Hybrid propulsion systems

Efficiency analysis and design methodology of hybrid propulsion systems

Master thesis

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Delft University of Technology

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Foreword

This thesis is written to obtain the Master of Science degree in Marine Engineering at Delft University of Technology. Its main goal is to develop a general design methodology for hybrid propulsion systems, with the focus on fuel efficiency. New technologies and possibilities mean that ship propulsion systems have become increasingly more complex to design, while fuel prices have been rising significantly over the last years.
Fuel efficiency has become a more and more important topic concerning ship design, but also in general view of the world since it influences the environmental issues. Not only should we look at alternative methods for energy production, we should also make out existing methods as efficient as possible. This is of course also interesting from an economic point of view, with the ever-rising fuel prices.

The research is conducted at MAN Diesel & Turbo in Augsburg. I would like to thank them for this opportunity. I would like to thank all the colleagues for their support and interest in this subject. In particular I would like to thank Friedrich Wirz and Bernd Friedrich for the formulation of the assignment and their recommendations how to approach this project.
Next to the mentors at MAN Diesel & Turbo I also would like to thank the support at Delft University of Technology. In particular I would like to give my gratitude to Hugo Grimmelius for his suggestions and his enthusiasm in the subject. Next to this I must express my thanks to Professor Stapersma for his guidance and ambition to get the most out of his students.
Last but not least I would like to use this space to give a small thank-you to friends and family who gave me the moral support to write this thesis and the people in Augsburg who made me feel at home during the process.

March 2013,

Bas Kwasieckyj

Summary

A hybrid propulsion system features both a diesel engine and an electric motor for propulsion. Electric auxiliary power is generated by diesel generator sets. The electric motor can also act as a generator, in which case the generators can produce less power. However, the main engines should then provide this power. These degrees of freedom with power generation raise the question how this division between power can be optimised in such a way that the engines are running with their optimal fuel efficiency.
This strongly depends on the operational profile of the vessel. A generalised method to determine the power generation for all operating modes with a focus on the lowest fuel consumption of the diesel engines has to be developed. All influencing characteristics concerning the components in the system will have to be evaluated. More specifically, all relevant efficiencies between the required propulsion and auxiliary power and the produced brake power of the engines should be investigated.

To optimise the system, all the degrees of freedom should be brought into one function. This general function can then be optimised using the gradient search method. The decision variables in this general function are the powers to be delivered by the main engines, the diesel generators and the electric machine that can operate in both directions. These variables should be solved for every operating mode in the operational profile.
A function with many degrees of freedom can have many local minima and after a mathematical verification this indeed proves to be the case. General optimisation algorithms have difficulties finding a global minimum. Depending on the starting point of the algorithm, i.e. the first set of initial values of the decision variables, the algorithm usually finds a local minimum. Therefore there should be a method that can find an initial set of values that is already close to the global minimum. Taking a full combinatorial set of values would be impractical, while taking random values does not guarantee that a global minimum would be found. The method of orthogonal arrays is chosen, that uses pairwise combinations between the decision variables to determine which combination leads to a feasible initial result. This way the number of initial test cases can be reduced significantly. From these sets of initial values the gradient search method will then find the global minimum.
After validation of the algorithm to some real life examples of ship profiles, this indeed proved to be the case. These tests also provided feasible results, indicating the combination between orthogonal array selection and gradient search optimisation works properly.

Index

1 Introduction

Ships use different kinds of propulsion systems. Which is the best for a particular type of vessel strongly depends on the requirements set early in the design stage. Within these requirements, a suitable propulsion concept can be selected and an efficiency estimate of this concept can be determined. The efficiency of the vessel not only lies in the efficiencies of the single components, but a great efficiency potential might also be found in the synergy between them.
Because the shipping industry requires vessels that perfectly suit its purpose, and there are so many purposes in the marine environment, almost every ship is just a single project instead of a series product. This means that for every vessel the design methodology has to be carried out every single time. Not only is this time consuming, it is a repetitive procedure with only several variations for each ship. This thesis will try to perform this efficiency analysis in a structured way, so the methodology can be used for many different ship types.
Within this thesis, only four-stroke diesel engines are considered as the prime mover, with the focus on hybrid propulsion systems. A hybrid system features both a diesel engine and an electric motor for propulsion.

Similar other studies have been performed for designing a propulsion system. After simulations for a container feeder, De Ruyck [2011] states that a reduction of 8% in energy consumption can be achieved by introducing a 500 kW PTI/PTO motor to a 4000 kW diesel engine. It must be noted that most of this improvement is due to the use of the combinator curve, allowing the propeller to operate on a lower speed with a higher efficiency. The actual improvement of the hybrid concept versus conventional mechanical propulsion is only 1,2%. This shows that the benefits from a hybrid propulsion concept are very delicate and are largely dependent on the operational profile of the vessel.
For a naval surface combatant, [van Es, 2011] had an extensive and detailed description of the components used, including for instance weight and size. However, the criteria for selecting the best propulsion system was very ship specific. These relations will not be investigated as such in this thesis.
In [van Straten & de Boer, 2012] this study was performed for a range of naval vessels. The optimisation was done for the operating mode with the highest timeshare in the whole operational profile. The study presented here aims at an optimisation of all operating modes in the profile combined.
It is interesting to note that both [van Es, 2011] and [van Straten & de Boer, 2011] have suggested a type of hybrid propulsion. For the surface combatant van Es [2011] suggested a combination of one or two gas turbines with electric motors on the shaft for

slow sailing (COmbined Diesel eLectric And Gasturbine or CODLAG). The example case described for an Offshore Patrol Vessel (OPV) in [van Straten & de Boer, 2012] has the lowest annual fuel consumption with a hybrid system (COmbined Diesel eLectric And Diesel or CODLAD)

[van Deursen, 2011] did have an optimisation over varying power demands. The diesel engine fuel consumption was assumed to be linear, which is a decent fit. The efficiency of various components was assumed to be a constant, which also gives reasonable results for most operating points.
In this research a method that is applicable for non-linear behaviour of both the fuel consumption and the efficiency of the components will be attempted. The design procedure has to be made applicable for a wide range of vessel types, power ranges and operational profiles, requiring a less physical model but a more mathematical model of the vessels operational characteristics, like in [van Deursen, 2011]. Due to the iterative calculation steps the amount of detail in the components is kept to a more practical limit.

In general, this thesis will attempt to answer the following fundamental questions:

What are the determining factors in deciding whether to use a hybrid propulsion concept?
Because the focus of this paper is on hybrid propulsion, the first question must be whether to utilise such a system in the first place or not. Many factors may influence the choice of a hybrid propulsion. This can be operational costs, such as fuel costs, or the different operating modes the vessel should be able to perform.
Whether a hybrid system is suitable for a specific ship, should therefore be determined first. Chapter 2 deals with the introduction of the hybrid propulsion concept. Next to this, the boundary conditions in which this research is carried out will also be explained in chapter 2.

What components are used within the concept and what are their characteristics?
When deciding which layout of the propulsion system is to be used, one must first determine of which components such a system is composed. Therefore all components that are often found in the hybrid propulsion system should be defined in an early stage. Not only the choice of which different components are used must be made clear, it is also important to know precisely when each type of machinery is required. This leads to the need for a definition –or at least a reliable estimate- of the power range, operating speeds and efficiency of each and every component.
Furthermore, the technical limitations that these components might have can play a large role in determining the use of them. Such limitations might be an upper limit for the amount of power it can produce or handle, a lower or upper limit for the rotational speed or other limitations that are only applicable to the specific type of machinery.

Chapter 3 gives an overview of all the possible components with all their relevant attributes.
These attributes will be the building blocks on which the design algorithm can calculate a suitable solution.

What determines the layout and the parameters within the design?
When the building blocks of the system are determined, the next question quickly arises: which components should be used for a particular vessel and how should they be dimensioned?
In chapter 4 a more analytical view of how to optimise the design of a hybrid propulsion concept is provided.
To determine the configuration, a structured methodology is needed that can be used for all kinds of vessels. This methodology then should be able to choose which components should be implemented and which are not needed. Besides this, a design algorithm should be created so that all the relevant design parameters can quickly be calculated.
The methodology should be constructed in a way that the design algorithm can easily be implemented in a software tool.

Are the results plausible?
When the algorithmic tool suggests the different concepts, a next question might be to check the plausibility of the results based on existing configurations. A good way of doing this is to test the tool on some sample configurations taken from actual vessels. Four typical vessels where a hybrid concept is either already chosen or would be a feasible option are given in chapter 5. In here, typical operational profiles and power demands of the particular vessels are subjected to the design algorithm to find out whether the suggested configuration is plausible. When selecting a propulsion concept, not only the efficiency is important, but also the investment costs. Therefore, the investment costs of all the components should also be included.

Which parameters have a greater influence on the concept and which less?
When the design algorithm provides a suitable propulsion concept, one is always free to decide whether the resulting concept is indeed an acceptable solution, or whether a modified concept would be better. Minor changes in either the input or for example the size of a single component could change the outcome significantly.
Therefore the results must be subjected to a sensitivity analysis. This analysis uses changes in several variables within the system to find to what effect the parameters influence the outcome. The validation of the results and the sensitivity analysis is also carried out in chapter 5.

2 The hybrid propulsion concept

Conventional diesel mechanical propulsion plants usually have the total amount of power installed to fulfil the operating mode with the highest power requirement. With conventional cargo vessels this fits the vessels operational profile, where the vessel is in transit for long periods of time.
When a vessel has operating modes where significantly less propulsion power is required, the diesel engine has to run on a low load. Diesel engines operate with the lowest specific fuel consumption at around 85% load. In general, a hybrid propulsion system offers potential during these part-load operating modes. A better performance of the diesel engines might counteract the introduced extra losses due to the increased number of components.
Below a certain load (around 50%), fouling inside the engine also becomes a problem due to incomplete combustion. This results in environmental issues and maintainability of the engine.
A hybrid propulsion plant combines features of a diesel mechanical system with features of a diesel electric plant. In its most basic form the configuration consists of a diesel engine connected to a gearbox, which in turn drives the propeller. Also connected to the gearbox is an electric machine (EM), which can operate in generating mode or in motoring mode. This gives the possibility for a Power Take Off (PTO) mode or a Power Take In (PTI) mode.
The option to use either the electric motor or the diesel engine for propulsion (or even both) makes it a hybrid system.

In short, a hybrid propulsion concept has the potential to extend the economically and environmentally attractive operating area of the vessel.

2.1 Operation modes

The possibility to operate the electric machine in generator or motor mode gives the hybrid concept several possible operating modes.

2.1.1 Power Take Off

In normal Power Take Off mode, the main engine drives the propeller through a gearbox. A generator is connected at the gearbox that provides the electric power for the vessel auxiliary load, see Figure 2.1. In this operation the main engine is the only prime mover in operation, which runs on a high and therefore efficient load. In case the main engine is running on HFO, fuel costs for electric power generation will often be lower than an auxiliary engine running on MDO or MGO. Because the main engine should be running, this mode is often used when a vessel is in transit at cruising speed.

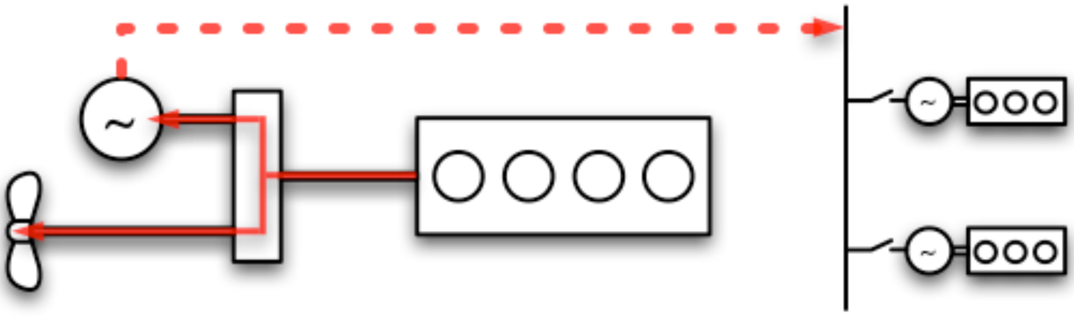

Figure 2.1: Energy flow in PTO mode

2.1.2 Boost Power Take In

In the booster mode the electric machine is in PTI mode and delivers extra power through the gearbox to the propeller. It is driving the propeller in parallel with the diesel engine, so extra power is transferred to the propeller. This power is provided by the diesel generators in the form of electrical power, see Figure 2.2. It has the advantage that power in electrical form can easily be transported throughout the ship through cables. Some vessels require high power for only short amounts of their total operational profile. Instead of installing a larger engine, the PTI can provide this extra power. This does require more or larger auxiliary engines.

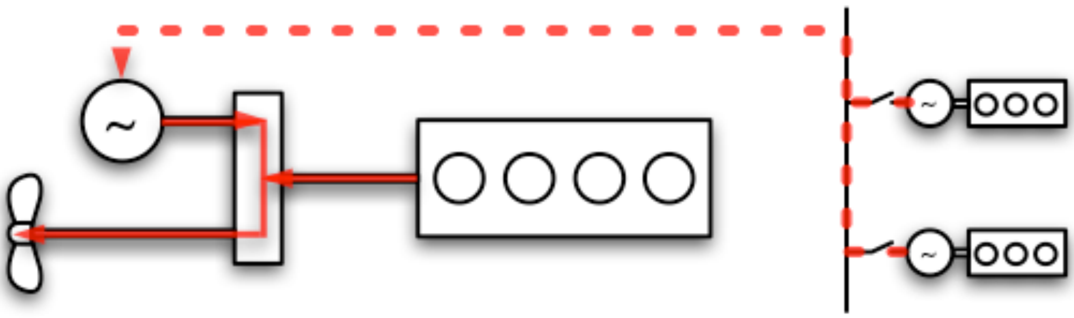

Figure 2.2: Energy flow in PTI boost mode

2.1.3 Slow Power Take In/Power Take Home

When the main diesel engine is not running a clutch can disconnect it from the gearbox. Propulsion power can still be provided by the electric motor, which uses the electric power delivered by the diesel generators (Figure 2.3). This could have two applications. One is for vessels that often sail at low speeds. At low load the diesel engine becomes less efficient and fouling starts to become an issue. The PTI motor will run on electric power that is provided by a flexible amount of diesel generators. Generally speaking,

more power is generated by the diesel generators, so larger or more auxiliary engines are required as opposed to a conventional diesel mechanic system.
The other application is an alternative propulsion capability. When the main engines fail for whatever reason, the ship is still capable of sailing on a slower speed using the electric motor. Class societies have set regulations regarding minimum speed, distance and/or power available.
The IMO has issued new SOLAS regulations that came into force in 2009 regarding safety on board passenger vessels. For given casualty scenarios, it sets requirements for safe evacuation of the passengers. One of these requirements is a redundant propulsion system or Power Take Home (PTH) function. This ensures the vessel will be able to return to port at limited speed after a defined amount of fire damage. At the moment there is no requirement regarding the speed, distance or available power. There is also no explicit requirement for the ship to be able to return to port in the case of flooding [Lloyds, 2010]. The regulations are applicable to passenger vessels constructed after July 1, 2010, with a length of over 120 meters or with three or more vertical fire zones[1].

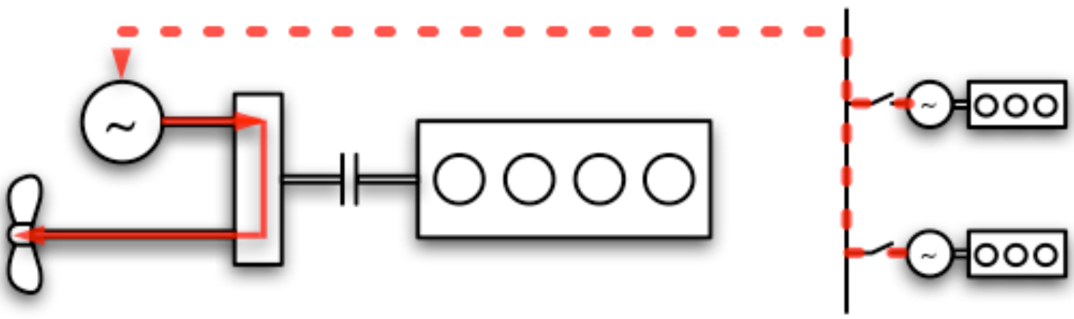

Figure 2.3: Energy flow in PTH mode

Any combinations between these three operating modes are possible. Of course, all these configurations are also possible on two or even more shaft applications. A two-shaft hybrid configuration could have an extra operating mode. If for any reason the port main engine fails, the starboard main engine can still drive the starboard propeller and a PTO. This electric power can then be transferred to the portside electric motor to drive the propeller. This so called cross-connection or electric-shaft between the two sides has the important advantage that the ship can still drive both sides to maintain manoeuvrability, without the need of using the auxiliary engines.

The electric motor can also be directly coupled onto the propeller shaft. This means that the motor has to run with the same relatively low speed as the propeller. To maintain the same power output, the torque produced by the motor has to be much higher. Because the torque of an electric motor is related to the current, the currents will increase. In order to withstand these higher currents the motor needs a higher number

[1] *SOLAS amendments Chapter II-1 (new regulation 8-1) and Chapter II-2 (regulations 21-22)*

of poles and therefore a larger construction in comparison to the high-speed / low-torque motors. The costs of such a motor will also be higher, as the purchase costs of an electric machine are largely dependent on the amount of copper used in the windings. For 4-stroke applications an electric motor on the propeller shaft is therefore not often used.

2.2 Benefit potential

In essence a hybrid propulsion concept becomes attractive when one of the following characteristics of a vessel's operational profile is the case:

- Large variations in both required propulsion and electrical power occur, with often a significant amount of low propulsion power demand.
- Maximum power for propulsion and electric loads do not occur simultaneously.
- The maximum electric power is determined by the auxiliary load and is not that large or constant that a fully diesel electric system would be a feasible solution.

2.2.1 Example ship types

Some examples of ship types where the above characteristics might be applicable are given in this paragraph.

Passenger vessels

Ferries or RoPax (combination between roll on/roll off and passenger) vessels that run on a fixed schedule sometimes have the need for a power boost in order to arrive on time in port. These vessels often require an alternative propulsion or PTH function that can also fulfil the Safe Return to Port regulations. A configuration with all operating modes might be installed on these vessels.

Cruise vessels often sail at night and lay in port during daytime. As the distance between the ports vary, the requested ship speed also varies. A PTI booster function can provide the extra speed required to arrive in port on time. A PTO might supply the relatively high hotel load. However, because of their size cruise vessels usually have a full diesel electric plant, omitting the need of a PTO.

Offshore vessels

Vessels with many varying loads, both for propulsion and electric load, benefit from the flexibility of the hybrid system. For example Anchor Handling Tug Support vessels (AHTS) have many different operating modes. When in transit, these vessels could use a PTO function. The required propulsion power is low with dynamic positioning (DP) in good weather, so a PTI could serve as the single propulsion motor. In bad weather the diesel engines could be started for more power. Also, sometimes the ship has the need to increase its bollard pull when towing. This condition might not happen very often, but the required power should be available. Instead of installing a larger engine, a PTI booster can deliver this extra torque. If such a vessel would operate in icy conditions the PTI motor can provide extra torque for propulsion. Because of the DP requirements and flexibility in positioning the propulsion motors on board the vessel, these vessels often have a full diesel electric propulsion system.

Chemical tankers

Conventional cargo vessels often equip a PTO in the form of a shaft generator. From the definition, this is not truly a hybrid concept. Chemical tankers that contain hazardous cargo, or cargo that cannot stay in the cargo tanks for too long could be equipped with a PTH function for the situation where the main propulsion fails.

Navy vessels

Offshore patrol vessels (OPV) usually sail at a low patrol speed. In this case the power can be supplied through the electric motor. When the vessel is required to respond to a call there is a need for high-speed operation. In this case the main engines could provide propulsion power with electric motor as a booster.
Frigates might have a more fluctuating electric power demand and sail at a steady cruising speed for longer periods of time. These vessels might benefit from a PTO as well as the PTI boost and PTI slow sailing.

Of course many other ship types can benefit from the hybrid propulsion concept, as long as their operational profiles demand a flexible system with high efficiency for all operating modes.

2.2.2 Emissions

Although compared to other prime movers the diesel engine is the most efficient, the use of heavy fuel oils have more impact regarding exhaust gas emissions. Global regulations concerning exhaust gas emissions have become more stringent in recent years. This not only demands more careful attention on the diesel engine as a single component, but also on the total power plant design. Hybrid propulsion can influence the emissions in a positive way.
During combustion in internal combustion engines, the carbon in the hydrocarbon fuels react with oxygen to form carbon dioxide (CO_2). It is not directly poisonous to life, but it is the most common greenhouse gas. It is thought to be a great influence on the global climate change. CO_2 emissions are completely dependent on the amount and type of fuel that is burnt. A more efficient propulsion plant with less fuel consumption will directly result in lower CO_2 emissions.

SO_x is the collective term for sulphur oxides, which are generally SO_2 and SO_3 and contribute to acid rain. As with CO_2, the formation of SO_x is directly dependent amount of fuel burnt, but also on the fuel type. Residual fuels contain higher concentrations of sulphur than distillate fuels. Therefore a more efficient system will result in less emission of SO_x. There is however a slight disadvantage when using the main engine to drive a PTO. When this engine is running on HFO, it might be cheaper than auxiliary engines on MDO or gas, but HFO contains more sulphur and therefore the sulphur emissions in the exhaust gas will increase.

Nitric oxide (NO) and nitrogen dioxide (NO_2) and in lesser quantities N_2O are known as NO_x. At high temperatures the nitrogen in the air reacts with the oxygen present. NO is formed during the combustion process and later converted to NO_2 outside the engine. NO_x contributes to acid rain and ozone depletion. NO_x emissions are partly dependent on the fuel. It is assumed that any nitrogen contained in the fuel will convert to NO [Klein Woud & Stapersma, 2008]. The amount of nitrogen in fuel is however quite low. Thus, the formation is more dependent on the process of combustion such as temperature and the presence of O_2 (air excess ratio). Furthermore, the formation of NO_x takes a certain amount of time. The formation of NO_x decreases with increasing engine speed. A hybrid propulsion concept might influence this formation by shifting more power from lower speed main engines to higher speed auxiliary engines. In PTO mode this effect is reversed.

Soot and particulate matter (PM) consist of carbon particles that are formed after incomplete combustion. Formations of these pollutants occur more frequently when an engine is running in part load. A hybrid system aims at operating the diesel engines in such a way that they operate in their optimum range, so part load running will not occur for longer periods of time. This will have a positive effect on the emissions of PM.

To summarise all the benefits of hybrid propulsion:

- Because the engines are running in the range of their optimal fuel consumption, the total efficiency of the plant can be increased. This is particularly the case with varying operating modes.
- Hybrid propulsion gives high flexibility in selection of operating modes. Also, redundancy in propulsion is easily achieved.
- Because of the better efficiency and therefore less fuel consumption, fuel related emissions such as SO_x and CO_2 could be reduced. PM emissions occur less because the engines do not often run on part load. However, a PTO driven by the main engines might increase SO_x and NO_x emissions.
- According to the particular operation type the auxiliary engines or the main engine run less operation hours per year. Also, operation is on higher loads, which in turn contributes to less required maintenance.
- Noise and vibration levels are reduced when engines run on fixed relatively high speeds. This is particularly important on board passenger vessels.

To make use of the potential of the hybrid propulsion system, careful attention has to be paid to the parameters within the design to match the operational profile.

2.3 General design methodology

In Figure 2.4 an overview of the design workflow is shown. The steps will be explained in this paragraph and in chapter 3.

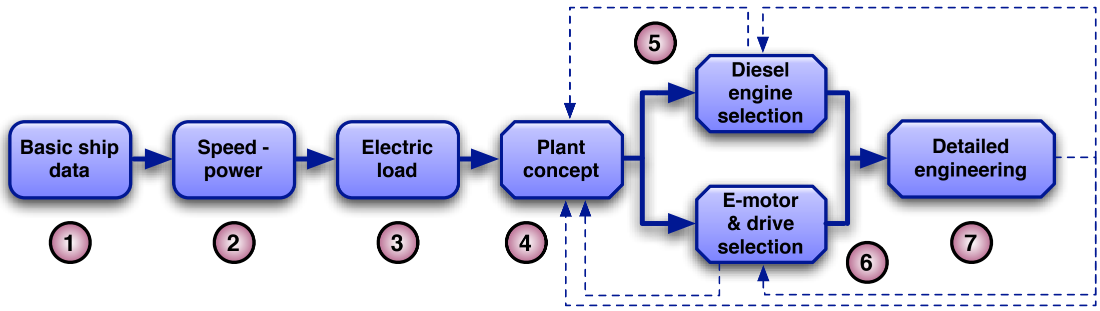

Figure 2.4: Design workflow of a hybrid propulsion concept

The type of vessel has an influence on the layout of the propulsion plant. For instance there could be class requirements for passenger vessels or ice going vessels. Qualitative requirements such as manoeuvrability might give the need for specific propulsion such as azimuth thrusters. This kind of basic data is the first step in designing a propulsion plant.

When basic data is known, the speed requirement is the first indication of the required installed power. Next to propulsion power, the power requirement for the auxiliary equipment must also be known before a plant concept can be designed. Step number 2 and 3 will be explained in more detail in paragraph 2.3.1 and 2.3.2 respectively. Together they form the operational profile of the vessel.

The plant concept design (step 4), together with the diesel engine selection (step 5) and electric machine selection (step 6) will be the main part of this thesis.

When the general outline of the propulsion plant is determined, several other aspects should be determined. These are for instance the specifics of the switchboard layout (frequency choice, voltage choice), calculation of short circuit currents and amount of harmonic distortion.
When these aspects are also determined, the detailed engineering (step 7) of all the auxiliary equipment and installation in the vessels engine room could be carried out.

The available diesel engines and electric motors might set limitations on the concept design. Also, calculations in the detailed engineering might influence the possibility of a certain concept. Therefore these last steps are more of an iterative procedure.

2.3.1 Resistance to propulsion power

A ship's propulsion plant has to overcome the ship's resistance. In general this consists of frictional resistance, residual resistance (such as wave resistance and eddy currents) and to a lesser extent the air resistance. Depending on hull form and speed the influence of the frictional resistance is usually the largest for slow vessels. For faster vessels, i.e. vessels with high Froude numbers, the wave resistance becomes far more influential.
For relatively low speeds, the total resistance is proportional to the square of ship speed. Because power is the product of speed and resistance, the power required for a ship is:

$$P_E \sim v_s^3 \tag{2.1}$$

Where P_E is the effective towing power of the ship, which is the power required to tow the ship through the water at a certain speed. For higher speeds, the curve might be steeper with v_s^4 or up to v_s^5. In figure 2.5 a qualitative ship resistance curve is given.
This power is of course not equal to the installed propulsion power. There are many losses that translate themselves into efficiencies. The propulsive efficiency η_D is defined as the ratio from effective towing power to the power delivered to the propeller shaft P_D:

$$\eta_D = \frac{P_E}{P_D} \tag{2.2}$$

It is this delivered power P_D that the propulsion plant must provide. This means that the effects in the propulsive efficiency are not included in the developed design in this thesis. A brief explanation of the relevant losses from P_E to the required installed power is given in appendix A.

From the concept design usually follows the P_D, which leaves the marine engineer to determine the required installed brake power P_B:

$$P_B = \frac{P_p}{\eta_{trm}} = \frac{P_p}{\eta_S \cdot \eta_{GB}} \tag{2.3}$$

Where η_{TRM} is the transmission efficiency. It contains both the shaft efficiency η_S and the gearbox efficiency η_{GB}. The shaft losses are caused by friction in the bearings and stern tubes and the gearbox losses by the friction between the teeth:

$$\eta_S = \frac{P_p}{P_S} \tag{2.4}$$

$$\eta_{GB} = \frac{P_S}{P_B}$$ [2.5]

P_B in [2.5] is the combined brake power that flows through the gearbox, i.e. the power from the connected main engines as well as the power from the electric machine.

