		19			
	DRAUGHT FOR/AFT	M		10.18	10.55		
	CORRESPOND DISPLACEMENT	MTs		50437			
	M/E SPEED	Kn		14.05			
	PROPELLER APARENT SLIP	%		16.70%			
VARIOUS M/E	ENGINE SPEED	R/MIN		104.22			
	SPEED SETTING	POS. BAR		105	RPM		
	ECR L.I./F.O LEVER	POS:		6.5	6.4		
	LOCAL L.I/SPILL V/V/GOV. INDEX	POS:		6.4	6.4		
	POWER	KW		4474			
	FUEL CONSUMPTION	MT/24 HRS		23.437			
	VIT DEVICE/FQS (+) Advance, (-) Retard	(+)/(-) CRANK ANGLE		3.0	0.8		
	T/C SPEED	RPM		15100			
	SCAVENGE PRESS (LCL/RMT)			0.16	0.15		
	AIR TEM: E/R/BEF. BLOWER	DEG C		44	44		
	AIR COOL TEMP BEF/AFTER	DEG C		150	47		
VARIOUS PRESSURES	PRESS DROP AIR COOLER	mmWG		115			
	BLOWER SUCTION	mmWG		50			
	AIR: VALVE AIR SPRING SUPPLY	Mpa		0.78			
	OIL: BEARING/CROSSHEAD	Mpa		0.44/1.24			
	V/V DRIVE ACTUATOR SUPPLY	Mpa		1.24			
	FUEL OIL AFTER FILTER	MPa		0.79			
	PISTON/JACKET/AIR CLR	MPa	0.43	0.32	0.16		
	SEA WATER	MPa		0.21			
		1	**2**	**3**	**4**	**5**	**6**
	CYL COMPRESSION PRESS (Mpa)	8	8	8.1	8	8.3	8.3
	CYL MAX COMBUSTION PRESS (Mpa)	11	11.2	11.1	11.4	11.5	11.5
VARIOUS TEMPERATURES	EXH T BEFR/AFTER T/C (LOCAL)	430				335	
	EXH T BEFR/AFTER T/C (REMT)	420				N/A	
	EXH T AFTER CYL (LOCAL)	355	365	370	375	370	365
	EXH T AFTER CYL (REMOTE)	351	352	372	367	366	363
		1	**2**	**3**	**4**	**5**	**6**
	PISTON COOLING IN:	44	44	44	44	44	44
	PISTON COOLING OUT(REMT):	62	62	63	61	62	60
	JACKET COOLING IN:	76	76	76	76	76	76
	JACKET COOLING OUT(LOCAL):	81	85	84	81	84	84
	JACKET COOLING OUT(REMT):	83	83	83	83	83	83
	PISTON UNDERSIDE TEMP.(L/R)	60/61	62/62	57/59	58/58	60/62	60/61
	T/C COOLING IN/OUT	44/54					
	FUEL OIL BFR H.P.PUMP/F.MTR	124/135					
	SEA WATER	29					
FUEL ANALYSIS	VISCOSITY AT 50°C	cSt	356.6	ALUM.+SILICON CONTENT		PPM	11
	DENSITY AT 15 °C	Kg/l	0.984	NCV		MJ/kg	41.04
	SULPHER CONTENT	%	0.7	CCAI		-	846
	VANADIUM CONTENT	PPM	46				
REMARKS: M/E Load Indicator fluctuate from 6.4-6.5, T/C rpm 15000-15200, M/E rpm 103.7-105.5							

APPENDIX VI:

ISO CORRECTION OF VARIOUS PERFORMANCE DATA

This Appendix`s purpose is to provide basic information about the concept of ISO correction methodology. The engine performance data varies in dependence with temperatures of intake air and scavenge air. Therefore, for the purpose of comparing and evaluating the engine performance data, certain reference conditions should be considered. Thus, the reference conditions should be either those at Shop Test or the ISO reference conditions.

- ISO reference conditions

Suction air temperature	25^{o}C
Charge air cooling water inlet	25^{o}C
Total barometric pressure	1bar = 750 mmHg
Suction air relatively humidity	60%

- ISO correction method with Shop Test reference conditions

  The following performance data are corrected:

  - Scavenge pressure
  - Combustion pressure (Pmax)
  - Compression pressure (Pcomp)
  - T/C revolution speed
  - Exhaust gas T/C inlet temperature
  - Exhaust gas T/C outlet temperature
  - Exhaust gas cylinder outlet temperature
  - Specific Fuel Oil Consumption (SFOC)

Due to the dependence of performance parameters on air temperature at T/C inlet and scavenging air temperature, corrections for these data are applied to the above performance data through correction coefficients. These coefficients are not provided to due to confidentiality of the data. However, the corrective equations have the following form:

$$X_{ISO} = X_{MEASURED} + \left(T_{Shop}^{air} - T_{meas.}^{air}\right) \cdot C_1 + \left(T_{Shop}^{scav.} - T_{meas.}^{scav.}\right) \cdot C_2$$

where:

X_{ISO} : Performance parameter corrected to Shop Test condition according to ISO methodology

$X_{MEASURED}$: Performance parameter reported in the M/E Performance Report (measured onboard)

T_{Shop}^{air} : Air temperature at T/C inlet (i.e. E/R temp.) at Shop Test

$T_{meas.}^{air}$: Measured air temp. at T/C inlet

$T_{Shop}^{scav.}$: Air temperature after the A/C at Shop Test

$\mathrm{T}_{meas.}^{scav.}$: Measured air temp. after A/C

In addition to the above corrections (ambient air temp. and scavenge air temp.), the exhaust gas temperatures are corrected for the usage of Heavy Fuel Oil (HFO) when applicable and the SFOC is corrected for the fuel`s calorific value.

Finally, the ISO-corrected values are compared with the relevant values calculated from Shop Test. Any deviation from reference values is evaluated and it could provide an indication whether or not it is caused by a change of ambient conditions or by an abnormality of the engine itself. Last but not least, in-site inspections are of utmost importance and should be linked with the evaluation procedure.

Abbreviations

NTUA	:	National Technical University of Athens
MOTHER	:	MOTor THERmodynamics
FOBAS	:	Fuel Oil Bunker Analysis Service (by Llyod`s Register)
TDC	:	Top Dead Center
BDC	:	Bottom Dead Center
CA	:	Crank Angle
M/E	:	Main Engine
IHP	:	Indicated Horse Power
BHP	:	Brake Horse Power
MEP	:	Mean Effective Pressure
BMEP	:	Brake Mean Effective Pressure
IMEP	:	Indicated Mean Effective Pressure
LCV	:	Low Calorific Value
NCV	:	Net Calorific Value
CCAI	:	Calculated Carbon Aromaticity Index
HFO	:	Heavy Fuel Oil
LO	:	Lub Oil
FW	:	Fresh Water
MEPIC	:	Main Engine Performance Index Calculation
LI	:	Load Indicator
NCR	:	Normal Continuous Rating
(C)MCR	:	(Contracted) Maximum Continuous Rating
E/R	:	Engine Room
ECR	:	Engine Control Room
V/V	:	Valve
T/C	:	Turbocharger
A/C	:	Air Cooler
VIT	:	Variable Injection Timing
FQS	:	Fuel Quality Setting
HTF	:	Heat Transfer Factor
SFOC	:	Specific Fuel Oil Consumption
SOI	:	Start Of static Injection
A/F	:	Air/Fuel ratio
HT	:	Heat Transfer

References

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