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Abstract:

By reducing ships' fuel consumption, shipowners can achieve lower operating costs while at the same time reducing their impact on the environment.

This report is a study of different measures that can be taken in order to improve ships' energy efficiency, and a study of how various engine parameters can be related to efficient running of the engine.

Several measures have been assessed and related to a specific ship's energy profile in order to find the fuel and diesel oil savings potential. The ship's profile has been updated largely based on the different manufacturers' own estimations of their product's effect on fuel consumption.

Engine data for a two-stroke diesel engine have been collected from a database and are presented compared to the overall efficiency of the engine. The parameters are presented with the goal of identifying which ones, if any, that most accurately indicate the condition of the engine in terms of efficiency.

For a complete summary, see page v.

Keyword:

energy efficient operation

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PREFACE

This report is the result of the work on my master's thesis at the Department of Marine Technology at the Norwegian University of Science and Technology, spring 2011.

The thesis is based on the "energy efficiency" part of the Norwegian Shipowner's Association's Energy Management in Practice project in which MARINTEK participates together with a group of shipowners.

I would especially like to thank Professor Magnus Rasmussen at the institute and Brage Mo at Sintef MARINTEK for their support. I would also like to thank Harald Rødseth at MARINTEK for assistance.

As my specialization is in the maintenance part of operation technology, I started this thesis with limited knowledge of machinery systems and engine performance. Many hours have been spent studying literature in order to get an overview on the subjects at hand, and many more have been spent collecting, sorting and organizing relevant data.

In the end I feel that I have gained useful knowledge on the topic of energy efficient operation of ships and also useful experience with independent work.

Trondheim, June 10^{th}, 2011

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SUMMARY

In chapter 2 (task 1), different energy saving measures are explored and related to the Grieg ship *Star Istind's* energy profile.

Star Istind's diesel and fuel oil consumption before implementation of measures:

- Annual HFO consumption = 10 843,3 tons
- Annual MDO consumption = 1 096,5 tons

The assessed measures are

- Mewis Duct
  - Improving flow into the propeller, eliminating losses due to vortices and erratic conditions
  - Estimated savings of 5 %
- Brunel EnviroMarine hull coating
  - Self-polishing coating which reduces friction on the hull
  - Estimated savings of 5,5 %
- Waste heat recovery system
  - Produces electrical power by utilizing exhaust gas heat
  - Partly replaces the auxiliary engines when in operation
  - Estimated production: 6,9 - 7,1 % of main engine output for a single-pressure steam system and 7,9 - 8,1 % of main engine output for a dual-pressure steam system
- Fuel homogenizer
  - Reduces the amount of sludge by breaking apart asphaltene agglomerations otherwise discarded as sludge
  - Improves the combustion process by reducing oil droplets' size
  - Estimated fuel consumption reduction of 2-2,4 %

Savings by the Mewis duct and the EnviroMarine coating result in reduced need for main engine power output. This new power output is the base for calculating the waste heat recovery system's power production. This results in a new MDO consumption corresponding with the reduced auxiliary power need. The new consumptions of HFO and MDO are finally reduced according to the estimated effects of the fuel homogenizer. The resulting consumptions after implementing all the measures are:

- Annual HFO consumption = 9 675,0 tons
  - **Reduction of 1 168,3 tons equaling 10,8 %**
- Annual MDO consumption = 225,8 tons
  - **Reduction of 870,7 tons, equaling 79,4 %**

In chapter 3 (task 2), various engine parameters are presented and compared to the engine's efficiency over time. The goal is to identify the ones that indicate engine efficiency the best.

Engine efficiency is calculated for all the dates using necessary data from the database. By plotting the efficiency over time in a graph, the efficiency and the different parameters are compared.

The results show that, while some of the parameters to a limited degree can indicate overall engine efficiency, the measurement data are too inconsistent to use any of them as definitive references.

The conclusion is that in order to get useful information from such data, more reliable measurement methods must be introduced. A common ground for the data to be presented on, such as an indexing of the measurements, would make it possible to get a more holistic picture of the engine performance.

Chapter 4 (task 3) is to be regarded as a general study of shaft generators and controllable pitch propellers. Shaft generators can be used to produce additional electrical power and are often used together with a variable pitch propeller. The controllable pitch propeller enables thrust changes at constant RPM by adjusting the pitch of the propeller blades.

There are two main types of shaft generators

- Generators producing electricity with frequency varying with engine RPM
- Generators producing constant frequency electricity

Controllable pitch propellers have the following characteristics:

- Enables the propeller to maximize thrust for every RPM
- Rapid maneuvering due to quick changes in thrust
- Reduced propeller efficiency due to the larger propeller boss
- More expensive; higher equipment cost, more complicated maintenance and higher spare part costs, need for drydock

A study conducted by MAN Diesel & Turbo shows that an arrangement with two auxiliary engines and one variable frequency shaft generator with a controllable pitch propeller would consume more lubricating and fuel oil compared to an arrangement with three auxiliary engines and a fixed pitch propeller. This specific study, however, is not applicable to all cases.

The conclusion is that it is the area of application that decides whether installing a shaft generator and/or a controllable pitch propeller results in higher fuel efficiency.

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CHAPTER 1:
INTRODUCTION

In the EMIP project, 5 Norwegian shipowners are working together to find ways to reduce the environmental impact caused by their seagoing fleets. The project group, “Workgroup 5” (WG5) includes BW Gas, Grieg Shipping Group, Höegh Autoliners, Klaveness and Wilh.Wilhelmsen .

A suggested 1 billion tons of CO_2 are released into the atmosphere by the global shipping fleet each year, and the number is expected to increase (1). By reducing their ships’ fuel consumption, shipowners can save money and at the same time do something about this depressing trend.

In the 2009 report by Grieg Shipping Group and the WG5, “A3 Energy Efficiency,” several different measures for reduced energy consumption are identified. These include a variety of suggestions, from technical devices and installations to operational measures and voyage adjustments. Some have already been tested and some are in the idea stage.

This report consists of three subjects related to energy efficient operation of ships.

Task 1 (chapter 2), will focus on promising or already implemented technical measures identified in the aforementioned report by GSG. Potential savings will be related to the energy profile of a GSG vessel to compare energy consumption before and after the implementation of the solutions. The measures studied are the Mewis duct, Brunel EnviroMarine hull coating, waste heat recovery system by MAN Diesel & Turbo, and a fuel homogenizer system.

Task 2 (chapter 3) is a study of engine data from a Höegh ship extracted from MARINTEK’s software TeCoMan. The focus will be on selection of parameters as indicators of a ship’s fuel efficiency.

Task 3 (chapter 4) is a study of different technologies for reduced fuel consumption. Shaft generators and variable pitch propellers are discussed. This part is not related to a specific case, but is to be regarded as a general assessment of the technologies.

CHAPTER 2:
IN-DETAIL ASSESSMENT OF MEASURES

In this part an in-detail assessment of promising or already implemented measures in the EMIP project will be presented.

The energy profile of a GSG ship, *Star Istind,* will be the basis when looking at the potential energy savings for the measures, so as to be able to compare the situation before and after the implementations. It is, however, difficult to assess the combined effects of solutions that have similar areas of function. One example of this is the combination of a duct to set up flow into the propeller and a boss cap fin to optimize the flow astern of the propeller. It is therefore suitable to look at measures that have minimal effect on each other.

2.1 EXAMPLE SHIP – *STAR ISTIND*

As mentioned, the basis of this part is the energy profile of GSG's *Star Istind*. This ship is one of Grieg's I-class vessels, which are categorized as general cargo/container ships.

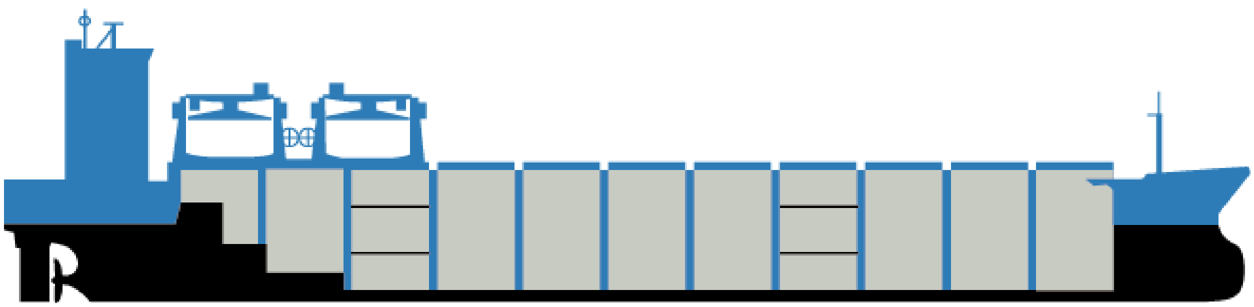

FIGURE 1: GRIEG'S I-CLASS (25)

***Star Istind* main characteristics**	
Draft summer	12,32 m
Length overall	198 m
Beam	31 m
Max speed	16 knots
Container capacity	1914 TEU
Deadweight	46 547 mt

TABLE 1: STAR ISTIND MAIN CHARACTERISTICS (25)

The ship is designed with two gantry cranes and removable hatch covers for general cargo, bulk and containers. Its main engine is a MAN Diesel & Turbo 6S60MC, which is a six-cylinder diesel combustion engine with a piston diameter of 60 cm and stroke length of 240 cm (2). At 100 % MCR with 96 RPM it produces 10 520 kW (3).

The ship is also equipped with an auxiliary engine arrangement comprising two Ulstein KRG-8 4-stroke engines producing 2 800 kW at 100 % MCR, and one Ulstein KRG-5 producing 795 kW at 100 % MCR.

2.1.1 ENERGY PROFILE

The energy profile of a ship is a detailed review of the ship's energy consumption. Different situations such as normal seagoing load, ballast and port maneuvering calls for different power needs. This is taken into consideration in the energy profile, where the energy production and consumption for all the different load conditions are accumulated to give a yearly total.

Star Istind's energy profile, as worked out by B. Mo and H. Rødseth at MARINTEK, is presented in Appendix A. Table 2 summarizes the effect production and consumption for each state of the ship's operation.

Extract from energy profile of *Star Istind*								
[kW]	State 1 (3000 h) Full load	State 2 (3000 h) Ballast	State 3 (1000 h) Intermediate	State 4 (20 h) Maneuvering in port	State 5 (580 h) Loading	State 6 (580 h) Discharging	State 7 (600 h) Waiting	State 8 (100 h) Restricted waters
Production	9996,2	9639,0	9281,9	2998,5	813,3	813,3	448,6	0,0
Transmission	578,0	578,0	578,0	2848,6	772,6	772,6	426,2	0,0
Consumption	9772,2	9422,2	9072,2	2820,1	325,2	325,2	421,9	0,0

TABLE 2: EFFECT SUMMARY OF STAR ISTIND (26)

The numbers in table 2 represent the effect produced and consumed in the various load conditions. The production category comprises the main engine, auxiliary engines and boilers, with only the main engine and auxiliary engines contributing in this case. Transmission is the effect that the switchboard is able to distribute to the consumers. This effect is produced by the auxiliary engines. The consumption is all the power consuming systems: propulsion, pumps, various equipment, heating, accommodation, thrusters etc.

The different states represent the ship's modes of operation through one year:

- **Full load** is for laden ship in open sea – approximately 3000 hours through the year.
- **Ballast** is the ship at ballast sailing in open sea – 3000 hours.
- **Intermediate** is sailing in waters where the ship is unable to maintain a constant speed, as in the ballast or laden states – 1000 hours.
- **Maneuvering in port** is the state in which only thrusters are used for maneuvering – 20 hours.
- **Loading/Discharging** is when the ship is in port, loading or discharging cargo. Some main engine pumps, accommodation and cargo handling equipment are in use – 580 hours each.
- **Waiting** is when the ship is idle in port, e.g. waiting for cargo or crew. Only accommodation and main engine pumps are operating in this state – 600 hours.
- **Restricted waters** – 100 hours.

Another table shows the amount of energy needed in the different states based on the effect produced and time spent in each:

Extract from energy profile of *Star Istind*									
[MJ]	State 1 Full load	State 2 Ballast	State 3 Intermediate	State 4 Maneuvering in port	State 5 Loading	State 6 Discharging	State 7 Waiting	State 8 Restricted waters	**Total (1 year)**
Production	107 958 472,8	104101329,9	33414729,0	215890,3	1698077,2	1698077,2	968956,9	0,0	**248 357 456,2**
Transmission	6242181,8	6242181,8	2080727,3	205095,8	1613173,3	1613173,3	920509,1	0,0	**173 038 69,2**
Consumption	105539760,0	101759760,0	32659920,0	203044,8	1597041,6	1597041,6	911304,0	0,0	**242 670 830,4**

TABLE 3: ENERGY SUMMARY OF STAR ISTIND(26)

Table 4 is an excerpt from the energy profile showing the breakdown of the production, transmission and consumption categories as presented in MARINTEK's energy profile.

Extract from energy profile of *Star Istind*				
Energy category	**Shortname**	**Total consumption (1 year)**		
		Unit	**Count**	**Total**
Production	Main Engine indicated	MJ	230142857,1	230142857,1
	Auxiliary Engines indicated	MJ	18214599,0	248357456,2
	Boiler	MJ	0,0	**248 357 456,2**
Transmission	Shaft motor	MJ	0,0	0,0
	Shaft generator	MJ	0,0	0,0
	Waste Heat Recovery	MJ	0,0	0,0
	Switchboard	MJ	17303869,1	**173 038 69,1**
Consumption	Shaft Power	MJ	225540000,0	225540000,0
	Main engine pumps over 50 kW (Including car hold vent fan)	MJ	2969983,2	228509983,2
	Main engine pumps under 50 kW	MJ	7883688,0	236393671,2
	Accomodation over 50 kW	MJ	2598393,6	238992064,8
	Accomodation under 50 kW	MJ	2681616,0	241673680,8
	Cargo Heat over 50 kW	MJ	0,0	241673680,8
	Cargo Heat under 50 kW	MJ	0,0	241673680,8
	Thrusters over 50 kW	MJ	150840,0	241824520,8
	Thrusters under 50 kW	MJ	532,8	241825053,6
	Cargo pumps over 50 kW	MJ	0,0	241825053,6
	Cargo pumps under 50 kW	MJ	0,0	241825053,6
	Cargo gear over 50 kW	MJ	140832,0	241965885,6
	Cargo gear under 50 kW	MJ	481320,0	242447205,6
	Ballast pumps over 50 kW	MJ	223624,8	242670830,4
	Ballast pumps under 50 kW	MJ	0,0	242670830,4
	Reefers over 50 kW	MJ	0,0	242670830,4
	Reefers under 50 kW	MJ	0,0	**242 670 830,4**

TABLE 4: CATEGORY BREAKDOWN OF ENERGY PROFILE (26)

The table gives an overview of where the energy is produced and spent. Here, only the year total is shown, but this breakdown also exists for states 1-8, as seen in Appendix A.

It is important to note that the table only gives the net energy produced by the engines, i.e. the usable energy. For fuel consumption calculations, one would have to take into consideration the energy lost between the fuel tank and the engine output.

Looking at the $\frac{switchboard}{AE\ indicated\ work}$ ratio, we find that the energy loss in the conversion from mechanical to electrical energy is 5 %, which is reasonable. Considering the electrical $\frac{consumption}{switchboard}$ ratio, the general electrical energy loss after the switchboard is about 1 %, which also seems reasonable. The overall efficiency for the auxiliary engines is therefore 94.05 %.

Similarly, the $\frac{shaft\ effective\ work}{ME\ indicated\ work}$ ratio shows a loss of 2 % between the engine and the shaft. This is due to small losses, such as friction, losses in couplings etc. In other words, the mechanical efficiency is 98 %.

2.1.2 CONSUMPTION CALCULATIONS

MARINTEK's energy profile includes the following assumptions (4):

- Main engine efficiency of 52 % which is the $\frac{work\ at\ output\ flange}{HFO\ energy\ input}$ ratio
- Main engine uses HFO with heating value of 40 000 kJ/kg
- Auxiliary engine efficiency of 40 % which is the $\frac{switchboard}{MDO\ energy\ input}$ ratio
- Auxiliary engines utilizes MDO with heating value of 42 700 kJ/kg

The main and auxiliary engines' efficiencies are within expectations for these machineries. The same goes for the heating values of HFO and MDO. The point is to compare the situation before and after implementing the measures, so these numbers work as reference.

From before we have mechanical efficiency of 98 % for the main engine, and mechanical → electrical power conversion efficiency of 94.05 % for the auxiliary engines. The fuel consumption in each state is calculated following these methods:

$$FOC_{HFO}[ton] = \frac{\frac{ME\ indicated\ work\ \cdot\ mechanical\ efficiency}{ME\ efficiency}[MJ]}{HFO\ heating\ value\ [MJ/ton]} \qquad [1]$$

$$FOC_{MDO}[ton] = \frac{\frac{AE\ indicated\ work\ \cdot\ conversion\ efficiency}{AE\ efficiency}[MJ]}{MDO\ heating\ value\ [kJ/ton]} \qquad [2]$$

Fuel consumption for all states, and in total, is presented as follows:

Consumption calculations										
		State 1	State 2	State 3	State 4	State 5	State 6	State 7	State 8	**Total, 1 year**
Main engine net production	MJ	101387755	97530612	31224490	0,0	0,0	0,0	0,0	0,0	**230142857**
Main engine efficiency	%	52	52	52	52	52	52	52	52	
Mechanical efficiency	%	98	98	98	98	98	98	98	98	
HFO heating value	MJ/kg	40,0	40,0	40,0	40,0	40,0	40,0	40,0	40,0	
Aux engine net production	MJ	6570718	6570718	2190239	215890	1698077	1698077	968957	0,0	**19912676**
Aux engine efficiency	%	40	40	40	40	40	40	40	40	
Conversion efficiency	%	94,05	94,05	94,05	94,05	94,05	94,05	94,05	94,05	
MDO heating value	MJ/kg	42,7	42,7	42,7	42,7	42,7	42,7	42,7	42,7	
Consumption, HFO	ton	4776,9	4595,2	1471,2	0,0	0,0	0,0	0,0	0,0	**10843,3**
Consumption, MDO	ton	361,8	361,8	120,6	11,9	93,5	93,5	53,4	0,0	**1096,5**

TABLE 5: FUEL CONSUMPTION FOR ALL STATES AND 1 YEAR TOTAL, HFO AND MDO

We have from the table that our reference fuel consumption is 10 843,3 tons of HFO and 1 096,5 tons of MDO per year.

2.2 SUGGESTED MEASURES; TECHNOLOGIES AND ECONOMIC CONSIDERATIONS

One issue when assessing the effects of implementing measures is the fact that many of the possible solutions may affect each other's impact on the performance of the ship. Exactly how the measures affect each other is difficult to say and one may only find out by conducting tests or, in the case of hydrodynamic measures, performing advanced computational fluid dynamics calculations.

Examples of such conflicting measures are combinations of different wake enhancing equipment on and around the propeller and various combinations of machinery and combustion enhancing equipment such as fuel homogenizer, fuel heating, or cooling water adjustments.

Therefore, this report is limited to a few selected measures that are not in direct conflict with each other to be able to give a realistic estimate of the potential energy savings.

2.2.1 MEWIS DUCT

The Mewis duct is a type of nozzle fitted ahead of the propeller. Its purpose is to adjust the flow into the propeller in order to reduce the formation of vortices and rotations in the ship's wake.

Technology

A problem with large vessels with high block coefficients is the generation of vortices near the stern. This results in unfavorable conditions for the propeller because of the turbulent flow and sub-optimal inflow of water into the propeller. The propeller also generates rotation in the slipstream, which is a source for energy loss. The Mewis duct seeks to solve these issues by providing a more homogenous inflow, creating better working conditions and higher efficiency for the propeller. The duct itself is also thrust-generating.

Figure 2 is a drawing of the stern of a ship and shows roughly how the duct affects the flow conditions around the propeller. The figure is not necessarily a realistic depiction, but it shows the working principle.

In addition, Mewis' system has a set of fins fitted inside the duct. This fin system sets up the pre-swirl and reduces the rotational energy loss in the slipstream.

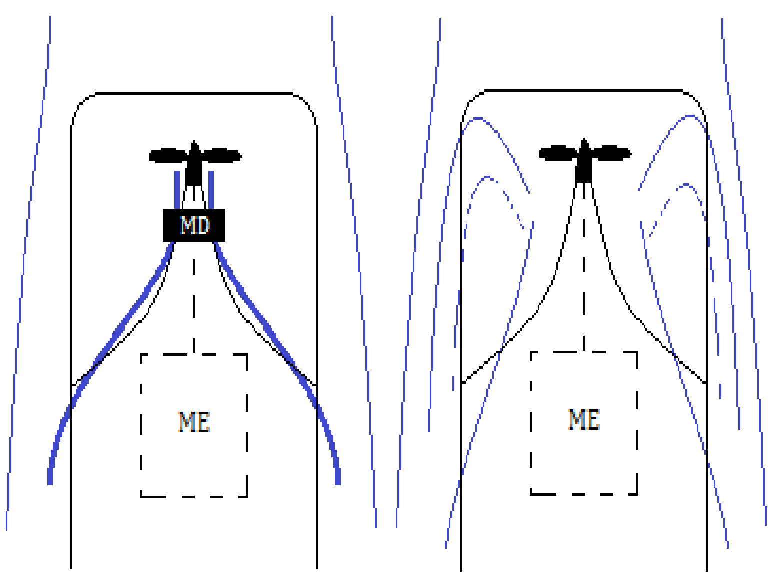

FIGURE 2: FLOW CONDITIONS WITH AND WITHOUT THE MEWIS DUCT

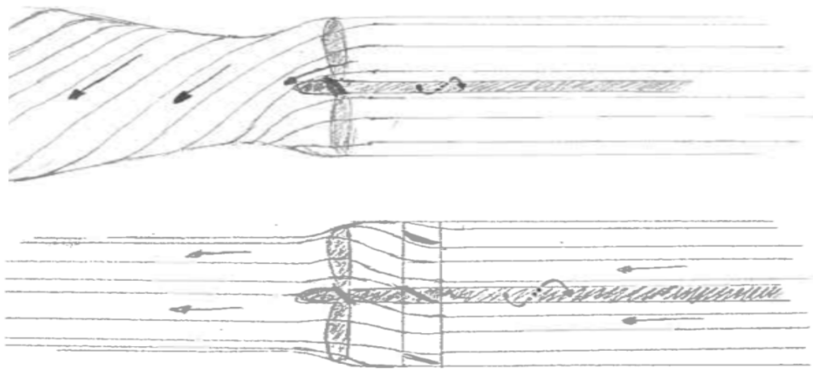

FIGURE 3: SLIPSTREAM CONDITIONS WITH (BOTTOM) AND WITHOUT FIN SYSTEM (4)

Figure 3 illustrates the working principle of a fin system set to prevent such loss astern of the propeller.