In Figure 2.5 a qualitative power-speed curve is presented, where the distinction between the operating modes in a hybrid system can be seen. In normal (1) operation the diesel engine is running and driving both propeller and PTO. If the vessel needs a power boost to reach even higher speeds than the diesel engine could produce, the electric motor acts as a booster function (2). At slow speeds (3) the required propulsion power is significantly reduced and operation with a smaller electric motor would suffice. These operating modes correspond to Figure 2.1, Figure 2.2 and Figure 2.3.

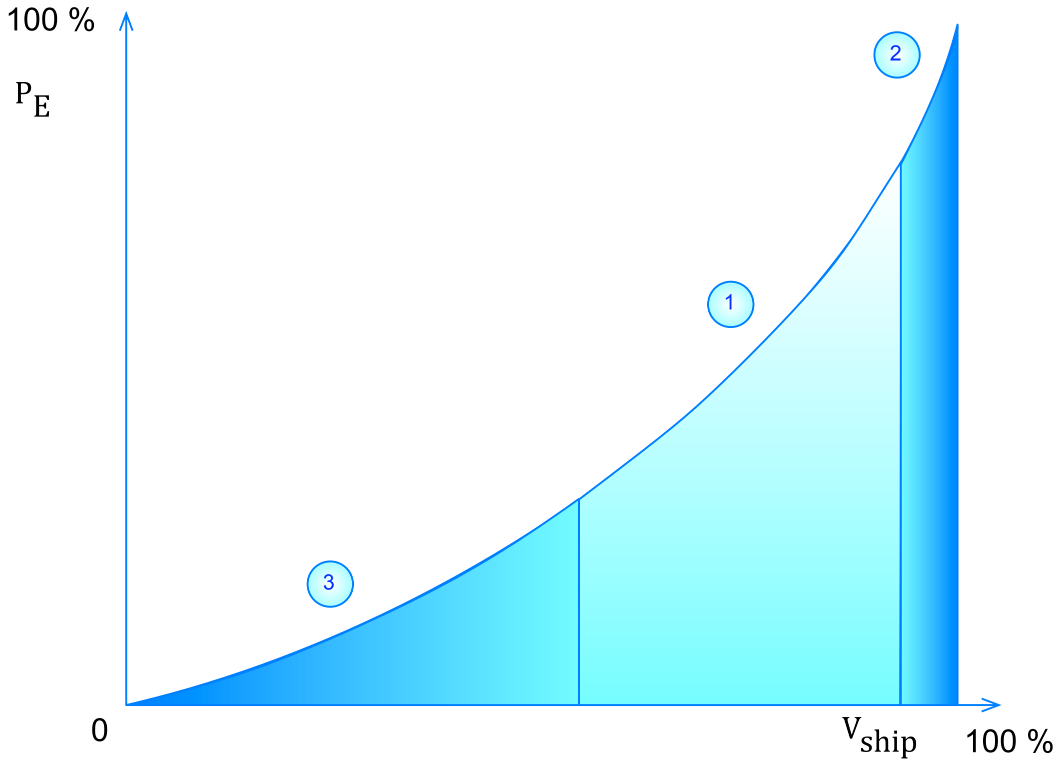

Figure 2.5: Speed-power curve with operation modes

2.3.2 Auxiliary power

To complete the power demand of a vessel the required auxiliary power P_{aux} should also be determined. In general the required auxiliary power is in electrical form.

Auxiliary power is dependent on the various operating conditions the vessels have. Every ship type has different requirements for auxiliary power. For instance, cruise vessels have large hotel loads such as air conditioning units and anchor handlers need electric power to operate the winches. Usually as a first estimate only the large consumers are specified; for other normal loads just fixed power demands are estimated. For a complete overview of the auxiliary power demand, usually an electrical load analysis is carried out, since most auxiliary power is electric power. In this analysis all electric consumers are listed with their estimated electrical power demand and simultaneous factor. This factor gives an indication whether the equipment is used continuously or intermittently. Then, the various operational conditions are described when the consumers are in operation. These combined give an overview of which consumers are in operation at which operational condition and the combined electrical load can then be estimated. These intermittent peaks are not included in the operational profile. Instead an average over the time interval is assumed. Large peaks should however be taken into consideration; the available electric power must be able to provide this without overloading.

A complete detailed electrical load analysis might be quite hard to carry out in the early design stage of a vessel. If there is no comparable ship type available, sometimes the use of empirical formulae are needed for an initial estimate. These have to be used with care, since these are often based on a certain ship type, having little resemblance to other ship types.
For an estimate [Deltamarin, 2009] was consulted. The estimates are dependent on the total installed main engine power P_B. The nominal load $P_{aux,0}$ for a general cargo vessel is:

$$P_{aux,0} = 120 + 0{,}8 \cdot P_B^{0{,}65} \qquad [2.6]$$

The auxiliary load when manoeuvring is:

$$P_{aux,man} = 1{,}3 \cdot P_{aux,0} \qquad [2.7]$$

And for main engine auxiliaries:

$$P_{aux,ME} = 0{,}07 \cdot P_B \qquad [2.8]$$

When the operational profile does not specify the auxiliary power demand, these formulae are used. The electrical load profile in combination with the propulsion profile gives the total operational profile.

2.3.3 Operational profile

The operational profile of a vessel gives the time intervals *n* where the ship requires a certain amount of power. The total amount of time can be different for different kind of applications, but for the purpose of this thesis it is one year. One year is also quite common in the first specifications of a vessel. Often the time intervals are presented in a percentage of the total time, but to be able to include this within the calculation the time intervals are presented in absolute figures; amount of hours per year *t* in [hr/yr]. The sum of these intervals is the total amount of hours per year, i.e. 8760.
Often the design is a first, so there is no actual data present on the operational profile. The profile is then based on previous experience of similar vessels or simply an estimate of the amount of time in which the ship will operate in every mode in the future. Four examples are given for four different kinds of vessels in chapter 5. For practical reasons *n* is limited to 10 operating modes in this thesis.
A distinction has to be made between propulsion power P_D and auxiliary power P_{aux} (Figure 2.6), as explained in the previous two paragraphs. The propulsion power is the power required by the propeller to reach a certain speed. The auxiliary power is required to operate the auxiliary machinery and hotel services.

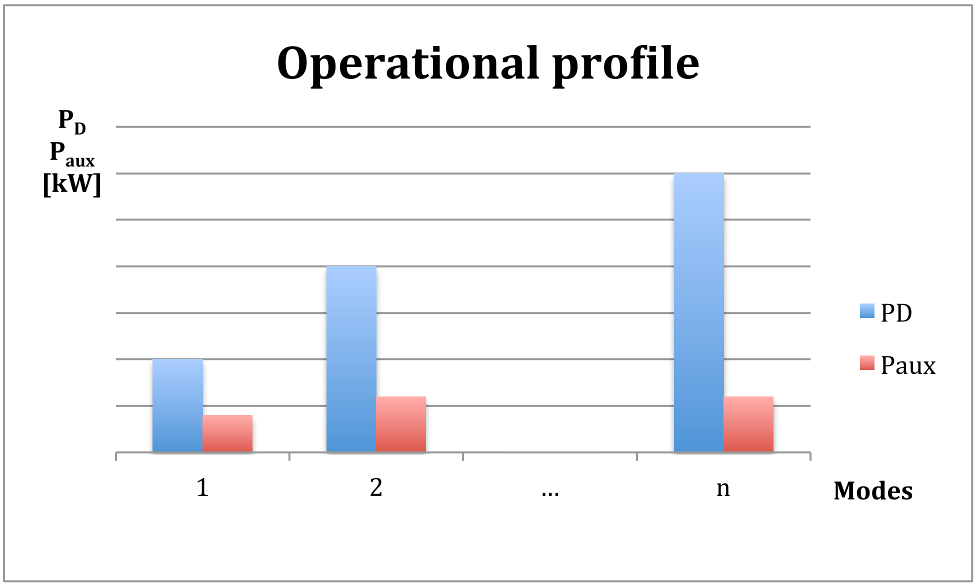

Figure 2.6: Example of operational profile

2.3.4 Scope of MAN Diesel & Turbo SE

Usually the ship basic data, power-speed estimation and the electrical load analysis are the input (provided by e.g. design offices or yards) for designing a propulsion power plant concept. The concept, engine and electric motor selection and basic engineering is then the responsibility of the marine engineers at, for instance, MAN Diesel & Turbo SE (MDT).

MDT is mainly known as a diesel engine manufacturer, but is becoming more involved as full propulsion system integrators. For complete propulsion packages MDT does the concept evaluation and system layout. MDT then uses sub suppliers for the detailed engineering, delivery and installation of the electrical equipment.

This thesis focuses on the power plant concept, engine selection and electric machine and drive selection.
The assignment is to find the best distribution between mechanical and electric power to fulfil the demand of propulsion power and auxiliary power P_{aux}. In particular, P_D is given as the power demand in the operational profile. This suggests that the propeller and hull effects that are included in η_D are already known. After all, the design process first determines P_E and then P_D. This also implies that a propeller has been selected before the engines are selected.
This is an important aspect in the design of the propulsion plant. With this method it is not possible to determine the optimal propeller matched to the given propulsion plant. It would actually be better to determine the propulsion plant first and then match the optimal propeller to this concept. Now the matching is done somewhat in reverse; given a hull shape, propeller and power demand, the optimal propulsion concept has to be determined.

2.4 Variable definition

There are many possibilities in the configuration of the propulsion plant. There can be variations on multiple engines on one shaft, multiple shafts, choice in number of diesel generators, installing an electric machine on the shaft that can produce power or take power and all combinations between these.
From a practical point of view not all combinations seem convenient. And from a mathematical point of view the amount of possibilities must be limited to a clear and finite set, in order for the algorithm to find a feasible and acceptable solution. The number of possibilities results in the number of degrees of freedom that are used in the design algorithm.

In order to validate the methodology with several ship examples, the developed algorithm must be implemented into a software tool. By MDT the preference is to use MS Excel. The reasons behind this are compatibility with other tools already in use, for example for life cycle costs analysis and the ability for easy exchange between other departments within MDT.
While the general design algorithm would be mathematically applicable for any number of degrees of freedom, implementation in Excel requires the number of variables to be limited to a finite set. A practical number will be defined in this paragraph.

There can be multiple main engines on one single shaft. It is assumed that two equal engines will both run on an equal load share. Whether engine 1 (ME1) or engine 2 (ME2) is running is irrelevant, since the available brake power is equal and the fuel consumption curve as well.
In the case where the two engines are not equal in size, the so-called father/son configuration, this assumption is not valid. In this case these engines will have a different brake power. In some modes it makes sense running one engine on an efficient setting, in some modes it is better to run the other one. For example, from 0 to 60% propeller power demand the smaller engine will run by itself, from 60 to 80% the larger engine will run by itself and above 80% both engines will run. The design algorithm in this study is expected to give these results. This separates the two main engine variables, making them independent of each other.
Although engine configurations with three engines on one shaft do exist, they are rare and require large gearboxes and are considered to be special cases. This methodology is limited to two main engines per shaft.
It must also be noted that in case of father/son configurations, the engine type (i.e. bore diameter) will normally be the same. Only the number of cylinders is different, because of spare parts and maintenance reasons. This means that the sfc-curve is the same for both engines. Technically speaking there might be a slight difference between the sfc-

curves, since engines with more cylinders tend to be slightly more efficient. The difference is minimal and therefore neglected in this study.

The number of installed diesel generator sets (DG) is variable. For redundancy reasons a minimum of two generators should always be installed. For practical and maintenance reasons it is common to install the same engine types and sizes for all generator sets. So in this study it is assumed that every diesel generator has the same brake power.
The required electric power from the diesel generators is in fact one single degree of freedom. Equal load sharing between the generators by the power management system will result in an equal load for all running generators. For example, for a certain electric power demand first two generators will run up to 90% load. As the power demand increases, a third generator will switch on, all running on 60%. This load sharing connects the different variables making them dependent on each other. So the required electric power from the diesel generators is one degree of freedom; the brake power of the generators will determine how many engines will run.

The electric machine (EM) can operate in two directions. In PTO mode the machine is operating as a generator, converting power generated by the main engine to electric power to the grid. In PTI mode the machine acts as a motor, taking electric power from the grid generated by the diesel generator sets and converting it to mechanical power to the propeller shaft. These are not two separable degrees of freedom, because it is one single machine. In other words, it is not possible to run with a PTO and a PTI at the same time. It is therefore taken as a single degree of freedom, where a positive value stands for generator mode (PTO) and a negative value for a motoring mode (PTI).

This brings the number of degrees of freedom to four: ($P_{ME1,n}$, $P_{ME2,n}$, $P_{DG,n}$ and $P_{EM,n}$). These stand for the actual power delivered by the components and are valid per operating mode *n*. This means the total number of degrees of freedom in the system is $4n$.
These determined four degrees of freedom are described for a single shaft vessel. Of course, multiple shafts are possible. Three shaft vessels do exist, but are quite rare and will be left out of the further analysis. Only single or two-shaft configurations will be considered.

With a two-shaft configuration many more operating possibilities arise. However, for normal operation a symmetrical loading on the two propellers is common. This reduces the need for operating the rudder(s) and improves manoeuvrability.
It is possible to install an electric motor on one shaft, but not on the other. Also, single shaft operation is possible, with the other shaft as a trailing shaft. While these options are very useful for redundancy (PTH mode), they are not considered to be standard operating modes. A ship is not designed to run in emergency mode for longer periods of

time. In other words, fuel consumption is never an issue in an emergency. However, exceptions to this rule do exist.
One other possibility is running with a so-called cross connection mode with the electric machines. In this case one shaft has a PTO mode. With this electrical power the electric machine of the other shaft runs in PTI mode, reducing the required power from the diesel engine. However, the increased electric losses do not weigh up to the better fuel consumption of the engines in most cases[2]. It is only used for redundancy or flexibility reasons. This mode is also not considered as a standard option in this analysis.

Reviewing the extra options with two shafts it makes more sense to make port and starboard shaft symmetrical. Making all variables independent of each other might result in implausible answers that are not feasible in practice. This symmetrical behaviour is only valid for the main engines and the electric machines. The diesel generators are separate from the shaft in the first place. Making the port and starboard parameters symmetrical does leave out the slow sailing option for one shaft (trailing shaft mode). But there is always the option for running the electric motor for that.

To summarise, the used degrees of freedom are summed in the table below and is also displayed in Figure 2.7:

	Single shaft	Two shaft	Comments
1	$P_{ME1,n}$	$P_{ME3,n} = P_{ME1,n}$	
2	$P_{ME2,n}$	$P_{ME4,n} = P_{ME2,n}$	
3	$P_{DG,n}$	$P_{DG,n} = P_{DG,n}$	
4	$P_{EM,n}$	$P_{EM,n} = P_{EM,n}$	> 0 is PTO; < 0 is PTI

Table 2.1: Number of degrees of freedom used

[2] *Improvement in fuel consumption for engines in the order of 2%, in the order of introduced losses 10%.*

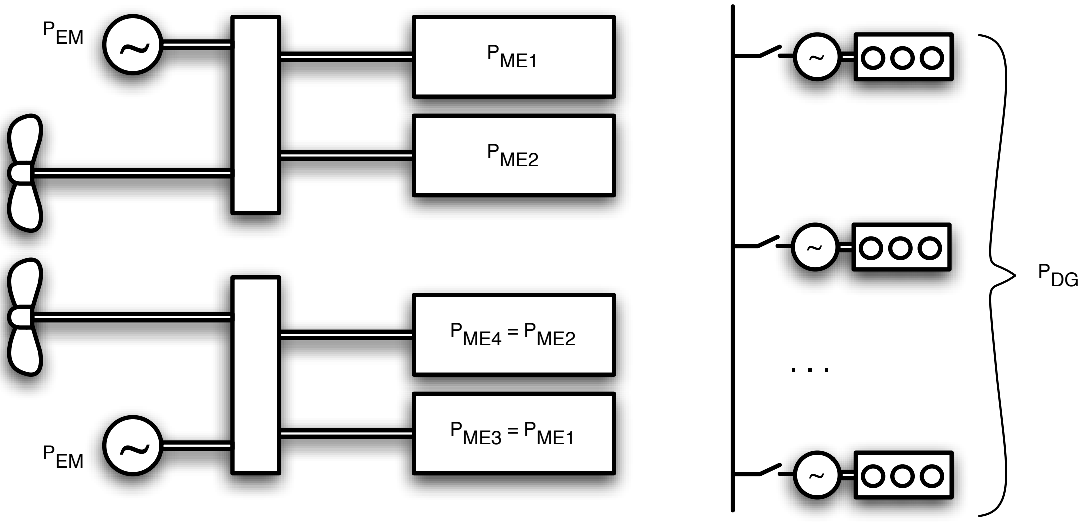

Figure 2.7: Degrees of freedom overview

The default setting for the type of gearbox is a single stage primary PTO gearbox. The gear ratios follow from the selected engine and the optimum propeller speed, if this is already known. It is not considered to be a parameter that influences the concept design in this study.

The same holds for using a frequency converter or not. Installing a frequency converter might have a large impact in the total efficiency including the propeller, but these effects are not taken into account since P_D is defined as power requirement.

2.5 Power demand

2.5.1 Energy flow in system

To give a complete overview of all the components that are required to transfer fuel oil into efficient and controllable propulsion, it can be useful to draw an energy flow diagram (EFD) as seen in [Klein Woud & Stapersma, 2008]. It shows which components transfer or convert energy so that it can be distributed to usable mechanical energy (Figure 2.8).

The energy source (ES) on board vessels is fuel oil, which can be MGO, MDO, HFO or gas, depending on the engine type. The diesel engine is used to convert the fuel to mechanical work (ES/M). Generally speaking this can also be any other prime mover. The gearbox (M/M) transfers this power to the propeller by reducing the engine speed to the propeller shaft speed that in turn provides the propulsion for the ship. Some shaft losses are also accounted for.

On the electric side, the diesel engines provide mechanical rotational energy that drives the generators (M/E) to generate electric power. Electric power is brought together and distributed in the main switchboard (MSB) to the consumers.

What makes the hybrid concept a hybrid, are the components between the gearbox and the switchboard. Electric power with a fixed frequency can be converted to any other frequency using a frequency converter (E/E). This gives the option to drive the electric machine (E/M or M/E) with a variable speed. The speed from the electric motor can be adapted through the gearbox to the propeller speed in either PTI boosting mode or as an alternative propulsion motor (APM). Vice versa, power delivered through the gearbox by the main engine can go to the electric machine in PTO mode. The generated electric power will be distributed through the switchboard to the consumers.

In the EFD the system boundary of this study is also shown. The fuel type is not included in the optimisation and neither is the propeller. This means that all diesel engines run on the same fuel type.

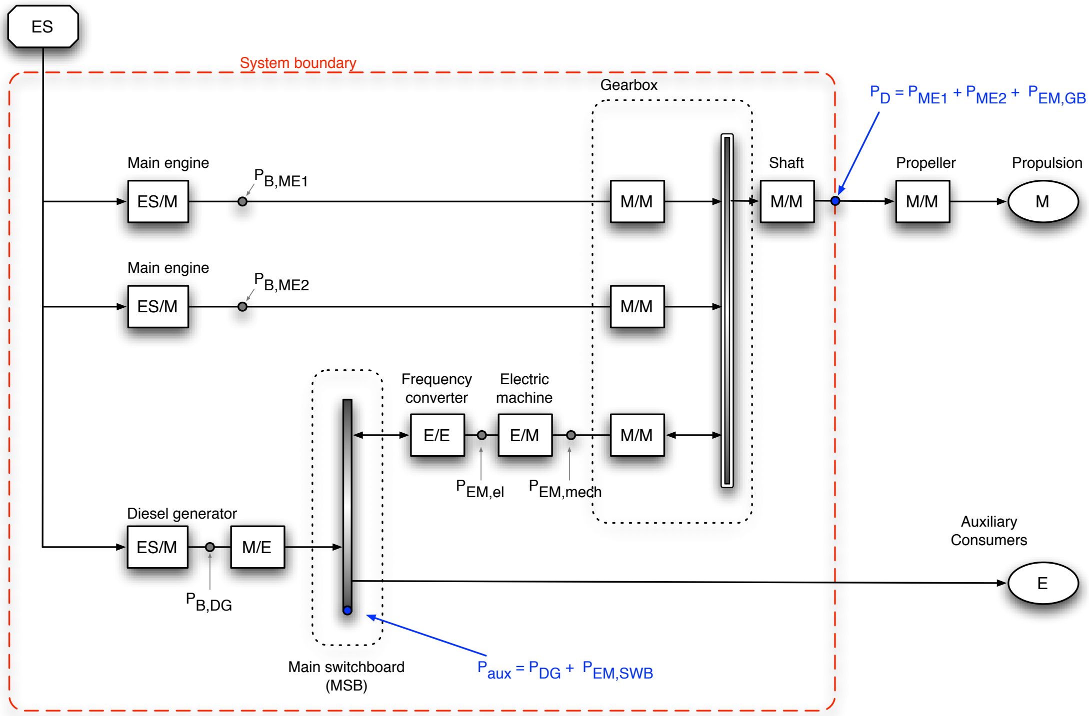

Figure 2.8: EFD of a general hybrid system layout

2.5.2 Power function

The purpose of the machinery installation is to meet the propulsion power P_D and auxiliary power P_{aux} demand as stated in the operational profile. All power has to be generated by the main engines and the diesel engines.

The propulsion power must be provided by the main engines and/or PTI motor. With the defined four variables this is:

$$P_{D,n} = P_{ME1,n} + P_{ME2,n} + P_{EM,GB,n} \tag{2.9}$$

The summation point is on the propeller shaft before the propeller (P_D). In this case, $P_{EM,n}$ will be named $P_{EM,GB,n}$. The electric power must be provided by the diesel generators or the PTO generator:

$$P_{aux,n} = P_{DG,n} + P_{EM,SWB,n} \tag{2.10}$$

In here the summation point is at the switchboard where $P_{EM,n}$ is now called $P_{EM,SWB,n}$. It is clear that the previously determined $P_{EM,n}$ can deliver power at the summation point of the switchboard or at the gearbox, even though it is essentially one variable apart from some losses. This means that the position of $P_{EM,n}$ changes depending its operation direction; in PTO mode it is in the switchboard and in PTI mode it is measured at the gearbox, where the following is valid:

$$P_{EM,n} = \begin{cases} P_{EM,SWB,n} & \text{for } P_{EM,n} > 0 \\ P_{EM,GB,n} & \text{for } P_{EM,n} < 0 \end{cases} \qquad [2.11]$$

Substituting [2.9] into [2.10] shows that all power demand is provided by the engines and that the electric machine is cancelled out:

$$\begin{aligned} &P_{EM,SWB,n} = P_{aux,n} - P_{DG,n} \\ &P_{D,n} = P_{ME1,n} + P_{ME2,n} - P_{aux,n} + P_{DG,n} \\ &P_{D,n} + P_{aux,n} = P_{ME1,n} + P_{ME2,n} + P_{DG,n} \end{aligned} \qquad [2.12]$$

This is true since all power must somehow be produced by the prime movers; the diesel engines. Writing [2.9] and [2.10] in vector form yields:

$$\begin{pmatrix} 1 & 1 & 0 & 1 \\ 0 & 0 & 1 & -1 \end{pmatrix} \cdot \underbrace{\begin{pmatrix} P_{ME1,n} \\ P_{ME2,n} \\ P_{DG,n} \\ P_{EM,n} \end{pmatrix}}_{X_n} = \begin{pmatrix} P_{D,n} \\ P_{aux,n} \end{pmatrix} \qquad [2.13]$$

Vector $\mathbf{X_n}$ is the vector that should be solved for every mode. Four variables and two equations give two degrees of freedom to find an optimum solution of the distribution of powers in the vector $\mathbf{X_n}$. When one main engine is used (i.e. $P_{ME2,n}$ is not included in $\mathbf{X_n}$), then there is just one degree of freedom. This is then $P_{EM,n}$.
Eq. [2.13] must always be valid for every operating mode. Solving and optimising [2.13] can be done in Excel, using the built-in tool Solver [Frontline Solvers, 2011]. This does add some complexity to the calculation methods. The standard solver has limitations that influence the possibilities of optimising the previous stated objective function. One important example is that the constraints cannot handle logic functions. This is the reason that $P_{ME1,n}$, $P_{ME2,n}$, $P_{DG,n}$ and $P_{EM,n}$ have to be determined to fulfil P_D and P_{aux} instead of the brake power of these components. This means that the conversion between P_D and P_{aux} to P_B cannot be included in the solver optimisation algorithm itself.

2.5.3 From effective to brake power

The degrees of freedom in the system correspond to the delivered power of the components, either at the propeller (P_D) or electrically at the switchboard (P_{aux}), see Figure 2.8.

The required brake power depends on the operating mode that influences the losses, so they are different for PTO or PTI mode. These losses translate themselves into efficiencies of the components. For every component all the relevant efficiencies can be grouped into one combined plant efficiency η_P. How these are constructed and what values are assigned to these parameters will be described in chapter 3.

Applying all these efficiencies in the Excel optimisation methods would require some logic statements in the setup. The solver cannot handle logic in the constraints [Frontline Solvers, 2008]. So the solver calculates with the delivered power of all decision variables. This is why the decision variables are the delivered power instead of the brake power in the optimisation algorithm. This is also true when the EM is not in operation.

With an electric machine the terminology for brake power might be misleading if the direction of the energy flow changes. Therefore, the term electric power $P_{EM,el}$ and mechanical power $P_{EM,mech}$ of the electric machine are introduced. Whether the machine is operating in generator mode or motor mode, these points remain the same.

If the EM is in PTO mode the power losses of the EM must also be delivered by the main engine. If the EM is in PTI mode, this does not influence the difference between P_B and P_{ME1} of the main engines:

$$P_{B,ME1,n} = \frac{P_{ME1,n}}{\eta_{P,ME1,n}} = \begin{cases} \left[P_{ME1,n} + \left(\dfrac{P_{EM,SWB,n}}{\eta_{FC} \cdot \eta_{EM,n}} - P_{EM,SWB,n} \right) \cdot \dfrac{1}{k} \right] \cdot \dfrac{1}{\eta_{trm,n}} & \text{for } P_{EM,n} \geq 0 \\ \dfrac{P_{ME1,n}}{\eta_{trm,n}} & \text{for } P_{EM,n} < 0 \end{cases} \quad [2.14]$$

Where k stands for the number of engines connected to the gearbox. One remark has to be made concerning the EM losses; they are always divided by two when k is 2. Even with a father/son configuration the losses are divided equally, so the share of the losses that the main engines have to provide is not proportional to engine brake power. The difference is however marginal. It is in the order of a few kW and this hardly influences the loading point and sfc of the engine.

These equations are also valid for $P_{B,ME2,n}$. Note that the $\eta_{trm,n}$ does not have to be the same in the different cases, since it also varies for the different modes.

For the diesel generator sets the efficiency of the electric machine η_{EM} and frequency converter efficiency η_{FC} have to be taken into account. When the EM is in PTO mode the only losses that have to be taken into account are the losses of the generator. When the

EM is in PTI mode the losses of the electric chain from the switchboard to the propeller have to be delivered by the diesel generators:

$$P_{B,DG,n} = \frac{P_{DG,n}}{\eta_{P,DG,n}} = \begin{cases} \dfrac{P_{DG,n}}{\eta_{gen}} & \text{for } P_{EM,n} \geq 0 \\ \left[P_{DG,n} - \left(\dfrac{P_{EM,GB,n}}{\eta_{trm,n} \cdot \eta_{EM,n} \cdot \eta_{FC}} - P_{EM,GB,n} \right) \right] \cdot \dfrac{1}{\eta_{gen}} & \text{for } P_{EM,n} < 0 \end{cases} \qquad [2.15]$$

In [2.15] for PTI mode, the DG delivers both power to the propulsion side and to the switchboard for auxiliary power. Note that $P_{EM,GB,n}$ is negative in this case.

3 Components

An overview of a hybrid propulsion plant is again presented as in Figure 2.1 but in some more detail including a frequency converter and supply transformer (Figure 3.1).
This chapter will describe these components in detail, to determine the application of the equipment and when to use it, their operating ranges and the possible limitations. The mentioned losses and efficiency in the calculation from effective power to brake power are explained in more detail.

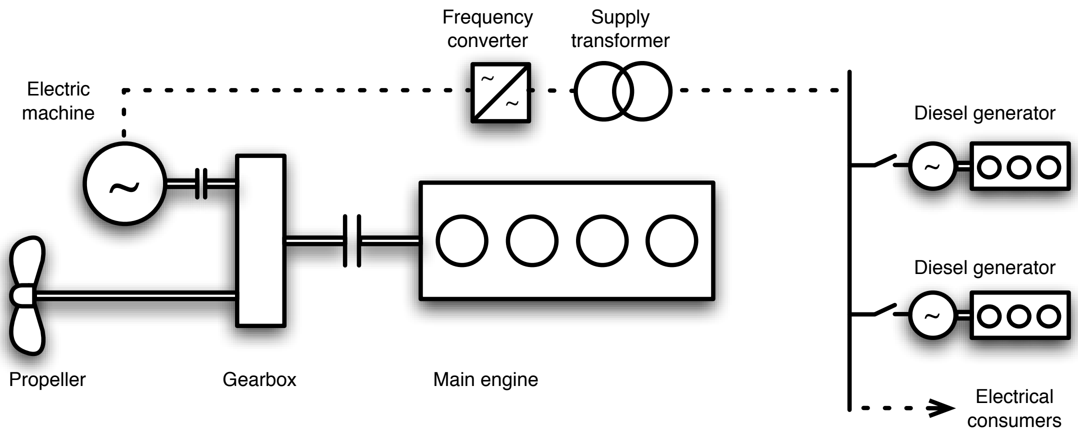

Figure 3.1: General hybrid system layout

3.1 Diesel engine

A diesel engine is a prime mover with the function to convert chemical energy stored in fuel to mechanical energy. Compared to other prime movers such as a gas turbine or steam turbine, the diesel engine has a high efficiency. Its low fuel consumption is the main reason it is widely used in marine applications.
A diesel engine is a reciprocating internal combustion engine, which converts chemical power to mechanical power at the output shaft in two steps: first the fuel will deliver thermal energy by means of combustion. This causes an expansion of the combustion gas in the cylinder, performing work on the piston. The work delivered to the pistons is translated into rotational work on the crankshaft, by transferring the reciprocating motion of the pistons to rotating motion on the crankshaft. This is the output shaft and can be connected to a propeller (for medium speed applications usually through a reduction gearbox) or a generator.
The diesel cycle consists of four processes: compression, expansion, gas exhaust and air inlet, which are also the four strokes of the piston. A 2-stroke engine also has these processes, but combines the gas exhaust and air inlet at the end of the expansion stroke

and start of the compression stroke, resulting in just two strokes. This thesis limits the wide range of diesel engines to 4-stroke medium speed engines (500 to 1000 rpm).

The type of engines, number and their brake power largely determines the initial purchase costs of the power plant. Because an engine runs at an optimum efficiency around 85% load, but moreover to prevent overloading in off-design conditions, an engine margin (EM) is chosen. See Figure 3.2.
This margin is the ratio of the continuous service rating (CSR) and the maximum continuous rating (MCR):

$$EM = \frac{CSR}{MCR} \tag{3.1}$$

Next to the EM, a sea margin (SM) is also specified:

$$SM = \frac{P_{B,service}}{P_{B,trial}} \tag{3.2}$$

This is a margin containing the differences in propulsion power between the actual service of the vessel and the trial conditions. These change because of fouling of the hull, change in displacement, sea state or water depth. These factors technically influence the towing power P_E, but as the propulsion efficiency η_D does not actually change during different service conditions, brake engine power can also be assumed [Klein Woud & Stapersma, 2008].

3.1.1 Power

The effective work on the piston is the product of the pressure during the expansion stroke and the stroke volume. The effective power or brake power is related to the effective work W_e on the piston. For the complete cycle:

$$W_e = p_{me} \cdot V_S = \frac{1}{i} \int_0^{2\pi k} M_{flange} \cdot d\alpha \tag{3.3}$$

And:

$$P_B = W_e \cdot \frac{n \cdot i}{k} \tag{3.4}$$

Where p_{me} is the mean effective pressure, V_s is the stroke volume, n is the engine speed in s^{-1} and i the number of cylinders. The factor $1/k$ indicates a 2-stroke engine (k = 1) or a 4-stroke engine (k = 2). M_{flange} is the torque delivered at the output flange and α is the crank angle.

Another definition of the brake power is:

$$P_B = M_B \cdot \omega = M_B \cdot 2\pi \cdot n \qquad [3.5]$$

Where M_B is the torque on the crankshaft averaged over time and ω is the rotational speed in rad/s.

An overview of the engine powers in the MDT catalogue is given in [MAN, 2011]. The engine type designation is also clarified here. The range is between 1290 kW and 21600 kW for single propulsion engines and between 500 kW and 11200 kW for generator sets. If higher powers are required in the hybrid propulsion plant, of course more engines could be installed.
For certain engines a distinction between generator load and propeller load is made. Maximum output power is set at 100% for driving a propeller. For driving a generator it can be at 110%, but only for a short time should a frequency drop occur.[3]
Other engines have a so-called navy profile. This allows them for example to run at 110% for 1 out of 6 hours according to the DNV High Speed Vessel Rules (HSVR) [MAN, various].
When designing the propulsion plant, a maximum of 100% excluding the engine margin should be maintained.

Although the engine is theoretically able to run at very low loads, the lower limit is usually set around 25%. At lower loads the engine runs less smoothly and fouling due to incomplete combustion becomes an issue. Operating at low loads should only be allowed for limited periods of time.

3.1.2 Speed

The nominal operating speed for each engine is also presented in [MAN, 2011]. The range lies between 400 and 1000 rpm and is an important factor concerning the construction of the reduction gearbox or the generator. The nominal speed is at 100% load. The speed can go up to 103% at idling operation. To overcome frequency drops in the net, engines driving a generator can go up to 108% for a limited amount of time.
The minimum speed is around 60% of nominal speed. Below this there might be loss of compression and failure of ignition.
With the speed limit together with the power limit and the turbocharger limit, the generalised operating envelope can be drawn as in Figure 3.2. Note that this envelope does not have the 60% speed limit.

[3] *Short is about ½ hour in 6 hours, according to rules in DIM ISO 8528-1*

In this figure some examples of the combinator curve are also shown. This is the operating curve of the engine to drive the propeller and combines both pitch control and speed control, so the whole area beneath the engine limits can be used. Therefore, only a controllable pitch propeller (CPP) can be used.
As a default, a combinator curve is used. Above 80% load the speed remains constant and the thrust can be increased by increasing the pitch. Below 80% and between 60% n* and 100% n* the power follows the propeller law:

$$n^* = \sqrt[3]{P^*} \qquad [3.6]$$

Where the asterisk stands for normalised speed and normalised power.

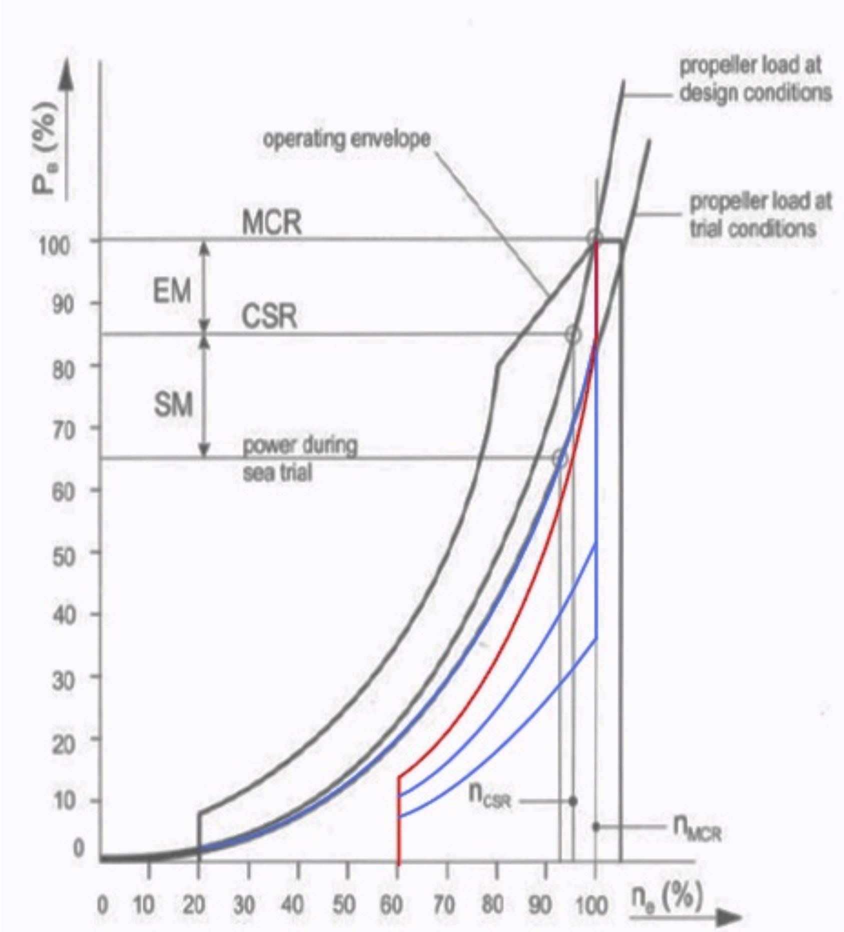

Figure 3.2: Engine operating envelope including example combinator curve. Edited from [Klein Woud & Stapersma, 2008]

3.1.3 Efficiency

The efficiency of a diesel engine depends on thermal losses in the exhaust gas and cooling water, some combustion losses and friction losses in the mechanical parts of the engine. The engine efficiency is the useful power that the engine delivers (P_B) divided by the heat input $\dot{Q}_f$ by the fuel:

$$\eta_e = \frac{P_B}{\dot{Q}_f} \qquad [3.7]$$

$\dot{Q}_f$ is related to how much fuel is burnt, but also how much energy is stored in the fuel. It can be approximated by:

$$\dot{Q}_f \cong \dot{m}_f \cdot h^L \qquad [3.8]$$

Where $\dot{m}_f$ is mass of fuel per second and h^L is the lower heat value of the fuel[4]. This lower heat value varies for different fuel types. For MDO as specified in the engine operating manuals according to ISO standards:

h^L = 42.700 *kJ/kg*

Engine manufactures such as MDT specify the performance often in terms of specific fuel consumption (sfc, or sfoc) in g/kWh. A low sfc corresponds to a higher efficiency. By definition it is calculated as:

$$sfc = \frac{\dot{m}_f}{P_B} \qquad [3.9]$$

Combining equations [3.7], [3.8] and [3.9] the engine efficiency can be expressed as sfc:

$$\eta_e = \frac{P_B}{\dot{Q}_f} = \frac{1}{h^L \cdot sfc} \qquad [3.10]$$

With the sfc usually specified in g/kWh and h^L in kJ/kg, this becomes:

$$\eta_e = \frac{3.600.000}{h^L \cdot sfc}$$

MDT Diesel engines are optimised at around 85% load. For 2-stroke engines this point can be shifted by regulating the airflow through the turbocharger settings. For 4-stroke medium speed engines used here all have their lowest sfc at 85% load.

[4] *This heat input is valid for the reference temperature and assumes all combustion products enter and leave the cylinder at equal temperatures.*

At part load the efficiency goes down. This can among others be explained by the fact that the mechanical friction losses are relatively higher at low loads.
The heat loss efficiency will go up at part load and together with the mechanical efficiency this gives the shape of the efficiency with the optimum at around 85% load.

The specific fuel consumption is different for every engine. For all engines in the 4-stroke MDT portfolio the sfc data is presented for a certain number of per cent load points. An example for the given sfc data is shown in Figure 3.3 for the L48/60 CR engine. A clear hump can be seen at 75%. To comply with the Tier II regulations by IMO concerning pollutant emissions, an adjustment on the camshaft at 75% has been made. This is to operate the engine with so-called Miller timing, where a different timing of the in- and outlet valves results in lower NO_x formation.
A complete overview of the used fuel consumption data is given in appendix B. The data is based on measurements on the engine test beds. For a propulsion engine the dependency between power and speed is based on the propeller law. This does give some deviation when a combinator curve is chosen for operation. For generator sets it is based on a fixed rotational speed *n*.
The sfc is converted to fuel flow $\dot{m}_f$ in by multiplying the sfc with the power. Since the sfc data is the same for an engine with different cylinder configurations, the fuel consumption is determined per cylinder. This can then be used for multiple configurations. The fuel consumption per cylinder in kg/h is:

$$m_{f,cyl} = \frac{sfc \cdot P^* \cdot P_{B,cyl}}{1000} \qquad [3.11]$$

The fraction of power P* stands for the actual power divided by the nominal brake power. This is equal to the previously mentioned load (P* = 1 is equal to 100% load):

$$P^* = \frac{P_{B,n}}{P_B} = \frac{P_{B,cyl,n}}{P_{B,cyl}} \qquad [3.12]$$

In Figure 3.4 the $\dot{m}_f$ is plotted against the cylinder power for the L48/60 CR engine. Fitting a curve to these data points can be done using polynomials or power functions. To fit the curve more precisely to the actual data, would require some high order polynomials. These high orders might not even have any real physical meaning. But more important, quick calculation with the sfc is required for every calculation step. Having a very complex function inside the objective function will lead to longer calculation times and is more prone to errors.
Therefore it is chosen to fit the existing data with two second order polynomials: one for the range between 0% to 75% load and one for the range between 75% to 100% load. The fuel flow per cylinder with a linear fit is then approximated by:

$$\dot{m}_{f,cyl} = a \cdot P^* + b \qquad [3.13]$$

And with the second-order polynomials:

$$\dot{m}_{f,cyl} = \begin{cases} a_1 \cdot \left(P^*\right)^2 + b_1 \cdot P^* + c_1 & \text{for } 0 < P^* < 0{,}75 \\ a_2 \cdot \left(P^*\right)^2 + b_2 \cdot P^* + c_2 & \text{for } 0{,}75 < P^* < 1 \end{cases} \qquad [3.14]$$

For further fuel consumption calculations the equations are based on the second order polynomials, but are similar to the linear equations. To determine the total fuel consumption of the main engines, this $\dot{m}_f$ must be multiplied by the number of cylinders i:

$$\dot{m}_{f,ME1} = \left(a_{ME1} \cdot \left(P^*_{ME1}\right)^2 + b_{ME1} \cdot P^*_{ME1} + c_{ME1} \right) \cdot i_{ME1} \qquad [3.15]$$

This is of course also valid for ME2. For the total fuel consumption of the diesel generators, next to the number of cylinders, [3.14] must also be multiplied by the number of running diesel generators N_{DG}:

$$\dot{m}_{f,DG} = \left(a_{DG} \cdot \left(P^*_{DG}\right)^2 + b_{DG} \cdot P^*_{DG} + c_{DG} \right) \cdot i_{DG} \cdot N_{DG} \qquad [3.16]$$

Since the absolute value of $\dot{m}_f$ is relatively high compared to the sfc values, the curve seems very linear. However, when one takes a more detailed look this is not the case. The fuel consumption differs by several kg/h per cylinder.
Special interest must be paid at 0% loading. Here there are two possible values, one for idling speed (around 8 kg/h for this engine) and one where the engine is switched off (0 kg/h). Since the design of the propulsion plant is done in a static way instead of dynamic, an idling engine should be avoided for continuous use. Therefore the fuel consumption is set to zero at zero load. This means this function becomes discontinuous around zero, which can be of influence in the optimisation algorithm.

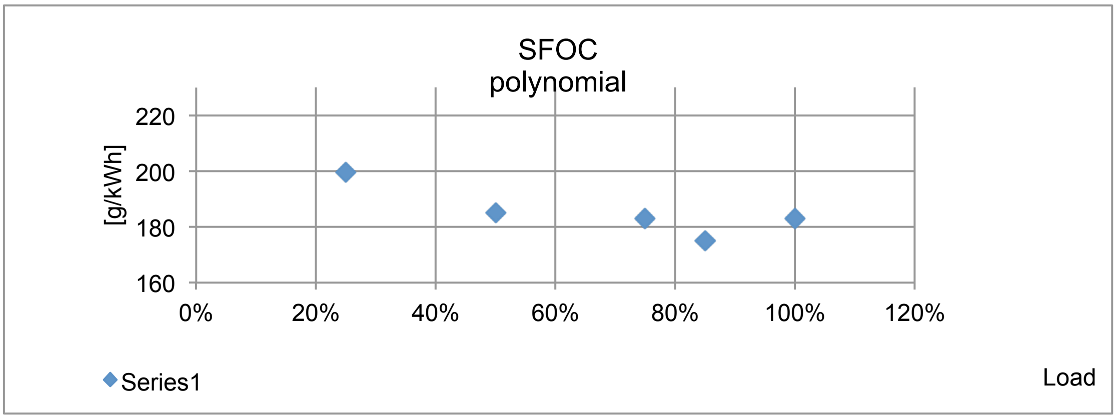

Figure 3.3: Sfc data for the L48/60 CR engine

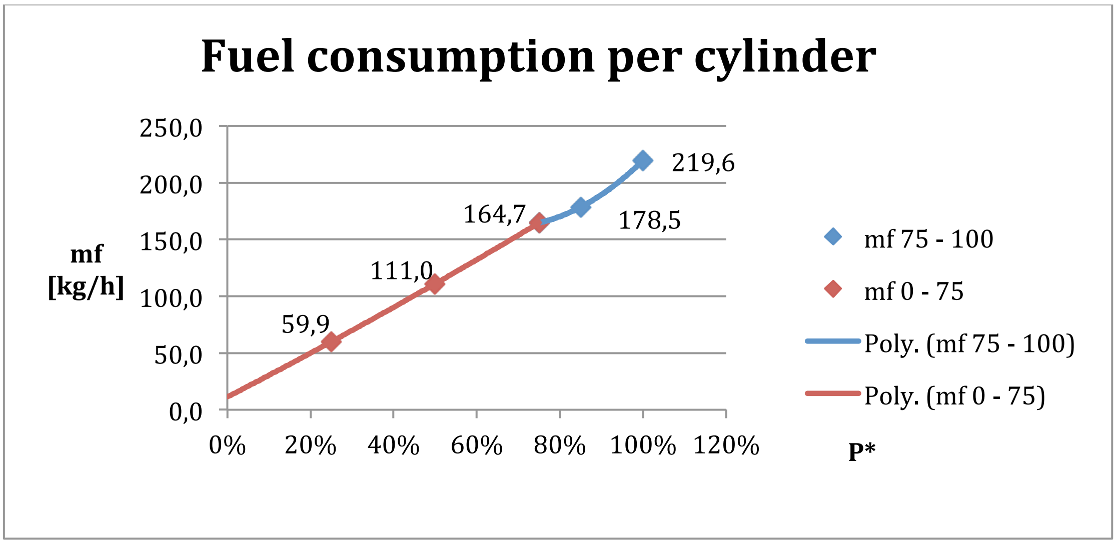

Figure 3.4: Fitted curve to fuel consumption of L48/60 CR engine

3.2 Electric machine

Next to the diesel engine selection, the size and type of electric motor is determined. This is either the amount of power supplied by the PTO, or the amount of power in PTI mode (Figure 2.5). This also influences the required installed engine power. When the amount of power required or supplied by the motor/generator is known, the required electrical power supplied by the diesel generator sets can be determined.
Compared to the prime mover that is restricted to 4-stroke medium speed diesel engines in the MDT portfolio in this study, there is a large selection of electric machines. To understand the difference in their applications and limitations, their working principle should be explained a little more in detail.

3.2.1 Types

An electric motor translates electric power into mechanical power. The outside stator is supplied with electricity to produce a magnetic field. A current also supplies the inner rotor, so that the rotating magnetic field generates torque on the output shaft. The way the stator and rotor are being supplied distinguishes the type of motor.

Electric machines all work on the same principle; creating a Lorentz force F_L on the rotor. When a current carrying conductor is placed in a magnetic field, a force will act on this conductor. The size of this force depends on the current I flowing through the conductor and the magnetic flux density Φ. The conductors on both sides of the rotor carry current in the opposite direction. The F_L then also acts in the opposite direction, creating the output torque:

$$M = K_M \cdot \Phi \cdot I \tag{3.17}$$

K_M is a constant for the motor depending on size, number of windings (pole pairs), and flux density variations in the motor. When high torque is required, for example for a motor on the propeller shaft, either the currents have to increase or the number of pole pairs, since this influences K_M. Either way the machine needs a larger construction and more copper. This increases the capital costs. The flux density Φ is dependent on the material and remains roughly the same for different motor types.

When a conductor (the rotor in this case) is moving in a magnetic field, an induction voltage E is generated. This so called electromotive force (EMF) gives the possibility for the electric machine to run in generator mode. The rotor is then connected to the output shaft of a diesel engine. The induction voltage is given by:

$$E = K_E \cdot \Phi \cdot n \qquad [3.18]$$

Where K_E is a constant for a certain machine depending on size, number of windings and flux density variations in the coil and n is the rotational speed of the coil in the magnetic field (rotor speed).
So when an electric machine creates a torque at the output shaft from a provided current, it is an electric motor. When a voltage is induced because of the provided rotation of the rotor, it is a generator.

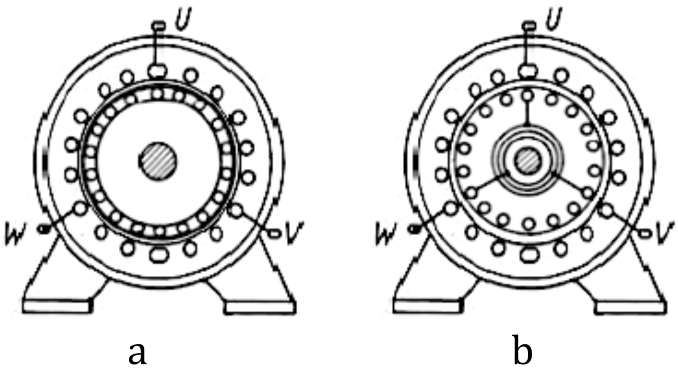

Figure 3.5: Basic construction of an induction motor (a) and synchronous motor (b). Edited from [Fischer, 2006]

Two main types of electric machines are commonly seen on board ships. In an induction motor the stator field windings are powered by an alternating current (AC). This creates a rotating magnetic field. The rotor windings are short-circuited by end rings. The rotor winding currents are then induced because of the relative speed of the rotor in the rotating magnetic field. If the rotor speed equals the speed of the magnetic field, no induction will take place. Therefore the output shaft always rotates a little slower than the synchronous speed of the grid. This is why the induction motor is also referred to as an asynchronous machine.
In a synchronous motor the stator is powered by AC, just like the induction motor. The difference is that the rotor windings are not short-circuited but excited by DC, through slip rings on the shaft. Modern electric machines have a small exciter generator that is attached to the motor. The voltage of this generator can be regulated and through a diode bridge any required DC can be produced. This construction omits the need for slip rings.

When the induction motor runs with a speed lower than the synchronous speed, torque is supplied to the rotor. When the motor runs with a higher speed than the synchronous speed, the slip becomes negative and the motor acts as a generator. See figure 3.6 for the four-quadrant operation of an induction machine and synchronous machine. In this figure a typical drive curve is given. Motoring operation is in the second quadrant; generator operation is in the fourth quadrant.

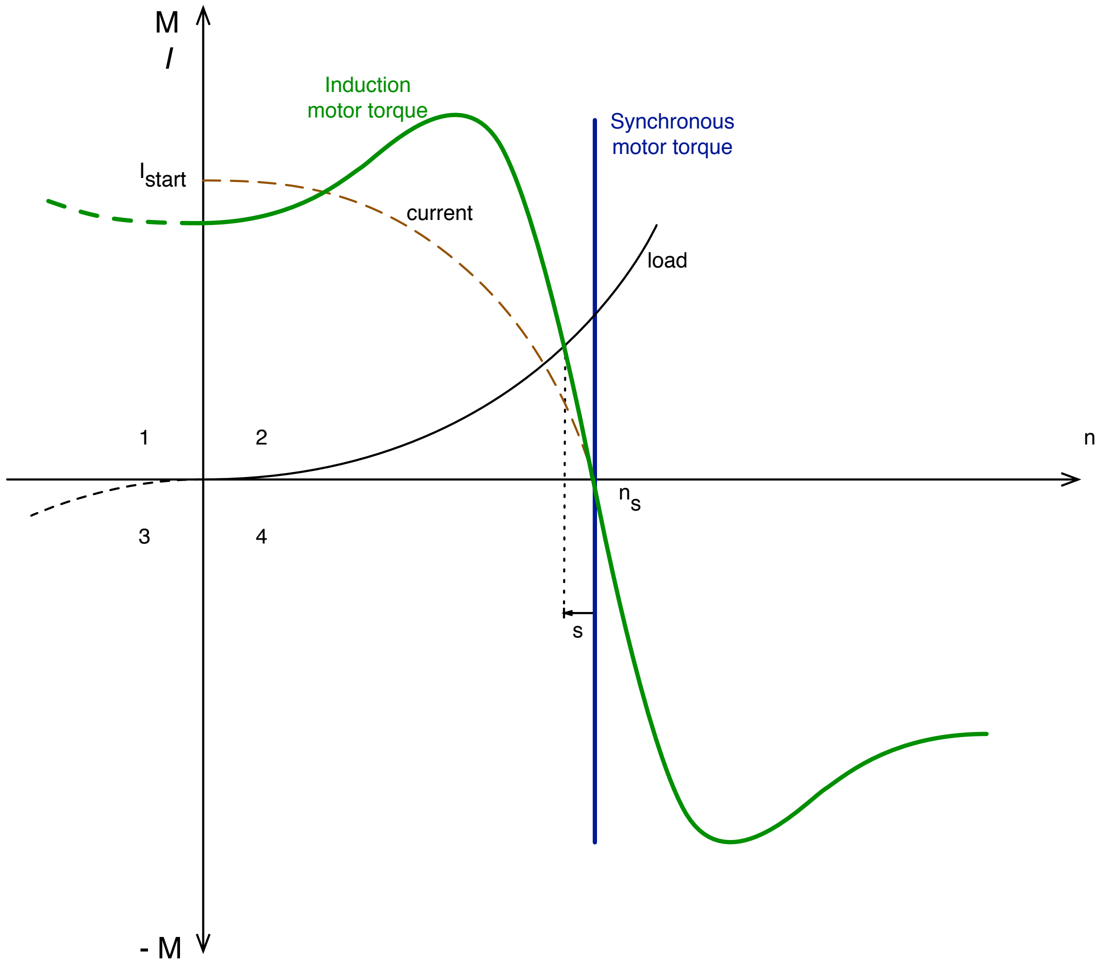

Figure 3.6: 4-quadrant operation of electric machine

Note that rotation in the opposite direction (-*n*) does not occur, so this is not shown in the graph. The slip at synchronous speed n_s is zero and maximum at n = 0; at start-up. A typical load curve is also presented. For simplicity it is drawn as a power curve through zero. However, even at zero pitch the propeller has to overcome the zero pitch losses. So the curve lies considerably higher if shaft speed is kept constant. It does not actually start horizontally either; also a certain moment is needed to overcome the friction to start turning from zero speed. This breakaway torque causes the curve to start a little higher than zero, move downward slightly and from there on continue as a power curve.