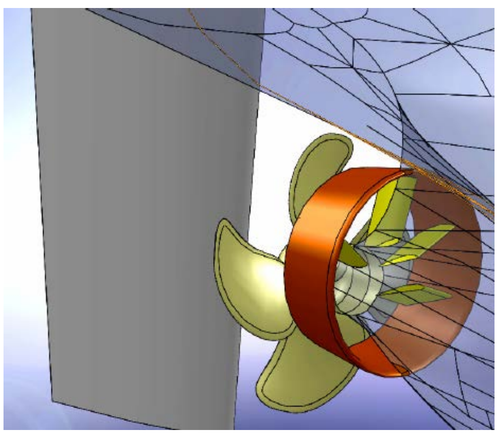

FIGURE 4: CAD DRAWING OF A MEWIS DUCT SETUP (4)

Figure 4 shows the Mewis Duct fitted ahead of the propeller. Note that the duct is slightly offset vertically compared to the propeller and that the fins are asymmetrical compared to the rest of the ship. This is due to optimization using CFD and model testing. These parameters are subject to change from ship to ship depending on the inflow and wake conditions. Other aspects favoring the Mewis duct system are that it is relatively quick and uncomplicated to install on both newbuildings and on vessels in operation, and that it contains no moving parts. Compared to similar measures, such as contra-rotating propellers, this is an advantage because of the minimal maintenance following installation (4).

Effects

Star Istind was fitted with a Mewis duct in October of 2009. Preliminary analyses have shown a fuel consumption reduction of about 5 %. Investment costs are around USD 275 000 (5).

2.2.2 BRUNEL ENVIROMARINE HULL COATING

Hull coating is something that greatly impacts a ship's ability to move through water. The reason coating is of such importance is that it prevents the forming of marine organisms, corrosion and other irregularities on the hull. Hull roughness will increase a ships' friction resistance significantly and up to 15 % per year if the hull is not cleaned (6). This part focuses on a particular hard coating system, Brunel's EnviroMarine, which has shown promising hydrodynamic and resistive properties.

Technology

EnviroMarine is a coating made up of a two-component resin. It is solvent-free, meaning for instance that there is no forming of pores which provide potential for marine growth. The coating has a coefficient of friction which is smaller than glass, and again this means marine growth will face difficult conditions. It also means that even at low speeds the coating is self-polishing. The coating is very resilient and is claimed to be tougher than concrete and more flexible than the steel on which it is applied (7). As of January 2011, Brunel EnviroMarine is the only antifouling coating certified by Det

Norske Veritas for a 10-year docking interval (8). As the coating does not release any substances or pollute in any way, it is also considered to be an environmentally friendly measure.

After application, the hull's roughness will be less than 20 µm. In comparison, normal figures for a ship's roughness after dry docking is in the range of 50 -150 µm (9). The coating's thickness is about 300 µm, each layer being 150 µm. Added weight after application is 348 g/m^2, area being the wet surface area of the ship. This weight is negligible because the ship's weight with EnviroMarine coating and its weight with conventional coating will be close to identical. Not negligible is EnviroMarine's added cost of about USD 15/m^2 (10) (7). A wet surface area guestimate of about 8 000 m^2 (11) gives an added cost of USD 120 000.

Effects

In a Grieg Shipping Group project a number of coating manufacturers were approached and fuel savings were estimated. Savings of 5,5 % were indicated for Brunel EnviroMarine (6). This number is due to the reduced friction on the hull of the ship, resulting in less marine growth and water resistance. Recalling that the friction resistance of a ship can increase with up to 15 % per year without cleaning, a self-cleaning hull as provided by EnviroMarine will have a large positive impact on fuel use.

2.2.3 WASTE HEAT RECOVERY SYSTEM

We have from chapter 2.1.2 that the main engine on *Star Istind* operates with an efficiency of 52 %. That means 48 % of the total energy is wasted before the propeller shaft.

The Sankey diagram in Appendix B illustrates where energy is lost, from fuel to shaft power output. The numbers can be summarized as in Table 6.

Losses in a MAN 6S60MC-C engine	
Fuel energy	100 %
Heat radiation	-0.8 %
Air cooler	-14.6 %
Exhaust gas	-25.0 %
Jacket water	-5.8 %
Lubricating oil cooler	-3.3 %
Shaft output	**50.5 %**

TABLE 6: LOSSES IN A MAN 6S60MC-C ENGINE (20)

The engine in the Sankey diagram and the table is of the same type as in our sample ship, *Star Istind*, but has a higher SMCR of 13 500 kW compared to our 10 520 kW SMCR (3). Although it is a different setup, we can expect the losses in our system to be roughly the same as in Table 6. As shown in the table, exhaust gas heat losses account for about half of the total losses. Being able to utilize some of this energy for electricity production could mean that the diesel generators would have to produce less electricity from their own source of energy.

Technology

The exhaust heat is normally used for cargo and accommodation heating purposes. This is achieved through a boiler that heats steam which is then distributed to the various systems.

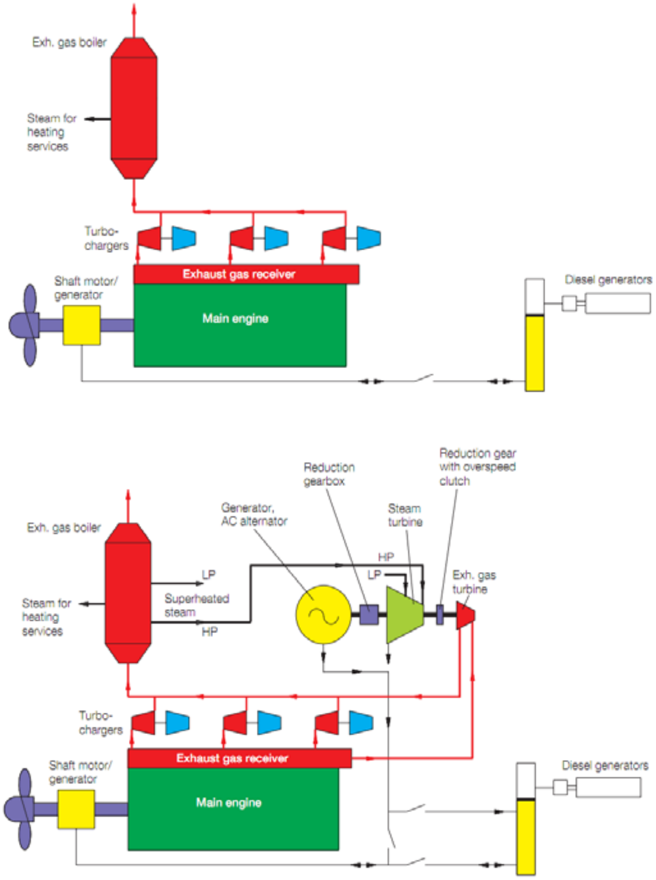

FIGURE 5: SETUP WITH (BOTTOM) AND WITHOUT HEAT RECOVERY (13)

Figure 5 (top) shows a conventional setup with exhaust gas driven turbochargers and exhaust gas boiler producing steam. The steam consumption on a ship varies with its size. For a vessel such as *Star Istind* the steam need is approximately 700 kg/h (13).Normal steam pressure is 5-7 bars (12).

Figure 5 (bottom) illustrates MAN Diesel & Turbo's power producing heat recovery system. The system roughly includes an exhaust gas boiler, an exhaust gas driven turbine, a steam turbine and the electrical power generator.

Part of the exhaust gas flow after the combustion process is bypassed the turbochargers and is led to and powers the exhaust gas power turbine. The results of this bypass are reduced amount of air in the cylinder and consequently reduced amount of exhaust gas. This leads to reduced energy loss over the turbochargers and thus increased exhaust gas temperature after the turbochargers. This heat produces superheated steam in the boiler. The steam is used to power a steam turbine which in turn powers the AC generator together with the exhaust gas turbine from the exhaust bypass, as shown in Figure 5.

At 50 % or less of SMCR, the exhaust bypass is closed resulting in lower exhaust gas temperature after the turbochargers. Therefore, in this case we will only apply the assumed effects of the thermo efficiency system (TES) on the states in which the engine load is >50 % SMCR.

MAN Diesel & Gas describes two systems for producing steam from the exhaust gas.

The first and simplest is the single-pressure steam system. In this system, water from the hot well is preheated in a low-temperature section of the exhaust gas boiler before being evaporated at 165°C and absolute pressure of 7 bars. This steam is contained in a steam drum and some of it is used for the various heating services onboard. The rest of the steam is circulated through the high temperature part of the exhaust boiler and led to the steam turbine powering the AC generator.

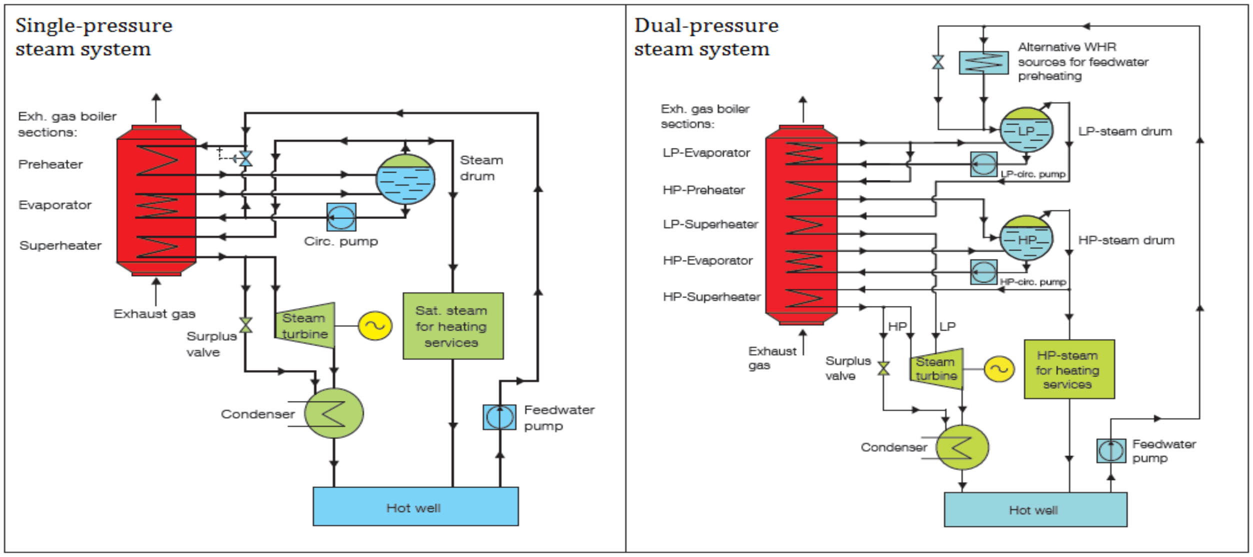

FIGURE 6: SINGLE- AND DUAL-PRESSURE STEAM SYSTEMS (13)

The second system is a dual pressure steam arrangement. Water from the hot well is transported via an external preheater before it is evaporated at low pressure (LP), 4 bars absolute, and 144°C in the exhaust boiler. This steam is contained in the LP steam drum. Some of the steam after the LP evaporation is diverted to the high pressure (HP) system and condensed under the increased pressure. This water is preheated in the boiler before being evaporated at 10 bar absolute and 180°C and contained in a HP steam drum. Some of this steam, as in the single-pressure system, is used for heating services, while the rest is superheated through the boiler and used to power the steam turbine. The steam in the LP drum is also superheated and used to power the steam turbine; however, due to its lower pressure it enters the steam turbine at a later stage than the HP steam.

The reason for the need for an external preheater for the LP flow in the dual-pressure system is the risk of oily soot deposits on the boiler tubes, which is the result of too low exhaust temperature in the boiler. Therefore the LP flow is not preheated in the boiler, something that would cause the exhaust temperature to decrease.

Similar for these two systems is the surplus valve arrangement which regulates the amount of superheated steam going into the steam turbine. This way, one can regulate the power output from the generator in order to produce the correct amount, e.g. when operating in parallel with auxiliary engines.

Effects

As previously stated, both the exhaust gas turbine and the steam turbine drive the generator. The total effective power output is therefore the combined power output of the two turbines minus the efficiency losses over the gears and the generator. The numbers in this section are all based on ambient air temperature of 25°C, which is the ISO reference condition (14).

At 100 % MCR, MAN Diesel & Turbo indicates an output from the exhaust gas turbine alone of maximum 4,6 % of the ME output for their 12K98ME engine producing 68 640 kW at 100 % MCR. This output percentage is subject to a slight decrease as the engine size is reduced, as with our example ship's 6S60MC producing 10 520 kW at 100 % MCR.

Recalling that the TES does not produce power in the case of the engine operating at 50 % MCR or less, the only states relevant to this part, and the only ones in which the ME produces power, are full load, ballast and intermediate. Following is an example showing what output we can expect from the current ME outputs and loads. From the energy profile we have:

State 1 Full load		**State 2 Ballast**		**State 3 Intermediate**	
Unit	Item	Unit	Item	Unit	Item
kW	9387,8	kW	9030,6	kW	8673,5

TABLE 7: POWER PRODUCTION FOR STATES 1-3 (26)

Considering that 100 % of MCR is 10 520 kW, the engine loads in these states are 89 %, 86 % and 82 % respectively.

MAN Diesel & Turbo's research indicates that the exhaust gas turbine power output roughly follows this formula (13):

$$\%\ output\ of\ ME\ at\ \boldsymbol{X}\ MCR \approx \frac{[output\ at\ 100\ \%\ MCR]\cdot(1-\sqrt{1-\boldsymbol{X}})}{\boldsymbol{X}} \qquad [3]$$

Recalling that MAN's 12K98ME engine's exhaust gas turbine output at 100 % MCR is 4,6 % of MCR, and considering the not insignificant difference in engine size, we will here assume that our system is capable of producing 4,2 % of MCR.

The formula (3) above, for state 1 at 89 % MCR, gives:

$$\%\ output\ of\ ME\ at\ 0{,}89\ MCR \approx \frac{(0{,}042)\cdot\left(1-\sqrt{1-0{,}89}\right)}{0{,}89} = 3{,}1\ \%$$

Similarly for states 2 and 3 we have 3,1 % and 2,9 % of ME output, respectively. These rough estimates are to be considered conservative.

For the steam turbine outputs, we have no formula to help us calculate the power output relative to the engine load. In this case we will look at data for other MAN engines and guestimate what the corresponding numbers for *Star Istind's* engine are. The following numbers are presented by MAN Diesel & Turbo for four different engine systems:

ME Type	6S90ME-C	7K98ME-C	12K90ME	12K98ME
kW, SMCR	29 340	39 970	54 840	68 640
Output at 85 % MCR	24 939	33 975	46 614	58 344
Single-pressure, % of ME output	4,5	4,8	4,8	4,9
Dual-pressure, % of ME output	5,4	5,9	6,0	6,0

TABLE 8: ELECTRICAL OUTPUT FROM STEAM TURBINE IN TES (13)

There is no clear connection between the engines' output and the TES' output, other than that the electrical output is between 4,5 and 4,9 % for single-pressure systems and between 5,4 and 6,0 % for dual-pressure systems, and that lower output on the ME means lower percentage output from the steam turbine. A conservative guestimate for *Star* Istind's smaller engine is therefore believed to be around 4 % power output for a single-pressure system and around 5 % for a dual-pressure system, also taking into account mechanical and conversion losses in the turbine and generator setup.

Accordingly, the total output from the heat recovery unit, comprising an exhaust gas power turbine and a steam turbine, driven by a single-pressure or a dual-pressure steam system is estimated to be

- 6,9 - 7,1 % of ME output for a single-pressure steam TES
- 7,9 - 8,1 % of ME output for a dual-pressure steam TES

in ISO ambient conditions.

The TES' power output directly reduces the auxiliary engine's power production with the same amount, resulting in MDO fuel savings and emission reductions. Installing a TES will affect the main engine efficiency by increasing its fuel consumption by up to 1,8 %; however, optimization of the ME can make it possible to gain the effects of the TES without fuel penalty. Hence, this increment is ignored.

As one would expect, the dual-pressure system comes with a higher installation cost because of its more complicated arrangement, but the payback time is approximately the same due to the higher power output from the dual-pressure system (13).

2.2.4 FUEL HOMOGENIZER

A fuel homogenizer is a system designed to reduce sludge and improve the combustion process, leading to decreased fuel use per kWh produced. The systems often comprise a reducer part, which reduces sludge, and an improver part, which improves the oil droplets before the combustion. Additionally, a water injection device can be installed in order to significantly reduce the emission of NOx in the exhaust. Here the focus is on a system utilizing a reducer and an improver.

Technology

One problem with marine fuel oil is the inevitable sludge. This sludge must be separated as it can be damaging to the engine if it enters the combustion process together with the ordinary fuel oil. However, the sludge also contains relatively large amounts of asphaltenes. Asphaltenes are agglomerations of tar-like oil substances with high calorific values. The asphaltenes are filtrated together with the sludge, meaning that their potential energy is lost before combustion takes place.

What a reducer does is to break apart the agglomerations of asphaltenes before the separator to make these particles smaller, enabling them to pass through the sludge separation process. This has proved to reduce the amount of sludge by as much as 80 % (15) (16).

The improver part of the system is installed before the engine and its purpose is to convert the flow of fuel oil into tiny droplets. This improves the combustion of the oil by allowing the oil droplets to be more evenly distributed in the cylinder and having better contact with oxygen, a result of the conversion from larger drops (smaller surface area) to smaller drops (increased surface area).

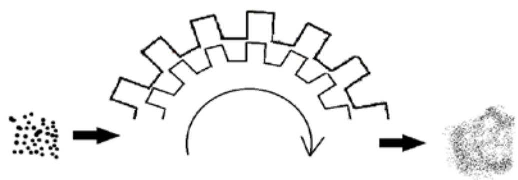

FIGURE 7: BASIC PRINCIPLE OF THE ROTOR-STATOR HOMOGENIZER ARRANGEMENT

The most common homogenizers are based on a conic rotor-stator arrangement with the addition of ultrasonic sound waves. The oil is pumped through the device and is exposed to heavy shear forces which together with the sound breaks apart asphaltene chains and reduces the droplets' size, as seen in figure 7. The rotor and stator are never in contact with one another, so the wear on the components is at a minimum. The droplets' size after the improver is reduced to around 3 µm, making for a better combustion (15) (16).

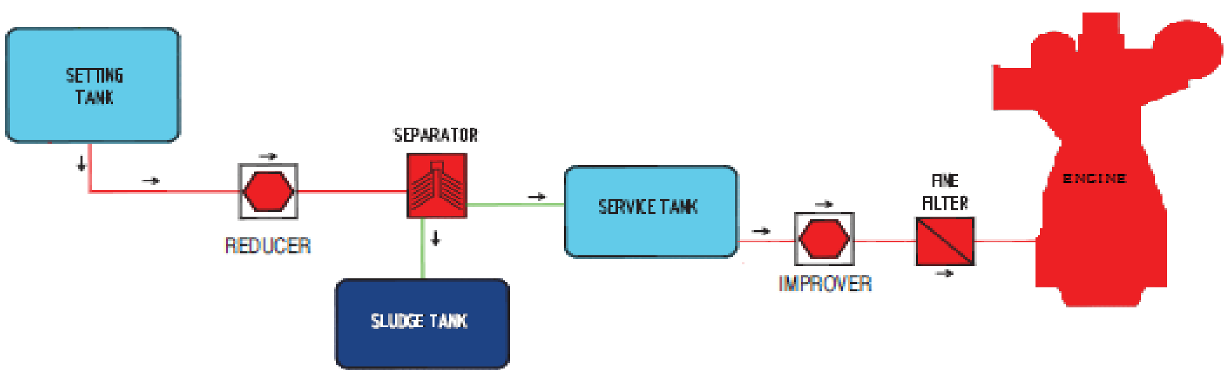

FIGURE 8: FUEL TREATMENT SYSTEM (15)

Figure 8 is a simplified sketch of the homogenizer system with its main parts, excluding pumps and valves. Fuel oil is pumped from the setting tank through the reducer where the usable asphaltene's sizes are reduced. The next step is the separator in which sludge is removed, and then fuel is pumped into the service tank. From there, fuel is sent through the improver, and finally through the fine filter, before entering the combustion process in the engine. If the system is to include a fresh water injector, this would be fitted before the improver in order for the water and fuel oil to be mixed. The homogenizer system described here can be used on both 2-stroke HFO engines and on 4-stroke MDO engines.

Effects

The first result of installing a homogenizer as described is that fuel oil asphaltenes that would otherwise be discarded as sludge now can be utilized. The amount of sludge is reduced. Secondly, the reduced size of the fuel droplets entering the cylinder leads to a more optimized and complete combustion of the fuel. In the end we will get better fuel efficiency, using fewer grams of fuel per kilowatt produced.

Homogenizer manufacturers indicate a 2-2,4 % gain on 2-stroke engines and about 1,25 % gain on 4-stroke engines. (15) (16). These gains translate to reduced fuel consumption.

2.3 UPDATED ENERGY PROFILE

Having assessed a few energy-saving measures that are likely candidates for implementation, these will now be related to *Star Istind's* energy profile. This provides an updated profile with aggregated effects from the measures described in chapter 2.2.

2.3.1 FUEL SAVING CALCULATIONS

The measures assessed in chapter 2.2 have different ways to improve the energy efficiency:

- **The Mewis duct and the coating technology** will improve the way the ship runs through water. This means that with the implementation of these solutions the ship can either travel faster with the same amount of power, or it can travel at the same speed with lower power, relative to before implementation. In this report it is assumed that the vessel will be sailing at the same speed, therefore lowering its power need in order to maintain that speed.
- **The thermo efficiency system** will take its power from the exhaust heat following the combustion process and will not affect other stages of the ME system in any significant way as this energy would be lost without heat recovery. Its power production will depend on the ME's power output and will directly reduce the AEs output. The best way to estimate this is therefore to first figure what power output and load is expected from the ME after the implementation of the Mewis duct and the coating.
- **The fuel homogenizer** will increase the amount of usable fuel by lowering the amount discarded as sludge. It will also improve the way the fuel is combusted. Here it is assumed that the homogenizer will enable the same amount of power to be produced by the engine with a smaller amount of fuel. This is best estimated when we know the ME's and AEs' fuel use with the above measures implemented.

1. Power reduction with Mewis duct

Tests conducted with the Mewis Duct installed have shown a 5 % reduction of the fuel use. We will now see what changes in power production corresponds to this reduction.

Our basis is the energy profile of *Star Istind* assumed not to include any form of efficiency increasing installations. Referring to the consumption calculations from chapter 2.1.2, we have a HFO consumption of 10 843,3 tons/year. 5 % of this is 542,1 tons/year. But in order to calculate how this translates into power reduction from the ME, we need to find the fuel savings for each of the different states and factor in HFO heating value, engine and shaft efficiencies, and the time spent in each state per year. The only relevant states here are the ones in which the ME produces power, i.e. states 1, 2 and 3. The expression will then become:

$$Power\ [kW] = \frac{Yearlyconsumption\ \left[\frac{ton}{year}\right] \cdot HFO\ heating\ value\ [\frac{kJ}{ton}] \cdot \frac{ME\ efficiency}{Shaft\ efficiency}}{State\ duration\ [s]} \qquad [4]$$

We can control that this will give us the correct ME power output using the consumption for state 1 from table 5:

$$Power\ [kW] = \frac{4\ 776{,}9\ \left[\frac{ton}{year}\right] \cdot 40 \cdot 10^6 \left[\frac{kJ}{ton}\right] \cdot \frac{0{,}52}{0{,}98}}{3\ 000[h] \cdot 3\ 600\frac{s}{h}} = 9\ 387{,}7$$

That number is 0,1 kW less than what the energy profile for *Star Istind* lists, but this is attributed to rounding off of various numbers in the calculation. So this method can be used.

We can see from the expression that 5 % reduction in fuel consumption equals 5 % reduction in power production. We then have the new ME power output for state 1:

$$9\ 387{,}8\ kW \cdot (1 - 0{,}05) = 8\ 918{,}4\ kW$$

Doing the same for states 2 and 3 gives:

Mewis duct:			
State	kW after	SMCR, kW	Load after
State 1	8 918,4	10 520,0	0,85
State 2	8 579,1	10 520,0	0,82
State 3	8 239,8	10 520,0	0,78

TABLE 9: POWER REDUCTION AND ME LOAD AFTER INSTALLATION OF A MEWIS DUCT

2. Power reduction with Brunel EnviroMarine coating

As with the Mewis duct, the coating also allows for a ME power output reduction. As mentioned in chapter 2.2.2, an estimated 5,5 % reduction in fuel consumption can be achieved. We will assume a more conservative estimate of 5 %.

Taking the power reduction from the Mewis duct, state 1 power will now be reduced by another 5 %:

$$8\ 918{,}4\ kW \cdot (1 - 0{,}05) = 8\ 472{,}4\ kW$$

And for all relevant states:

Brunel Enviromarine coating			
State	kW after	SMCR, kW	Load after
State 1	8 472,4	10 520,0	80,1 %
State 2	8 150,1	10 520,0	77,4 %
State 3	7 827,8	10 520,0	74,4 %

TABLE 10: POWER REDUCTION AND ME LOAD AFTER APPLICATION OF ENVIROMARINE COATING

3. Power production from the thermo efficiency system

Now that we have found the power outputs and engine loads we have a platform to calculate what we can expect the thermo efficiency system to produce. As previously stated, the TES will only produce power when the ME operates at 50 % load or more, and that means states 1, 2 and 3. This power output will replace the AEs' power production, reducing the consumption of MDO.

As in chapter 2.2.3, we have the following formula provided by MAN for estimation of power output from the exhaust gas turbine:

$$\%\ output\ of\ ME\ at\ \boldsymbol{X}\ MCR \approx \frac{[output\ at\ 100\%\ MCR] \cdot (1 - \sqrt{1 - \boldsymbol{X}})}{\boldsymbol{X}} \qquad [3]$$

where **X** is the load for the state in question and 100 % MCR is 10 520 kW. We have assumed a TES exhaust turbine output of 4,2 % at 100 % MCR, so for state 1 we then have, for the exhaust gas turbine part of the TES:

$$\%\ output\ of\ ME\ at\ 0{,}8\ MCR \approx \frac{0{,}042 \cdot \left(1 - \sqrt{1 - 0{,}8}\right)}{0{,}8} = 2{,}9\ \%$$

Application of this estimation to the other states gives:

Exhaust gas turbine output from TES		
State	Load	Output, % of ME output
State 1	0,80	2,9 %
State 2	0,77	2,8 %
State 3	0,74	2,8 %

TABLE 11: EXHAUST GAS TURBINE OUTPUT FROM TES

In chapter 2.2.3 we estimated that a single-pressure steam system will give an output from the steam turbine of 4 % of MCR, and a dual-pressure system results in an output of 5 %.

Combining the exhaust turbine output and the steam turbine output, we get the following output from a single-pressure TES:

Single-pressure steam TES			
State	% of ME output	ME output, kW	TES output, kW
State 1	6,9 %	8 472,4	584,6
State 2	6,8 %	8 150,1	554,2
State 3	6,8 %	7 827,8	532,3

TABLE 12: EXPECTED POWER FROM SINGLE-PRESSURE STEAM TES

From *Star Istind's* energy profile, we have the original power production. Following is a table showing the reduction percentage of the AE power production when using a single-pressure TES:

Power reduction on AE with single-pressure TES				
State	AE output before TES, kW	TES output, kW	Difference, kW	Reduction %
State 1	608,4	584,6	23,8	96,1
State 2	608,4	554,2	54,2	91,1
State 3	608,4	532,3	76,1	87,5

TABLE 13: POWER REDUCTION ON AE WITH SINGLE-PRESSURE TES

What table 13 shows is that the single-pressure TES will cover almost all of the auxiliary power needed in states 1, 2 and 3. The power not covered by the TES must still be produced, and one way to achieve this could be to use a shaft generator to cover the difference. This means that the ME have to produce a little more power, but the increased load on the engine will not affect the above calculations in any significant way (the difference is less than 1 % of the ME output in all states). On the other hand, the installation of a shaft generator is a rather costly solution to cover 23-76 kW.

The obvious solution is therefore to install a dual-pressure TES which will be able to cover all the power needs previously met by the AE:

Dual-pressure steam TES			
State	% of ME output	ME output, kW	TES output, kW
State 1	7,9 %	8 472,4	669,3
State 2	7,8 %	8 150,1	635,7
State 3	7,8 %	7 827,8	610,6

TABLE 14: EXPECTED POWER OUTPUT FROM A DUAL-PRESSURE STEAM

The TES can be controlled in order to give the exact power output needed.

The costs of investment of the single- and dual-pressure systems are relative to the size of the engine, and the bigger the engine, the lower the investment cost in USD per kW. Seeing as our engine is just over 10 000 kW, a TES is likely to have a longer payback time than engines with significantly higher SMCR. Assuming that the only AE running in the states in question is the small 795 kW AE, installing a TES may render this engine unnecessary as TES output fully replaces that of the AE, so the payback time may potentially be reduced. One must keep in mind, though, that the TES only produces power in states 1, 2 and 3. For the other states requiring auxiliary power, one must still be able to cover this power with the other AEs.

4. Consumption reduction by fuel homogenizer

The fuel homogenizer system will affect the engine in a way that enables the same amount of power to be produced by less fuel. Therefore, before trying to estimate the impact of a fuel homogenizer we must first find the fuel consumption after the implementation of the measures previously discussed.

The assumptions made and the method of calculating the consumption is the same as in chapter 2.1.2.

Power and consumption after implementing of Mewis duct, Enviromarine coating and TES											
		Unit	State 1	State 2	State 3	State 4	State 5	State 6	State 7	State 8	**Total, 1 year**
	Duration	h	3000	3000	1000	20	580	580	600	100	
Production	Main Engine	kW	8472,4	8150,1	7827,8	0	0	0	0	0	
	Auxiliary Engines	kW	0	0	0	2998,5	813,3	813,3	448,6	0	
	Heat recovery	kW	608,4	608,4	608,4	0	0	0	0	0	
Consumption	HFO, ME	ton	4311,1	4231,8	1354,8	0	0	0	0	0	**9897,7**
	MDO, AE	ton	0	0	0	10,7	84,5	84,5	48,2	0	**228,1**

TABLE 15: CONSUMPTION AFTER IMPLEMENTING OF MEWIS DUCT, ENVIROMARINE COATING AND TES

Now we have the new base consumption of HFO and MDO, and are able to apply the effects of the homogenizer.

Estimates of 2-2,8 % fuel reduction for HFO systems and 1,25 % for MDO systems are indicated by manufacturers. Again we will use the more conservative estimations of 2,25 % and 1 % respectively. This results in the following new fuel oil consumptions for HFO and MDO:

$$FOC_{HFO,new} = 9897{,}7\ ton \cdot (1 - 0{,}0225) = 9675{,}0\ tons$$

$$FOC_{MDO,new} = 228{,}1 ton \cdot (1 - 0{,}01) = 225{,}8\ tons$$

Conclusion: Fuel savings in tons and dollars

The yearly saved fuel consumption is the difference between the previously calculated consumption, without any of the energy efficiency measures, and the new consumption including those measures:

$$\Delta FOC_{HFO} = 10843{,}3\ ton - 9675{,}0\ ton = 1168{,}3\ ton$$

$$\Delta FOC_{MDO} = 1096{,}5\ ton - 225{,}8\ ton = 870{,}7\ ton$$

The reduction in HFO consumption is 10,8 % while the reduction in MDO consumption is 79,4 %. These reductions are significant. Especially the reduction of MDO is large, which is due to the assumption that the running of an auxiliary engine is not needed in states 1, 2 and 3 (which is most of the year). This may not be a realistic assumption, but the fact is, according to the calculations made, that the heat recovery system covers all of the auxiliary power needs during these states.

The numbers used in this chapter are all based on various manufacturers' estimates or results from testing. Most of the time conservative estimates have been chosen in order to get as realistic figures as possible. It is however difficult to say how different measures affect each other without conducting more thorough testing. Therefore, these numbers are to be considered rough theoretical estimates.

The potential savings in monetary terms varies, as the fuel oil price, and the general oil price, is fluctuating. That means that the higher the fuel oil price, the greater the potential savings will be.

The table below presents the expected savings for low, neutral and high oil prices (17):

Estimated annual USD savings			
	HFO price, USD	MDO price, USD	
Low estimate	450	700	(June 2010)
Neutral estimate	550	850	(February 2011)
High estimate	650	1000	(April 2011)
	HFO savings, tons	MDO savings, tons	
	1168,3	870,7	
	HFO savings, USD	MDO savings, USD	**Total savings, USD**
Low estimate	525 735	609 490	**1 135 225**
Neutral estimate	642 565	740 095	**1 382 660**
High estimate	759 395	870 700	**1 630 095**

TABLE 16: ESTIMATED ANNUAL SAVINGS IN USD WITH ENERGY SAVING MEASURES

This shows that even with a low oil price estimate, the savings will be substantial.

2.4 DISCUSSION

Mewis duct

The Mewis Duct is a quite simple system which is easy to install. The installation can be done while the ship is in a planned drydock, minimizing downtime. The setup also includes no moving parts, meaning it cannot completely break down (that is; unless the ship is run aground, in which case a broken duct is probably one of the smaller concerns). The maintenance needed is in the form of cleaning the duct and fins, something which can be done along with hull cleaning at no significant extra cost or time. On all foils there is a chance of cavitation, but it is here assumed that this will not be a problem due to the customization of the duct. Additional maintenance because of the duct can therefore be neglected. The investment cost of installing a Mewis duct in the form of claimed payback time is low. It is also easy to calculate the cost before deciding to make the investment, as the installation is similar from ship to ship. On the other hand, extensive CFD calculations and model testing may be necessary in order to get good estimations of the potential fuel savings.

The Mewis Duct, if thoroughly tested on models and in CFD calculations, is regarded as a quite safe and certain way of lowering the fuel consumption. Also, the consumption reduction from the preliminary analyses is actually based on testing with *Star Istind*, making 5 % an accurate estimation for this report.

Improved hull coating: Brunel EnviroMarine

It is not completely certain if 5,5 % is a realistic number as no clear test results from large scale applications have been presented. Additionally, as the product is relatively new, one does not know if its effect holds up for as long as claimed by the manufacturer. It is also difficult to say what a 10-year docking interval certification means in terms of what the condition of the coating will be at the end of that interval. Assuming that the manufacturer is not overly optimistic on behalf of their product's lifetime, the extra maintenance needed is minimal. This is because conventional coatings are re-applied in drydock anyway. Provided that Brunel's coating outlasts these coatings by up to 5 years, it may even be possible to prolong the drydock cycle and thus lowering downtime and drydock costs for the ship's lifetime.

Waste heat recovery system

Installing a new heat recovery system is a major task, consuming a lot of time and resources. The investment of the system itself is also likely to be significant, though no exact numbers are available concerning a price tag. It is also a question whether a small engine, such as the engine in our example ship, is suitable for this kind of installation, as the investment becomes relatively cheaper with increasing engine sizes. This means that a shipowner would have to be certain of the benefits obtained with such a system on their particular ship before deciding to install it.

Equipment such as the waste heat recovery system necessarily means more maintenance, as the engine system becomes more complicated with the installation of complex units. The risk of soot deposits on the boiler tubes, especially present in the dual-pressure system, means that inspection and cleaning must be conducted more often than before. In general, to maintain an effective heat recovery system, keeping tubes and heating surfaces clean is of major importance. Another aspect of the TES is the introduction of an additional turbine and generator system. This system includes many moving parts and is therefore subject to wearing and thus frequent inspections and maintenance tasks. There is also the risk of the system breaking down, with the complications brought by such event.

The numbers estimated for power production from the TES may be inaccurate due to them being based on much larger engines' outputs and rough calculations from the engine loads.

Fuel homogenizer

Like the TES, a fuel homogenizer system introduces more units on which maintenance is inevitable. The effects of the homogenizer lie in its ability to reduce the oil droplets' size and depend on the proper functioning of the rotor-stator arrangement. As with all moving/rotating equipment, wearing will occur despite there being no contact between the rotor and the stator. The condition of these two parts will probably determine the effectiveness of the homogenizer, making them subject to inspection and cleaning on a regular basis. On the other hand, the treatment of the fuel and consequent improved combustion may reduce the maintenance needed on the cylinders, pistons and other related parts in the engine. The tiny gap between the rotor and stator also makes them vulnerable if solid particles of a certain size are to slip past the filters. This is of course not supposed to happen if the fuel is filtered correctly. A failed filter is serious for any fuel system, but the consequences of a failed filter are even larger for a system fitted with a homogenizer than without one.

The origins of the savings percentages provided by the manufacturers are unknown, and these numbers are for that reason to be considered uncertain. However, as with the heat recovery system, the effects of a homogenizer should be possible to examine before deciding to install the system on a ship. The installation itself should not be a big concern as the homogenizer system is compact in size and does not intervene significantly with other systems.

Another aspect is how the fuel treatment affects the conditions after combustion, i.e. exhaust gas temperature/heat recovery. As there are no indications of any changes in exhaust temperature, this has been ignored.

In general

In this chapter, different means of reducing a ship's fuel oil consumption were assessed. The measures were chosen based on how much they would affect each other when combining them. Four measures which affect different parts of the ship and machinery system were chosen on the assumption that this would minimize their effect on each other, making the results more realistic and useful.

The consumption calculations were based on the assumption that the hydrodynamic measures examined, Mewis duct and Brunel coating, would reduce the need for engine power. In the calculations, the speed would remain the same and the power would be reduced according to the savings estimations. After that, the TES output was calculated based on the new power production of the ME, giving new consumptions for HFO and MDO. Finally the effect of the homogenizer was applied, resulting in final results for fuel and diesel oil consumption after the implementation of all four measures.

One thing that was ignored is the fact that a reduced load on the engine could result in lower fuel efficiency, hence affecting all the calculations. It was decided that the results of examining the effects of such a load reduction would be insignificant relative to the work it would require. The reduction in load could also result in more wear on the cylinders and engine in general, thus higher maintenance costs. An assumption that the engine could be optimized for the new load was therefore made.

A problem with the general approach in this chapter is that it is strictly on a theoretical level. The effects of combining several measures may very well have a different outcome compared to the calculations; it is difficult to say for sure without thorough testing. As mentioned earlier in this section, this was attempted to be avoided by selecting different measures.

These assessments were also only conducted on one ship. The various manufacturers' own estimations of their products' energy savings are very general and often vague in terms of the type of ship the estimations are based upon. This makes for potentially large deviations of the energy saving effects between ships. This variation is difficult to assess without extensive testing.

CHAPTER 3:

ENGINE DATA AS EFFICIENCY INDICATORS

This chapter is a study of various engine data gathered from a Höegh vessel. These data will be analyzed in an attempt to identify the most accurate or best suited parameters with regards to determining the engine efficiency. The data was collected from the TeCoMan database (18).

3.1 ENGINE CONDITION AND PERFORMANCE

Marine machineries are complex systems and their condition relies on a number of parameters. In order to maintain efficient operation of an engine it is essential to monitor its combustion conditions, cylinder condition and overall engine condition. The following are the key parameters in such performance observation, according to MAN Diesel & Turbo (14):

- Barometric pressure
- Engine speed
- Ship draft
- Mean indicated pressure
- Compression pressure
- Maximum combustion pressure
- Fuel pump index
- Exhaust gas pressures
- Exhaust gas temperatures
- Scavenge air pressure
- Scavenge air temperature
- Turbocharger speed
- Back pressure of exhaust gas in exhaust pipe after turbocharger
- Air temperature before turbocharger filters
- Δp over air filter
- Δp over air cooler
- Air and cooling water temperatures before and after scavenge air cooler

All these parameters will in one way or another affect the losses in the diesel process, and identifying which ones that have the largest impact on the engine's efficiency will make engine condition monitoring easier.

3.2 EXAMPLE SHIP – *HÖEGH ASIA*

The data used in this section is gathered from a Leif Höegh vessel, *Höegh Asia*. This ship is a Ro-Ro cargo vessel running on a MAN B&W 7S60MC engine with an MCR of 14 314 kW (19).

***Höegh Asia* main characteristics**	
Draft	10,0 m
Length overall	228,7 m
Breadth	32,3 m
Max speed	20,1 knots
Container capacity	7 800 CEU
Deadweight	27 000 mt

TABLE 17: HÖEGH ASIA MAIN CHARACTERISTICS (28)

3.2.1 MEASUREMENT DATA AND EFFICIENCY CALCULATIONS

The collection of data is from the period between July 2000 and July 2009, and the measurements have been taken with irregular intervals ranging from one month to several months apart. There are 96 dates averaging 10,7 readings per year. The measurement data collection consists of:

- Cylinder compression pressure
- Cylinder maximum pressure
- Cylinder mean indicated pressure (MIP)
- Fuel heat value
- Fuel oil consumption
- Specific fuel oil consumption
- Indicated power
- Indicated effective power
- Ship speed
- Various other measurements for scavenging air, turbochargers, exhaust gas temperatures etc.

To find out which of these affect the performance the most, engine efficiency for all the different dates must be calculated based on numbers from the data collection. These efficiencies can then be compared to the values of the different parameters, giving an indication of that parameter's effect on performance.

The total efficiency of an engine is calculated as follows

$$Total\ efficiency = mechanical\ efficiency\ \cdot thermal\ efficiency, \qquad [5]$$

where

$$mechanical\ efficiency = \frac{measured\ shaft\ power}{indicated\ engine\ power} \qquad [5.1]$$

and

$$thermal\ efficiency = \frac{indicated\ engine\ power}{fuel\ input} \qquad [5.2]$$

The indicated engine power is calculated from the mean indicated pressures of the cylinders:

$$Indicated\ power\ from\ cylinder\ x, P_{i,x}[kW] = \frac{\pi \cdot {D_{cyl}}^2}{4} \cdot S_{cyl} \cdot \frac{RPM_x}{60} \cdot MIP_x \qquad [6]$$

where D_{cyl} is the cylinder diameter and S_{cyl} is the cylinder length. This gives the total indicated engine power when adding up the indicated power from cylinders 1 through 7. The data collection contains measurements for the shaft power, while the fuel input is calculated from heating value and consumption per second, both varying over time:

$$Fuel\ input\ power\ [kW] = fuel\ heating\ value\ \left[\frac{kJ}{kg}\right] \cdot fuel\ consumption[\frac{kg}{s}] \qquad [7]$$

As an example, the total engine efficiency can be calculated for the newest measurement date. The following data are found in the collection:

- Date: July 8th, 2009
- Cylinder 1-7 RPM = 104 RPM (average) = 1,73 s^{-1}
- Cylinder 1-7 MIP = 15,75 bar (average) = 1575 kPa
- Fuel heating value = 40,45 kJ/kg
- Fuel consumption = 2 383,64 kg/h = 0,662 kg/s
- Measured shaft power = 11 507 kW
- Cylinder diameter = 0,600 m
- Cylinder length = 2,292 m

From this the indicated engine power is found:

$$P_i = \frac{\pi \cdot 0{,}6[m]^2}{4} \cdot 2{,}292[m] \cdot 1{,}73[s^{-1}] \cdot 1\ 575[kPa] \cdot 7[cylinders] = 12\ 360\ kW$$

The fuel input power:

$$Fuel\ input\ power = 40\ 450\left[\frac{kJ}{kg}\right] \cdot 0{,}662\left[\frac{kg}{s}\right] = 26\ 778\ kW$$

The thermal efficiency:

$$Thermal\ efficiency = \frac{12\ 360\ kW}{26\ 778\ kW} = 0{,}46$$

The mechanical efficiency:

$$Mechanical\ efficiency = \frac{11\ 507\ kW}{12\ 360\ kW} = 0{,}93$$

And the total efficiency:

$$Total\ efficiency = 0{,}46 \cdot 0{,}93 = 0{,}43$$

This method of calculating the engine efficiency can only be done when RPM readings exist. This is not the case with *Höegh Asia's* data set, where only a portion of the dates have measurements for RPM. Also, because there are shaft power measurements for the whole time period, there is no need to go through the steps of mechanical and thermal efficiency. Instead we can look at $\frac{shaft\ measured\ power}{fuel\ input\ power}$, which will give identical results to the method shown above.

Measurements for specific fuel oil consumption, SFOC [g/kWh], are also found. This can be a good indicator for the efficiency of the engine as well, with small SFOC numbers meaning less fuel consumed per kWh produced and vice versa for high numbers.

To easily find high and low efficiency points, the efficiencies are plotted against the dates in a graph. SFOC is also plotted in order to get more reliable readings. If high efficiency numbers are coupled with low SFOC numbers the chance that they are faulty measurements is smaller.