3.2.2 Power

The power of electric machines is the product of the line voltage U and the line current I. For a 3-phase system there are three lines, most often connected in a delta configuration. With the line values of U and I the total active power of an electric machine is:

$$P = \sqrt{3} \cdot U \cdot I \cdot \cos\varphi \tag{3.19}$$

The $cos\varphi$ is called the power factor and is a result of the phase shift φ between the voltage and current. The larger the phase angle φ the smaller the power factor.
For synchronous machines at zero torque the poles on the rotor are exactly opposite of the stator poles. If torque increases, the rotor magnetic field starts lagging behind the stator magnetic field. The DC to the rotor needs to be supplied by an excitation unit that can control the leading or lagging power factor. The induction motor on the other hand always has a lagging power factor because of the induced voltage.

Induction motors are found up to about 25 MW and synchronous motors up to about 50 MW (ABB, Converteam). The synchronous motor has a slightly better efficiency, but is more expensive. A trade-off between these factors generally results in induction motors being chosen for a power range below 7 MW (ABB, 2006). This limit was also set by the use of a pulse width modulation (PWM) converter, but as the technology for converters has advanced over the years this does not always apply anymore. Frequency converters are discussed more in detail in chapter 3.4.
The power range of the investigated vessels in chapter 5 means that an induction motor will be chosen as default.

It must be mentioned that a power specification for a particular electric motor or generator is always given as the delivered power of the machine. This means that a motor is specified in terms of its mechanical power $P_{EM,mech}$ and a generator as its electrical power $P_{EM,el}$.

3.2.3 Speed

The output speed of the electric motor is determined by the frequency f of the AC supply and the number of poles p:

$$n_s = \frac{2 \cdot 60 \cdot f}{p} \tag{3.20}$$

The factor 60 is because n_s is in rpm (min^{-1}) and f in Hz (s^{-1}). This gives the synchronous speeds shown in Table 3.1. Asynchronous speeds are about 1 – 4 % lower because of the slip s.

	2	4	6	8	10	12	[poles]
50 Hz	3000	1500	1000	750	600	500	[rpm]
60 Hz	3600	1800	1200	900	720	600	[rpm]

Table 3.1: Synchronous speeds of electric machines

For generators this means that the input speed and number of poles determines the frequency. Omitting a frequency converter sets requirements for the driving diesel engine, especially in case of a PTO.
Very large synchronous motors can have many more poles to obtain lower speeds down to about 80 rpm. They can be used to drive a propeller without the use of a reduction gearbox.
With a frequency converter the output AC frequency can be varied so all rotational speeds can be achieved.

3.2.4 Efficiency

A schematic overview of which losses occur in an electric machine is given in

Figure 3.7. P_{in} is the power that is supplied to the motor, so $P_{EM,el}$. P_i is the inner power that is transmitted between the stator and the rotor. P_{out} is the mechanical output power delivered by the electric motor $P_{EM,mech}$. P_{cu1} and P_{cu2} are the copper losses at the stator and at the rotor side. P_{Fe} are the iron losses that consist of hysteresis losses (when changing the direction of the iron crystals due to magnetisation) and eddy current losses in the material. [Fischer, 2006] describes a method to distinguish the different losses.
The copper losses increase with the load, or more specifically, with the current:

$$P_{cu,1} = m \cdot I_1^2 \cdot R_1 \quad [3.21]$$

$$P_{cu,2} = m \cdot I_2^2 \cdot R_2 \quad [3.22]$$

R is the resistance in the windings and is a constant for a given size and material and *m* is the number of windings (3 in most cases). So the copper losses increase with increasing current i.e. increasing torque.

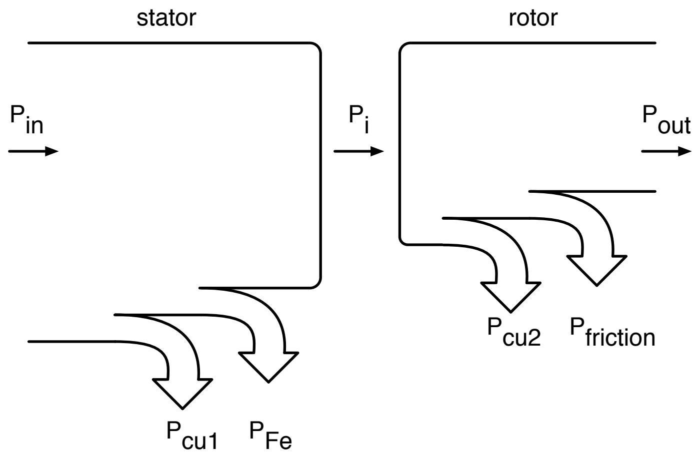

Figure 3.7: Power losses in electric machine

The iron losses are also load dependent in theory, although for practical reasons it is often assumed they are more or less constant for a given voltage and material [Fischer, 2006], [ABB, 2010]:

$$P_{Fe} = m \cdot \frac{U_1^2}{R_{Fe}} \qquad [3.23]$$

Friction losses $P_{friction}$ are combined extra losses that consist of bearing losses, windage losses, air resistance on the rotor and parasitic power for internal cooling fans. Larger motors are also water-cooled, which also needs circulation and power to operate the pump. They increase with increasing speed and increasing power; they are more speed dependent than torque dependent. In total the power losses are:

$$P_{loss,EM} = P_{cu,1} \cdot P_{cu,2} \cdot P_{Fe} \cdot P_{friction} \qquad [3.24]$$

This leads to the definition of the efficiency of the electric machine:

$$\eta_{motor} = \frac{P_{EM,el} - P_{loss,EM}}{P_{EM,el}} \qquad [3.25]$$

$$\eta_{gen} = \frac{P_{EM,mech} - P_{loss,EM}}{P_{EM,mech}} \qquad [3.26]$$

The η_{gen} for the diesel generator sets is known and also listed in [MAN, 2011]. The losses are the same, since they depend on internal resistance, currents and voltage that are the same in either direction. This means that depending on the power direction, the efficiencies change slightly. However, the difference is very small and it suffices to state that η_{motor} and η_{gen} are equal for one machine.

A typical power-efficiency curve is presented in Figure 3.8 based on calculated values of a synchronous machine from ABB [ABB, 2010]. It has 1325 kVA with a power factor of 0,8.
Since P_{Fe} is assumed constant, the losses increase relatively at low loads. This explains the downward efficiency curve at low loads. The slight downward curve at higher loads might be due to less effective cooling.

When an electric machine is selected, it will most likely operate in the higher load region. However, very low load can occur and the efficiency belonging to this part load can be of influence to the design. Therefore, the part load efficiency must be made variable. To accurately model the part load losses, parameters like *m* and *R* must be known. Since the type and size of the electric machine is not yet known at this stage, a part load efficiency must be assumed.
In [Stapersma, 1994] a general curve for non-linear part load behaviour is formulated. It assumes a parabolic curve through the nominal point. For the electric machine the fit between P^*_{in} and P^*_{out} is approximated by [3.27], where the asterisk again stands for normalised power.

$$P^*_{in} = 1 - a\left(1 - P^*_{out}\right) + b\left(1 - P^*_{out}\right)^2 \qquad [3.27]$$

And since efficiency is by definition the ratio between power output and power input, this gives a part load efficiency of:

$$\eta^*_{EM} = \frac{P^*_{out}}{P^*_{in}} = \frac{P^*_{out}}{1 - a\left(1 - P^*_{out}\right) + b\left(1 - P^*_{out}\right)^2}$$

$$\eta_{EM} = \eta^*_{EM} \cdot \eta_{EM,0} = \frac{P^*_{out}}{1 - a\left(1 - P^*_{out}\right) + b\left(1 - P^*_{out}\right)^2} \cdot \eta_{EM,0} \qquad [3.28]$$

Where:

$$\eta^* = \frac{\eta}{\eta_0} \qquad [3.29]$$

The coefficients a and b are chosen such that the efficiency drops to about 0,93 at 50% load and around 0,89 at 25% load. This means that a = 1 and b = 0,01. This graph is given in
Figure 3.9 and is consistent with the measured data from ABB in Figure 3.8. Data for the lowest loads was unfortunately not available [ABB, 2010].
A synchronous motor has a slightly higher nominal efficiency compared to an induction motor. This can be explained by the fact that the external excitation for the rotor can be controlled very precisely. In this thesis the nominal efficiency $\eta_{EM,0}$ is assumed to be 96% for induction machines and 97% for synchronous machines.

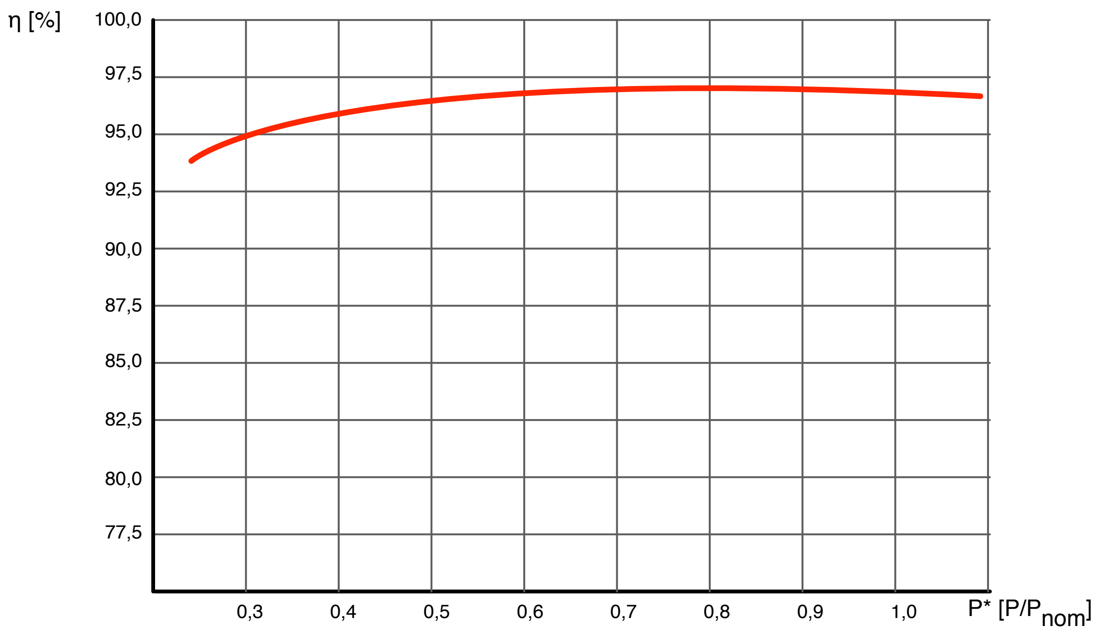

Figure 3.8:Efficiency curve for a 1325 kVA synchronous motor, p.f. 0,8. Based on [ABB, 2010]

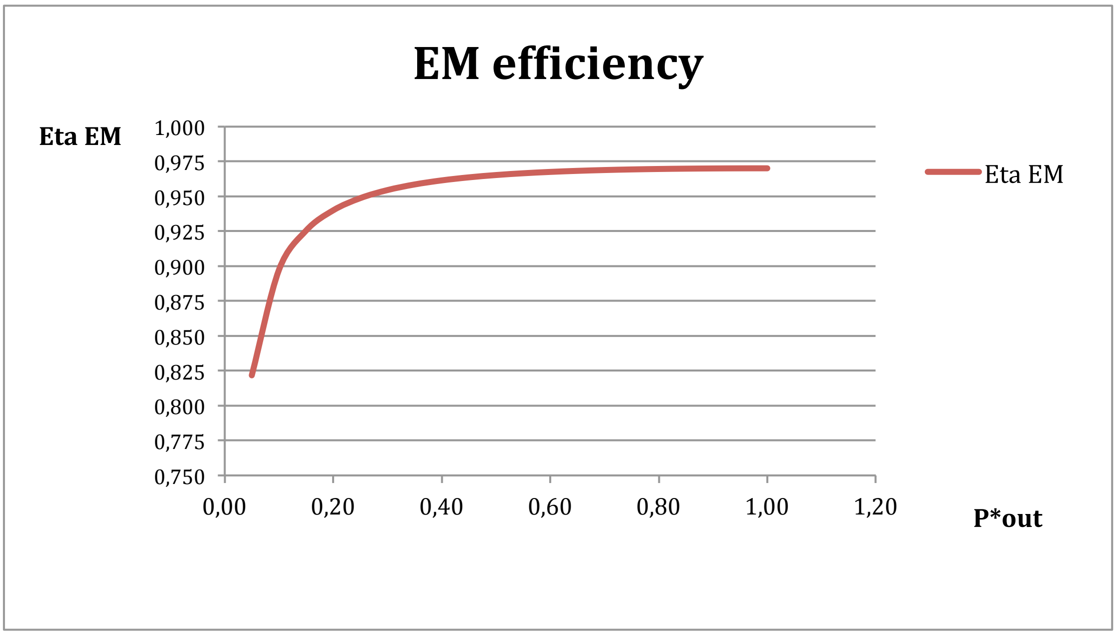

Figure 3.9: Assumed efficiency curve for electric machine

3.3 Gearbox

The high flexibility in a hybrid system sets requirements for the complexity of the gearbox. A gearbox has multiple functions, where any combination of these features is possible:

- Change rotational speed from the input shaft to the output shaft
- Couple multiple drives to one shaft (diesel engine/PTI to propeller) or one drive to multiple outputs (diesel engine to propeller/PTO)
- In case a FPP is used with a non-reversible engine: reverse direction of the propeller
- Switch between number of inputs, outputs and/or rotational speeds

3.3.1 Gearbox configuration

The vast majority of gearboxes found on ships are parallel configurations that are constructed of connecting wheels and pinions with teeth. On a basic single gear the engine drives a pinion with a small diameter and small number of teeth. This pinion drives the main wheel with a larger diameter and number of teeth. Because the teeth mesh into each other, their circumference speed is the same. But because of the different diameters, the rotational speed of the output wheel is reduced. The reduction ratio i is defined as:

$$i = \frac{n_e}{n_p} \qquad [3.30]$$

With n_e the input speed of the diesel engine and n_p the propeller speed. The gearbox transfers power, so it could also be stated that:

$$M_s = \eta_{GB} \cdot i \cdot M_B \qquad [3.31]$$

The high-speed side is therefore also referred to as the low torque side (brake torque at engine side, M_B) and the low speed side as the high torque side (shaft torque, M_S).

Marine gearboxes often have angled teeth that form part of a helical shape. Instead of just one, there are always more teeth in contact. This ensures better distribution of the forces on the wheels. Because of this, these gears can withstand higher torque and higher speeds, resulting in higher power. A slight downside can be that because there is more mechanical friction, the efficiency is slightly lower [Muhs, 2007]. More important is that this also generates an extra axial force with possible resulting axial vibrations. To cancel out these forces, a double helical gear with opposed angles is used. When vibrations are an issue, for example on naval vessels or yachts, these types of gears can be installed.

The offset distance between the in- and outgoing shaft can be horizontal or vertical. For smaller single stage applications a vertical offset is often found. For multiple input and/or output it would make more sense installing a horizontal offset configuration. Three types of gearbox configurations are found in a hybrid concept, with the distinction being made between the PTO on the primary wheel or a secondary wheel.

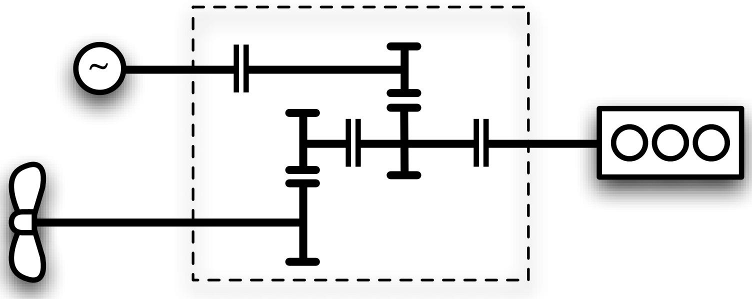

Figure 3.10: Primary PTO/PTI configuration

As a default the primary PTO/PTI gearbox is used as in Figure 3.10. With a primary PTO/PTI configuration the power take off is at the first wheel from the diesel engine, hence the name primary. It is possible to declutch the propeller side and run the PTO by the diesel engine, for example on DP operation or in port. It is also possible to power the propeller by the PTI if the main engine can be disengaged by a clutch.

3.3.2 Power

The power that a gearbox can transmit is not really a limiting factor. Applications of over 60 MW can be found. Often the power is given in dimension of torque, or kW/rpm. This range is also very wide, for example from 1 to 110 found in the Renk product range.
The teeth are made of hardened steel to prevent small particles of metal to end up in the lubrication oil. The efficiency of the gearbox is partly dependent on the purity of the oil. Attention must be paid whether the power is transferred in PTO, PTI or both modes. If power is transmitted in one direction only, just one side of the teeth need to be hardened. If the gearbox should be able to transmit power in reverse operation, both sides of the teeth should be hardened. Note that reverse operation only applies to the power direction, not to the direction of rotation of the wheels.

3.3.3 Speed

Gearbox manufacturers give a maximum gear ratio *i* of about 10 for single stage units and about 25 for two-stage units. For the medium speed engines at MDT with a maximum speed of 1000 rpm, a single stage reduction gear is usually sufficient to reach propeller speeds around 100 rpm.

Electric motors can have higher speeds though. In that case sometimes a second reduction stage is required.

Within the speed ranges of the diesel engine, electric motors and propellers there are no limitations for the gearbox.

3.3.4 Efficiency

Rewriting [3.31] gives the efficiency of the gearbox:

$$\eta_{GB} = \frac{M_S}{i \cdot M_B}$$

ref [3.31]

Losses in a gearbox consist of load dependent losses and no load losses. Load dependent losses occur because of friction in the power transmitting components such as bearings and meshing teeth. The teeth are moving over each other with friction, which causes some heat generation.

No load losses occur by splashing the lubrication oil and churning, making the oil warmer. These losses are more speed dependent. For low load and high speed these losses are dominant. For higher loads and continuous operation the bearing and gear losses are more dominant [Höhn et al, 2007].

Single power losses are small, but all together they contribute to some heat development and a reduced efficiency.

Lubrication of the gears clearly improves the efficiency by reducing the friction between the teeth, but also introduces some churning and stirring losses. Lubrication of the meshing teeth can be achieved by running the gear through an oil sump, where the rotating teeth pick up oil. Larger gearboxes found on ships almost always have forced lubrication by spraying the oil onto the wheels. This reduces the churning losses, but require some extra power for driving the oil pump. This can be done mechanically with an attached pump at the gearbox, or a separate pump is installed.

[Muhs, 2007] gives example how nominal efficiency can be calculated. As mentioned before, the frictional power losses dominate the efficiency at higher loads. The losses due to churning or the auxiliary pump are small and are not incorporated in the nominal efficiency.

The nominal efficiency consists of the efficiency between the meshing teeth η_t, in the wheel bearings η_b and at the shaft seals η_s. These make up the total nominal gearbox efficiency:

$$\eta_{GB,0} = \eta_t \cdot \eta_b \cdot \eta_s \qquad [3.32]$$

[Muhs, 2007] gives estimations of these values:
η_t = 0,99
η_b = 0,99
η_s = 0,98

For a single stage gearbox with two wheels and two shafts this gives:

$$\eta_{GB,0} = \eta_t \cdot \eta_b^2 \cdot \eta_s^2 = 0{,}932 \qquad [3.33]$$

Based on [Klein Woud & Stapersma, 2003] and gearbox manufacturers such as Renk the nominal efficiency $\eta_{GB,0}$ of a single stage gearbox lies in the range of 0,98 – 0,99 and for multiple stage around 0,95 – 0,98. This implies that the values chosen by [Muhs et al, 2007] might be a bit low.

By careful design, high number of teeth and high rotational speeds (> 10 m/s) the η_t can get up to 0,995 [van Heesewijk, 1982]. With this in mind some new values for the efficiencies are used here:

η_t = 0,995
η_b = 0,998
η_s = 0,995

This gives some more reasonable results for a single stage gearbox:

$$\eta_{GB,0} = \eta_t \cdot \eta_b^2 \cdot \eta_s^2 = 0{,}981 \qquad [3.34]$$

For a primary PTO gearbox depicted in fig 3.10 the efficiency is:

$$\eta_{GB,0} = \eta_t^2 \cdot \eta_b^4 \cdot \eta_s^3 = 0{,}967 \qquad [3.35]$$

These figures are more consistent with what other literature and manufacturers state.
If for instance the propeller is driven by just the PTI motor and the engine is clutched off, the efficiency for a primary PTO gearbox decreases by one factor η_s. In other words, it

depends on the configuration and the settings of the clutches what the nominal gearbox efficiency will be.

At part load the efficiency will change. From the definition of efficiency in [3.36] and [3.29] the part load efficiency of the gearbox is given in [3.37]:

$$\eta = \frac{P - P_{loss}}{P} = 1 - \frac{P_{loss}}{P} \qquad [3.36]$$

With [3.29] this can be rewritten as:

$$\begin{aligned} 1-\eta_{GB} &= \frac{P_{loss}}{P} \\ \frac{1-\eta_{GB}}{1-\eta_{GB,0}} &= \frac{P^*_{loss}}{P^*} \\ \eta_{GB} &= 1 - \frac{P^*_{loss}\cdot\left(1-\eta_{GB,0}\right)}{P^*} \end{aligned} \qquad [3.37]$$

Where the gearbox efficiency η_{GB} is as described before and P^* is the power normalised to nominal power. P^*_{loss} is described in [Stapersma, 1994]:

$$P^*_{loss} = aP^* + bM^* + cn^* \qquad [3.38]$$

Where M^* is the torque normalised to nominal torque and n^* is the speed normalised to nominal speed. The coefficients a, b and c are chosen as 0,4, 0,4 and 0,2 respectively, based on experience.

The power flow through the gearbox and therefore also the power losses are a product of effort and flow, i.e. torque and speed respectively. With the power as a product of torque and speed (P* = M*·n*) eq. 3.28 can be written as:

$$P^*_{loss} = aM^*n^* + bM^* + cn^* \qquad [3.39]$$

The power flow P^* is known from the decision variables and the installed engine power, so to determine the torque M^*, n^* must be known. The selection of the operating curve has an influence on this. As a default, the combinator curve with constant speed above 80% is used. Above 80% load the speed remains constant and the thrust can be increased by increasing the pitch. Below 80% load and between 60% n^* and 100% n^* the power follows the propeller law:

$$n^* = \sqrt[3]{P^*}$$ ref [3.6]

Below 60% n^* the speed remains constant. Various curves are drawn in Figure 3.11, that corresponds to the gearbox efficiency. Since the part load efficiency of the gearbox is a function of M^* and n^*, and n^* is related to P via the propeller law, a plot can be made that shows η as a function of normalised power P^*, see Figure 3.12.

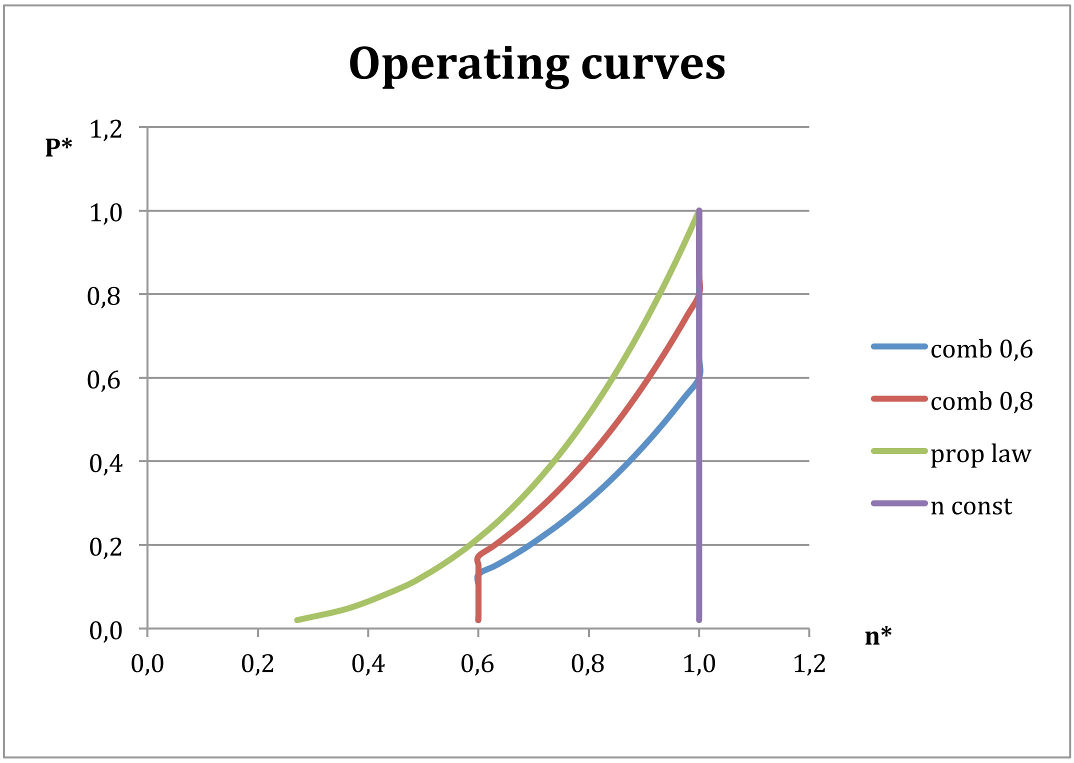

Figure 3.11: Examples of operating curves of propeller

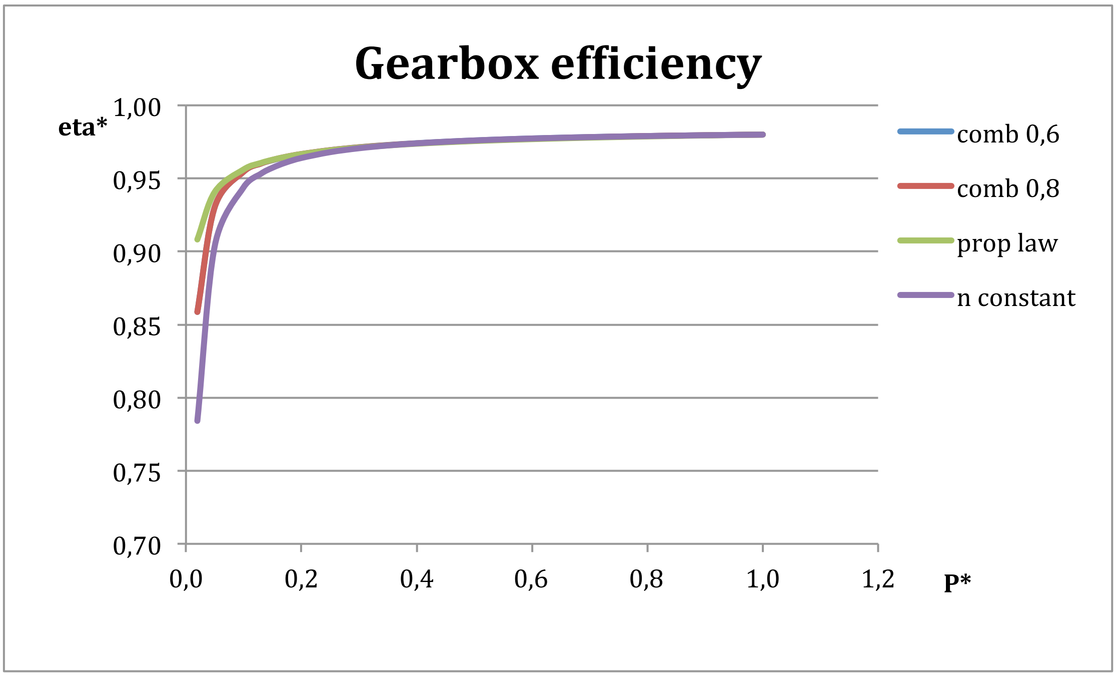

Figure 3.12: Gearbox efficiency corresponding to different operating curves

3.4 Electrical equipment

If an electric motor has to operate with varying speeds, there is a need for a drive with a variable frequency. A fixed pitch propeller might need a frequency converter to keep the output frequency in generator mode constant, even with variable shaft speed. A frequency converter in combination with a CPP introduces the use of the combinator curve. According to a case study done by Rolls-Royce [2010] on a platform supply vessel, a hybrid system running on a combinator curve saves about 5% of fuel compared to a fixed speed system. Even though the efficiency of a converter lies in the region of 97%-98%, there still remains a significant improvement in overall efficiency.
A fixed input frequency of 50 or 60 Hz will be converted to a variable frequency that determines the motor speed according to [3.20].
Two commonly used converters in the power range of medium speed diesel engines are the Pulse Width Modulation (PWM) converter and the Current Source Inverter (CSI) converter, which both have the same principle diagram as in Figure 3.13. A 3-phase AC will be converted to a DC. This DC link acts as an energy buffer. Then through switching elements the DC is converted to a varying AC that drives the motor.