Due to insufficient and clearly faulty measurements, not all dates are represented with efficiency calculations. Following are the graphs for efficiency and SFOC plotted against the dates from which efficiencies have been calculated:

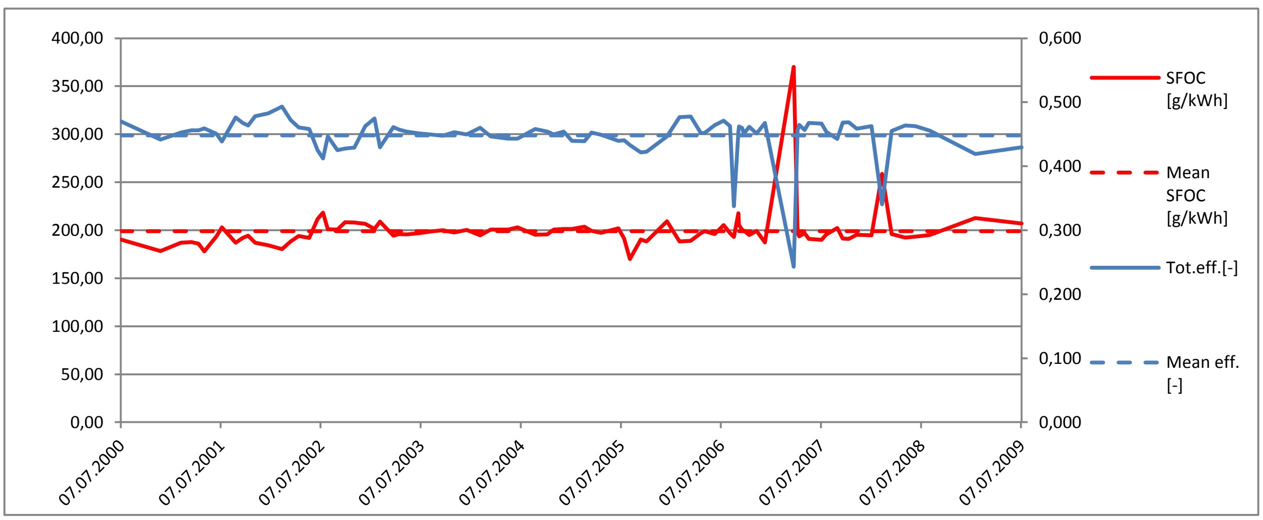

FIGURE 9: EFFICIENCY AND SFOC GRAPHS

The graphs, for the most part, clearly show that the SFOC and efficiency values correspond to each other.

A larger, easier to read version of figure 9 can be found in Appendix C.

To assess the different parameters' impact on performance they will be compared to their mean values for the dates chosen. Based on MAN's listed key parameters for performance observations, and limitations in the measurement data, these parameters will be examined:

- Compression pressure (p_{comp})
- Maximum combustion pressure (p_{max})
- Exhaust gas temperature (t_{exhv})
- Scavenging air receiver pressure (p_{scav})
- Mean indicated pressure, MIP (p_i)
- Fuel pump index
- Exhaust gas pressure (p_{exh})
- Turbocharger speed
- Δp over air filter (Δp_f)
- Δp over air cooler (Δp_c)
- Scavenging air inlet air temperature ($t_{scavinl}$)
- Scavenging air outlet air temperature ($t_{scavoutl}$)

Additionally, some of the parameters need to be corrected to ISO ambient conditions. According to MAN Diesel & Turbo (14), these are:

- Compression pressure, p_{comp}
- Maximum combustion pressure, p_{max}
- Exhaust gas temperature, t_{exhv}
- Scavenging air pressure, p_{scav}

The reason for this is that these parameters largely depend on the ambient conditions. If they are not corrected relative to these conditions, they will be incomparable. Unfortunately, the measurements needed to perform the corrections are limited to those of May 2006 onwards leaving only 30 useful measurements. So, for these four parameters, there is a limited base of comparison between efficiencies and measurements.

ISO values relevant to these parameters are (20):

- Cooling air and water inlet reference temperatures, t_{ref} = 25°C
- Ambient pressure, p_{ref} = 1000 mbar = 1 bar

MAN Diesel & Turbo's corrections are performed in the following way:

A_{corr} = (t_{meas} - t_{ref}) · F · (K + A_{meas})

where

A_{corr} = the correction to be applied to the parameter (p_{max}, t_{exhv}, p_{comp}, p_{scav})

t_{meas} = measured air inlet temperature (t_{inl}) or cooling water inlet temperature ($t_{coolinl}$)

t_{ref} = reference t_{inl} or $t_{coolinl}$ (= 25°C)

F_1, F_2 = constants (see table 18)

K = constant (see the table 18)

A_{meas} = the measured parameter to be corrected (p_{max}, t_{exhv}, p_{comp}, p_{scav})

FIGURE 10: MAN DIESEL & TURBO METHOD OF CORRECTING TO ISO AMBIENT CONDITIONS (14)

Table of constants for correcting to ISO ambient conditions			
Parameter to be corrected	F_1: for air inlet temp.	F_2: for cooling water inlet temp.	K
t_{exhv}	$-2{,}446 \cdot 10^{-3}$	$-0{,}590 \cdot 10^{-3}$	273
p_{scav}	$+2{,}856 \cdot 10^{-3}$	$-2{,}220 \cdot 10^{-3}$	1 bar
p_{comp}	$+2{,}954 \cdot 10^{-3}$	$-1{,}530 \cdot 10^{-3}$	1 bar
p_{max}	$+2{,}198 \cdot 10^{-3}$	$-0{,}810 \cdot 10^{-3}$	1 bar

TABLE 18: TABLE OF CONSTANTS FOR CORRECTING TO ISO AMBIENT CONDITIONS (14)

Referring to figure 10, the t_{meas} data are the ones that are needed from the database, and that can only be found after May 2006.

As an example, the most recent compression pressure measurement is here corrected for the ambient conditions:

- Date: July 9th, 2009
- Measured compression pressure, p_{comp} = 98,43 bar
- Air inlet temperature $t_{inl(air)}$ = 37 °C
- Cooling water inlet temperature $t_{inl(water)}$ = 48 °C
- ISO reference temperatures, $t_{inl(air,\ water)}$ = 25 °C

$$A_{corr(air)} = (37 - 25) \cdot 2{,}954 \cdot 10^{-3} \cdot (1 + 98{,}43) \approx 3{,}52\ bar$$

$$A_{corr(water)} = (48 - 25) \cdot (-1{,}530 \cdot 10^{-3}) \cdot (1 + 98{,}43) \approx -3{,}50\ bar$$

$$P_{comp,corr} = P_{comp} + A_{corr(air)} + A_{corr(water)} = 98{,}43\ bar + 3{,}52\ bar - 3{,}50\ bar = 98{,}45\ bar$$

So the corrected pressure in this case is slightly higher than indicated by the measurement.

3.2.2 PARAMETER ANALYSIS

Over the next pages the available key parameters will be compared to the ship efficiency data in the same time period. This way one may be able to identify which parameters give the best idea of overall engine performance and condition.

ISO corrected compression pressure, p_{comp}

The compression pressure is the pressure after the air in the cylinder has been compressed prior to combustion. This pressure indicates the amount of air in the cylinder. Higher pressure equals more air and a better combustion process.

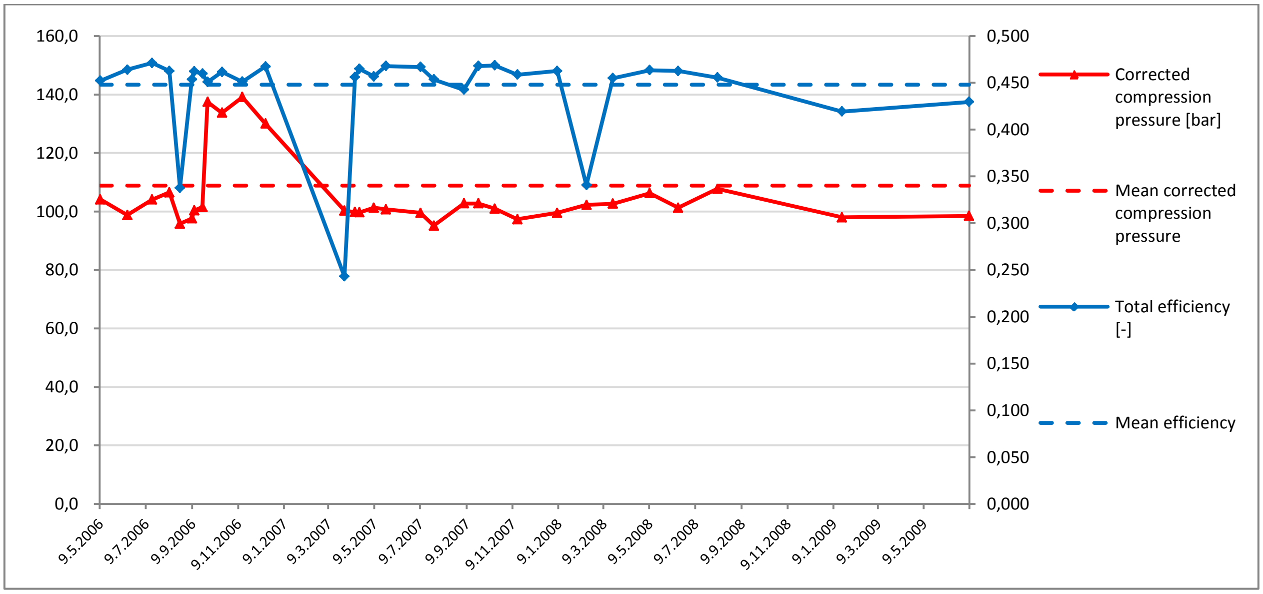

FIGURE 11: COMPRESSION PRESSURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

Figure 11 shows the corrected measurements for compression pressure. The mean compression pressure line would probably be a little lower if the measurement base was larger, making the impact that the high points have on the mean value smaller.

Due to the limited number of data available for comparison it is difficult to see a clear relation between engine efficiency and compression pressure for the whole period. However, ignoring the large deviations from their mean values, one may distinguish a relation between the high points in efficiency and high points in pressure, and the same for low points. Also, seeing as the large deviations in pressure from the average value come at different points than the drops in efficiency, one can assume that these are the results of faulty readings or other circumstances not significantly affecting the efficiency.

According to MAN Diesel & Turbo (14), reduced compression pressure is likely due to either of the following:

- Reduced scavenging air pressure
- Leaks in piston rings

- Burnt piston crown
- Worn cylinder liner
- Leaking exhaust valve
- Error in timing of exhaust valve

The dips in pressure we can see on the graph, albeit rather small, could therefore be the results of one or more of these causes.

ISO corrected maximum combustion pressure, p_{max}

The combustion pressure is the pressure occurring after the compressed air and fuel mix in the cylinder ignites. This pressure forces the piston back, creating the shaft power.

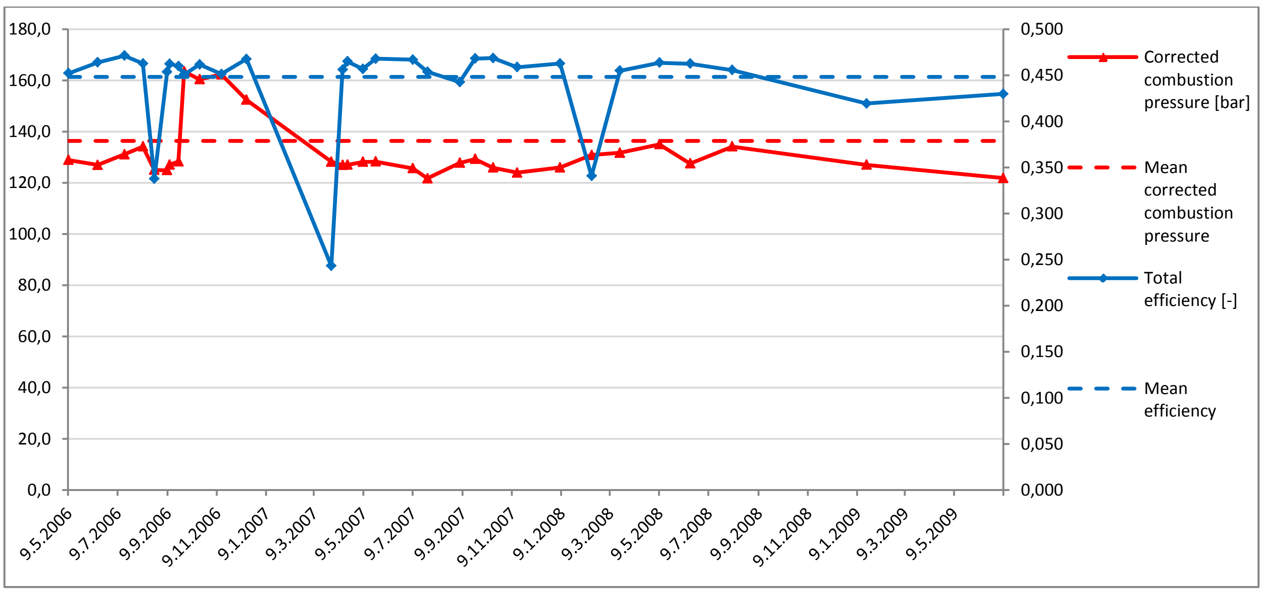

FIGURE 12: MAXIMUM COMBUSTION PRESSURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

This figure shows that the combustion pressure graph's form is nearly identical to that of the compression pressure. This is natural as the combustion pressure is directly dependent on what pressure exists in the cylinder before the fuel is ignited. Therefore, the same causes of deviation from the mean compression pressure can be applied to the combustion pressure.

Another different factor for the combustion pressure compared to compression pressure is the injected fuel's effect. The combustion pressure is affected by the fuel, whereas the compression pressure is measured before the fuel is injected. The fuel pump index could therefore be an indication of how the combustion process will look like, and it may be interesting to compare it to the combustion pressure:

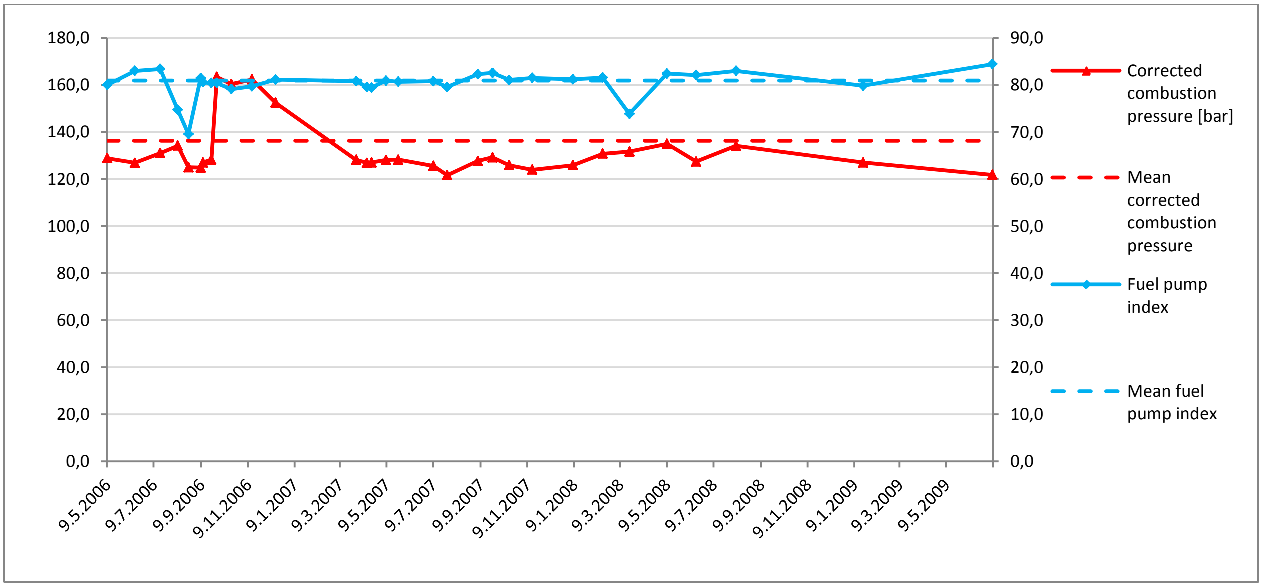

FIGURE 13: MAXIMUM COMBUSTION PRESSURE (LEFT AXIS) COMPARED TO FUEL PUMP INDEX

Again, it is difficult to clearly see the relation between the two graphs. Deviations of the fuel pump index values is an indication of the condition of the fuel injection equipment, where a worn fuel pump or leaking valves translates to a higher fuel pump index. Accordingly, a higher fuel pump index is assumed to lead to a lower combustion pressure. Keeping in mind the limited measurement data for these ambient condition corrected parameters, it is hard to reach a conclusion based on these few readings. Later we will be able to compare the fuel pump index to engine efficiency for the whole time period.

ISO corrected exhaust gas temperature, t_{exhv}

The exhaust gas temperature is the temperature measured after the exhaust valve. This temperature should indicate if a complete combustion has taken place. Naturally, a higher exhaust gas temperature means that more energy is lost in the exhaust gas and that less energy is used to drive the propeller shaft.

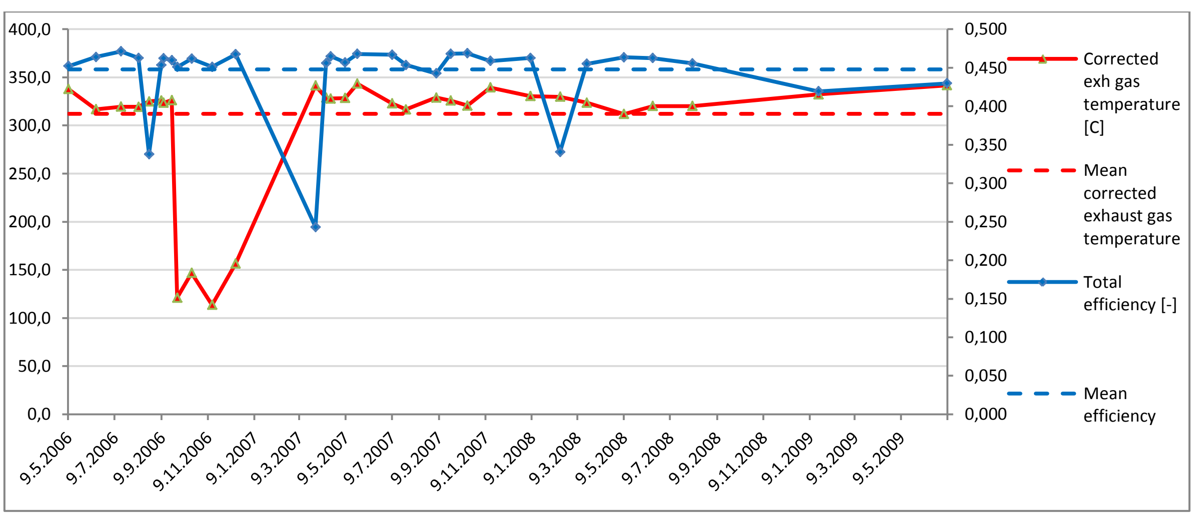

FIGURE 14: EXHAUST GAS TEMPERATURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

Aside from the most dramatic deviations in exhaust gas temperatures, we can see that at some of the high efficiency points we have a relatively low exhaust gas temperature, and the opposite for low efficiency points.

It is also interesting to see that the big drop in exhaust gas temperature occurs at the same points as the high points in compression and combustion pressure. At the same time, these deviations do not seem to affect engine efficiency; they occur at points where the efficiency is within reasonable values.

There is no clear explanation for these drops in temperature and the corresponding jumps in pressure for the dates concerned, which are 29.9.2006, 18.10.2006, 14.11.2006 and 15.12.2006. The indicated power, SFOC values, efficiencies, measured mean indicated pressures and other relevant parameters are all within reasonable limits of their expected values. The only abnormality is a very likely faulty measurement of cylinder 3's RPM on 29.9.2006, reading 404 RPM where the mean is about 104 RPM. This, however, does not explain any of the other date's deviations. The conclusion is that they are the result of faulty readings due to measurement equipment failure. It is difficult to say why both the temperature readings and the pressure readings are off, if they are not dependent on each other in some way.

Man Diesel & Turbo's (14) list of possible causes for increased exhaust temperature comprises:

- Fuel injection equipment failures
    - Leaking/errors on fuel valves
    - Worn fuel pumps
- Cylinder condition
    - Piston rings failure (blow-by)
    - Exhaust valve leaks
- Fouling of the air or water sides in the air cooler
- Fouling of turbine or compressor in turbocharger
- Extreme climatic conditions
- Fuel oil type/quality

ISO corrected scavenging air receiver pressure, p_{scav}

The scavenging air is the air that is injected into the cylinder in each cycle. It pushes the exhaust gas out, clearing the cylinder of any remaining gases after combustion. In order to clear the cylinder, the scavenging air must have a higher pressure than the pressure in the exhaust outlet. The scavenging air pressure is the result of the turbocharger compressing the air in the scavenging air receiver, which is the stage before the air is let into the cylinder. (20) (21).

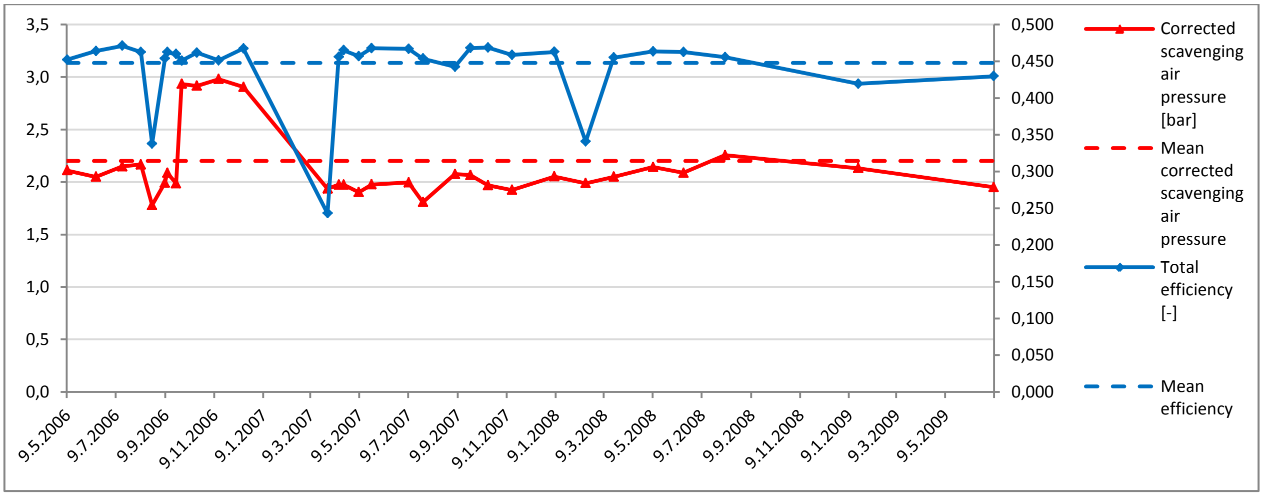

FIGURE 15: SCAVENGING AIR PRESSURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

A lower scavenging pressure would mean that the cylinder is less likely to be cleared of gases before another combustion cycle, thus resulting in a lower efficiency. This graph does not clearly indicate that a higher scavenging air pressure equals higher engine efficiency. This could be due to the cylinder being cleared just as good with a scavenging air pressure of e.g. 2,0 bars compared to a higher pressure.

Again, we are seeing some large deviations in the pressure. As in the previous parameters, these are assumed to be the results of measurement equipment irregularities.

Changes in scavenging air pressure, according to MAN Diesel & Turbo (14), can be seen in connection with:

- Air cooler condition
- Turbocharger condition
- Camshaft timing

Seeing as the scavenging air pressure is closely related to the turbocharger condition, the TC speed is here compared to the scavenging pressure:

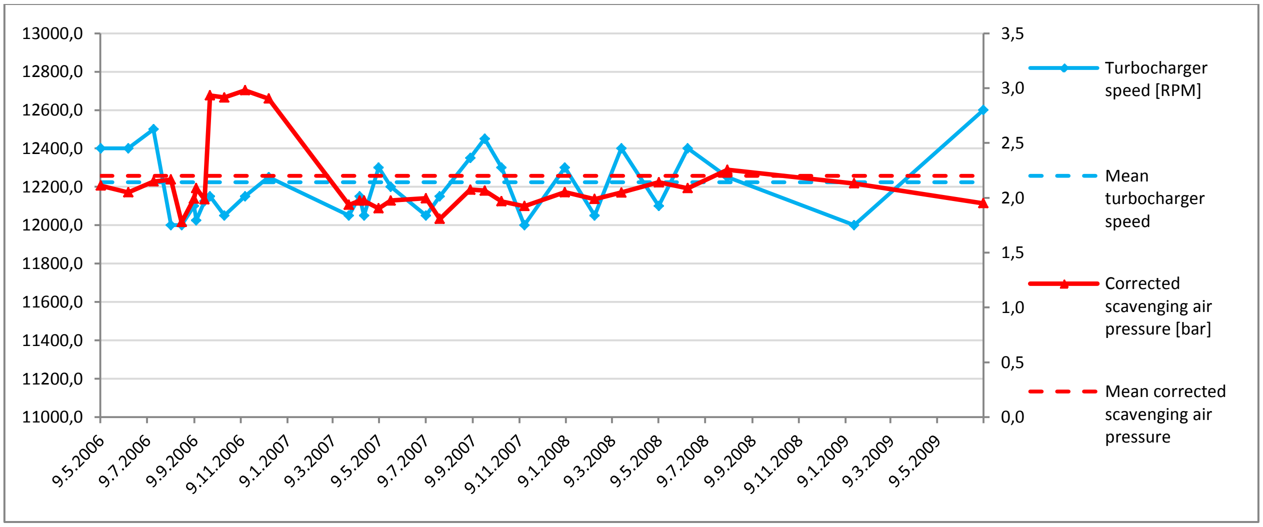

FIGURE 16: TURBOCHARGER SPEED (LEFT AXIS) COMPARED TO SCAVENGING AIR PRESSURE

The scavenging air pressure should rise with higher TC speeds. Again, the limited measurement base makes it hard to identify a definitive relation between the graphs for TC speed and scavenging pressure. However, the increases and decreases of the two seem to follow each other more often than not.

The parameters examined over the next few pages will have a lot more readings as they do not need to be corrected to ISO ambient conditions. This may make it easier to see patterns in the relations between the different parameters and engine efficiency.

Mean indicated pressure, MIP

The mean indicated pressure (indicated mean effective pressure) of a cylinder is the average pressure over a cylinder during one cycle of the combustion process. This is closely related to the work performed by the cylinder, as seen in equation 6 for indicated power, P_i.

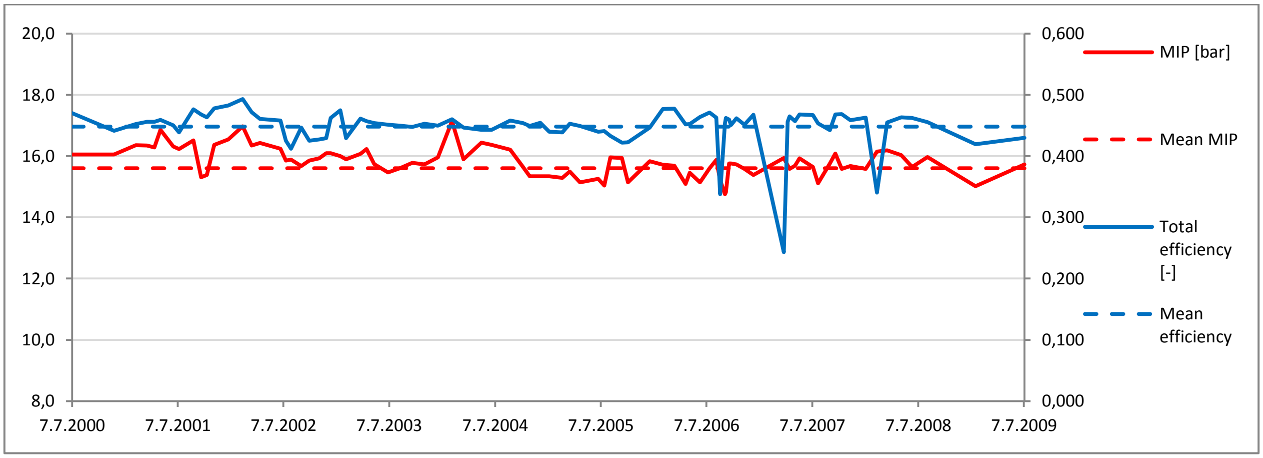

FIGURE 17: MEAN INDICATED PRESSURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

For some of the readings, the MIP follows the trend of the efficiency. For many others, however, the trends of MIP values and the efficiency seem to be unrelated to each other.

The mean indicated pressure in the cylinder will depend on the compression pressure and the maximum combustion pressure, as shown in the figure below where the three graphs follow approximately the same trend:

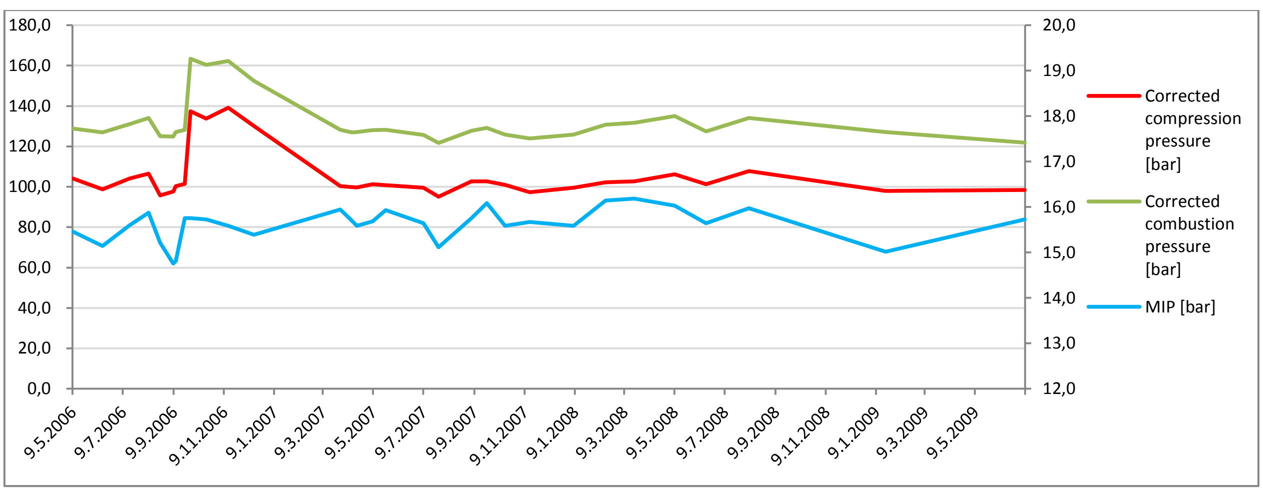

FIGURE 18: COMPRESSION PRESSURE AND COMBUSTION PRESSURE (LEFT AXIS) COMPARED TO MIP

The MIP should be a good indication of efficiency, along with compression and combustion pressure, of the cylinder and overall combustion process condition. However, with the data collected here, it is difficult to distinguish a clear relation between the MIP and the engine efficiency.

Fuel pump index

Previously we compared fuel pump index to maximum compression pressure. The connection between the two parameters was not very clear visually. Remembering that a high fuel pump index means a deteriorating fuel pump system, we will now see it compared to the overall engine efficiency for the whole time period:

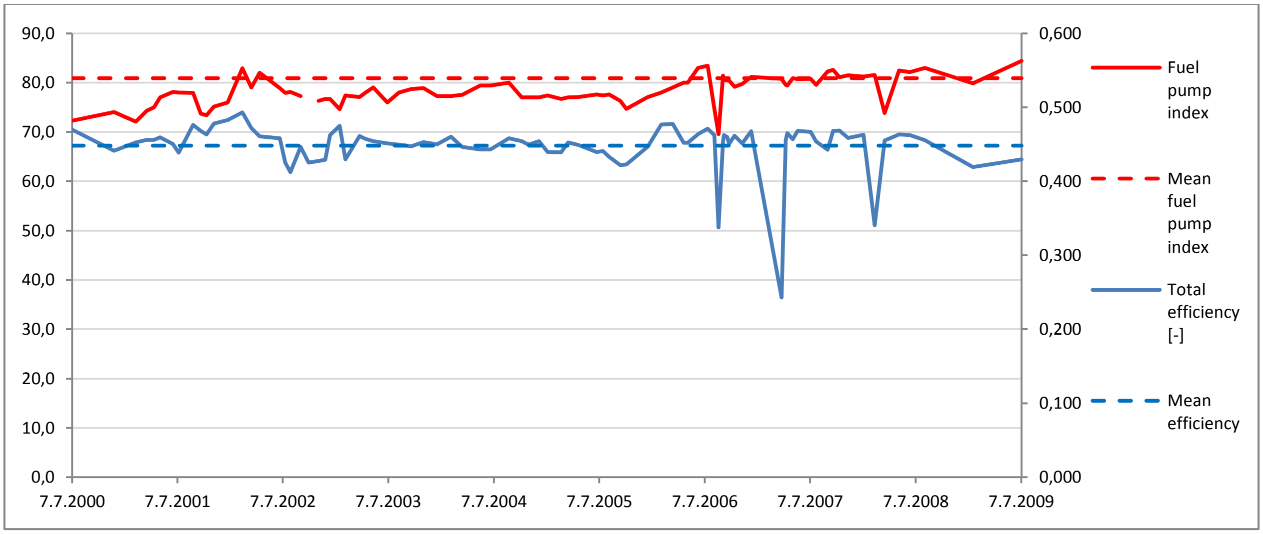

FIGURE 19: FUEL PUMP INDEX (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

It is difficult to conclude with the assumption that a high fuel pump index equals lower efficiency, as the high and low points of these graphs are inconsistent with each other. In this case it might be a good idea to double check the fuel pump index graph against the SFOC graph from the beginning of this chapter. This is presented in Appendix D. The comparison reveals no obvious connection to be seen between SFOC and fuel pump index.

Exhaust gas pressure

The exhaust gas pressure is the pressure held by the exhaust gas after combustion has taken place. As previously stated, this pressure must be lower than the scavenging air pressure in order for the cylinder to be cleared of remaining exhaust gas. This is also the case (for the available scavenging pressure readings), as shown in the figure below.

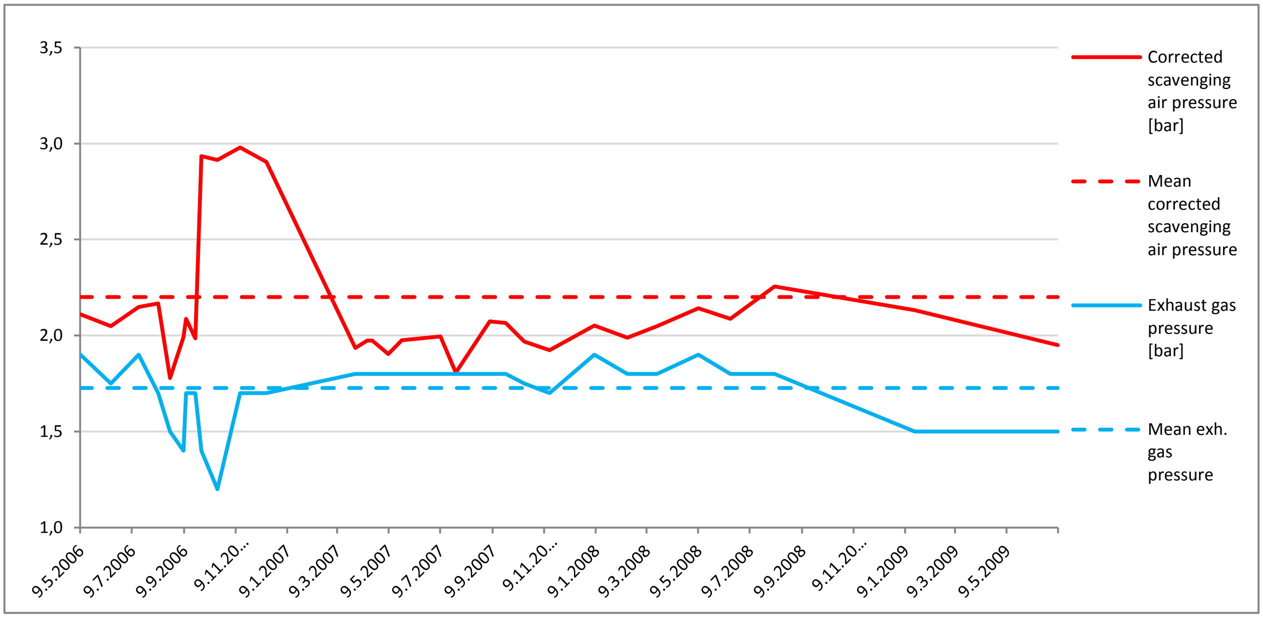

FIGURE 20: EXHAUST GAS PRESSURE COMPARED TO SCAVENGING AIR PRESSURE

In Appendix E, the difference between the scavenging air pressure and exhaust pressure has been compared to the engine efficiency, and no relation can be seen between them. The figure below compares exhaust pressure to engine efficiency.

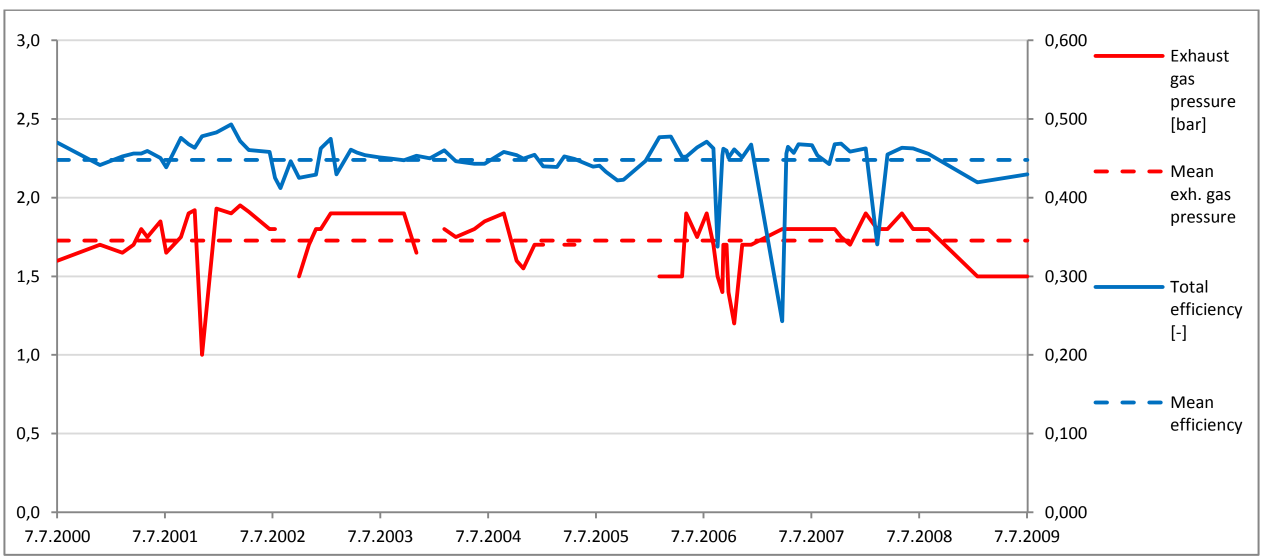

FIGURE 21: EXHAUST GAS PRESSURE (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

The missing exhaust gas readings do not make it any easier to draw any conclusions based on these graphs. However, the inconsistencies in high and low points of the exhaust pressure compared to those of the efficiencies imply that there is no direct or obvious relation, according to the collected data.

Turbocharger speed

The turbocharger speed is something that is not directly affecting the conditions in the cylinder. However, it, builds the scavenging pressure, thereby contributing to the scavenging and eventually compression pressure in the cylinder.

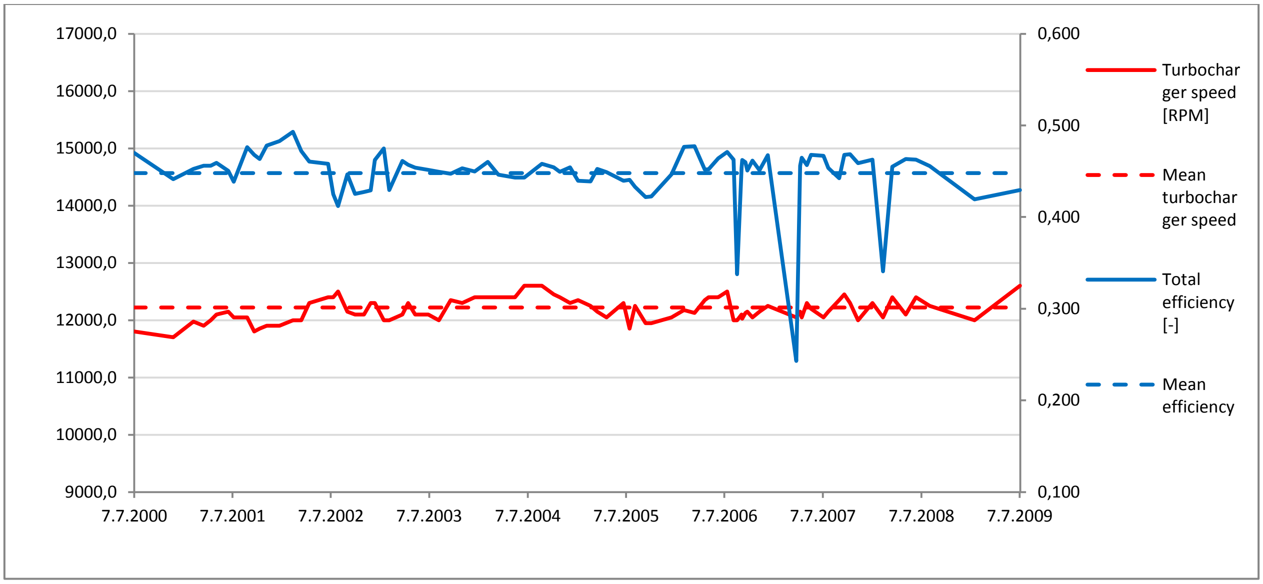

FIGURE 22: TURBOCHARGER SPEED (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

A high TC speed could mean a higher efficiency, as more air would have been pushed into the cylinder. In this visualization, a distinct relation between the two parameters is not found.

Δp turbocharger air filter

Deviations in pressure drop across the air filter can be an indication of the filter's cleanliness. This could influence the scavenging pressure and the engine efficiency. (14)

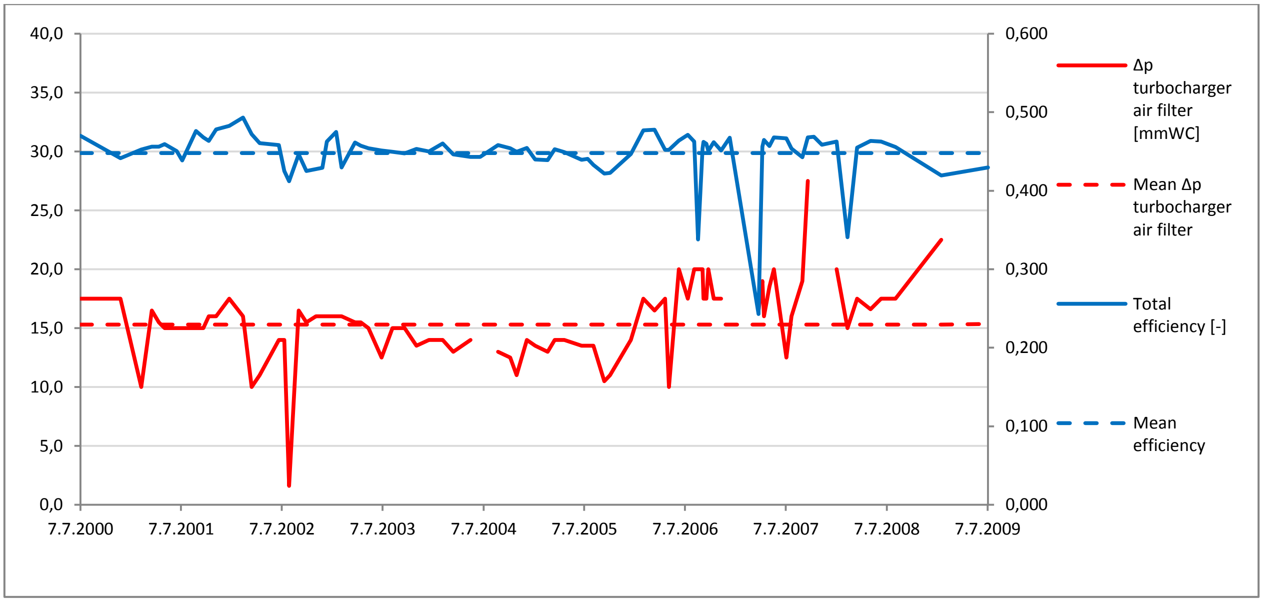

FIGURE 23: ΔP ACROSS TURBOCHARGER FILTER (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

A high Δp would imply that air is not easily passing through the filter, thereby affecting the scavenging air pressure. High points in the pressure drop should therefore come with low points in efficiency. The figure does not clearly show this, which is likely due to the pressure drop of the air filter not directly influencing the combustion process. This parameter is therefore mostly suited to assess the air filter's condition and not so much the efficiency or overall condition of the engine.