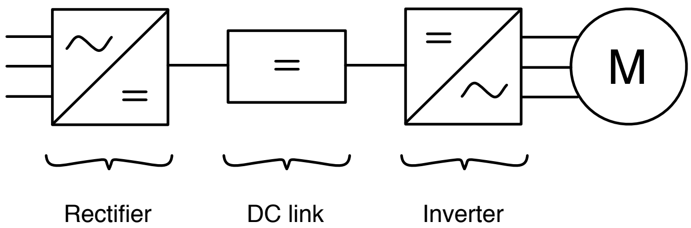

Figure 3.13: Principle of frequency converter with DC link

3.4.1 Frequency converter types

The ways the rectifier and inverter are put together and controlled determine the type of frequency converter.

Pulse width modulated converter

A PWM converter normally consists of an uncontrolled rectifier with capacitator and a PWM inverter. In this converter the voltage is held constant. The capacitator in the DC link smoothens the voltage by accumulating some energy. Therefore this type of converter is often called a voltage source inverter (VSI).
With an uncontrolled rectifying bridge, the converter cannot generate negative voltage. This means that it can only operate in the motoring quadrants as seen in Figure 3.6. If needed, the converter can be provided with a so-called active frond end, that has

switching transistors instead of passive diodes. This makes it possible to deliver power back to the grid, for example in PTO operation.

CSI converter

A CSI converter consists of a controllable rectifier and an inverter. By controlling the switching rate of the thyristors in the controlled rectifier, any DC voltage and current can be produced This wave is not so smooth, so some energy buffer is required in the form of an inductor that smoothens the current. In the DC link the current is kept constant, hence the name current source inverter (CSI). The varying DC makes it possible to create a varying output voltage and frequency by the inverter bridge.

The input AC commutates the thyristors on the rectifier side. Therefore this converter also is referred to as a load-commutated inverter (LCI). This natural commutation is only possible with a leading power factor (i.e. current phase leads the voltage phase). With a synchronous motor this is possible, because the excitation can be controlled. With an induction motor, this cannot be done, so a CSI converter can only be applied with synchronous motors. Therefore the default converter is a PWM converter.
Because the rectifier bridge is controlled, it has a varying power factor that varies with the desired frequency. Also, it enables delivery of power back to the grid, in a 4-quadrant operation.

3.4.2 Power of converters

The capacitator in the DC buffer link in the PWM converter is able to deliver reactive power. This power is needed to start induction motor as stated earlier in this chapter. Because the converter does this, the generators can be dimensioned smaller. This makes the PWM converter applicable to both induction motors and synchronous motors. The power limit used to be around 8 MW, but modern transistors such as the IGBT make it possible to go higher up to 30 MW [Adnanes, 2003]. However, the PWM converter is usually chosen in combination with a induction motor and this motor size is limited to 7 MW due to economic reasons. It has a high and constant power factor of around 0,95, instead of the CSI converter that has a varying power factor.
The CSI converter does not need very fast switching elements and can be equipped with ordinary thyristors. This makes it possible to go to very high powers, up to 100 MW [Adnanes, 2003]. In summary, generally PWM converters are chosen in combination with induction motors up to about 7 MW and above that a CSI converter in combination with a synchronous motor is chosen.

3.4.3 Efficiency of converters

The frequency converter has losses in the rectifier, in the DC bridge and in the inverter. These losses are due to heat development in the material; copper losses ($P = I^2R$). In the rectifier and inverter part there is also a voltage drop at the diodes and thyristors that

also contributes to a small loss. These losses are load dependent. Next to this there is usually a no-load loss in the form of a running cooling fan.
[Ross et al, 2010] mentions the distribution between these three to be 45% conduction (IR), 45% switching losses (I^2R) and 10% steady losses:

$$P_{loss,FC} = P_{nom} \cdot \left(1-\eta_{nom}\right) \cdot \left(0{,}1 + 0{,}45 I^* + 0{,}45\left(I^*\right)^2\right) \quad [3.40]$$

Converter manufacturers all specify a nominal efficiency between 0,97 and 0,98. For a constant voltage the current I^* is proportional to power P^*, so according to [3.40] it can be stated that the efficiency will drop for low loads. However, the range between 10% and 100% load is within 1% of nominal efficiency for a nominal efficiency of 98%, see Figure 3.14. Since the design of the vessel is still in a concept phase and the actual flowing currents are not known yet (partially because the voltage level is yet to be determined), no useful assumptions can be made regarding the efficiency for the converters. As a default, a fixed η_{FC} is assumed at 0,98.

The use of more advanced switching elements might increase the efficiency. Some heat loss is generated in the switching elements and the faster they switch, the more losses can occur. This heat should be dissipated by forced air-cooling or water-cooling.
It should be noted however, that also the power factor of the synchronous converter varies with lower load.

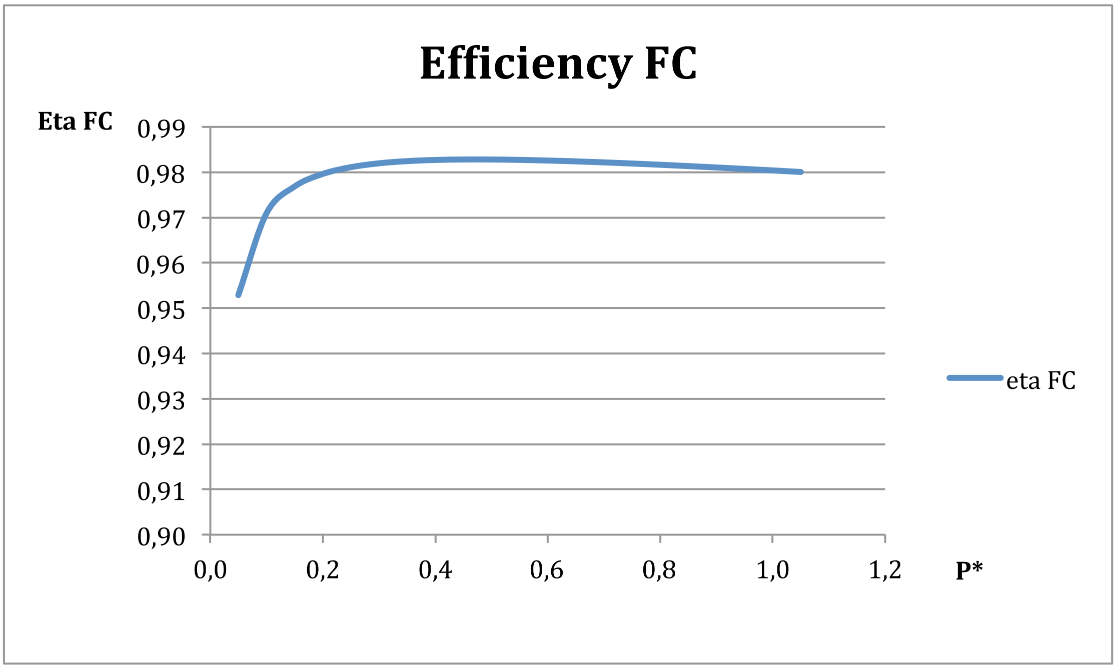

Figure 3.14: Efficiency of frequency converter

3.4.4 Main switchboard

A main component that cannot be excluded in this chapter is the main switchboard. The function of the main switchboard is to distribute all electric power generated by the generators to the consumers. These switches can be circuit breakers or simple contactors. Simple breakers can be hand operated, but larger ones are usually operated by electric motors.

As a rule of thumb, these standards are used at MDT for total installed generator power:

Installed power	Recommended Voltage level
< 10 – 12 MW_{el}:	440 V
< 13 – 15 MW_{el}:	690 V
< 48 MW_{el}:	6.600 V
< 130 MW_{el}:	11.000 V

Table 3.2: Voltage choice based on total installed electric power

The range above 48 MW is not in the range of MDT's medium speed engines, and probably a hybrid system would not be feasible here. 130 MW is exceptionally high, used in for example cruise vessels and large offshore drilling ships. For these powers a full diesel electric system would be more appropriate.

There are several reasons to switch to a higher voltage level. For the same amount of power, a higher voltage means a lower current, see [3.19]. The current determines the size of the switching gear and the diameter of the cables. Besides this, the short circuit currents are lower. In general, less copper is needed to distribute all electric power, which will be cheaper. The downside is that all these switching components should be able to handle medium voltage. This means overall larger switching gear because of insulation and a higher price for all components. Next to this, transformers are needed to reduce voltage for converters and lower distribution boards. Transformers for higher voltage are larger and heavier.
These are the reasons that above 13 – 15 MW installed electric power a medium voltage system is more attractive.

Another selection can be made between 50 Hz or a 60Hz system. On board vessels usually a 60 Hz grid is selected. According to [3.20] the frequency has an influence on the output speed of an electric motor. For the same power, a higher speed motor has lower torque, and therefore lower currents flowing. This means a smaller and cheaper construction is possible.

Some consumers are large single consumers such as propulsion motors. Others are grouped onto lower distribution boards. These distribution boards often have a lower voltage (i.e. 230 V).
Important equipment such as emergency lighting and navigational equipment can also be fed from an emergency switchboard. This switchboard is then located separately from the main switchboard.

Switching losses in the switchboard have similar behaviour as in converters. There are some copper losses that increase with increasing currents. These are however so small that they do not influence the design of the complete plant significantly and the efficiency is assumed to be constant at 1,0 in this plant design methodology.

3.5 Propeller

Although it is left outside the design scope of this thesis, the propeller plays a major role in a propulsion system. The main function of a propulsor is to generate thrust to propel the vessel through the water. Generally it is mounted aft of the vessel to have a good flow and achieve the best efficiency. The most common type of propulsor is a screw propeller. It will usually be made for each ship specifically, to fit the flow conditions around the hull and the propulsion engines.

The propeller pitch is the increase in axial direction over one full revolution. When the forward displacement over one revolution is smaller than the pitch, the propeller will develop thrust. The axial displacement per revolution (V/n) is therefore an important parameter and can be made dimensionless as the advance ratio J:

$$J = \frac{v_A}{n_p \cdot D} \tag{3.41}$$

The thrust and torque can be made non-dimensional with the speed, propeller diameter and the density of the water:

$$K_T = \frac{T}{\rho \cdot n_p^2 \cdot D^4} \tag{3.42}$$

$$K_Q = \frac{Q}{\rho \cdot n_p^2 \cdot D^5} \tag{3.43}$$

These parameters are a measure of the propellers performance and efficiency. The propeller's open water efficiency (defined in appendix A) can also be expressed in terms of J, K_T and K_Q:

$$\eta_O = \frac{P_T}{P_O} = \frac{1}{2\pi} \cdot \frac{T \cdot v_A}{Q \cdot n_p} = \frac{1}{2\pi} \cdot \frac{K_T \cdot J}{K_Q} \tag{3.44}$$

For a certain pitch/diameter ratio, an open water diagram can be drawn to show this efficiency for different advance ratios, seen in Figure 3.15 with multiple *P/D* ratios. Generally, the maximum efficiency of a FPP is around 0,7. Although the construction of the propeller can be quite complex, this piece of equipment actually has no moving parts.

3.5.1 Controllable pitch propeller

The controllable pitch propeller (CPP) has been mentioned several times before in the previous chapters. There are two ways of controlling the ships speed with a CPP: either by pitch control at a constant rotational speed, or by operating on the combinator curve. A CPP is suitable for many different operating conditions, because it can vary the thrust at constant speed. By a hydraulic mechanism in the hub and through the shaft of the propeller, the blade pitch can be varied. This results in varying *P/D* ratios, giving a very flexible open water diagram. In a CPP there is less effective blade area because the hub is slightly larger to accommodate the pitch changing mechanism. Next to this, the expanded blade area ratio cannot be as large as is possible with an FPP because clearance for negative pitch is necessary. Therefore the maximum efficiency of a CPP is somewhat lower than that of a FPP, for the same diameter. The advantages of a CPP are not in efficiency but more in manoeuvrability. Next to this, the lower limit for the speed of a 4-stroke diesel engine require a propeller that can produce small thrust, even at higher speeds. This makes the CPP the obvious choice for 4-stroke applications.

Another reason for choosing a CPP is when applying a PTO without a frequency converter. In this case the engine speed should be kept constant to provide a constant frequency on the ships grid. In order to be still able to change the thrust of the vessel, the pitch angle will be changed. This is not always favourable; when low thrust is required a low pitch angle must be applied. With still a high rotative speed this results in a low efficiency and the risk of pressure side cavitation.
A third great advantage with a CPP has less to do with the combination of a hybrid drive, but more with flexibility in operation. In particular the manoeuvrability is improved compared to a FPP, because the reactions of the pitch changes are faster than changes of engine speeds. This especially is the case when manoeuvring at low speeds. The CPP can quickly change the thrust direction, instead of reversing the propeller direction at the gearbox or with the engine direction. Sometimes it is needed to generate a large thrust without any ship speed for example with tugboats. A large thrust can be generated with a large pitch, where a FPP can only change the thrust by increasing the propeller revolutions.

The design of the propeller is either constrained by the optimum diameter or optimum speed. Usually the largest diameter is chosen depending on the available space beneath the hull, so the optimal speed can be determined. To find the best propeller properties first the required thrust for a certain design speed must be known. The ships thrust can also be made non-dimensional and expressed in terms of J:

$$K_{T,ship} = \frac{T}{\rho \cdot v_A^2 \cdot D^2} \cdot \frac{v_A^2}{n_p^2 \cdot D^2} = \frac{T}{\rho \cdot v_A^2 \cdot D^2} \cdot J^2 \qquad [3.45]$$

This curve can also be drawn in the open water diagram. See Figure 3.15 for an example. This figure is based on the Wageningen-B series, i.e. a series of fixed pitch propellers. This does not completely represent a CPP. For a CPP the design point can be represented by an open water diagram, but for all other operating points this has no purpose. The actual efficiency depends on the shape of the combinator curve and is very much dependent on the pitch and speed combined. Often a corrected open-water diagram is used for a CPP that gives reasonable outcomes compared to measured data [Krueger, 2005].

The $K_{T,ship}$ curve intersects the K_T curves of the propeller for several P/D ratios. There is an optimum η_0 belonging to a certain P/D ratio, shown in red (Kurve der Wirkungsgrade). This is the design point of the propeller.

3.5.2 Power

Generally speaking propellers can be constructed for all powers available in marine engines. The largest propeller to this day is an 11 m diameter propeller that can handle 90 MW [MAN, 2010]. The restrictions are due to the strength in the blade roots. This somewhat limits the power of the CP propeller, since the bolts on the blades can handle less tensional and shear forces than a fixed blade. The largest CPP built to date is about 40 MW.

If propulsion power cannot be handled by one propeller diameter, a two-shaft system is required. This is also the case if more manoeuvrability is required.

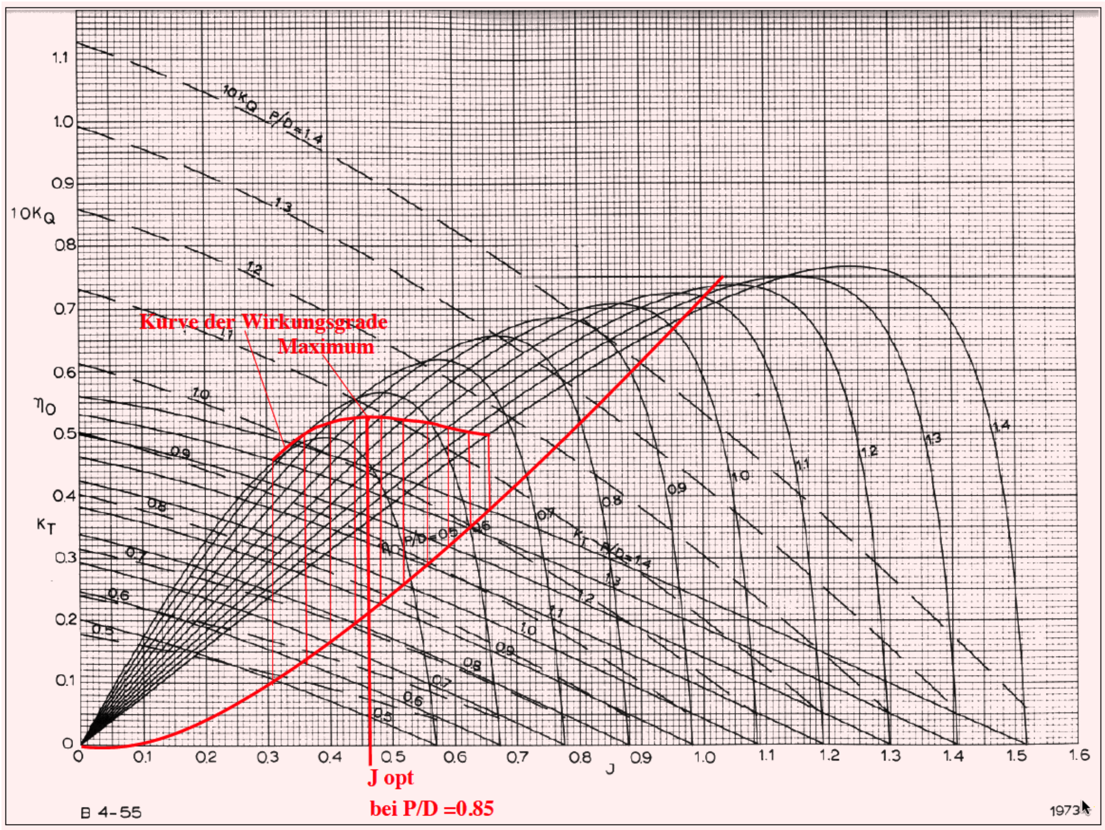

Figure 3.15: Open water diagram for the Wageningen B 4-55 series. From [Krueger, 2005]

3.5.3 Speed

Just as with power, the speed of propellers is limited because of the forces at the blade roots. Another important factor concerning the speed of the propeller is cavitation. This is the situation where water in liquid form changes to vapour due to very low pressures. These low pressures are caused by very high local velocities, usually at the forward side of the blades and at the blade tips. When these vapour bubbles come to the aft and high-pressure side of the blade, they implode leaving marks on the material. It causes erosion on the propeller and introduces noise and vibrations. High-speed propellers have a higher tip velocity and therefore are more likely to be subjected to cavitation. Cavitation limits the product of rotational speed and propeller diameter.

The speed limitation in combination with a better efficiency at lower speeds, sets the requirements for a reduction gearbox with medium speed engines.

3.5.4 Efficiency on combinator curve

The efficiency of a CPP is slightly lower than an FPP because of the less effective blade area due to the larger hub. At low pitch, the propeller has a relatively larger boundary

layer with a smaller thrust. These are called the zero pitch losses. One way to overcome these efficiency losses at low loads is to install a two-stage gearbox.
Many factors influence the propeller efficiency. One important factor is the inflow conditions before the propeller. Because every vessel is different, every design will require its own well-fitted propeller. With a two-shaft concept, the wake after the ship is different, resulting in a little lower efficiency for the propellers.

With a CPP, both pitch and speed can be adjusted, as long as the propeller fulfils the thrust demand. When pitch is reduced at constant rotational speed, the intersection with K_T now occurs at a different value for *J*. This results in a less optimal η_0 than the design pitch condition in most cases.
With a variable propeller speed, the design point can remain at a certain *P/D* ratio. The propeller speed just has to vary proportionally with the different ships speed, or more in particular, with v_A. So by varying the propeller speed instead of the pitch, η_0 can remain relatively high.
To match the engine envelope, particularly of 4-stroke engines, pitch control has to be used to fulfil all thrust demands. This is for instance the case for low vessel speed when the engine speed cannot go below 60% n_e. Another instance is in the high loads up to the design point, where pitch control ensures a combinator curve further away from the engine surge limit.

The general trend is that below a certain rotational speed, using the combinator curve becomes beneficial over fixed speed operation. The trade-off is of course the investment costs of frequency converters that are required for PTO operation or electric propulsion. Only when including the propeller in the design scope, i.e. use P_E as a main requirement instead of P_D, do these effects have an influence.

3.6 Investment costs

Next to the operational costs that are mainly dependent on the fuel costs, the capital costs should also be taken into account when designing a propulsion power plant. The focus of this thesis is on fuel efficiency, but a brief study on the investment costs of the components can also influence the final decision.

Diesel engine prices drop for larger bore diameters. Next to this, V-engines with a higher power density are less expensive than line-engines. For the larger bore diameters this difference becomes noticeable. In the table the distinction is made at a bore diameter of 32 cm.

An estimation of investment costs is based on supplier data and MDT experience. A more extensive research on investment costs of components is carried out in [van Es, 2011] and also served as a basis for these figures, listed in table 5.1. It must be mentioned that these prices are just a rough estimate, no real conclusions can be derived from these figures; it just serves as a basis for comparison between the different configurations.
Although the costs are influenced by several factors such as weight, size and power, the costs indicated here are only given in €/kW for easy calculation. Especially the gearbox costs are more dependent on weight than power.

Component	Costs [€/kW]	Remarks
Diesel engine 4-stroke	360	Line type
Diesel engine 4-stroke	340	V-type < 32 bore
Diesel engine 4-stroke	280	V-type ≥ 32 bore
Diesel generator set	400	< 32 bore
Diesel generator set	360	≥ 32 bore
Electric machine	50	Induction
Single stage gearbox	30	Extra input adds ± 15%
Frequency converter	120	Both PWM and LCI
Frequency converter	135	With active front end
CPP + shaftline	100	

Table 3.3: Indication for investment costs of components

4 Optimisation potential

In order to find the optimal design parameters within a hybrid propulsion design, all relevant variables declared in chapter 2 should be brought into one function. This function can then be optimised for lowest fuel consumption.

4.1 Objective function

With the power demand functions determined in paragraph 2.5 and the components in chapter 3, it is now possible to set up the function that can be optimised for lowest fuel consumption. In optimisation terminology this function is called the objective function. It is a function of several variables (in $\boldsymbol{X_n}$), called the decision variables.

4.1.1 Annual fuel consumption

The fuel consumption comes from the prime movers, i.e. the main diesel engines and the diesel generator sets. The general total annual fuel consumption M_{fuel} is:

$$M_{fuel} = \sum_n \left[\left(\dot{m}_{f,ME1,n} + \dot{m}_{f,ME2,n} + \dot{m}_{f,DG,n} \right) \cdot t_n \right] \qquad [4.1]$$

The $\dot{m}_f$ comes from the fitted 2nd order polynomials in [3.15] and [3.16]:

$$\dot{m}_{f,ME1} = \left(a_{ME1} \cdot \left(P^*_{ME1} \right)^2 + b_{ME1} \cdot P^*_{ME1} + c_{ME1} \right) \cdot i_{ME1} \qquad \text{ref [3.15]}$$

$$\dot{m}_{f,DG} = \left(a_{DG} \cdot \left(P^*_{DG} \right)^2 + b_{DG} \cdot P^*_{DG} + c_{DG} \right) \cdot i_{DG} \cdot N_{DG} \qquad \text{ref [3.16]}$$

Combining [4.1] with [3.15] and [3.16] the total annual fuel consumption can be calculated:

$$M_{fuel} = \sum_n \left[\left[\begin{array}{l} \left(a_{ME1} \left(P^*_{ME1,n} \right)^2 + b_{ME1} P^*_{ME1,n} + c_{ME1} \right) \cdot i_{ME1} + \\ \left(a_{ME2} \left(P^*_{ME2,n} \right)^2 + b_{ME2} P^*_{ME2,n} + c_{ME2} \right) \cdot i_{ME2} + \\ \left(a_{DG} \left(P^*_{DG,n} \right)^2 + b_{DG} P^*_{DG,n} + c_{DG} \right) \cdot i_{DG} \cdot N_{DG} \end{array} \right] \cdot t_n \right] \qquad [4.2]$$

The objective function has to be a function of the decision variables, so the brake power has to be expressed in terms of actual delivered power with the applicable plant efficiency according to [2.14] and [2.15]:

$$M_{fuel} = \sum_n \left[\left[\begin{array}{l} \left(a_{ME1} \cdot \left(\dfrac{P_{ME1,n}}{\eta_{P,ME1,n} \cdot P_{B,ME1}} \right)^2 + b_{ME1} \cdot \left(\dfrac{P_{ME1,n}}{\eta_{P,ME1,n} \cdot P_{B,ME1}} \right) + c_{ME1} \right) \cdot i_{ME1} + \\ \left(a_{ME2} \cdot \left(\dfrac{P_{ME2,n}}{\eta_{P,ME2,n} \cdot P_{B,ME2}} \right)^2 + b_{ME2} \cdot \left(\dfrac{P_{ME2,n}}{\eta_{P,ME2,n} \cdot P_{B,ME2}} \right) + c_{ME2} \right) \cdot i_{ME2} + \\ \left(a_{DG} \cdot \left(\dfrac{P_{DG,n}}{\eta_{P,DG,n} \cdot P_{B,DG}} \right)^2 + b_{DG} \cdot \left(\dfrac{P_{DG,n}}{\eta_{P,DG,n} \cdot P_{B,DG}} \right) + c_{DG} \right) \cdot i_{DG} \cdot N_{DG} \end{array} \right] \cdot t_n \right] \quad [4.3]$$

Eq. [4.3] is the objective function that has to be optimised for lowest fuel consumption, by changing the decision variables.

Note that the $P_{EM,n}$ parts are not included in the objective function. Still, M_{fuel} is a function of **X**, where $P_{EM,n}$ is included. This is because of the boundary conditions to [4.3], which are given in [2.13].

4.1.2 Selection of components

To check which engine fits best and to limit the amount of variables used in the objective function, a pre-selection of the component configuration must be performed. Four separate configurations can be chosen for comparison.
In each configuration one or two main engine types (bore) can be chosen. The engine type determines the correct fuel consumption curve parameters *a*, *b* and *c*. The choice between a linear fit and a non-linear fit can also be made. With a linear fit, only parameters *a* and *b* are used. Every engine has an available cylinder configuration that must also be pre-selected. This gives the total P_B for the main engine(s). To prevent overloading a certain engine margin is chosen. The default is 10%, which gives the maximum available brake power at 90% MCR.
The same selection can be performed for the DG's. But next to this, the number of installed DG sets must be selected. This gives a maximum available brake power and fuel consumption curve. Note however, that the number of installed DG's does not mean that there are multiple P_{DG}'s as decision variables, as explained in paragraph 2.4.
As a default, an electric machine is installed so hybrid is possible. Setting the EM at zero for all modes gives a diesel mechanic mode. Setting all the main engines to zero means only DG power is available, creating a diesel electric mode. Appendix C gives an example of the requested input in Excel.

4.2 Optimisation method

From the objective function and the shape of the sfc curves it can be concluded that the system is non-linear. This leaves out many efficient optimisation methods that are based on linear systems. For non-linear systems there are even more methods, all with their own complications. There is no single algorithm that can always be used to solve optimisation problems with non-convex functions [Hillier, 2010]. To select a suitable optimisation algorithm, first it must be known whether the objective function is convex or not.

4.2.1 Convexity of function

Nonlinear optimisation algorithms are generally unable to distinguish between a local minimum and a global minimum. Therefore is must be known whether the determined local minimum is also definitely the global minimum. This can be proven when the second derivative of the function is always larger than or equal to zero. This kind of function is called a convex function. Similarly, if the second derivative is always smaller than or equal to zero the function is called concave. A convex function has the characteristic that a line joining any two points on the graph always lies above the graph (Figure 4.1). There is only one minimum point: the global minimum.