Air cooler: Δp_{air}

Similar to the pressure drop across the turbocharger air filter, the pressure drop across the scavenging air cooler is an indication of fouling on the air side of the cooler. This is one of the indicators of air cooler condition.

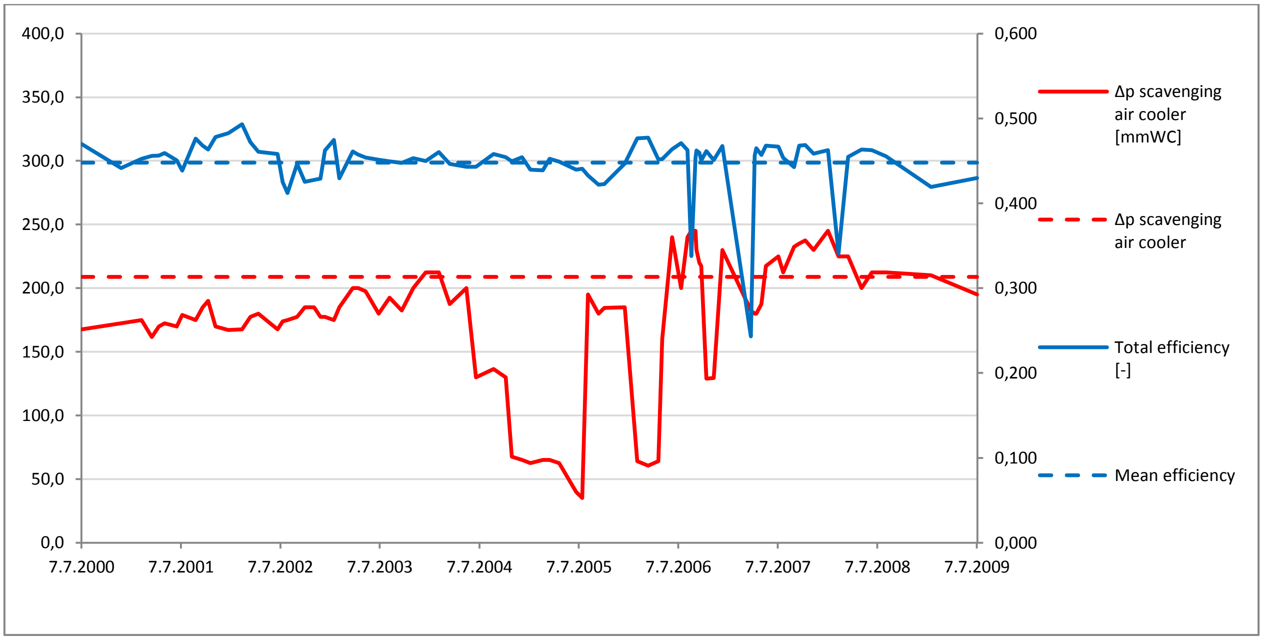

FIGURE 24: ΔP ACROSS SCAVENGING AIR COOLER (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

As with pressure drop across the turbocharger air filter, high points in Δp should correspond to low points in efficiency. The graphs in the figure above seem to be somewhat stable for a while, without having a clear relation between them, until they both deviate and become unstable and incomparable about halfway through. Based on this visualization, the air cooler pressure drop is not a good indication of engine efficiency.

Air cooler: Temperature difference between air outlet and water inlet, $\Delta t_{(air\text{-}water)}$

According to MAN Diesel & Turbo, this is the most essential parameter concerning the air cooler's cooling ability. A high $t_{air,outlet}$ combined with a low $t_{water,inlet}$ means that the air was not properly cooled, as opposed to a low $t_{air,outlet}$ and a relatively higher $t_{water,inlet}$ where more heat must have been transferred from the air to the cooling water. Cooler air means more air can be compressed in the cylinder.

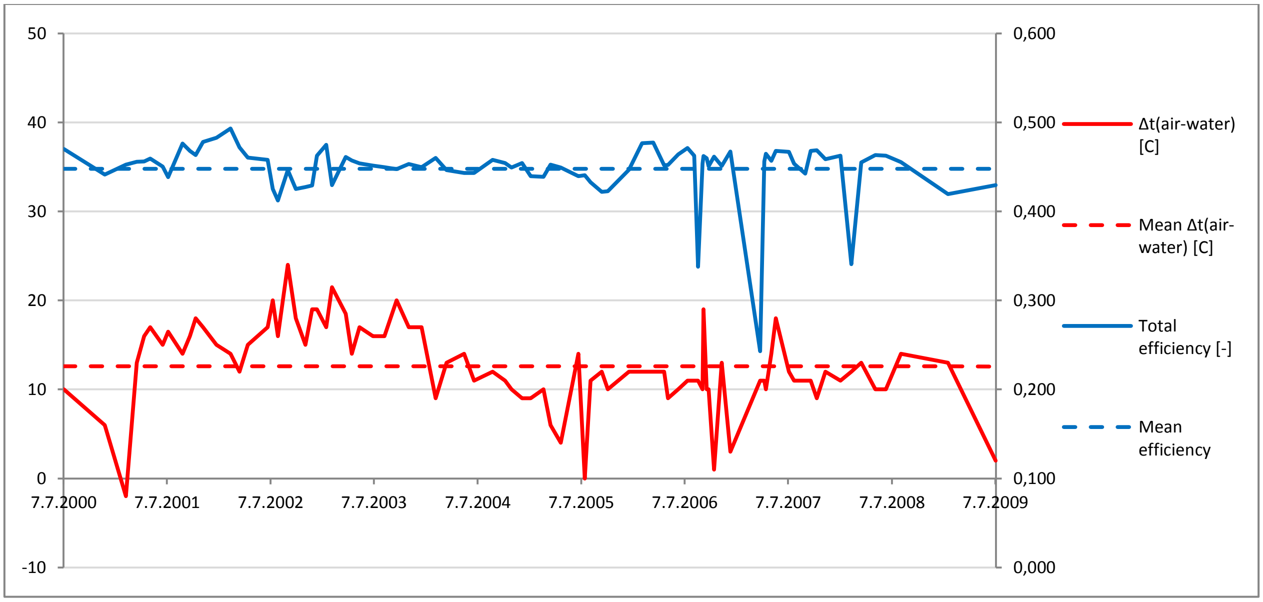

FIGURE 25: TEMPERATURE DIFFERENCE BETWEEN COOLER AIR OUTLET AND WATER INLET (LEFT AXIS) COMPARED TO ENGINE EFFICIENCY

As with the pressure difference across the air cooler, ideally we should have opposite high and low points for the graphs. This figure shows a slight relation between the $\Delta t_{(air\text{-}water)}$ and efficiency for a period, before they seem to get inconsistent and messy towards the end of the time period.

Other observations

In the various graphs in this section there are a lot of spikes in both positive and negative direction. There is no explanation for most of these spikes, as they occur at different points and independently of each other. The most obvious reasons for them are faulty measurements, misreadings or errors in registering. Plotting all of the graphs in one chart (Appendix F) further underlines the fact that the extremes are mostly independent of each other.

3.3 Discussion

To look at the key parameters individually apparently does not serve any serious purpose if the objective is to determine their effect on the engine efficiency. While some of the graphs in the figures in this section seem to follow the efficiency for small periods of time, especially the parameters directly related to cylinder work (e.g. the different pressures), for the most part any distinct relation is not seen. This can be attributed to the fact that, individually, the parameters will have only small impacts on the efficiency. In order to make use of the measurements, one must look at them in relation to each other.

Appendix G shows a small extract of the dataset from *Höegh Asia.* This makes it clear that, by only looking at parameters' deviation from their mean values, one cannot really determine anything concerning engine efficiency. Every date has at least one parameter deviating negatively (marked in red), most dates have several. There is no clearer pattern in the table than in the graphs in this section. This underlines that changes in one parameter do not necessarily affect the engine efficiency significantly. If these measurements were to be aggregated, they would likely prove to indicate the condition of the engine in a better way than any of them can do individually.

By themselves, most of the parameters which are not directly related to the cylinder condition, such as TC speed and pressure drop over air filter and cooler, if measured correctly, are mostly suitable to determine the condition of that particular part (i.e. turbocharger, air filter and air cooler).

A problem with this section is the lack of information about how the parameters have been measured, how the measurements are collected and registered, and in what way the human factor affects the readings. Errors in measurements, like many of the spikes we have seen in the different graphs, make it especially difficult to assess their impact on efficiency. Measurement reliability plays a big role in this work.

While the instruction manual for this engine series contains reference values for some of the parameters, *Höegh Asia's* engine does not seem to follow those figures. Therefore, the mean values of the parameters were chosen as guidelines for acceptable values. An obvious disadvantage with this method is that extreme values, likely due to faulty readings, affect the mean value deciding if a parameter is within acceptable limits or not. Most of these extreme values have been ignored when calculating the mean, so this should not be a very big problem.

Information about maintenance would also be useful, where one could see the condition of the parameters before and after maintenance tasks. Some of the parts have long periods operating at conditions where the readings are below their ideal or mean values, followed by a period where the values are fine. That could mean that they have undergone maintenance and that operation has been improved because of this.

CHAPTER 4:
A STUDY OF RELEVANT TECHNOLOGIES

This chapter is a study of various technologies that can be used in order to have a more energy efficient operation of a ship. The different systems will not be related to any specific ship, but rather be discussed as suggestions for higher efficiency.

4.1 SHAFT GENERATOR ARRANGEMENTS

Shaft generators, or power take off (PTO) systems, are a way to use the ME as an electrical power generator. The basic principle is that, through reduction gears, one can use some of the shaft power to generate electrical energy to be used for various auxiliary functions onboard the ship. In this part different setups and uses of shaft generators will be discussed.

4.1.1 TECHNOLOGIES

There are two main types of shaft generator arrangements:

- Constant frequency shaft generators
- Generators which frequency varies with engine speed

Gear constant ratio (GCR) power take off

This is the basic shaft generator. There is no speed control of the generator, or frequency control of the power produced, meaning that the GCR system's produced electrical power frequency is depending on what RPM the ME is running at.

The GCR system is normally used together with a controllable pitch propeller (CPP), as it requires constant speed on the ME to produce stable frequency current. The speed of the vessel can then be adjusted by changing the propeller pitch instead of the engine speed. Even with a variable pitch propeller, though, the power output will not be completely stable. It can, however, be used to produce power with floating frequency, e.g. 50-60 Hz to be utilized by various consumers.

As the current is not stable without a frequency controller, the GCR shaft generator cannot be used in parallel with auxiliary engines. The shaft generator is therefore often the sole source of power production on a ship during voyage. The efficiency of a generator of this type is around 92 % (22).

Constant frequency power take off

A shaft generator of the constant frequency type will have a gearbox arrangement and/or frequency converting system which is able to control the power output frequency over a wide range of engine speeds. The most used generator of this type is the shaft mounted constant frequency electrical (CFE) generator. The electrical power is often produced at frequencies varying with engine speed before a frequency converter changes it to constant frequency. This way the CFE is able to operate in parallel with AEs at a speed range between 75 % and 100 % of the ME's SMCR. At lower speeds the electrical output is reduced correspondingly. This generator will operate with an efficiency of approximately 88 % (22).

Another type of constant frequency shaft generators is one produced by Renk in Germany, the Renk constant frequency (RCF) generator. RCF comprises a planetary gearbox where a hydrostatic motor varies the speed of the ring gear, hence controlling the speed of the output shaft to produce power at constant frequency. The hydrostatic motor is driven by a pump on the input shaft, and is controlled by an electronic control unit to match the frequency of the power produced to that of the AEs. The generator can operate in parallel with AEs in the range between 70 % and 100 % of the ME's SMCR. Its efficiency lies between 88 % and 91 %, depending on engine speed (22).

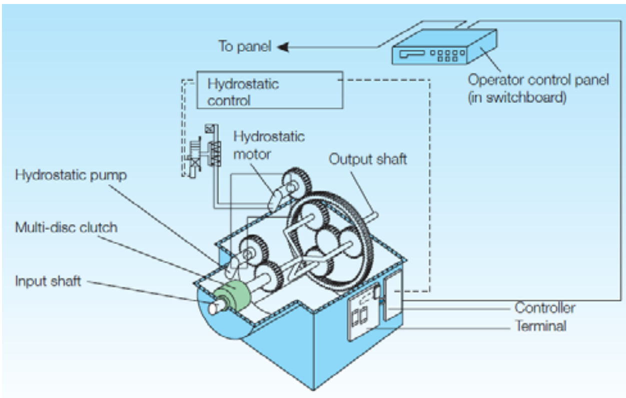

FIGURE 26: RCF DESIGN PRINCIPLE (22)

As previously stated, a shaft generator with variable frequency power output is best suited with a CPP arrangement. These arrangements, however, have reduced propeller efficiency because of increased propeller boss size. A constant frequency generator may therefore be preferred as it can operate in combination with a fixed pitch propeller (FPP) at high propeller efficiency.

4.1.2 Applications of shaft generators

GCR systems are often used on small to medium sized container vessels with CPP arrangement. Their output is normally in the 500 kW to 1 800 kW range. The CFE is more often used on larger container vessels and other ships with the biggest engines, together with an FPP (22).

A clutch arrangement on the propeller shaft aft of the shaft generator will enable the propeller to disconnect from the ME, allowing the ME to be used entirely for electrical production in combination with a shaft generator. This can be a useful feature when the ship requires extensive electrical power while in dock or during other operations when the ship is not moving (e.g. shuttle tankers which have high electricity consumption).

Some types of shaft generators can also be used as auxiliary propulsion power; emergency propulsion or alternative propulsion. In these cases the shaft generator is used as an electrical motor powered by electricity produced by the AEs.

4.1.3 Advantages and disadvantages of shaft generators

Man Diesel & Turbo lists a number of advantages and disadvantages of shaft generators (22):

Advantages

- Small space requirement
- Relatively low investment cost; GCR is inexpensive while RCF and CFE are more costly
- Low installation costs; installation is relatively quick and simple
- Shaft generators, as main engines, are considered highly reliable; long lifetime
- Low maintenance man-hour and spare part cost

Disadvantages

- No power production in harbor if no clutch is installed
- Higher load on ME means higher SFOC and lubricating oil consumption
- Reduced propeller and fuel efficiency with GCR; ME speed must remain constant even at reduced ship speed
- GCR unable to run in parallel with AEs
- More complicated shaft arrangement

Economic comparison: Alternatives with and without a shaft generator

MAN Diesel & Turbo have conducted a study comparing two identical container ships with different arrangements.

Ship 1's arrangement comprises a main engine with SMCR of 11 060 kW with a GCR shaft generator and 2 auxiliary engines.

Ship 2 has the same ME as no. 1, with an SMCR of 9 760 kW and with 3 auxiliary engines, no shaft generator.

Conditions are:

- Load profile for ship 2:
  - 90 % load for 15 % of time at sea
  - 80 % load for 40 % of time at sea
  - 70 % load for 35 % of time at sea
  - 60 % load for 10 % of time at sea
- Load profile for ship 1 is increased to compensate for reduced propeller efficiency with GCR and CPP
- Time at sea: 250 days/year
- Time in port: 115 days/year
- Electric load at sea: 900 kW
- Electric load in harbor: 500 kW

These conditions result in an additional fuel and lubrication oil cost of 8 630 USD per year for ship 1. So the shaft generator arrangement actually consumes more fuel than ship 2 with only AE setup.

On the other hand, reduced maintenance and spare part costs amounts to 20 000 USD per year for ship 1 with the shaft generator, so in total the operating cost will be lower for the shaft generator alternative.

MAN Diesel & Turbo's detailed calculations for these scenarios can be found in Appendix H.

4.2 CONTROLLABLE PITCH PROPELLER SYSTEMS

Controllable pitch propellers were only just mentioned in the previous part. Their main function is that one can increase and decrease the hydrodynamic lift on the propeller blades by varying the propeller blades' pitch.

4.2.1 TECHNOLOGY

An oil distributor (OD) box controls the propeller pitch hydraulically. A type of piston pushes a crankpin located inside the propeller boss. This crankpin is connected via a control rod to the blades. The oil pressure decides which direction the piston is pushed, ahead or astern, thus deciding the pitch of the propeller blades. The hydraulic system is located inside the propeller shaft. A sketch of the working principle is shown in figure 27. The propeller boss must be large enough to house the pitch control mechanisms. The OD can be shaft mounted in the case of a direct driven propeller (i.e. on two-stroke engines without reduction gear) or mounted in front of/inside the gearbox in the case of geared engines (21) (23).

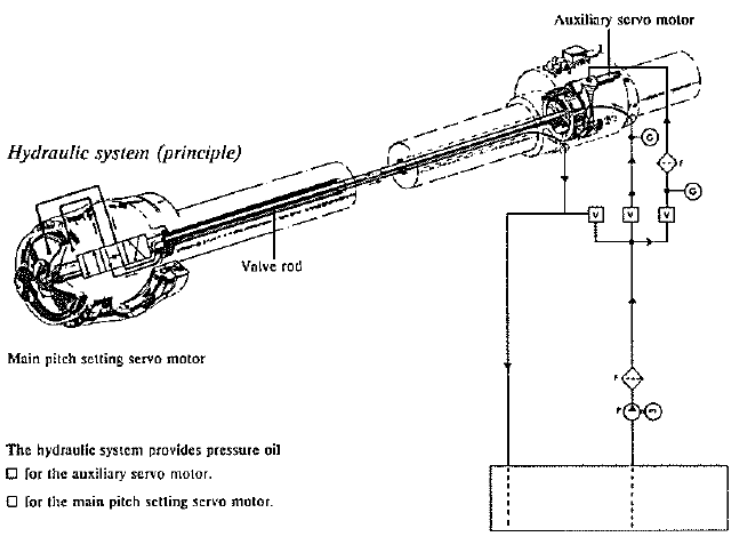

FIGURE 27: WORKING PRINCIPLE OF THE CONTROLLABLE PITCH PROPELLER (21)

4.2.2 ADVANTAGES AND DISADVANTAGES OF CONTROLLABLE PITCH PROPELLERS

Advantages

- Rapid maneuvering; the engine runs at constant load and speed while the pitch control varies the thrust instantaneously
- There is never need to shut down the engine even if no thrust is needed

- Higher average fuel efficiency; the ME is able to maintain a constant, optimal load with varying ship speed
- The propeller is able to maximize the thrust based on the engine RPM
- As discussed in chapter 4.1, advantageous for PTO systems (20) (24)

Disadvantages

- The larger propeller boss reduces the propeller efficiency
- A CPP arrangement is on average "3-4 times more expensive" than FPPs
- More moving parts and advanced systems means that it is more prone to failures
- An unexpected failure of the system can have disastrous results (e.g. if the propeller fails to give thrust when needed)
- Complicated and expensive maintenance (20)

Economic considerations

No exact numbers are available to assess the financial benefits of a CPP. The more advanced system means that the investment costs are higher; by "3-4 times" according to the author of *Diesel Engines I & II*, Kees Kuiken (20). A CPP arrangement is also likely to have higher maintenance and spare part costs. Maintenance tasks on the propeller system may require drydocking of the ship, resulting in unnecessary downtime if the failure is unexpected.

A disadvantage of the CPP is the reduced propeller efficiency compared to an FPP of the same diameter because of the larger propeller boss on the former. However, a variable pitch means that, in the case of a varying engine speed, the propeller can adjust the pitch in order to have maximum thrust for any engine RPM. So, the CPP can both adjust the thrust at constant optimal ME speed and maximize thrust with varying engine speed. This means that increased fuel efficiency can be achieved with the right application of CPPs.

4.3 Discussion

This task was originally meant to contain an assessment of different technologies' ability to reduce fuel consumption. The combination of waste heat recovery systems, shaft generators, controllable pitch propellers and frequency converters was highlighted. Waste heat utilization was thoroughly discussed earlier in the report. It was decided that frequency converters are a part of some of the shaft generator systems and, due to the fact that electrical systems are far outside of the author's area of expertise, a further study of the subject was dropped.

Shaft generators

This report focuses on energy saving and efficiency and not only on financial benefits. MAN Diesel & Turbo's study clearly shows that, while it may be financially beneficial to install a shaft generator, it will not necessarily mean reduced fuel costs. The study, however, is based on a setup with a GCR requiring constant ME speed and a CPP. The result is reduced propeller efficiency and constant load on the ME even at reduced vessel speeds. With the more advanced CFE or RCF arrangements we could see a different outcome because the engine would be able to vary its load a lot more for varying vessel speeds. On the other hand, these are more expensive and the maintenance costs are likely to be higher than with the simpler GCR.

Whether a shaft generator system is advantageous or not will vary with ship type and the shipowner's intention. While some owners may be environmentally conscious and will base their decisions on the potential reduction in fuel consumption, others are probably only looking for ways to improve their end result on their financial statements.

A situation as in MAN Diesel & Turbo's study obviously does not favor a shaft generator when looking strictly at energy efficiency. In a different setting, e.g. onboard a shuttle tanker having a high demand for electrical power for long periods of time, a shaft generator could on the other hand improve overall energy efficiency.

Controllable pitch propellers

The CPP and the FPP have slightly different areas of application.

In the case of a cargo ship spending most of its time on long transport legs at constant speed, a fixed pitch propeller would be preferred. This is due to it having a higher efficiency and that speed variations mostly occur near and in ports. The FPP is optimized for the expected ship speed and its efficiency makes it the more energy efficient alternative for a vessel of this type.

The CPP is suited when there is a high demand for maneuverability, either in the form of ME running at constant speed or when the speed is varying, or together with a shaft generator of the GCR type.

In the end it is the application of the propeller system that decides whether or not it will be an energy efficiency improvement or not.

Chapter 5:
Conclusion

In this report, various measures for improved energy efficiency have been assessed and engine data have been examined with the aim of identifying efficiency indicators.