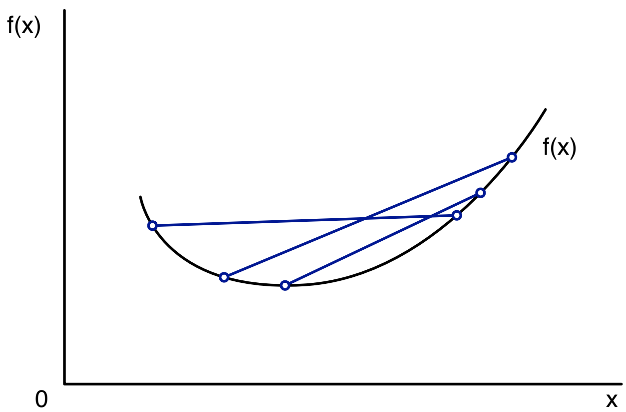

Figure 4.1: Example of a convex function

Checking whether a function is concave or convex can help understanding the behaviour of the function and help determining a correct optimisation method. The target function of annual fuel consumption should be a convex function; it should be minimised. The theory of a convex function is as follows:

"A function with many variables and continuous first and second order partial derivatives is convex if the Hessian matrix H(**X**) is positive semi-definite for all $\mathbf{X} \in S$."

The Hessian matrix of a function with m (= 4n) variables is a matrix consisting of all the function's second order partial derivatives:

$$H(\boldsymbol{X}) = \begin{vmatrix} \dfrac{\partial^2 M_{fuel}}{\partial P_{ME1,1}^2} & \dfrac{\partial^2 M_{fuel}}{\partial P_{ME1,1}\partial P_{ME2,1}} & \cdots & \dfrac{\partial^2 M_{fuel}}{\partial P_{ME1,1}\partial P_{DG,n}} \\ \dfrac{\partial^2 M_{fuel}}{\partial P_{ME2,1}\partial P_{ME1,1}} & \dfrac{\partial^2 M_{fuel}}{\partial P_{ME2,1}^2} & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots \\ \dfrac{\partial^2 M_{fuel}}{\partial P_{DG,n}\partial P_{ME1,1}} & \cdots & \cdots & \dfrac{\partial^2 M_{fuel}}{\partial P_{DG,n}^2} \end{vmatrix} \quad [4.4]$$

The notation $\mathbf{X} \in$ S means all $\mathbf{X}$ from the set S. In this case the $\mathbf{X}$ stands for the vector $\boldsymbol{X_n}$ determined in paragraph 2.4. The set S is the range in which the variables can operate, i.e. the range of the engines. This varies from zero to the CSR point.

Positive semi-definite means in a way that the matrix is regarded as positive as one would call a single real number positive. This does not mean all entries in the matrix should be positive however. It is positive semi-definite if the eigenvalues are greater or equal than zero for $\mathbf{X} \in$ S.

Eq. [4.3] is the equation that should be differentiated to construct the Hessian matrix. The first entry is:

$$\frac{\partial M_{fuel}}{\partial P_{ME1,1}} = \left[2 \cdot a_{ME1} \cdot \left(\frac{1}{\eta_{P,ME1,1} \cdot P_{B,ME1}} \right)^2 \cdot P_{ME1,1} + b_{ME1} \cdot \left(\frac{1}{\eta_{P,ME1,1} \cdot P_{B,ME1}} \right) \right] \cdot i_{ME1} \cdot t_1 + 0 + 0$$

$$\frac{\partial^2 M_{fuel}}{\partial P_{ME1,1}^2} = 2 \cdot a_{ME1} \cdot i_{ME1} \cdot t_1 \cdot \left(\frac{1}{\eta_{P,ME1,1} \cdot P_{B,ME1}} \right)^2 \quad [4.5]$$

Similar partial derivatives can be taken for all entries in the matrix. For example:

$$\frac{\partial^2 M_{fuel}}{\partial P_{ME2,1}^2} = 2 \cdot a_{ME2} \cdot i_{ME2} \cdot t_1 \cdot \left(\frac{1}{\eta_{P,ME2,1} \cdot P_{B,ME2}} \right)^2 \quad [4.6]$$

It turns out that taking the first partial derivatives of this function to one variable and the second derivative to another variable, the result is always zero. The resulting Hessian matrix is one where only the diagonal entries are non-zero:

$$H(\boldsymbol{X}) = \begin{vmatrix} 2 \cdot a_{ME1} \cdot i_{ME1} \cdot t_1 \cdot \left(\frac{1}{\eta_{P,ME1,1} \cdot P_{B,ME1}} \right)^2 & 0 & \ldots & 0 \\ 0 & \neq 0 & \ldots & \ldots \\ \ldots & \ldots & \ldots & \ldots \\ 0 & \ldots & \ldots & 2 \cdot a_{DG} \cdot i_{DG} \cdot N_{DG} \cdot t_n \cdot \left(\frac{1}{\eta_{P,DG,n} \cdot P_{B,DG}} \right)^2 \end{vmatrix}$$

[4.7]

It is relatively straightforward to determine whether it is positive semi-definite or not and to check whether it is convex or not. The eigenvalues of a diagonal and linear independent matrix are the entries on the diagonal.
This means the function is convex for $\mathbf{X} \in S$ if all parameters *a* are larger than zero and if the applicable η_P is a constant. This is not the case, since some of the fuel consumption fits have a negative *a* value. But more important is the η_P, which varies with different power flow and engine configuration and is therefore also a function of **X.** In these examples it was treated as a constant.

This means that the Hessian matrix is <u>not</u> positive semi-definite and therefore the function $M_{fuel}(\mathbf{X})$ is <u>not</u> convex. This leads to many local minima, while the global minimum is not that straightforward to determine.
Intuitively one might expect such a result simply by looking at the functions that make up the target function; the fuel consumption curves based on the sfc curves. Their minimum lies around 85% load, above and below that the curve moves upward.

4.2.2 Search method

With many directions to go to find the global minimum, the rates at which the variables can be changed must be known. In a minimum (or maximum) point, all the partial derivatives of all the variables must be 0. The value of the partial derivatives is a measure of how fast the variable is changing at that point. So this directly gives a measure in which direction to move to reach the optimum the fastest. This is the basis of the Generalised Reduced Gradient (GRG) search method used by the solver algorithm.
The objective function is differentiable and continuous, so it has a gradient $\nabla M_{fuel}(\mathbf{X})$ at each point **X**. At a certain point **X'** the gradient is the vector:

$$\nabla M_{fuel}\left(\boldsymbol{X}\right)=\left(\frac{\partial f}{\partial x_1},\frac{\partial f}{\partial x_2}\dots\frac{\partial f}{\partial x_m}\right) \text{ at } \boldsymbol{X}=\boldsymbol{X'}$$ [4.8]

The infinitesimal change in the position of **X'** that maximises the rate at which $M_{fuel}(\mathbf{X})$ decreases is the change that is proportional to $\nabla M_{fuel}(\mathbf{X})$. This means that the rate at which $M_{fuel}(\mathbf{X})$ decreases is maximised if the changes in the position of **X** are in the direction of the gradient $\nabla M_{fuel}(\mathbf{X})$.

To put it in other words: at point **X'** the algorithm determines how steep every variable is and takes the next iteration step in the steepest direction. At this new iteration point it evaluates the gradient again. It continues to do this until the gradient is zero. In this case a local minimum is found. In most cases the gradient will not be completely zero, but the algorithm keeps performing iteration steps until the changes are smaller than a defined stopping rule.

If the first order derivatives from the gradient search method do not provide accurate enough information for what direction to follow, a second order differentiation is carried out. The solver can switch between these first and second order derivatives automatically. Note the analogy with the Hessian matrix that also consists of the second order derivatives. The Hessian matrix in this case consists only of the diagonal as seen in [4.7]. In fact, a diagonal matrix multiplied by a vector will result in a vector. So the algorithm calculates the Hessian matrix at every iteration point to determine the direction of the next step. Calculating a complete Hessian matrix is time consuming, so often an approximation of the Hessian is carried out if this approximation provides enough information[5].

These methods actually only work for unconstrained options. This basically means that the only constraint is that the variables cannot be negative. Unfortunately this is not the case when applying it here, since the P_{EM} can be negative. For constrained optimisation problems there is no single algorithm available that always finds a local minimum [Hillier, 2010].

The GRG method first reduces the problem to an unconstrained optimisation problem so that the gradient search method is applicable. It does this by introducing an extra set of variables into the constraints. These variables are called the Lagrange multipliers. For every constrained variable a new variable λ is added to make the new function, in the most basic form:

$$\Lambda\left(\boldsymbol{X},\lambda\right)=M_{fuel}\left(\boldsymbol{X}\right)+\lambda\left[g\left(\boldsymbol{X}\right)-c\right]$$ [4.9]

[5] *The use of second order derivatives was developed by Newton, and is therefore often called the Newton's method or quasi-Newton method.*

The g(**X**) is some constraint function limited to the constraint *c*. These are for example given in [2.13] and the constraint that engines cannot deliver more than their brake power. The Λ is the symbol for the Lagrangian. The GRG method solves this set of equations for the variables in **X** and for the Lagrange Multipliers. The value of the Lagrange multipliers (if not 0) is a measure of how the constraints actually constrain the problem. This will be explained further in the sensitivity analysis.
If $M_{fuel}(\mathbf{X})$ is a minimum for the original constrained problem, there exists a λ such that the set (**X**, λ) is a point where all the partial derivatives are 0. To be more precise, the Lagrangian method is the basis for the KKT (Karush-Kuhn-Tucker) conditions. It falls beyond the scope of this thesis to go into detail for these conditions, but they can be found in [Hillier, 2010].

It might be clear that an objective function with many variables and complex constraints is hard to solve. At best a local minimum will be found.

4.2.3 Implementation in software

The Excel solver has many options, but also many limitations to what kind of problems it can handle. In this paragraph the setup of the solver will be described, as well as the problems concerning these limitations.

All calculations concerning the objective function occur in the cells in the spreadsheet. The solver only needs to know what variables can be altered, what the goal is and what the objective function is. The objective function does not need to be just in one cell reference; Excel can handle all cell references and dependencies that make up the total objective function. This makes the setup for the constraints somewhat easier.

As mentioned all the above calculations are being performed in the spreadsheet cells, making the setup of the constraints more clear. The constraints are listed in Table 4.1. The important dependencies are listed in Table 4.2. These are the two main equations in [2.13].

Variable	Constraint
$\mathbf{P_{ME1,n}}$	≥ 0
$\mathbf{P_{ME2,n}}$	≥ 0
$\mathbf{P_{DG,n}}$	≥ 0
$\mathbf{P_{ME1,n}}$	$\leq \eta_{P,ME1,min} \cdot Load_{max,ME1} \cdot P_{B,ME1}$
$\mathbf{P_{ME2,n}}$	$\leq \eta_{P,ME2,min} \cdot Load_{max,ME2} \cdot P_{B,ME2}$
$\mathbf{P_{DG,n}}$	$\leq \eta_{P,DG,min} \cdot Load_{max,DG} \cdot P_{B,DG}$

Table 4.1: Constraints on the decision variables

In the constraints the $\eta_{P,min}$ is the smallest total plant efficiency that occurs in the configuration for a particular component. The $\eta_{P,min}$ depends on a certain operation of the components, but it also influences the decision which operation to use in one particular operating mode. Every iteration performs these calculations in a certain order, before moving on to the next iteration step. The calculation step might take the 'old' plant efficiency and uses that value for the 'new' operation. This might result in slightly wrong values and can cause the engines to operate above their maximum allowable loading. To be safe, the lowest possible $\eta_{P,min}$ is therefore used in the constraints.

Variable	Dependency
$\mathbf{P_{EM,n}}$	$= P_{aux,n} - P_{DG,n}$
$\mathbf{P_{EM,n}}$	$= (P_{ME1,n} + P_{ME2,n}) - P_{D,n}$

Table 4.2: Dependencies between the decision variables

Another problem occurred with the total plant efficiency that changes for every operation. It depends on the values of the decision variables but at the same time influences the optimisation algorithm as to what the values of the decision variables should be. This creates an algebraic loop in the sheet that can cause problems in the iteration steps. Therefore at every solver iteration step 100 small sheet iterations are performed, before going to the next iteration step in the solver.

The stopping rule for the solver is set at 0,01%. If the last 5 iteration steps change the outcome by less than this percentage, the solver regards the result as the final solution.

4.3 Selecting initial values

Non-convex functions have the characteristic that there are multiple local minima, and that it is difficult to determine whether one of these local minima is actually the global minimum that is preferred. However, there are many different search algorithms that are good in finding a local minimum. It all depends on the starting point. From this point, a local minimum can be found. Therefore some insight is needed in the system and the objective function to find a set of initial values that can be used as a first estimate of the desired minimum point.

One way of selecting many different scenarios as an input for the gradient search method is to select multiple levels for each design parameter and perform a full combinatorial test of these levels. Every decision variable can for instance have a minimum value, a medium value and a maximum value based on the limitations of the installed equipment. A maximum of 10 distinct operating modes and 4 decision variables gives a total of 40 solvable variables. If three levels are chosen this leads to 3^{40} possible combinations resulting in an impractical number of solver tests to be carried out.
Another possibility is to take random combinations of the parameters and levels. The amount of test scenarios is then variable; the more possibilities that are tested, the greater the chance that a good result will be among them. The downside is that the closer the number of tested combinations is to 3^{40} the greater the chance that the best initial values will be selected. With fewer tests, there is no solid indication that a solution close to the global minimum is among them.
This paragraph describes how the initial values are selected and deals with methods to reduce the number of tests.

4.3.1 Orthogonal arrays

There are several ways of deciding which is a good first solution with the minimal amount of test scenarios. This study makes use of orthogonal arrays developed by Taguchi [Ross, 1996]. The use of orthogonal arrays provides a systematic way of testing many variable interactions in a system. It is a statistical method to cover all pairwise combinations of these variables using the least possible test scenarios. The statistical benefit of these arrays was developed mainly by Dr. Genichi Taguchi [Ross, 1996]. Therefore the method of this statistical testing of different variable combinations is also often referred to as the Taguchi Methods for experimental design.
The Taguchi method investigates how the different variables in a system affect the outcome. In this case the different variables are given in $\mathbf{X_n}$ and the outcome is the annual fuel consumption M_{fuel}. Instead of testing all possible combinations between the

variables, the Taguchi method tests pairs of combinations. This is a method to determine which variables influence fuel consumption the most with a minimum amount of testing. An orthogonal array is in fact just a two-dimensional table with the characteristic that every combination of the variables is represented at least once. The variables are presented in the different columns and the number of test cases *s* are presented in the rows. The entries in the table are the number of levels each variable can have. The tables have the quality that by choosing any two columns, all two-way combinations between all levels of all variables is presented at least once, after testing all cases in the array. To select the suitable array, first the number of decision parameters of the system is required. This is *m* and is known from the operational profile. Next, the number of levels must be selected. Based on these two parameters the suitable array can be selected. Small arrays can be done by hand, but larger ones require some deterministic algorithms and the process of elimination. The arrays in this study are found in [Ross, 1996].

4.3.2 Selecting suitable levels

The orthogonal array methods are based on no interactions between the variables; they should be linearly independent of each other. This is actually not the case within a singular operating mode. After all, the electric machine couples the main engines to the diesel generators through the electric power, in whichever direction. If one decision variable is chosen the others are fixed. Creating independent values for the levels for every decision variable will lead to infeasible solutions of the objective functions.
Within one operating mode the decision variables are dependent. However the different operating modes are independent of each other. The decision variables in $\mathbf{X_n}$ are dependent, but the different vectors of $\mathbf{X_n}$ are independent of each other. So the previously determined *m* variables are not all independent and an array with *m* variables is not suitable. In fact there are only *n* modes that are independent. But still all *m* variables must be represented in the array. For this reason *n sets* of variables are taken, with each set containing the 4 decision variables in $\mathbf{X_n}$.

The number of levels of the values is set to three. With the decision variables now taken as *n* sets, the number of solver tests for a full combinatorial that have to be carried out is reduced to 3^n with a maximum of 3^{10}. This is still a very large number for the Excel solver to handle. So a suitable orthogonal array must be selected.

Determining the values of the levels requires insight in the system. The limitations of the components were described in chapter 3. A logical first estimate would be to choose the minimum as 0, a medium value and a high value that for instance is the nominal brake power of an engine. However, this cannot be achieved because of the dependency between the variables.
Therefore it is decided not to take the limitations of each decision variable itself, but to take several pre-selected values for the complete vector $\mathbf{X_n}$. This is now the ‘variable’ in

the orthogonal array and the vectors $\mathbf{X_n}$ are combined with each other in the different test cases (see Table 4.3). Every value, or every level of $\mathbf{X_n}$, must be a feasible solution of the objective function itself.

Solver tests	X_1	X_2	...	X_n
1	1	1	...	1
2	1	2	...	2
3	1	3	...	3
...	...	...	...	...
s	3	3	...	1

Table 4.3: Layout of an orthogonal array

The selected levels are based on the different possibilities of the component configuration: fully diesel mechanic, a hybrid and a fully electric system. This gives a good spread between the possible values of $\mathbf{X_n}$. Both the extremes and an intermediate value are represented.

A fully diesel mechanic mode does not equip the electric machine in either PTO or PTI mode. All propulsion power is provided by the main engines and all electric power by the diesel generators. Diesel mechanic corresponds to number 1 in the arrays.

A hybrid mode can have either a PTO mode or a PTI mode and is corresponding to number 2 in the array. It is decided to run in PTI mode until a certain amount of propulsion power demand, selectable by the user and in PTO mode for all power above this level. The default value for this changeover parameter α is 25% of installed main engine power. This value is based on some research in various ship types that have a PTI motor installed for slow sailing. A PTI-booster mode will be selected if the main engine power is insufficient for the propulsion power demand.

In fully diesel electric mode the main engines are not in operation. All electric power and propulsion power is generated by the diesel generator sets and the propulsion power is then provided by the electric motor. This mode corresponds to a number 3 in the arrays.

What is initially selected as the installed configuration can limit the selected levels. For instance, the amount of installed $P_{B,DG}$ might not be sufficient to fulfil the P_D demand in a fully diesel electric mode. Selecting the suitable initial values for $\mathbf{X_n}$ is therefore limited to what limitations the selected configuration gives.

To summarise, the number of decision variables is still *m*, but in the orthogonal array this leads to *n* sets of 4 decision variables. The number of levels is 3, so according to [Ross, 1996] the following arrays should be selected:

n	Array
2, 3 or 4	L9
5, 6, 7 or 8	L18
9 or 10	L27

Table 4.4: Selection of suitable array

The digit in the array corresponds to the number of solver tests *s*. This means that the number of solver tests is reduced from 3^n to 9, 18 or 27, depending on *n*. These orthogonal arrays are presented in appendix C.

For up to ten operating modes, the minimum amount of test cases for still a uniformly distributed amount of combinations is 27. There is of course nothing against adding more initial solutions. This can make sense from an engineering point of view. While the orthogonal array takes all sorts of combinations between diesel mechanic, hybrid and diesel electric, it sometimes makes sense to take just one sort of configuration and test it for all modes.
From the used arrays can be seen that the diesel mechanic mode is already represented, since all first test cases are composed of number 1's. A full test case of just number 2's (hybrid mode) is not included, so that one is added to the array. Also, the hybrid mode always takes into account a PTI mode for slow sailing. This means that the electric machine and gearbox must be able to handle this energy flow. In many cases just a PTO generator will be installed, without a PTI function. An initial test case where just a PTO is installed is also added. A full diesel electric case is also not included in the standard array, but this is not really necessary. If both the main engines are not included in the desired configuration, according to [2.13] all propulsion power and electric power must be delivered by the diesel generators, resulting in a fully electric mode. Therefore all three standard arrays have two added solver test cases. This means the number of simultaneous solver tests that are carried out are:

n	Array	s
2, 3 or 4	L9	11
5, 6, 7 or 8	L18	20
9 or 10	L27	29

Table 4.5: Number of solver test cases

For every of the 11, 20 or 29 test cases the total annual fuel consumption is calculated. This will give a first intermediate result of the performance. The best ones are then put as an input for the actual optimisation method.

Running the solver takes a couple of seconds. Running 29 solvers for 4 different configurations can easily take up to 10 minutes. To prevent long calculation times at

every solution, it is decided to analyse which final solution resulted from which initial solution.
The complete optimisation procedure for all possible initial solutions from the arrays has been carried out several times for a range of different configurations and operation profiles. After extensive testing it was found that the majority of the best *final* solutions resulted from one of the best *initial* solutions.

Therefore only the best 5 are selected for further optimisation in the solver. This results 5 x 4 = 20 different end solutions for 4 different configurations, instead of 29 x 4 = 116 different solutions. Not only does this reduce the calculation time significantly, it also makes post-processing clearer.

All the described procedures still do not guarantee the end result will be the global minimum. There is always the possibility that the chosen initial results do not find the global minimum. Due to the fairly flat shapes of the sfc curves the function will not have these clear local minima. When testing these steps for many different kinds of operational profiles and configurations, the result provided by the algorithm always proved to be the global minimum (or at least very close to this minimum, within 0,01%).

4.4 Sensitivity analysis

Sensitivity analysis is very important in an optimisation problem. In general one would like to know what would happen to the found solution if several parameters and assumptions in the problem were changed. Often this kind of analysis is also called what-if analysis or post-optimality analysis. This shows how the uncertainty of the results can be assigned to the uncertainty of the input parameters. A stable result is one that when the input parameters are changed, the result does not change that much. The sensitive parameters are those that cannot be changed without changing the outcome. Finding these parameters is therefore an important part in interpreting the results.
Every decision variable in the solver model has so-called dual variables. These come from introduced variables in the solver algorithm.

First there are the reduced gradients (reduced costs in linear problems). These tell how much the parameter can be decreased in order to keep the optimal solution. If it is negative, the value can be increased. High values imply a stable solution.
Second there are Lagrange multipliers, already mentioned in paragraph 4.2.2. These are a measure of how the value is restricted by the constraints. It tells the rate at which the objective function can be decreased by slightly changing the value of the constraints. Or in other words, how does the result benefit from loosening the constraints. Nonzero values indicate that the decision variable is equal to its constraint. Moving the variable away from the constraint will worsen the objective function's result and relaxing the constraint will improve the result. These values apply to the simple upper and lower bound constraints, listed in Table 4.1.

Unfortunately there are more constraints to this problem that are not single linear bounds. These are the dependencies between the EM and the diesel engines listed in Table 4.2.
Changing one variable in $\mathbf{X}_n$ without changing the other three will result in an infeasible solution. This means that a single reduced gradient is always zero, implying that the corresponding variable cannot be changed since this will not result in a feasible solution. The Lagrange multipliers also will not provide any useful information. One single decision variable in $\mathbf{X}_n$ is always constrained, as long as the other variables do not change accordingly.
It would be better to change the parameters again as sets, like with the modification of the Taguchi arrays. However, a systematic way of changing this is not possible in the solver environment. And since the search algorithm was not developed from the base up, it is not really possible to alter the methods used to create the sensitivity analysis. This means that the sensitivity reports created by the Excel solver cannot be used.

Therefore the sensitivity analysis must be carried out manually. Not every parameter can be altered at will. The brake power of a main engine is fixed and a maximum loading is usually specified. The fuel consumption curve of each engine is based on fixed data, so these coefficients are really treated as constants. It is possible to influence this by selecting a larger or smaller engine, but this is already being compared to each other in the different configurations. The parameters that are really based on assumptions are the ones with the greatest uncertainty and therefore the ones with the potential to be the most sensitive (or the least stable). These might be the efficiencies of the gearbox (η_{GB} and its constituents η_t, η_b and η_s), generator and motor (η_{EM}) and the frequency converter (η_{FC}). The other main assumption is the change over from PTI to PTO mode in a hybrid system, parameter α. It is only used as an initial value to have some feasible initial results in the Taguchi arrays for an input into the search algorithm. However, if the objective function has many local minima, the initial starting point is very important for the algorithm to find the correct global minimum.

5 Sample configurations

The developed design methodology will be subjected to some real life examples to test its validity and plausibility. The presented results are only serve for information and should not be treated as binding information concerning MDT engine efficiency.
Four vessels where a hybrid system might be applicable will be tested. Some of these vessels are already built; others are in their design stage. The operational data required for input was not always available in detail. In those cases some assumptions have been made. The focus is not on how accurate these assumptions are, but on the plausibility of the results of the design algorithm.

5.1 Chemical tanker

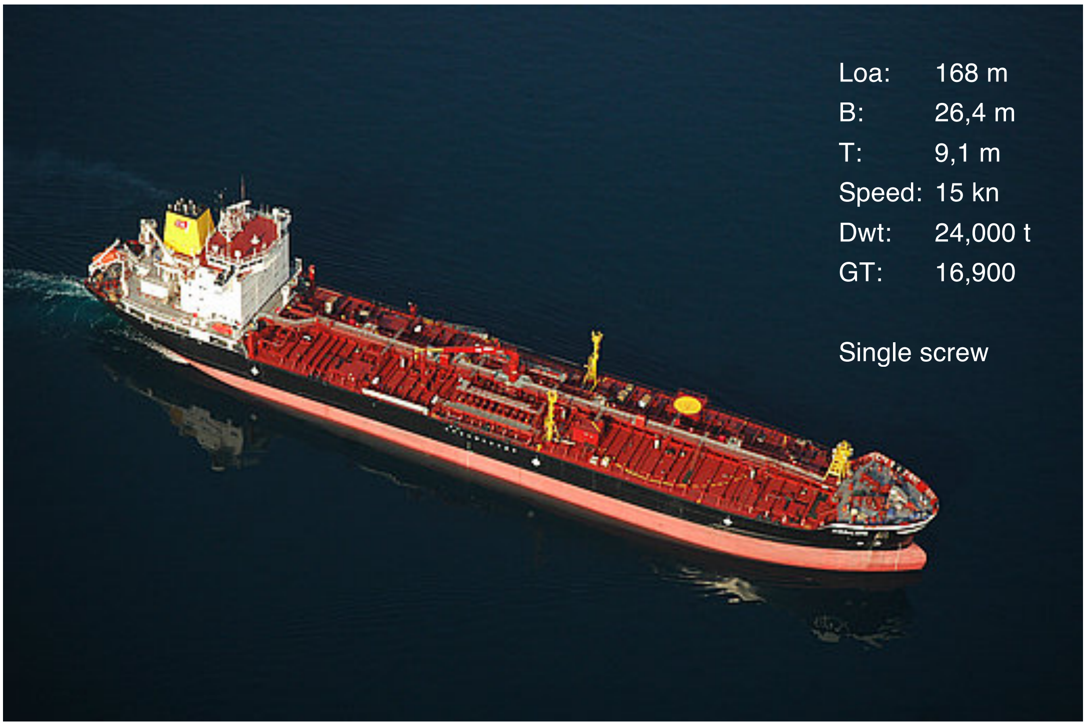

Figure 5.1: Chemical tanker reference vessel MT Avalon

Conventional cargo vessels usually feature a PTO in the form of a shaft generator. Chemical tankers often contain hazardous cargo and need a PTH function when the main propulsion fails. This is a requirement by some classification societies. Therefore these kinds of vessels can benefit from an electric machine that can operate in PTI and PTO mode. This particular vessel has an ice class, with the requirement of a PTI boost function for extra power in icy situations.

5.1.1 Operational profile

A tanker vessel runs on cruising speed for long periods of time. Generally there is not much manoeuvring time needed when going in and out of port. The port times of this vessel are split into two: discharging and loading. This vessel has cargo pumps for cargo discharging. When discharging, the electric power demand will be much higher than for loading.

An estimation of the operational profile was provided by the yard in delivered power P_D so that the algorithm has the correct values as an input. To complete the required input data, it is assumed that this vessel has about 30% yearly port time, which is a fair assumption for these types of vessels. The P_{aux} in loading and offloading conditions were specified. Only the continuous hotel load was specified, so assumptions based on [2.7] and [2.8] were made for manoeuvring and engine auxiliaries. An estimated 7% of the main engine power is assumed. The reference ship had a brake power of 8000 kW installed as the main engine, resulting in a $P_{aux,ME}$ of 560 kW. This is valid for all speeds. The electrical load during manoeuvring is set to 130% normal electric load. The continuous hotel load is 360 kW. This means the electrical load when manoeuvring is $1.3 \cdot (360 + 560) = 1196$ kW.

This gives the total operational profile, listed in Table 5.1 and Figure 5.2.

		P_D [kW]	P_{aux} [kW]	Time [hr/year]
Mode 1	**Port discharge**	0	2160	1139
Mode 2	**Port loading**	0	360	1139
Mode 3	**Manoeuvring**	340	1196	263
Mode 4	**< 5 kn**	340	920	87
Mode 5	**9 kn**	1991	920	307
Mode 6	**11 kn**	2910	920	307
Mode 7	**13 kn**	4851	920	1226
Mode 8	**14 kn**	5821	920	4292

Table 5.1: Operational profile of the chemical tanker

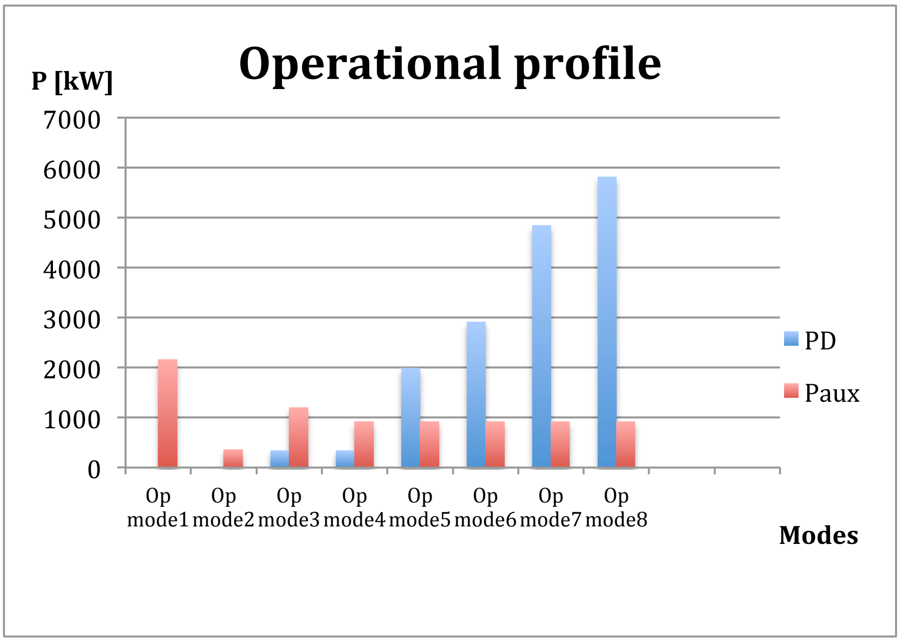

Figure 5.2: Operational profile of the chemical tanker

5.1.2 Suggested configurations

In this power range there are several main engine possibilities. The goal of these suggested configurations is not to compare different engines (this can easily be done by looking at the sfc data in appendix B), but to compare different engine room layouts and power distribution. Therefore a better comparison can be made by selecting one engine type. A suitable bore type is the 32/44 CR engine, which is available in line and V-type.

Configuration 1

This configuration will feature a single 14-cylinder V32/44 CR engine with a brake power of 7840 kW. This will be capable of delivering the largest propulsion power including auxiliary power in the form of a PTO.

The diesel generator sets will have to provide for the high auxiliary load in mode 1 when the vessel is discharging in port. Two 6 cylinder L21/31 DG's are sufficient, but one extra is installed for redundancy. One or two larger ones were considered, but the lower loading point in the other operating modes and higher capital costs made this choice less favourable than the L21/31 engines. An overview of this data is presented in Table 5.2 and Figure 5.3. Note that the electrical power to and/or from the electric machine does not have a direction. Whether it will operate in PTO or PTI mode will be determined by the design algorithm. This is also the case for all other tested configurations that feature an electric machine.

The electric machine can be an induction motor in combination with a PWM converter. This converter should transfer power in both PTO and PTI mode, so an active front end is required. The total estimated investment costs are based on Table 3.3 and are:

Main engine 1	2,195	k€	
Diesel generators	1,584	k€	
Electric machine	50	k€	
Gearbox	242	k€	
PWM Converter	135	k€	
CPP + shaftline	998	k€	+
Total	5,204	k€	

Again it must be mentioned that these costs might not be very accurate, but only serve as a comparison between the tested configurations.

The results are presented in Table 5.3. The results including gearbox and electric machine efficiency are presented in appendix D for all tested configurations.

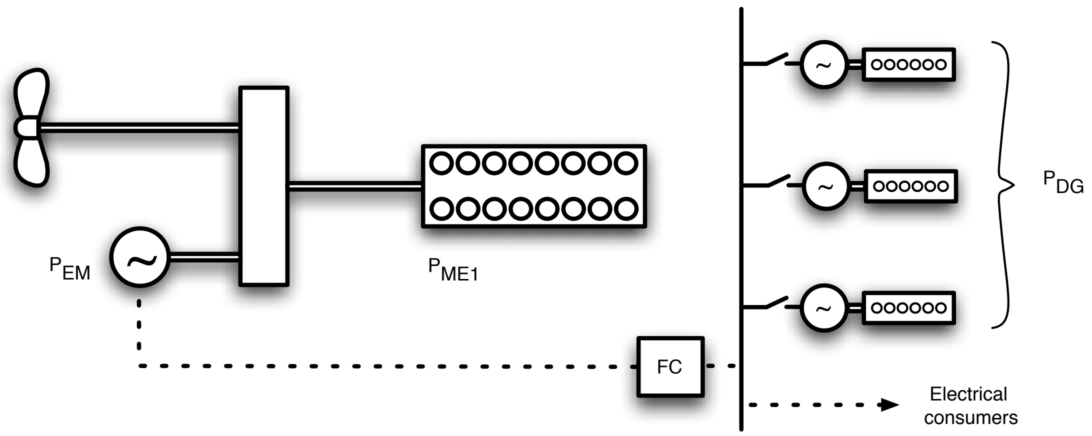

Figure 5.3: Tanker configuration 1 layout

Main engine 1		
Number	1	
Type	V32_44CR	
Cylinders	14	
Cyl power	560	kW
Max load	90%	
Pb engine	7840	kW
n nom	750	rpm

Diesel generator sets		
No. DG	3	
Type	L21_31_G	
Cylinders	6	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1320	kW
Pbrake (2)	2640	kW
Pbrake (3)	3960	kW
Pb total	3960	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	1000	kW

Table 5.2: Input data for tanker configuration 1

Best result Configuration 1

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	2160	2274	86,1%	189,1	35,8	1139	489,8	900	2
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	360	379	28,7%	203,3	12,8	1139	87,7	900	1
	EM	0	0						0	
Op mode3	ME 1	0	0	0,0%	0,0	0,0	263	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	1536	1698	64,3%	190,2	26,9	263	84,9	900	2
	EM	-340	-417						720	
Op mode4	ME 1	0	0	0,0%	0,0	0,0	87	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	87	0,0	0	
	DG	1260	1407	53,3%	192,2	22,5	87	23,5	900	2
	EM	-340	-417						720	
Op mode5	ME 1	2911	3129	39,9%	188,5	42,1	307	181,1	552	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	0	0	0,0%	0,0	0,0	307	0,0	0	0
	EM	920	978						530	
Op mode6	ME 1	3830	4083	52,1%	183,4	53,5	307	229,9	603	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	0	0	0,0%	0,0	0,0	307	0,0	0	0
	EM	920	978						579	
Op mode7	ME 1	5771	6095	77,7%	178,6	77,8	1226	1334,9	690	
	ME 2	0	0	0,0%	0,0	0,0	1226	0,0	0	
	DG	0	0	0,0%	0,0	0,0	1226	0,0	0	0
	EM	920	978						662	
Op mode8	ME 1	6741	7098	90,5%	175,9	89,2	4292	5357,3	750	
	ME 2	0	0	0,0%	0,0	0,0	4292	0,0	0	
	DG	0	0	0,0%	0,0	0,0	4292	0,0	0	0
	EM	920	978						720	

Total fuel consumption 7789,1 t/yr

Table 5.3: Results for tanker configuration 1

Having an engine that is capable of delivering both propulsion power and PTO power results in a PTO being chosen for the transit modes 5, 6, 7 and 8. This is a result that could be expected. Since the electric machine also has a PTI function, it can be selected for the low propulsion demands in modes 3 and 4. If this propulsion power would be provided by the main engines, it would lead to an overall increase of 20 t/yr.

An unexpected value occurs in operating mode 8, where the engine loading exceeds the set maximum of 90% MCR. This is due to the drawback that the solver cannot handle logic functions in the constraints. The gearbox and electric machine efficiency depend on the load, so they change according to the iteration steps in the solver algorithm. But in the constraints setup these η_{GB} and η_{EM} must be an appropriate fixed value. This value is chosen to be the lowest η_{TRM} and η_{EM} that occurs in the previous iteration step, although it might change in the next step. This can lead to a higher loading than the engine is allowed to. In the result tables all results are correct, but the solver might use an 'old' efficiency in its constraint calculations.

Configuration 2

In this configuration the same brake power is installed over two main engines, the 7 cylinder L32/44 CR. This might be a more flexible layout for the lower propulsion power demands. At transit speed both engines will run. The same DG configuration is chosen as in configuration 1.

Two 7-cylinder L-type engines are of course more expensive than one single V-type engine with the same brake power. The gearbox should be larger with extra wheels and pinions for the extra main engine. This would add 15% to the price. The frequency converter would again need an active front end, but the electric machine can be the same type. This results in the following investment costs:

Main engine 1	1,411	k€	
Main engine 2	1,411	k€	
Diesel generators	1,584	k€	
Electric machine	65	k€	
Gearbox	279	k€	
PWM Converter	175	k€	
CPP + shaftline	998	k€	+
Total	5,923	k€	

The final solution of this tested configuration is presented in Table 5.5.

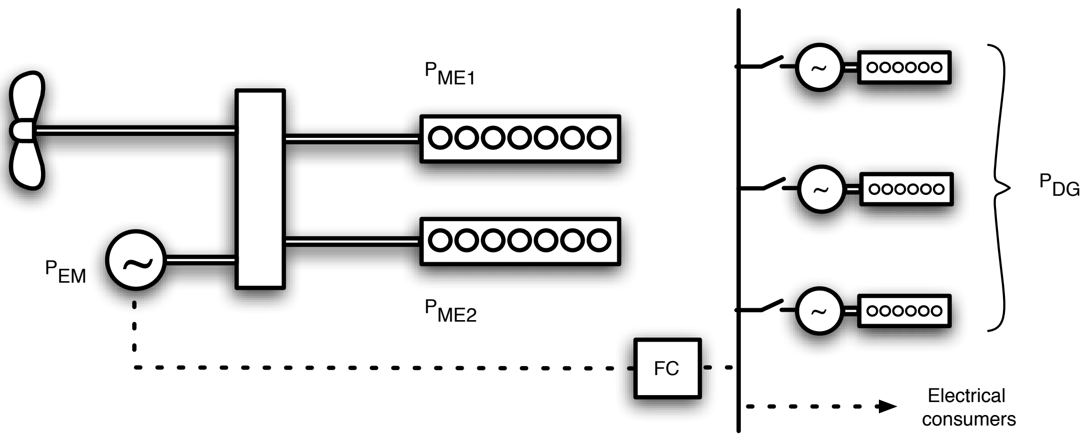

Figure 5.4: Tanker configuration 2 layout

Main engine 1			Main engine 2		
Number	1		Number	1	
Type	L32_44CR		Type	L32_44CR	
Cylinders	7		Cylinders	7	
Cyl power	560	kW	Cyl power	560	kW
Max load	90%		Max load	90%	
Pb engine	3920	kW	Pb engine	3920	kW
n nom	750	rpm	n nom	750	rpm

Diesel generator sets			Other input		
No. DG	3		PTI selected if PD <	0,25	PB ME
Type	L21_31_G		EM nominal speed	720	
Cylinders	6		Include FC	Yes	
Cyl power	220	kW			
Max load	100%		Combinator curve		
Pbrake (1)	1320	kW	Constant n below	60%	ne
Pbrake (2)	2640	kW	Constant n above	80%	load
Pbrake (3)	3960	kW			
Pb total	3960	kW	Nominal efficiencies		
Eta gen	0,95		Shaft	0,99	
n nom	900	rpm	EM	0,96	
			FC	0,98	
			trm	0,96	
			Maximum EM power	1300	kW

Table 5.4: Input data for tanker configuration 2

Best result Configuration 2

		Best Delivered kW	= 2 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	2160	2274	86,1%	189,1	35,8	1139	489,8	900	2
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	360	379	28,7%	203,3	12,8	1139	87,7	900	1
	EM	0	0						0	
Op mode3	ME 1	0	0	0,0%	0,0	0,0	263	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	1536	1710	64,8%	190,2	27,1	263	85,5	900	2
	EM	-340	-429						720	
Op mode4	ME 1	0	0	0,0%	0,0	0,0	87	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	87	0,0	0	
	DG	1260	1420	53,8%	192,1	22,7	87	23,7	900	2
	EM	-340	-429						720	
Op mode5	ME 1	1991	2116	54,0%	182,9	55,3	307	118,8	611	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	920	968	73,4%	189,2	30,5	307	56,2	900	1
	EM	0	0						0	
Op mode6	ME 1	2910	3072	78,4%	178,3	78,2	307	168,2	691	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	920	968	73,4%	189,2	30,5	307	56,2	900	1
	EM	0	0						0	
Op mode7	ME 1	2886	3086	78,7%	178,1	78,5	1226	673,8	693	
	ME 2	2886	3086	78,7%	178,1	78,5	1226	673,8	693	
	DG	0	0	0,0%	0,0	0,0	1226	0,0	0	0
	EM	920	978						665	
Op mode8	ME 1	3371	3591	91,6%	176,0	90,3	4292	2712,7	750	
	ME 2	3371	3591	91,6%	176,0	90,3	4292	2712,7	750	
	DG	0	0	0,0%	0,0	0,0	4292	0,0	0	0
	EM	920	978						720	

Total fuel consumption 7859,1 t/yr

Table 5.5: Results for tanker configuration 2

It can be seen that there are similar loading points of all engines as configuration 1, resulting in a similar overall fuel consumption. As in configuration 1, modes 3 and 4 are with a PTI. Running these modes in PTO will lead to only a marginal difference in fuel consumption, in contrast to configuration 1. However, this would lead to a more practical installation of electrical equipment and gearbox. For instance, a gearbox designed for just PTO without PTI does not require hardened teeth on both sides.
There is a difference in the use of a PTO in mode 5 and 6. In this configuration, operation with a PTO in mode 5 and 6 would cause one single engine to overload, so both engines have to run on half load with an increased sfc.
Modes 7 and 8 have the largest timeshare, so differences in these modes would have the largest effect on the absolute fuel consumption. Although the selected operation mode of the electric machine is the same, the larger gearbox losses result in more fuel consumption for these modes.

Overall, the gearbox efficiency is less compared to configuration 1, as can be seen in appendix D. The potential of having two engines run independently is not utilised with this profile and engine selection, while the gearbox efficiency brings a negative effect.

Configuration 3

The same two engines as in configuration 2 are selected, but with a different number of cylinders creating a father/son type layout. The total installed main engine brake power is still the same. Also the installed DG sets remain the same.

The investment costs are the same as in configuration 2, but with different costs for the main engines:

Main engine 1	1,612	k€	
Main engine 2	1,210	k€	
Diesel generators	1,584	k€	
Electric machine	65	k€	
Gearbox	279	k€	
PWM Converter	175	k€	
CPP + shaftline	998	k€	+
Total	5,923	k€	

The results for this configuration are shown in

Table 5.7.

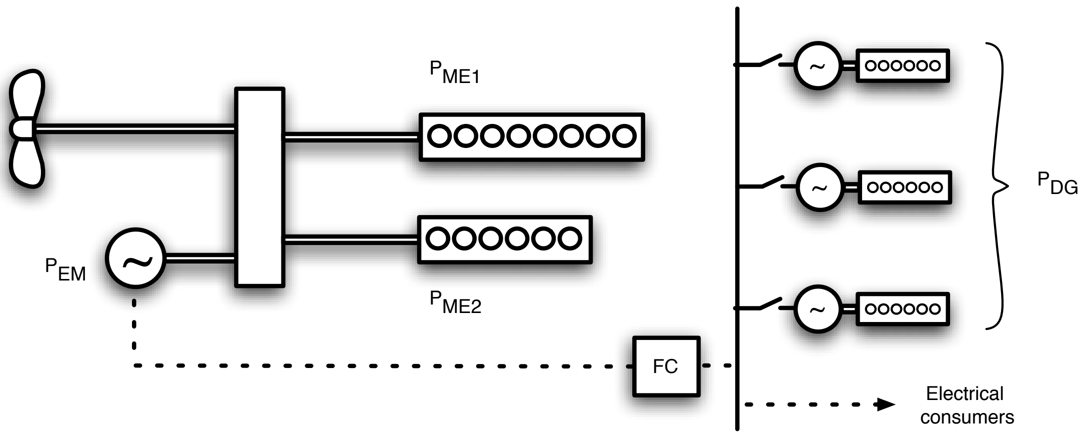

Figure 5.5: Tanker configuration 3 layout

Main engine 1		
Number	1	
Type	L32_44CR	
Cylinders	8	
Cyl power	560	kW
Max load	90%	
Pb engine	4480	kW
n nom	750	rpm

Main engine 2		
Number	1	
Type	L32_44CR	
Cylinders	6	
Cyl power	560	kW
Max load	90%	
Pb engine	3360	kW
n nom	750	rpm

Diesel generator sets		
No. DG	3	
Type	L21_31_G	
Cylinders	6	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1320	kW
Pbrake (2)	2640	kW
Pbrake (3)	3960	kW
Pb total	3960	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	1300	kW

Table 5.6: Input data for tanker configuration 3

Best result Configuration 3

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	2160	2274	86,1%	189,1	35,8	1139	489,8	900	2
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1139	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1139	0,0	0	
	DG	360	379	28,7%	203,3	12,8	1139	87,7	900	1
	EM	0	0						0	
Op mode3	ME 1	1536	1693	37,8%	189,9	40,2	263	84,6	542	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	0	0	0,0%	0,0	0,0	263	0,0	0	0
	EM	1196	1272						521	
Op mode4	ME 1	1260	1394	31,1%	196,0	34,1	87	23,8	508	
	ME 2	0	0	0,0%	0,0	0,0	87	0,0	0	
	DG	0	0	0,0%	0,0	0,0	87	0,0	0	0
	EM	920	978						488	
Op mode5	ME 1	2911	3123	69,7%	180,7	70,5	307	173,3	665	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	0	0	0,0%	0,0	0,0	307	0,0	0	0
	EM	920	978						638	
Op mode6	ME 1	3830	4081	91,1%	175,9	89,7	307	220,4	750	
	ME 2	0	0	0,0%	0,0	0,0	307	0,0	0	
	DG	0	0	0,0%	0,0	0,0	307	0,0	0	0
	EM	920	978						720	
Op mode7	ME 1	3298	3522	78,6%	178,1	78,4	1226	769,3	692	
	ME 2	2473	2649	78,9%	178,0	78,6	1226	578,3	692	
	DG	0	0	0,0%	0,0	0,0	1226	0,0	0	0
	EM	920	978						665	
Op mode8	ME 1	3852	4100	91,5%	176,0	90,2	4292	3096,7	750	
	ME 2	2889	3083	91,7%	176,0	90,4	4292	2328,7	750	
	DG	0	0	0,0%	0,0	0,0	4292	0,0	0	0
	EM	920	978						720	

Total fuel consumption 7852,5 t/yr

Table 5.7: Results for tanker configuration 3

As in configuration 1 and 2, the auxiliary power in modes 1 and 2 are delivered by the DG's. It is interesting to see that with this father/son configuration modes 3 and 4 are now operated with a PTO. The difference with configuration 2 is however neglectable and it can be shown that this difference is within the set accuracy of 0,01%. So modes 3 and 4 with PTO or PTI are quite similar for both configuration 2 and 3. However, in configuration 3 one must be careful with interpreting the engine load of ME1. In this particular case it would make more sense to run the smaller 6-cylinder engine instead of the 8-cylinder engine for modes 3 and 4. The difference is around 10 t/yr. This is not a substantial increase but nonetheless larger than the accuracy setting. It can be explained by the initial value selection in the orthogonal arrays. These are based on rules and one of these rules is that the main engine loading always starts with P_{ME1}. One method is to select the 6-cylinder engine as ME1 and the 8-cylinder engine as ME2. This would work for modes 3 and 4, but the changes in the initial values also influence the other operating modes. Specifically, in mode 6 this engine configuration would not allow single engine operation. Therefore it must be noted that this is a shortcoming that has to be monitored in the results. The influence of the initial values following the orthogonal arrays can be seen here; a different configuration with the same father/son engines would lead to two different local minima.

In mode 6 the larger engine is capable of delivering the PTO power, although this would increase the engine loading slightly beyond the set 90% MCR power. This effect has been described in configuration 1 of this vessel.
In mode 7 and 8 both engines run with a different absolute power, but proportional to their installed brake power. So they run with the same relative loading, resulting in the same sfc and the same fuel consumption per cylinder.

After testing and discussing these three configurations, it can be seen that configuration 1 results in the lowest annual fuel consumption. The lower gearbox efficiency in configuration 2 and 3 compared to configuration 1 counteract the better relative loading of the two engines.
Next to this, it can be noted that the PTI in modes 3 and 4 does not reduce fuel consumption significantly, while it does require a more complex installation (but not considered here). Based on these input settings and boundary conditions the most practical installation is a diesel mechanic system with a PTO. The PTI is only required as a standby torque boost for when this vessel is operating in heavy ice. This is however not included in the stated operational profile, so the potential for a hybrid system is low.

5.1.3 Sensitivity analysis

In configuration1 variations in the changeover parameter α were made for a check of the robustness of this result. A higher α means a PTI will be selected for mode 5 and 6 in the initial results from the orthogonal arrays. However, in the end results the PTI was only selected for modes 3 and 4. This suggests that the minimum found with the default setting α = 0,25 is the same local minimum as with other values of α. Changing this parameter also did not influence the difference in fuel consumption compared to configurations 2 and 3. This means α is not a sensitive parameter.
Slightly higher or lower values for the assumed efficiencies such as $\eta_{EM,0}$, η_{FC}, and $\eta_{GB,0}$ changed the outcome proportionally as expected, but did not change the final choice of configuration. This means these parameters are also not considered sensitive and configuration 1 is a robust solution.

Next to variations in the assumed variables, changes can also be made to the given data in the operational profile. It could be that to maintain the same speed, the actual P_D is higher than estimated due to excessive fouling, more icy conditions or heavier seas. This could result that in modes 7 and 8 the PTO power can no longer be delivered by the main engines. The DG's must now deliver the P_{aux}, leading to a higher overall fuel consumption. The same changes occur when the P_{aux} is slightly higher than expected. Then the main engines are not able to deliver both propulsion and auxiliary power in modes 7 and 8.
This is the case for all three tested configurations, because the installed brake power is equal. This means a larger main engine should be installed with higher investment costs. Another option is to set the NCR point to 93% MCR instead of 90%, but the question is whether the owner would want this. In any case, the single V-engine is better than two main engines.

5.2 AHTS

Figure 5.6: AHTS reference vessel

AHTS vessels have a very variable operational profile for various operational requirements. The vessel has a high share of manoeuvring modes compared to conventional cargo vessels. Often a high bollard pull is required for tug operations. In many cases a fully diesel electric system is installed when these vessels have a DP system. This is because of the high electric load for the multiple thrusters.
The reference vessel has not been built yet, but initially a full diesel electric system is designed. This paragraph will compare a diesel electric system to a hybrid system.

5.2.1 Operational profile

The reference AHTS vessel had 12 different modes specified. However, this study limits the number of distinct operating modes to a maximum of 10. This means that some modes have to be combined. Its DP and standby operation were specified for low, medium and high environmental conditions. These were combined to a low and high condition, so a total of 10 modes are specified. This example has well specified required power, both for propulsion and electric demand. Therefore not many assumptions had to be made for estimating the auxiliary demand. The profile is described in Table 5.8 and displayed in Figure 5.7.

The high P_{aux} in the DP modes is because of the 4 thrusters this vessel has, next to the two main propellers. This is also the case when the vessel is in anchor handling mode. In addition, a large electric load is required to operate the winches.
The bollard pull is the largest propulsion demand. In this case there is no advance velocity to the propeller. Since the advance ratio J is approximately zero, the torque that the engine has to deliver is very high. For these calculations it is assumed that the 9600 kW is the required delivered power P_D.

		P_D [kW]	P_{aux} [kW]	Time [hr/year]
Mode 1	**Port**	0	150	438
Mode 2	**Transit 16 kn.**	4500	650	3854
Mode 3	**Transit towing**	5000	1050	263
Mode 4	**Anchor handling**	4000	3050	964
Mode 5	**Bollard pull**	9600	1150	88
Mode 6	**DP low**	210	2490	1050
Mode 7	**DP high**	4100	3550	263
Mode 8	**Standby low**	420	890	1314
Mode 9	**Standby high**	1000	1120	438
Mode 10	**Fire Fighting**	4500	5100	88

Table 5.8: Operational profile of the AHTS vessel

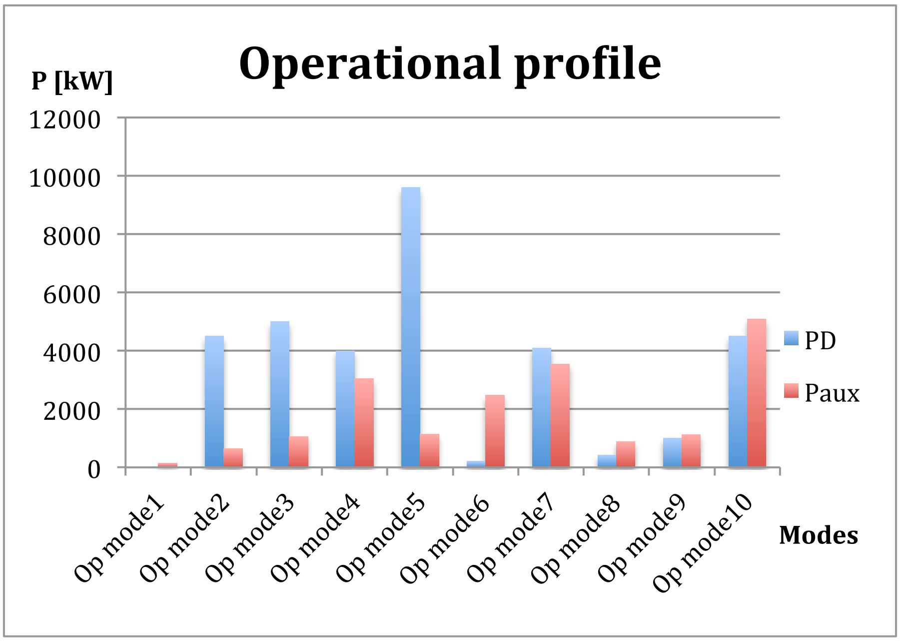

Figure 5.7: Operational profile of the AHTS vessel

5.2.2 Suggested configurations

This variable profile in combination with a DP system often suggests a full diesel electric system. This test case will investigate whether a hybrid system is a viable option.
A DP system sets requirements to the engine room layout such as extra redundancy and separate engine rooms. These requirements are not further treated in the choice of configuration.