The first part saw the assessment of various measures related to *Star Istind's* energy profile. The results show that, by implementing a few measures, a significant amount of fuel and diesel oil can be saved; at least theoretically. What the results are in practice requires more testing of the actual implemented measures.

It remains to be seen whether the measures assessed, along with other solutions not discussed in this report, will be implemented on a large scale. One alternative is slow steaming; reducing the speed of the ship and the engine load to an extent that allows for a significant reduction of fuel consumption. With engine manufacturers' introduction of engines with dual optimization, ships are able to maintain high efficiency and reliability at both low and high speeds. In times of limited demand for sea transport, this may be a desirable option. However, when there is a need for speed, implementing energy saving measures is a viable option.

The second part was an attempt to find the engine parameter best suited as an indicator of the engine's operation with regards to efficiency. The data for *Höegh Asia* was extracted from the TeCoMan database and comprised a number of different parameters that each potentially indicated the engine's efficiency. A few of these parameters seemed somewhat related to efficiency when compared over time, but most of them looked to be unrelated. Insufficient measurements and uncertainties regarding measurements methods and their consistency made it difficult to reach a conclusion based on the engine data. Hence the conclusion is that looking at the parameters separately does not give a clear picture of the engine efficiency.

The last part of the thesis is to be considered a general study of different technologies. As such, no conclusion has been reached other than the fact that these systems may or may not reduce fuel consumption depending on the area of their use.

CHAPTER 6:
FURTHER WORK

There are still things that can be done with regards to assess the effects of implementing energy saving measures on ships. The assessments in this report are based on one ships' energy profile and, for the most part, the various energy saving device manufacturer's own estimations for potential savings.

The same method as in this report can easily be applied to other ships' energy profiles, but the potential savings will vary between different ship types. When a manufacturer claims that savings of 5 % are possible with their product, their number is often based on guesswork, estimations and, at best, product testing with a very limited number of systems. How their product behaves and performs on several different systems is therefore difficult to say. Individual testing with the different technologies is probably the only way to conclude on the potential benefits for specific ships.

Regarding engine data as efficiency indicators, the conclusion was that looking at them individually did not serve much purpose. The measurements were too inconsistent and insufficient to be definitive indications. As of now, deviations in the parameters are mostly useful for deciding that particular part's condition and not so much the condition of the engine as a whole.

In order for the engine parameter measurements to be utilized in the desired way, i.e. as total engine condition indicators, one must find a way to get them on common ground and turn them into useful information. A performance index where all relevant measurement data are aggregated and treated to get a holistic indication of the engine performance would be ideal.

Measurement accuracy is a big issue, and eliminating sources for error in readings could make the data more reliable. Examples of such sources of error are human factors (manual readings, registering) and unreliable equipment. Automating the measurement process and using more reliable equipment could eliminate these.

GLOSSARY

AE	Auxiliary engine
CFD	Computational fluid dynamics
CFE	Constant frequency electrical
CPP	Controllable pitch propeller
EMIP	Energy management in practice
FOC	Fuel oil consumption
FPP	Fixed pitch propeller
GCR	Gear constant ratio
GSG	Grieg Shipping Group
HFO	Heavy fuel oil
MDO	Marine diesel oil
ME	Main engine
MIP	Mean indicated pressure
PTO	Power take off
RCF	Renk constant frequency
SFOC	Specific fuel oil consumption
SMCR	Specified maximum continuous rating
TES	Thermo efficiency system

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Appendix A - Star Istind's energy profile

Accounting list

Reference no.	System index	Energy category	Shortname	State 1 Full load (Normal seagoing) Unit	Item	Sum up	Unit	Item	Sum up	State 2 Ballast Unit	Item	Sum up	Unit	Item	Sum up
1	1.1		Main Engine	kW	9 387,8	9 387,8	MJ	101387755,10	101 387 755,1	kW	9 030,6	9 030,6	MJ	97 530 612	97 530 612,2
2	1.2	Production	Auxiliary Engines	kW	608,4	9 996,2	MJ	6570717,70	107 958 472,8	kW	608,4	9 639,0	MJ	6 570 718	104 101 329,9
3	1.3		Boiler	kW	0,0	9 996,2	MJ	0,00	107 958 472,8	kW	0,0	9 639,0	MJ	0	104 101 329,9
4	2.1		Shaft motor	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0	0,0
5	2.2	Transmission	Shaft generator	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0	0,0
6	2.3		Waste Heat Recovery	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0	0,0
7	2.4		Switchboard	kW	578,0	578,0	MJ	6242181,82	6 242 181,8	kW	578,0	578,0	MJ	6 242 182	6 242 181,8
8	3.1		Shaft Power	kW	9 200,0	9 200,0	MJ	99360000,00	99 360 000,0	kW	8 850,0	8 850,0	MJ	95 580 000	95 580 000,0
9	3.2.1		Main engine pumps over 50 kW	kW	105,5	9 305,5	MJ	1139400,00	100 499 400,0	kW	105,5	8 955,5	MJ	1 139 400	96 719 400,0
10	3.2.2		Main engine pumps under 50 kW	kW	278,6	9 584,1	MJ	3008520,00	103 507 920,0	kW	278,6	9 234,1	MJ	3 008 520	99 727 920,0
11	3.3.1		Accomodation over 50 kW	kW	89,0	9 673,1	MJ	961200,00	104 469 120,0	kW	89,0	9 323,1	MJ	961 200	100 689 120,0
12	3.3.2		Accomodation under 50 kW	kW	80,0	9 753,1	MJ	864360,00	105 333 480,0	kW	80,0	9 403,1	MJ	864 360	101 553 480,0
13	3.4.1		Cargo Heat over 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
14	3.4.2		Cargo Heat under 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
15	3.5.1		Thrusters over 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
16	3.5.2	Consumption	Thrusters under 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
17	3.6.1		Cargo pumps over 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
18	3.6.2		Cargo pumps under 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
19	3.7.1		Cargo gear over 50 kW	kW	0,0	9 753,1	MJ	0,00	105 333 480,0	kW	0,0	9 403,1	MJ	0	101 553 480,0
20	3.7.2		Cargo gear under 50 kW	kW	19,1	9 772,2	MJ	206280,00	105 539 760,0	kW	19,1	9 422,2	MJ	206 280	101 759 760,0
21	3.8.1		Ballast pumps over 50 kW	kW	0,0	9 772,2	MJ	0,00	105 539 760,0	kW	0,0	9 422,2	MJ	0	101 759 760,0
22	3.8.2		Ballast pumps under 50 kW	kW	0,0	9 772,2	MJ	0,00	105 539 760,0	kW	0,0	9 422,2	MJ	0	101 759 760,0
23	3.9.1		Reefers over 50 kW	kW	0,0	9 772,2	MJ	0,00	105 539 760,0	kW	0,0	9 422,2	MJ	0	101 759 760,0
24	3.9.2		Reefers under 50 kW	kW	0,0	9 772,2	MJ	0,00	105 539 760,0	kW	0,0	9 422,2	MJ	0	101 759 760,0

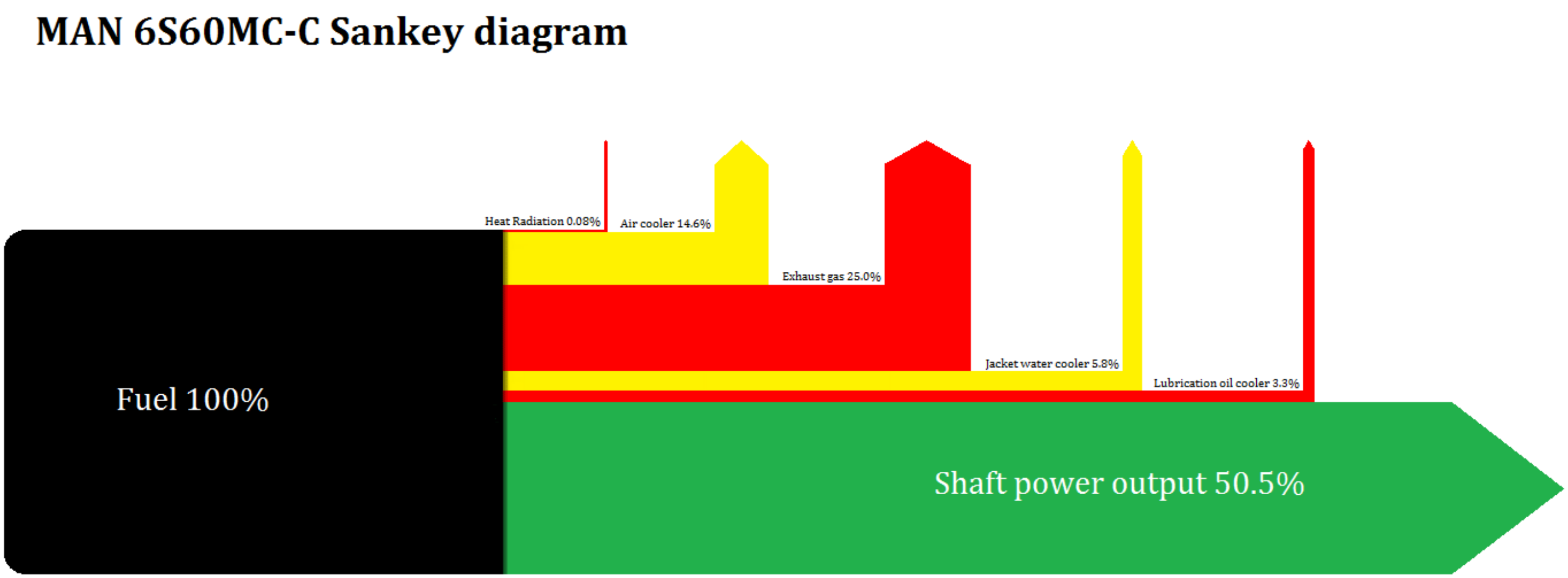
Reference no.	State 3 Intermediate Unit	Item	Sum up	Unit	Item	Sum up	State 4 manoeuvring (Leaving port with thruster) Unit	Item	Sum up	Unit	Item	Sum up
1	kW	8 673,5	8 673,5	MJ	31224489,80	31 224 489,8	kW	0,0	0,0	MJ	0,00	0,0
2	kW	608,4	9 281,9	MJ	2190239,23	33 414 729,0	kW	2 998,5	2 998,5	MJ	215890,27	215 890,3
3	kW	0,0	9 281,9	MJ	0,00	33 414 729,0	kW	0,0	2 998,5	MJ	0,00	215 890,3
4	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0,00	0,0
5	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0,00	0,0
6	kW	0,0	0,0	MJ	0,00	0,0	kW	0,0	0,0	MJ	0,00	0,0
7	kW	578,0	578,0	MJ	2080727,27	2 080 727,3	kW	2 848,6	2 848,6	MJ	205095,76	205 095,8
8	kW	8 500,0	8 500,0	MJ	30600000,00	30 600 000,0	kW	0,0	0,0	MJ	0,00	0,0
9	kW	105,5	8 605,5	MJ	379800,00	30 979 800,0	kW	241,5	241,5	MJ	17388,00	17 388,0
10	kW	278,6	8 884,1	MJ	1002840,00	31 982 640,0	kW	269,6	511,1	MJ	19413,60	36 801,6
11	kW	89,0	8 973,1	MJ	320400,00	32 303 040,0	kW	89,0	600,1	MJ	6408,00	43 209,6
12	kW	80,0	9 053,1	MJ	288120,00	32 591 160,0	kW	52,3	652,5	MJ	3768,00	46 977,6
13	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	0,0	652,5	MJ	0,00	46 977,6
14	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	0,0	652,5	MJ	0,00	46 977,6
15	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	2 095,0	2 747,5	MJ	150840,00	197 817,6
16	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	7,4	2 754,9	MJ	532,80	198 350,4
17	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	0,0	2 754,9	MJ	0,00	198 350,4
18	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	0,0	2 754,9	MJ	0,00	198 350,4
19	kW	0,0	9 053,1	MJ	0,00	32 591 160,0	kW	65,2	2 820,1	MJ	4694,40	203 044,8
20	kW	19,1	9 072,2	MJ	68760,00	32 659 920,0	kW	0,0	2 820,1	MJ	0,00	203 044,8
21	kW	0,0	9 072,2	MJ	0,00	32 659 920,0	kW	0,0	2 820,1	MJ	0,00	203 044,8
22	kW	0,0	9 072,2	MJ	0,00	32 659 920,0	kW	0,0	2 820,1	MJ	0,00	203 044,8
23	kW	0,0	9 072,2	MJ	0,00	32 659 920,0	kW	0,0	2 820,1	MJ	0,00	203 044,8
24	kW	0,0	9 072,2	MJ	0,00	32 659 920,0	kW	0,0	2 820,1	MJ	0,00	203 044,8

Reference no.	State 5 Loading (Cargo handling) Unit	Item	Sum up	Unit	Item	Sum up	State 6 Discharging Unit	Item	Sum up	Unit	Item	Sum up
1	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0
2	kW	813,3	813,3	MJ	1 698 077	1 698 077,2	kW	813,3	813,3	MJ	1 698 077	1 698 077,2
3	kW	0,0	813,3	MJ	0	1 698 077,2	kW	0,0	813,3	MJ	0	1 698 077,2
4	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0
5	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0
6	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0
7	kW	772,6	772,6	MJ	1 613 173	1 613 173,3	kW	772,6	772,6	MJ	1 613 173	1 613 173,3
8	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0
9	kW	70,7	70,7	MJ	147 691	147 691,2	kW	70,7	70,7	MJ	147 691	147 691,2
10	kW	219,3		MJ	457 898	605 589,6	kW	219,3		MJ	457 898	605 589,6
11	kW	82,2	152,9	MJ	171 634	777 223,2	kW	82,2	152,9	MJ	171 634	777 223,2
12	kW	220,3		MJ	460 056	1 237 279,2	kW	220,3		MJ	460 056	1 237 279,2
13	kW	0,0	152,9	MJ	0	1 237 279,2	kW	0,0	152,9	MJ	0	1 237 279,2
14	kW	0,0		MJ	0	1 237 279,2	kW	0,0		MJ	0	1 237 279,2
15	kW	0,0	152,9	MJ	0	1 237 279,2	kW	0,0	152,9	MJ	0	1 237 279,2
16	kW	0,0		MJ	0	1 237 279,2	kW	0,0		MJ	0	1 237 279,2
17	kW	0,0	152,9	MJ	0	1 237 279,2	kW	0,0	152,9	MJ	0	1 237 279,2
18	kW	0,0		MJ	0	1 237 279,2	kW	0,0		MJ	0	1 237 279,2
19	kW	65,2	218,1	MJ	136 138	1 373 416,8	kW	65,2	218,1	MJ	136 138	1 373 416,8
20	kW	0,0		MJ	0	1 373 416,8	kW	0,0		MJ	0	1 373 416,8
21	kW	107,1	325,2	MJ	223 625	1 597 041,6	kW	107,1	325,2	MJ	223 625	1 597 041,6
22	kW	0,0		MJ	0	1 597 041,6	kW	0,0		MJ	0	1 597 041,6
23	kW	0,0		MJ	0	1 597 041,6	kW	0,0		MJ	0	1 597 041,6
24	kW	0,0	325,2	MJ	0	1 597 041,6	kW	0,0	325,2	MJ	0	1 597 041,6

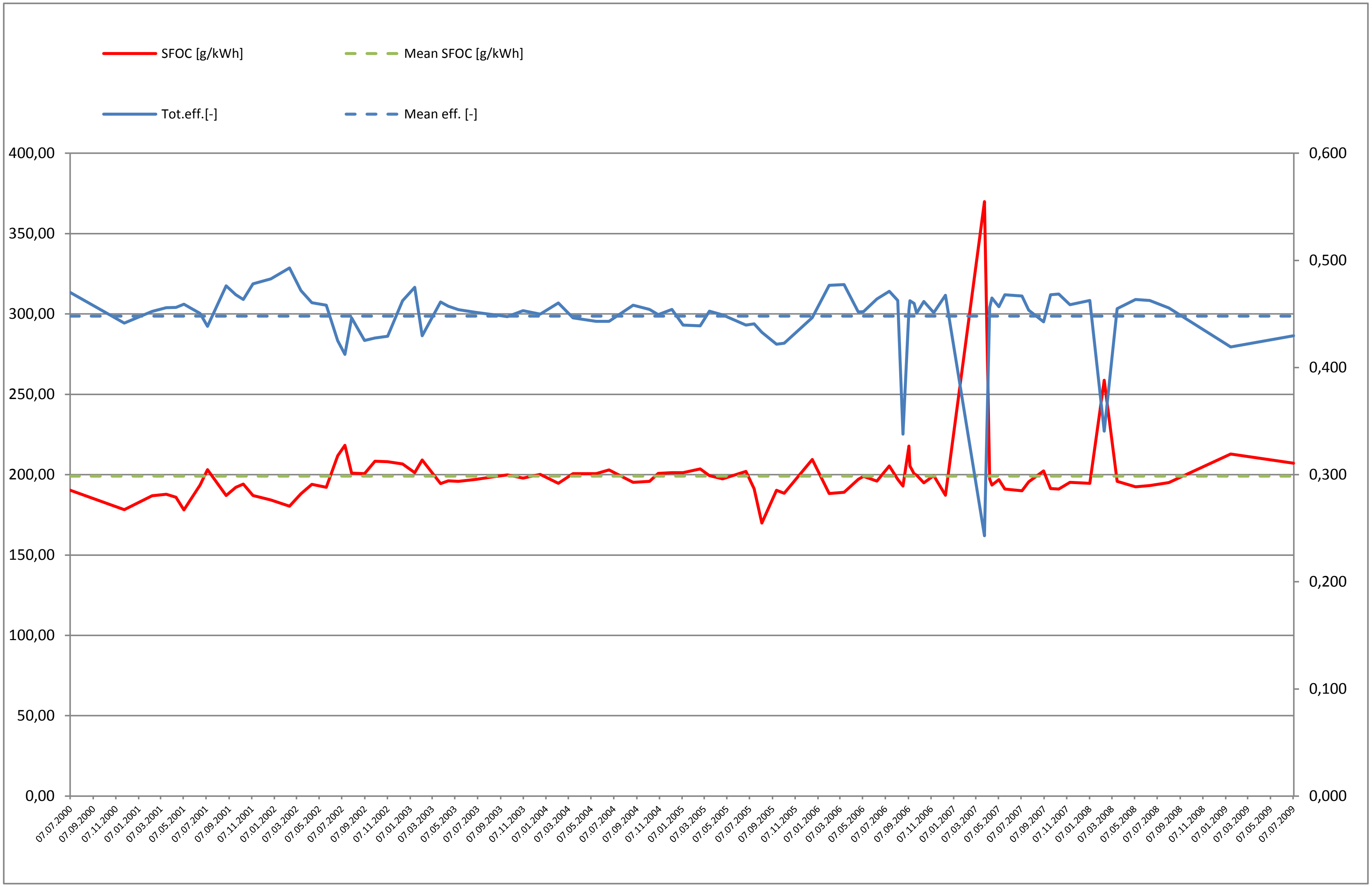
Reference no.	State 7 Waiting (Rest in port) Unit	Item	Sum up	Unit	Item	Sum up	State 8 Restricted waters Unit	Item	Sum up	Unit	Item	Sum up	Total consumption unit	Item	Sum up
1	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0	MJ	230 142 857,1	230 142 857
2	kW	448,6	448,6	MJ	968 957	968 956,9	kW	0,0	0,0	MJ	0	0,0	MJ	18 214 599,0	248 357 456
3	kW	0,0	448,6	MJ	0	968 956,9	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	248 357 456
4	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	0
5	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	0
6	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	0
7	kW	426,2	426,2	MJ	920 509	920 509,1	kW	0,0	0,0	MJ	0	0,0	MJ	17 303 869,1	17 303 869
8	kW	0,0	0,0	MJ	0	0,0	kW	0,0	0,0	MJ	0	0,0	MJ	225 540 000,0	225 540 000
9	kW	67,7	67,7	MJ	146 304	146 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	2 969 983,2	228 509 983
10	kW	178,9	246,7	MJ	386 496	532 800,0	kW	0,0	0,0	MJ	0	0,0	MJ	7 883 688,0	236 393 671
11	kW	82,2	328,9	MJ	177 552	710 352,0	kW	0,0	0,0	MJ	0	0,0	MJ	2 598 393,6	238 992 065
12	kW	93,0	421,9	MJ	200 952	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	2 681 616,0	241 673 681
13	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	241 673 681
14	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	241 673 681
15	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	150 840,0	241 824 521
16	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	532,8	241 825 054
17	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	241 825 054
18	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	241 825 054
19	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	140 832,0	241 965 886
20	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	481 320,0	242 447 206
21	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	223 624,8	242 670 830
22	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	242 670 830
23	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	242 670 830
24	kW	0,0	421,9	MJ	0	911 304,0	kW	0,0	0,0	MJ	0	0,0	MJ	0,0	242 670 830

Source: MARINTEK's energy profile for Star Istind

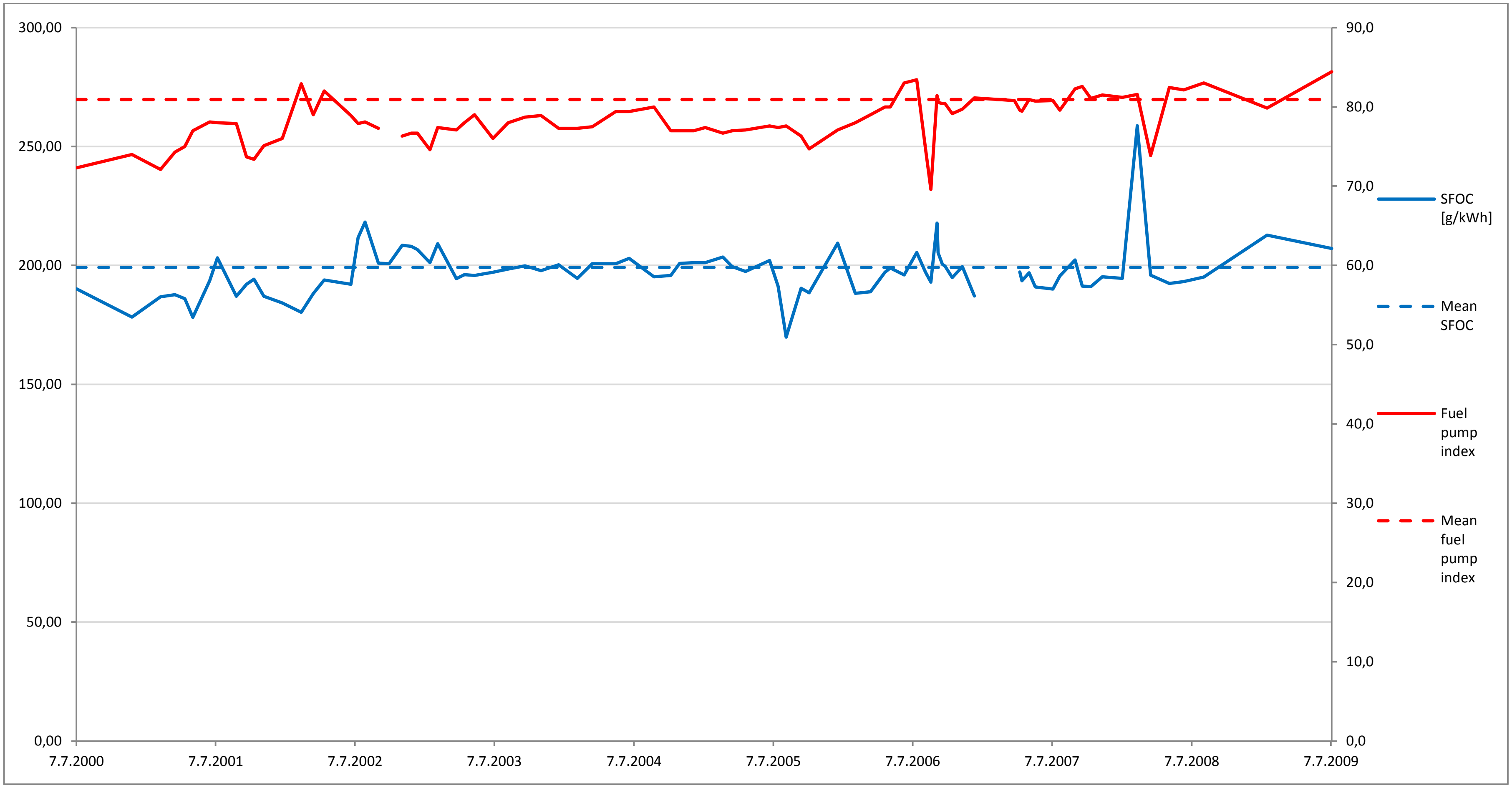
APPENDIX B - SANKEY DIAGRAM FOR MAN 6S60MC-C TWO-STROKE ENGINE



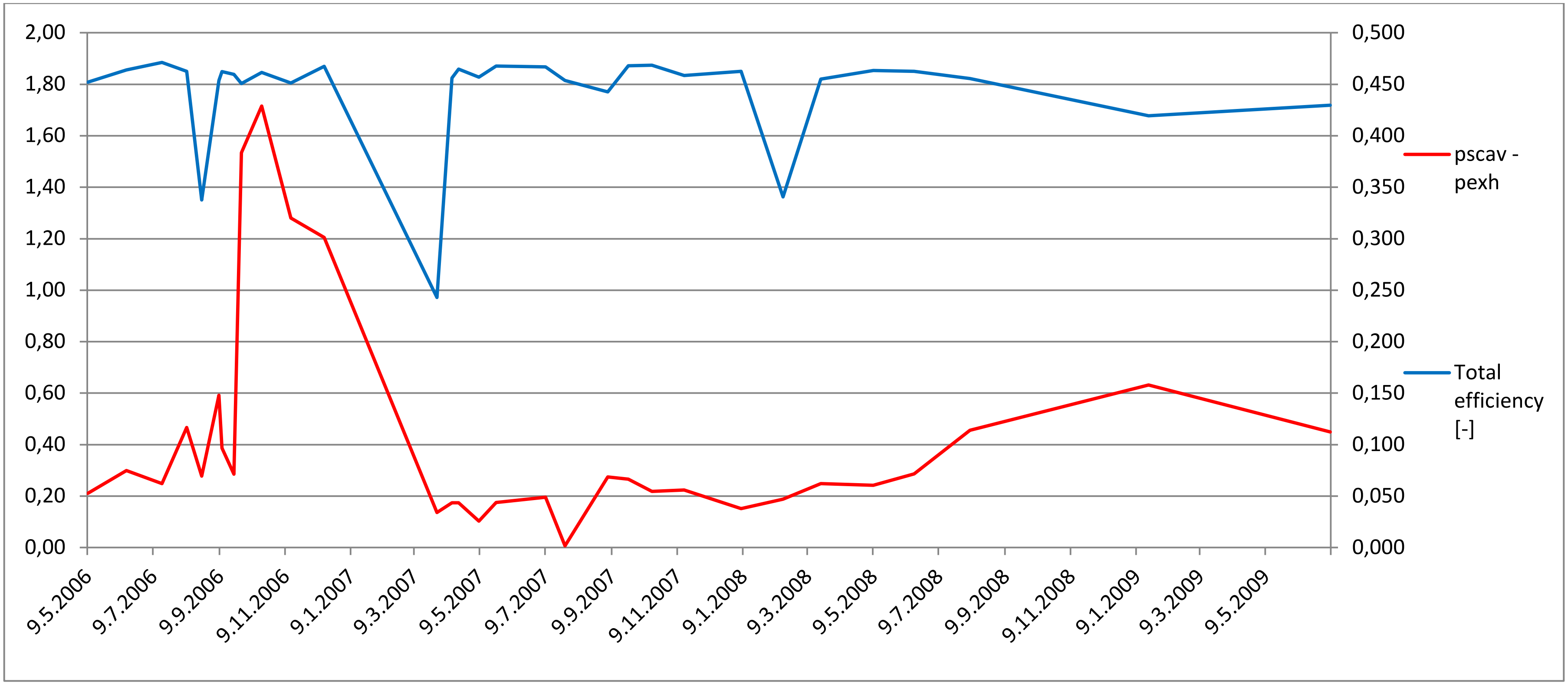
APPENDIX C - EFFICIENCY AND SFOC



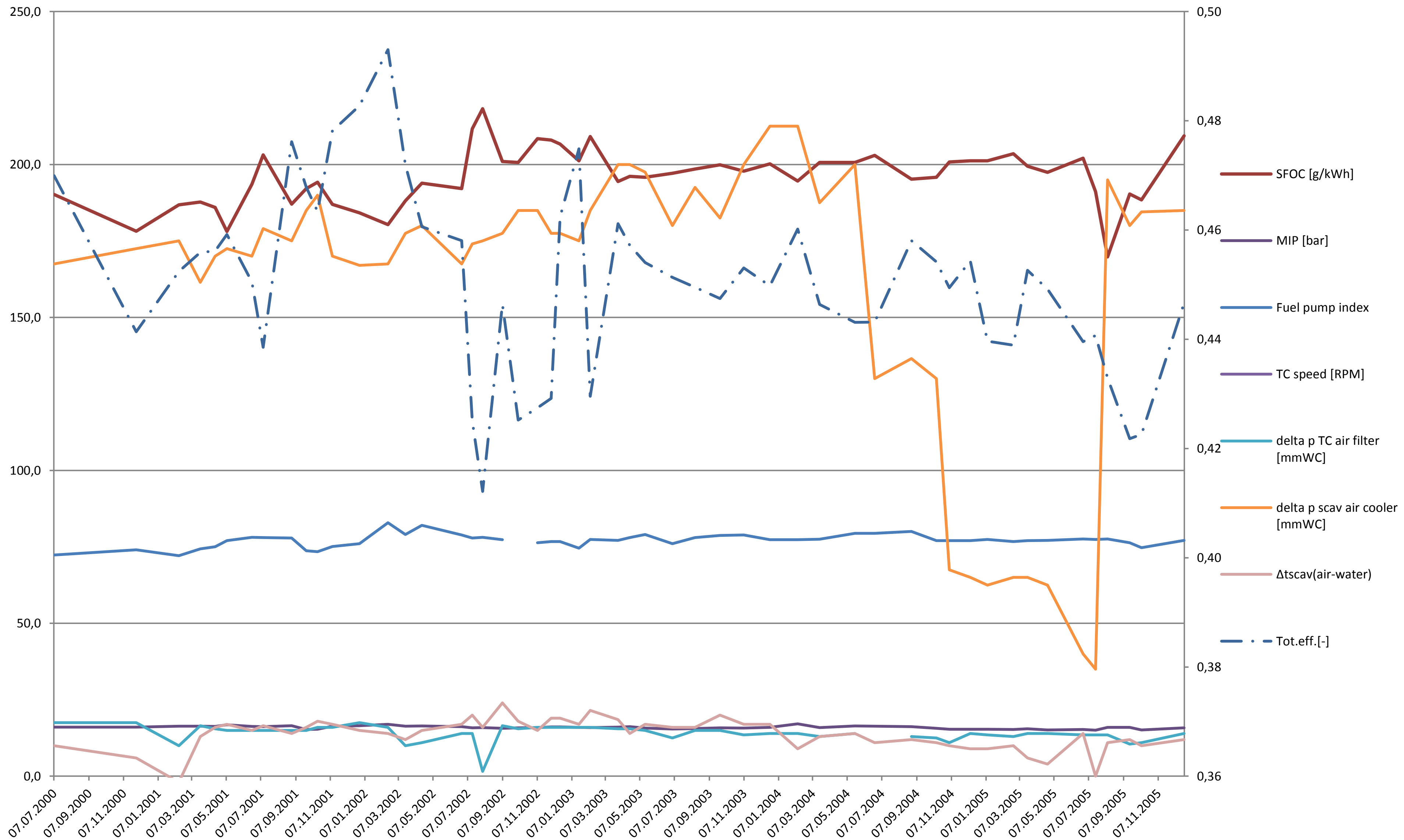
APPENDIX D - FUEL PUMP INDEX AND SFOC



APPENDIX E - $\Delta P_{SCAV-EXH}$ AND EFFICIENCY



APPENDIX F - EFFICIENCY AND OTHER PARAMETERS



Appendix G - Extract from dataset

Date ↓	Tot.eff.[-]	SFOC [g/kWh]	MIP [bar]	Fuel pump index	TC speed [RPM]	Δp TC air filter [mmWC]	Δp scav air cooler [mmWC]	Δtscav(air-water)
Mean →	0,45	199,1	16,0	78,3	12317,9	15,3	175,0	12,6
08.07.2009	0,43	207,1	15,7	84,4	12600,0		195,0	2,0
20.01.2009	0,42	212,8	15,0	79,9	12000,0	22,5	210,0	13,0
07.08.2008	0,46	195,1	16,0	83,0	12250,0	17,5	212,5	14,0
16.06.2008	0,46	193,2	15,6	82,1	12400,0	17,5	212,5	10,0
09.05.2008	0,46	192,5	16,0	82,4	12100,0	16,6	200,0	10,0
21.03.2008	0,45	195,8	16,2	73,9	12400,0	17,5	225,0	13,0
15.02.2008	0,34	258,8	16,1	81,6	12050,0	15,0	225,0	12,0
07.01.2008	0,46	194,6	15,6	81,2	12300,0	20,0	245,0	11,0
15.11.2007	0,46	195,2	15,7	81,5	12000,0		230,0	12,0
16.10.2007	0,47	191,1	15,6	81,1	12300,0		237,5	9,0
24.09.2007	0,47	191,3	16,1	82,6	12450,0	27,5	235,0	11,0
05.09.2007	0,44	202,3	15,8	82,3	12350,0	19,0	232,5	11,0
27.07.2007	0,45	195,6	15,1	79,6	12150,0	16,0	212,5	11,0
09.07.2007	0,47	190,0	15,6	80,8	12050,0	12,5	225,0	12,0
24.05.2007	0,47	191,0	15,9	80,7	12200,0	20,0	217,5	18,0
08.05.2007	0,46	196,9	15,7	80,9	12300,0	18,5	187,5	14,0
19.04.2007	0,46	193,5	15,6	79,4	12050,0	16,0	180,0	10,0
13.04.2007	0,46	197,2	15,7	79,6	12150,0	19,0	180,0	11,0
30.03.2007	0,24	370,0	15,9	80,8	12050,0	16,5	182,5	11,0
15.12.2006	0,47	187,2	15,4	81,1	12250,0		230,0	3,0
14.11.2006	0,45	199,4	15,6	79,7	12150,0	17,5	129,5	13,0
18.10.2006	0,46	194,9	15,7	79,1	12050,0	17,5	129,0	1,0
29.09.2006	0,45	199,7	15,8	80,4	12150,0	20,0	217,5	10,0
22.09.2006	0,46	200,5	15,8	80,4	12125,0	17,5	220,0	10,0
11.09.2006	0,46	205,4	14,8	80,6	12025,0	17,5	230,0	19,0
08.09.2006	0,45	217,8	14,8	81,4	12100,0	20,0	245,0	10,0
23.08.2006	0,34	192,9	15,2	69,6	12000,0	20,0	245,0	11,0
09.08.2006	0,46	197,1	15,9	74,7	12000,0	20,0	240,0	11,0
17.07.2006	0,47	205,4	15,6	83,4	12500,0	17,5	200,0	11,0
14.06.2006	0,46	196,0	15,1	83,0	12400,0	20,0	240,0	10,0
09.05.2006	0,45	198,9	15,5	80,0	12400,0	10,0	160,0	9,0
25.04.2006	0,45	197,1	15,1	80,0	12350,0	17,5	64,0	12,0
18.03.2006	0,48	189,0	15,7	79,0	12125,0	16,5	60,5	12,0
06.02.2006	0,48	188,3	15,7	78,0	12175,0	17,5	64,0	12,0
22.12.2005	0,45	209,4	15,8	77,1	12050,0	14,0	185,0	12,0
08.10.2005	0,42	188,4	15,1	74,7	11950,0	11,0	184,5	10,0
17.09.2005	0,42	190,3	15,9	76,3	11950,0	10,5	180,0	12,0
09.08.2005	0,43	169,8	16,0	77,6	12250,0	13,5	195,0	11,0
19.07.2005	0,44	191,2	15,0	77,4	11850,0	13,5	35,0	0,0
27.06.2005	0,44	202,0	15,3	77,6	12300,0	13,5	40,0	14,0
25.04.2005	0,45	197,4	15,1	77,1	12050,0	14,0	62,5	4,0
21.03.2005	0,45	199,5	15,5	77,0	12150,0	14,0	65,0	6,0
24.02.2005	0,44	203,5	15,3	76,7	12250,0	13,0	65,0	10,0
09.01.2005	0,44	201,2	15,3	77,4	12350,0	13,5	62,5	9,0
10.12.2004	0,45	201,2	15,3	77,0	12300,0	14,0	65,0	9,0
03.11.2004	0,45	200,8	15,3	77,0	12400,0	11,0	67,5	10,0
11.10.2004	0,45	195,8	15,6	77,0	12450,0	12,5	130,0	11,0
28.08.2004	0,46	195,2	16,2	80,0	12600,0	13,0	136,5	12,0
24.06.2004	0,44	203,0	16,4	79,4	12600,0		130,0	11,0
20.05.2004	0,44	200,7	16,4	79,4	12400,0	14,0	200,0	14,0
19.03.2004	0,45	200,7	15,9	77,5	12400,0	13,0	187,5	13,0
09.02.2004	0,46	194,6	17,1	77,3	12400,0	14,0	212,5	9,0
22.12.2003	0,45	200,2	16,0	77,3	12400,0	14,0	212,5	17,0
06.11.2003	0,45	197,8	15,7	78,9	12300,0	13,5	200,0	17,0
25.09.2003	0,45	199,9	15,8	78,7	12350,0	15,0	182,5	20,0
12.08.2003	0,45	198,5	15,6	78,0	12000,0	15,0	192,5	16,0
03.07.2003	0,45	197,1	15,5	76,0	12100,0	12,5	180,0	16,0
16.05.2003	0,45	195,8	15,7	79,0	12100,0	15,0	197,5	17,0
19.04.2003	0,46	196,1	16,2	78,0	12300,0	15,5	200,0	14,0
29.03.2003	0,46	194,4	16,1	77,1	12100,0	15,5	200,0	18,5
08.02.2003	0,43	209,1	15,9	77,4	12000,0	16,0	185,0	21,5
19.01.2003	0,47	201,2	16,0	74,6	12000,0	16,0	175,0	17,0
17.12.2002	0,46	206,7	16,1	76,7	12300,0	16,0	177,5	19,0
01.12.2002	0,43	208,0	16,1	76,7	12300,0	16,0	177,5	19,0
07.11.2002	0,43	208,4	15,9	76,3	12100,0	16,0	185,0	15,0
04.10.2002	0,43	200,7	15,9		12100,0	15,5	185,0	18,0
06.09.2002	0,45	201,0	15,7	77,3	12150,0	16,5	177,5	24,0
02.08.2002	0,41	218,2	15,9	78,1	12500,0	1,6	175,0	16,0
15.07.2002	0,43	211,7	15,9	77,9	12400,0	14,0	174,0	20,0
26.06.2002	0,46	192,1	16,2	78,9	12400,0	14,0	167,5	17,0
17.04.2002	0,46	193,9	16,4	82,0	12300,0	11,0	180,0	15,0
19.03.2002	0,47	188,2	16,3	79,0	12000,0	10,0	177,5	12,0
16.02.2002	0,49	180,3	17,0	82,9	12000,0	16,0	167,5	14,0
28.12.2001	0,48	184,2	16,5	76,0	11900,0	17,5	167,0	15,0
10.11.2001	0,48	187,0	16,4	75,1	11900,0	16,0	170,0	17,0
15.10.2001	0,46	194,2	15,4	73,4	11850,0	16,0	190,0	18,0
25.09.2001	0,47	192,1	15,3	73,7	11800,0	15,0	185,0	16,0
30.08.2001	0,48	187,0	16,5	77,9	12050,0	15,0	175,0	14,0
11.07.2001	0,44	203,1	16,2	78,0	12050,0	15,0	179,0	16,5
21.06.2001	0,45	193,5	16,3	78,1	12150,0	15,0	170,0	15,0
08.05.2001	0,46	178,1	16,9	77,0	12100,0	15,0	172,5	17,0
17.04.2001	0,46	186,0	16,3	75,0	12000,0	15,5	170,0	16,0
22.03.2001	0,46	187,7	16,3	74,3	11900,0	16,5	161,5	13,0
12.02.2001	0,45	186,8	16,4	72,1	11975,0	10,0	175,0	-2,0
29.11.2000	0,44	178,2	16,1	74,0	11700,0	17,5	172,5	6,0
07.07.2000	0,47	190,2	16,1	72,3	11800,0	17,5	167,5	10,0

Negative deviation relative to mean value marked in red.

APPENDIX H I) - ECONOMIC CONSIDERATIONS: ALTARNATIVE WITH GCR PTO

Machinery arrangement

7S50MC•C with PTO	
SMCR (kW)	11,060
Efficiency PTO (%)	92
2 x 7L16/24H gensets	

Basic data

Total days at sea	250
Total days in harbour	115
HFO price (USD/ton)	140
System oil price (USD/ton)	800
Cylinder oil price (USD/ton)	900

Load pattern

Load case	1	2	3	4	5
Hours	900	2,400	2,100	600	2,760
Propulsion power (kW)	8,780	7,833	6,916	5,142	0
PTO mech. power (kW)	975	975	975	975	0
Main engine power (kW)	9,755	8,808	7,891	6,117	0
MEP (bar)	16.8	15.1	13.6	10.5	0
SFOC (g/kWh)	168.4	168.0	168.2	170.4	0
Genset power (kW_{el})	0	0	0	0	500

Fuel oil consumption

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Main engine power (kW)	9,755	8,808	7,891	6,117	0		
SFOC (g/kWh)	168.4	168.0	168.2	170.4	0		
Genset power (kW_{el})	0	0	0	0	500		
SFOC (g/kWh_{el})	0	0	0	0	202		
HFO (tons/year)	1,571	3,773	2,961	664	296	9,265	1,297,100

(SFOC: ref. LCV = 42,700 kJ/kg)

(HFO: ref. LCV = 40,200 kJ/kg)

System oil consumption

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Main engine (kg/24h)	31	31	31	31	0		
Genset (kg/24h)	0	0	0	0	12		
System oil (tons/year)	1.2	3.1	2.7	0.8	1.4	9.2	7,360

Cylinder oil consumption (based on 1.02 g/kWh and Alpha ACC with 3% sulphur)

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Cylinder oil (tons/year)	9.0	21.6	16.9	3.7	0.0	51.2	46,080

Total cost per year (USD) excl. maintenance cost 1,350,540

Source: *MAN Diesel & Turbo: Shaft Generators* (22)

APPENDIX H II) - ECONOMIC CONSIDERATIONS: ALTARNATIVE WITHOUT GCR PTO

Machinery arrangement

7S50MC•C without PTO	
SMCR (kW)	9,760
Efficiency PTO (%)	–
3 x 7L16/24H gensets	

Basic data

Total days at sea	250
Total days in harbour	115
HFO price (USD/ton)	140
System oil price (USD/ton)	800
Cylinder oil price (USD/ton)	900

Load pattern

Load case	1	2	3	4	5
Hours	900	2,400	2,100	600	2,760
Propulsion power (kW)	8,784	7,808	6,832	4,880	0
PTO mech. power (kW)	0	0	0	0	0
Main engine power (kW)	8,784	7,808	6,832	4,880	0
MEP (bar)	16.2	14.8	13.2	9.9	0
SFOC (g/kWh)	166.6	165.5	164.9	168.6	0
Genset power (kW_{el})	900	900	900	900	500

Fuel oil consumption

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Main engine power (kW)	8,784	7,808	6,832	4,880	0		
SFOC (g/kWh)	166.6	165.5	164.9	168.6	0		
Genset power (kW_{el})	900	900	900	900	500		
SFOC (g/kWh_{el})	205	205	205	205	202		
HFO (tons/year)	1,576	3,765	2,925	642	296	9,204	1,288,560

(SFOC: ref. LCV = 42,700 kJ/kg)

(HFO: ref. LCV = 40,200 kJ/kg)

System oil consumption

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Main engine (kg/24h)	31	31	31	31	0		
Genset (kg/24h)	24	24	24	24	12		
System oil (tons/year)	2.1	5.5	4.8	1.4	1.4	15.2	12,160

Cylinder oil consumption (based on 1.49 g/kWh at nominal MCR and reduced proportional to MEP at part load)

Load case	1	2	3	4	5	Total	Cost (USD)
Hours	900	2,400	2,100	600	2,760		
Main engine power (kW)	8,784	7,808	6,832	4,880	0		
Cylinder oil (tons/year)	8.1	19.1	14.6	3.0	0.0	44.8	40,320

Total cost per year (USD) excl. maintenance cost 1,341,040

Source: *MAN Diesel & Turbo: Shaft Generators* (22)