Configuration 1

First, the diesel electric system will be tested as a basis for comparison of other configurations. No main engines will be installed, because all power must come from diesel generator sets. A maximum of four DG's can be selected. It is decided that three engines should be enough to provide for the largest power demand. A fourth one can then be standby or in maintenance. Two engines should be capable of providing the power in the transit modes 2 and 3. A third one will be needed in the occasion for bollard pull in mode 5 or fire fighting mode in mode 10. Four 8-cylinder L32/44 CR diesel generator sets are selected with a total engine brake power of 17,920 kW. Table 5.9 gives an overview of the input data. Table 5.10 gives the results for this configuration. The indicated power in the tables is the total power of the two shafts combined. Full results including the gearbox and EM efficiency are again presented in appendix D.

A single stage reduction gearbox will be used for the electric motor. Since this in only in motoring mode, no active front end is required for the converter. The displayed cost price is the total price for the symmetrical two shafts. The total estimated investment costs are:

Diesel generators	6,451	k€	
Electric machines	520	k€	
Gearboxes	312	k€	
PWM Converter	1,248	k€	
CPP + shaftline	1,646	k€	+
Total	9,819	k€	

Diesel generator sets		
No. DG	4	
Type	L32_44CR_G	
Cylinders	8	
Cyl power	560	kW
Max load	100%	
Pbrake (1)	4480	kW
Pbrake (2)	8960	kW
Pbrake (3)	13440	kW
Pbrake (4)	17920	kW
Pb total	17920	kW
Eta gen	0,965	
n nom	720	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	10400	kW

Table 5.9: Input data for AHTS vessel configuration 1

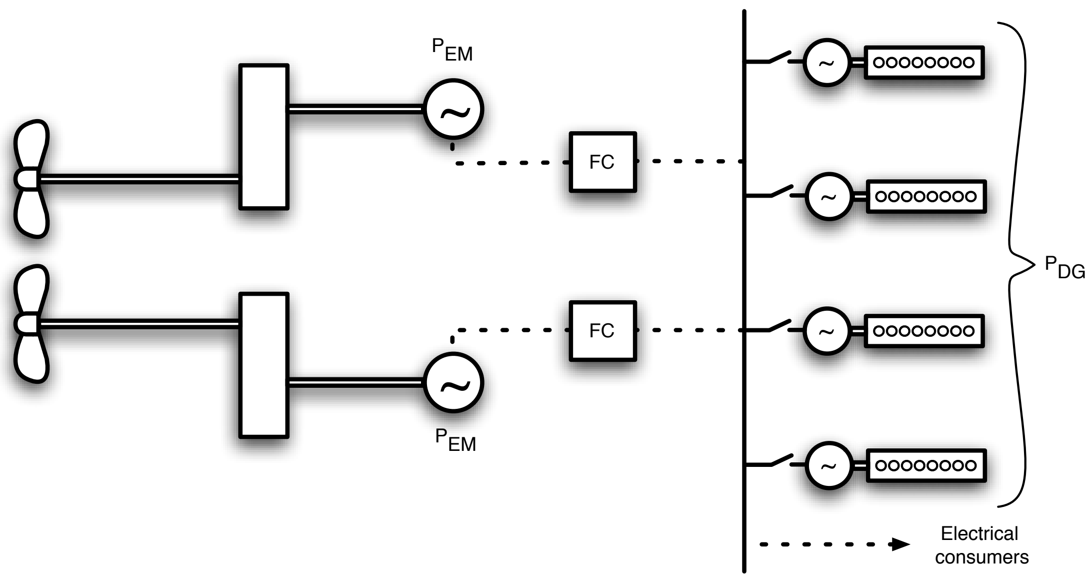

Figure 5.8: AHTS vessel configuration 1 layout

Best result Configuration 1

		Best	= 1							
		Delivered	Brake	load	sfc	Mf cyl	time	fuel	speed	DG's
		kW	kW	%	g/kWh	(kg/hr)	hr/yr	t/yr	rpm	running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	150	155	3,5%	526,3	10,2	438	35,8	720	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	3854	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3854	0,0	0	
	DG	5150	5836	65,1%	181,1	66,0	3854	4072,9	720	2
	EM	-4500	-4982						720	
Op mode3	ME 1	0	0	0,0%	0,0	0,0	263	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	6050	6810	76,0%	179,8	76,5	263	321,9	720	2
	EM	-5000	-5521						720	
Op mode4	ME 1	0	0	0,0%	0,0	0,0	964	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	964	0,0	0	
	DG	7050	7765	86,7%	175,8	85,3	964	1315,9	720	2
	EM	-4000	-4443						720	
Op mode5	ME 1	0	0	0,0%	0,0	0,0	88	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	10750	12081	89,9%	175,8	88,5	88	186,9	720	3
	EM	-9600	-10508						720	
Op mode6	ME 1	0	0	0,0%	0,0	0,0	1050	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1050	0,0	0	
	DG	2700	2958	66,0%	181,0	66,9	1050	562,2	720	1
	EM	-210	-365						720	
Op mode7	ME 1	0	0	0,0%	0,0	0,0	263	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	7650	8395	93,7%	176,4	92,5	263	389,4	720	2
	EM	-4100	-4551						720	
Op mode8	ME 1	0	0	0,0%	0,0	0,0	1314	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1314	0,0	0	
	DG	1310	1538	34,3%	192,7	37,0	1314	389,5	720	1
	EM	-420	-594						720	
Op mode9	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	2120	2424	54,1%	182,9	55,4	438	194,1	720	1
	EM	-1000	-1219						720	
)p mode10	ME 1	0	0	0,0%	0,0	0,0	88	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	9600	10447	77,7%	178,6	77,8	88	164,2	720	3
	EM	-4500	-4982						720	

Total fuel consumption 7632,9 t/yr

Table 5.10: Results for AHTS vessel configuration 1

A full electric concept does not show any surprises in terms of distribution of power. The highest load is for the bollard pull in mode 5 and can be delivered by three DG's. A 7-cylinder version of this engine was also tested. The overall fuel consumption will drop about 8 t/yr, but this also results that all four DG's would be used to provide for the power in modes 5 and 10. The question is whether this bollard pull or this fire-fighting mode will actually be used for all that time. For this short time, it would not be a big problem running on all four DG's.
Regarding investment costs, this does have a significant influence; the costs of the diesel generator sets can be reduced by a total of 806 k€.

For harbour mode 1 the engine loading is very low. For these low loads it is better to install a smaller harbour generator set but this is not taken into account here.

Configuration 2

This variable load profile might need an extended operating envelope at low engine speeds. An engine with a sequential turbocharger has a power reserve for manoeuvring or DP modes. This configuration will have enough installed main engine power to handle the high loads in modes 5 and 10. For this reason the maximum allowable engine loading is set to 100%, since it will not be used frequently. The DG power will be enough to provide for all auxiliary power. Four DG's will be installed to flexibility in the number of engines running.

The gearbox in this configuration requires an extra set of wheels and pinions for the electric motor. The converter does need an active front end:

Main engines 1	4,950	k€	
Diesel generators	2,816	k€	
Electric machines	170	k€	
Gearboxes	386	k€	
PWM Converters	459	k€	
CPP + shaftlines	1,646	k€	+
Total	10,427	k€	

This might not be a very energy efficient configuration but it will be interesting to see the difference between configurations with more electric power. Table 5.12 gives the results for this configuration.

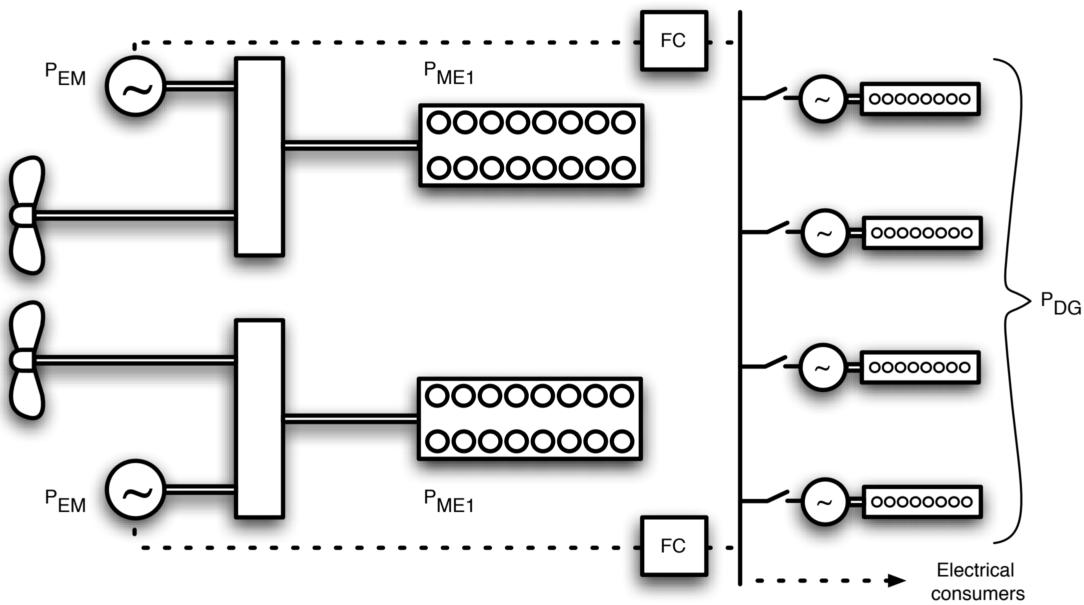

Figure 5.9: AHTS vessel configuration 2 layout

Main engine 1		
Number	2	
Type	V28_33D_STC_Ferry	
Cylinders	16	
Cyl power	455	kW
Max load	100%	
Pb engine	7280	kW
Pb total	14560	kW
n nom	1000	rpm

Diesel generator sets		
No. DG	4	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pbrake (4)	7040	kW
Pb total	7040	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	5500	kW

Table 5.11: Input data for AHTS vessel configuration 2

Best result Configuration 2

		Best	= 1							
		Delivered	**Brake**	**load**	**sfc**	**Mf cyl**	**time**	**fuel**	**speed**	**DG's**
		kW	**kW**	**%**	**g/kWh**	**(kg/hr)**	**hr/yr**	**t/yr**	**rpm**	**running**
Op mode1	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	150	158	9,0%	260,0	5,1	438	18,0	900	1
	EM	0	0						0	
Op mode2	ME 1	5150	5467	37,6%	196,5	67,1	3854	4140,3	721	
	ME 2	0	0	0,0%	0,0	0,0	3854	0,0	0	
	DG	0	0	0,0%	0,0	0,0	3854	0,0	0	0
	EM	650	735						519	
Op mode3	ME 1	6050	6412	44,0%	194,2	77,8	263	327,5	761	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	0	0	0,0%	0,0	0,0	263	0,0	0	0
	EM	1050	1153						548	
Op mode4	ME 1	7050	7501	51,5%	192,8	90,4	964	1394,3	802	
	ME 2	0	0	0,0%	0,0	0,0	964	0,0	0	
	DG	0	0	0,0%	0,0	0,0	964	0,0	0	0
	EM	3050	3253						577	
Op mode5	ME 1	10750	11287	77,5%	190,3	134,3	88	189,1	919	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	0	0	0,0%	0,0	0,0	88	0,0	0	0
	EM	1150	1258						661	
Op mode6	ME 1	0	0	0,0%	0,0	0,0	1050	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1050	0,0	0	
	DG	2700	2976	84,5%	189,0	35,1	1050	590,4	900	2
	EM	-210	-337						720	
Op mode7	ME 1	7650	8138	55,9%	192,4	97,9	263	411,9	824	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	0	0	0,0%	0,0	0,0	263	0,0	0	0
	EM	3550	3780						593	
Op mode8	ME 1	0	0	0,0%	0,0	0,0	1314	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1314	0,0	0	
	DG	1310	1534	87,2%	189,2	36,3	1314	381,5	900	1
	EM	-420	-568						720	
Op mode9	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	2120	2439	69,3%	189,6	28,9	438	202,5	900	2
	EM	-1000	-1197						720	
Op mode10	ME 1	9600	10206	70,1%	192,6	122,9	88	173,0	888	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	0	0	0,0%	0,0	0,0	88	0,0	0	0
	EM	5100	5421						640	

Total fuel consumption 7828,5 t/yr

Table 5.12: Results for AHTS vessel configuration 2

A lower overall sfc of the V28/33 engine compared to the L21/31 engine (see appendix B), results in as much main engine power as possible. This means a PTO in modes 2, 3, 4, 5, 7 and 10. Only where the required propulsion power P_D is sufficiently low, like in modes 6, 8 and 9, is the low loaded main engine less favourable than the L21/31 diesel generator sets.

Although a full diesel electrical configuration has more nominal losses, the flexibility in the number of running DG's combined with this varying operational profile gives it potential. Given the L32/44 CR engines in configuration 1 with their lower overall sfc curves, it can be seen that this is indeed the case.

Since this mode is capable of handling all power in diesel mechanic mode, it might be interesting to see how much fuel this would consume. It would increase fuel consumption by about 305 t/yr (not shown here), an increase of 3,9% over the hybrid mode and an increase of 6,3% over the diesel electric mode in configuration 1.

Configuration 3

Compared to configuration 2 with quite a lot of installed engine power, it might be better to reduce the main engine power for modes 2 and 3 where the vessel is in transit. The main engines can be much smaller. Two 9-cylinder L27/38 engines will suffice. This means that the installed DG power must be increased to deliver the auxiliary power required for the boost in mode 5.

This configuration has a larger electric machine and needs a larger converter. This leads to the following investment costs:

Main engines 1	2,203	k€	
Diesel generators	2,816	k€	
Electric machines	278	k€	
Gearboxes	386	k€	
PWM Converters	749	k€	
CPP + shaftlines	1,646	k€	+
Total	8,078	k€	

The results of this configuration are presented in Table 5.14.

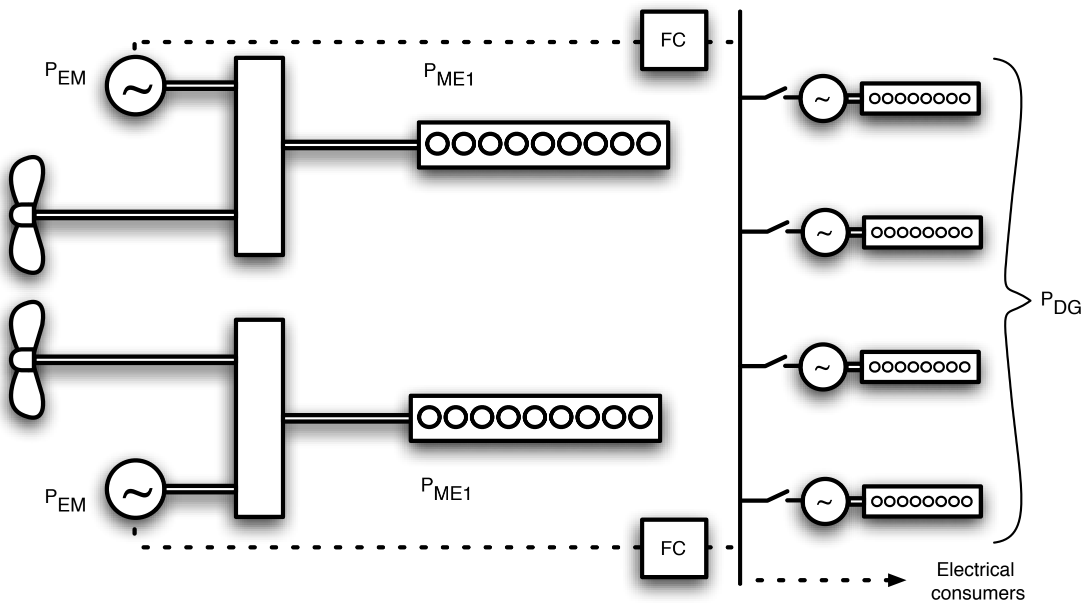

Figure 5.10: AHTS vessel configuration 3 layout

Main engine 1		
Number	2	
Type	L27_38	
Cylinders	9	
Cyl power	340	kW
Max load	90%	
Pb engine	3060	kW
Pb total	6120	kW
n nom	800	rpm

Diesel generator sets		
No. DG	4	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pbrake (4)	7040	kW
Pb total	7040	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	5550	kW

Table 5.13: Input data for AHTS vessel configuration 3

Best result Configuration 3

		Best Delivered kW	= 3 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	150	158	9,0%	260,0	5,1	438	18,0	900	1
	EM	0	0						0	
Op mode2	ME 1	4500	4711	77,0%	184,8	96,8	3854	3356,1	733	
	ME 2	0	0	0,0%	0,0	0,0	3854	0,0	0	
	DG	650	684	38,9%	196,8	16,8	3854	519,1	900	1
	EM	0	0						0	
Op mode3	ME 1	5000	5228	85,4%	185,0	107,5	263	254,4	800	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	1050	1105	62,8%	190,5	26,3	263	55,4	900	1
	EM	0	0						0	
Op mode4	ME 1	4000	4194	68,5%	186,1	86,7	964	752,5	705	
	ME 2	0	0	0,0%	0,0	0,0	964	0,0	0	
	DG	3050	3211	91,2%	189,9	38,1	964	587,7	900	2
	EM	0	0						0	
Op mode5	ME 1	5288	5524	90,3%	185,7	114,0	88	90,3	800	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	5462	6253	88,8%	189,5	37,0	88	104,3	900	4
	EM	-4312	-4790						720	
Op mode6	ME 1	0	0	0,0%	0,0	0,0	1050	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1050	0,0	0	
	DG	2700	2976	84,5%	189,0	35,1	1050	590,4	900	2
	EM	-210	-337						720	
Op mode7	ME 1	5288	5615	91,8%	186,0	116,0	263	274,7	800	
	ME 2	0	0	0,0%	0,0	0,0	263	0,0	0	
	DG	2362	2487	70,6%	189,4	29,4	263	123,9	900	2
	EM	1188	1297						720	
Op mode8	ME 1	0	0	0,0%	0,0	0,0	1314	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1314	0,0	0	
	DG	1310	1534	87,2%	189,2	36,3	1314	381,5	900	1
	EM	-420	-568						720	
Op mode9	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	2120	2439	69,3%	189,6	28,9	438	202,5	900	2
	EM	-1000	-1197						720	
Op mode10	ME 1	5288	5606	91,6%	186,0	115,8	88	91,7	800	
	ME 2	0	0	0,0%	0,0	0,0	88	0,0	0	
	DG	4312	4539	86,0%	189,1	35,8	88	75,5	900	3
	EM	788	879						720	

Total fuel consumption 7478,0 t/yr

Table 5.14: Results for AHTS vessel configuration 3

The smaller main engines are not able to deliver the electric power via the PTO in modes 2, 3 and 4. This would result in overloading. The main engines now operate with a good relative loading resulting in a lower sfc point. This is also the case with the diesel generators. Even with multiple engines running, this still results in a lower fuel consumption in these modes.

As expected, the propulsion power for mode 5 has to come in the form of a PTI boost. This means that the less efficient DG's have to provide more power. For the small timeshare in the profile, this mode does not have a large influence. The less installed main engine power results in a better loading point for modes 2, 3 and 4 with a much higher timeshare.

Another difference compared to configuration 2 is that the main engines now cannot handle all PTO power in mode 7. The main engines go to their maximum allowable loading and even exceeding it due to the effect described in the previous paragraph with tanker configuration 1. But the more efficient L27/38 main engines over the V28/33 engines still result in less fuel consumption per year. A similar effect occurs in mode 10.

Configuration 4

The previous configurations had one engine per shaft. Perhaps it is also interesting to install two engines per propeller shaft, so four main engines in total. Four of the smallest bore propulsion engines will be tested: the 7-cylinder L21/31. These have a higher sfc than the L27/38 engine, but more flexibility and perhaps higher loading of the engines might be beneficial for overall fuel consumption. The main engines are now equal in bore size to the DG's. A total of 8 engines will have to be installed with this configuration.

Main engines 1	1,084	k€	
Main engines 2	1,084	k€	
Diesel generators	2,816	k€	
Electric machines	278	k€	
Gearboxes	444	k€	
PWM Converters	743	k€	
CPP + shaftlines	1,646	k€	+
Total	8,095	k€	

Table 5.16 presents the results for this configuration.

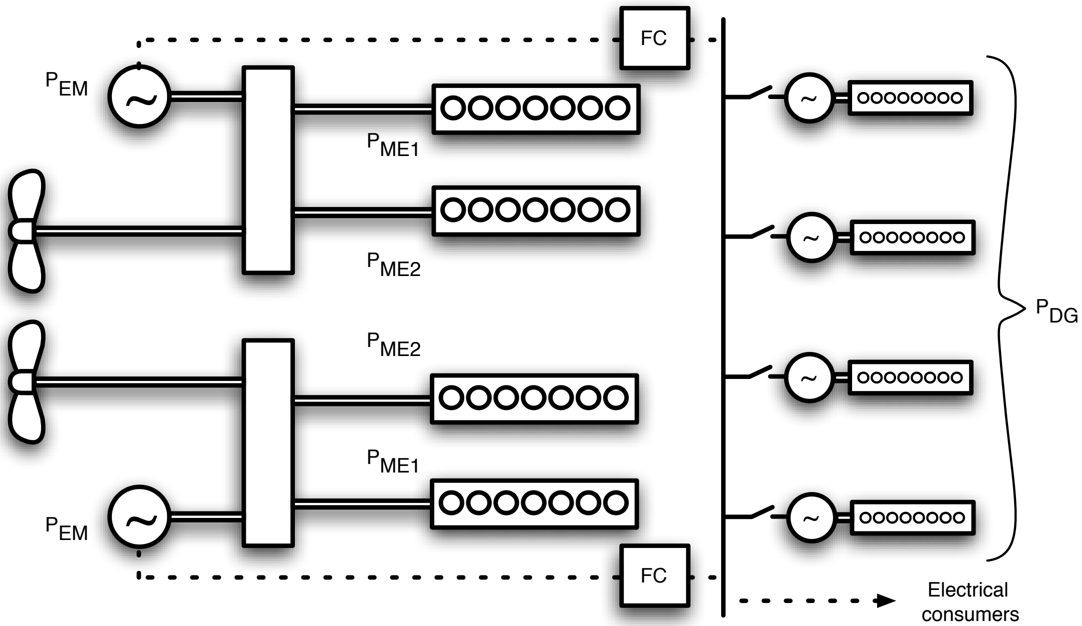

Figure 5.11: AHTS configuration 4 layout

Main engine 1		
Number	2	
Type	L21_31	
Cylinders	7	
Cyl power	215	kW
Max load	90%	
Pb engine	1505	kW
Pb total	3010	kW
n nom	1000	rpm

Main engine 2		
Number	2	
Type	L21_31	
Cylinders	7	
Cyl power	215	kW
Max load	90%	
Pb engine	1505	kW
Pb total	3010	kW
n nom	1000	rpm

Diesel generator sets		
No. DG	4	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pbrake (4)	7040	kW
Pb total	7040	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	5500	kW

Table 5.15: Input data for AHTS vessel configuration 4

Best result Configuration 4

		Best Delivered kW	= 2 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	150	158	9,0%	260,0	5,1	438	18,0	900	1
	EM	0	0						0	
Op mode2	ME 1	2250	2392	79,5%	191,8	65,5	3854	1767,7	926	
	ME 2	2250	2392	79,5%	191,8	65,5	3854	1767,7	926	
	DG	650	684	38,9%	196,8	16,8	3854	519,1	900	1
	EM	0	0						0	
Op mode3	ME 1	2500	2653	88,1%	192,4	72,9	263	134,2	1000	
	ME 2	2500	2653	88,1%	192,4	72,9	263	134,2	1000	
	DG	1050	1105	62,8%	190,5	26,3	263	55,4	900	1
	EM	0	0						0	
Op mode4	ME 1	2000	2130	70,8%	192,1	58,5	964	394,5	891	
	ME 2	2000	2130	70,8%	192,1	58,5	964	394,5	891	
	DG	3050	3211	91,2%	189,9	38,1	964	587,7	900	2
	EM	0	0						0	
Op mode5	ME 1	2601	2750	91,4%	192,9	75,8	88	46,7	1000	
	ME 2	2601	2750	91,4%	192,9	75,8	88	46,7	1000	
	DG	5549	6416	91,1%	189,9	38,1	88	107,2	900	4
	EM	-4399	-4945						720	
Op mode6	ME 1	0	0	0,0%	0,0	0,0	1050	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1050	0,0	0	
	DG	2700	2992	85,0%	189,0	35,3	1050	593,7	900	2
	EM	-210	-352						720	
Op mode7	ME 1	2601	2802	93,1%	193,3	77,4	263	142,4	1000	
	ME 2	2601	2802	93,1%	193,3	77,4	263	142,4	1000	
	DG	2449	2578	73,2%	189,2	30,5	263	128,2	900	2
	EM	1101	1207						720	
Op mode8	ME 1	0	0	0,0%	0,0	0,0	1314	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1314	0,0	0	
	DG	1310	1552	88,2%	189,4	36,7	1314	386,3	900	1
	EM	-420	-585						720	
Op mode9	ME 1	0	0	0,0%	0,0	0,0	438	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	438	0,0	0	
	DG	2120	2461	69,9%	189,5	29,2	438	204,3	900	2
	EM	-1000	-1218						720	
Op mode10	ME 1	2601	2797	92,9%	193,2	77,2	88	47,6	1000	
	ME 2	2601	2797	92,9%	193,2	77,2	88	47,6	1000	
	DG	4399	4630	87,7%	189,3	36,5	88	77,1	900	3
	EM	701	789						720	

Total fuel consumption 7743,3 t/yr

Table 5.16: Results for AHTS vessel configuration 4

Configuration 4 is practically the same as configuration 3, but with the installed main engine power now divided over two engines per shaft. Multiple smaller engines give more flexibility in choosing the number of running engines, with a negative effect of a smaller gearbox efficiency as is also described in the previous paragraph with the tanker vessel. Interestingly the flexibility does not come from the choice to run just one engine per shaft or two. Most modes operate with either none or both main engines per shaft running, meaning all four engines are running.
Compared to configuration 3 the same choice is made in running a PTO or PTI in all the different operating modes. This makes sense, since the installed main engine power and the installed generator power is roughly the same. The potential in the flexibility of running engines is not utilised with this P_D demand. All PTO modes must operate with both engines running on a relatively high load. This can be an expected result when looking in the loading of the main engines in configuration 3 in Table 5.14; the potential of multiple engines does not show here. But on the other hand, the negative effect of the gearbox efficiency does become clear, so the total annual fuel consumption in configuration 4 is higher than in configuration 3. This is clearly recognisable in e.g. transit mode 2 with the highest timeshare. A gearbox efficiency of 0,939 compared to an efficiency of 0,955 results in an increase of almost 200 t/yr. See appendix D for detailed information concerning gearbox efficiency.

Compared to configuration 2 there is some advantage with multiple smaller engines, but this is mainly because of the better loading point in mode 2.

5.2.3 Sensitivity analysis

Changes were made to all relevant component efficiencies to find their sensitivity. Just as with the tanker vessel described in the previous paragraph, changing the various electric efficiencies only changed the outcome proportionally, but did not alter the choice of configuration. The gearbox efficiency however had a larger effect. Lowering η_{GB} would lead to configuration 3 to become even more favourable over configuration 4. But the total fuel consumption increased. This effect is less in a diesel-electric system with a less complex gearbox like in configuration 1. The nominal gearbox efficiency has to be lowered below 0,9 for the full diesel electric vessel to be the most favourable in terms of fuel consumption. This is a value that is not very realistic anymore with modern gearboxes.

Configuration 3 had the best results, so based on this system some variations of the changeover parameter α were made. Lowering it would lead to a set of initial values that operate with a PTI only for the lower power demands. This means that for example mode 9 will not be operated in PTI mode, resulting in an increase of fuel. Lowering α will lead to another local minimum to be found that is not the global minimum.

Increasing α would mean a more diesel electrically operated vessel. Again, other local minima were found with overall worse results that the first result. An effect with more electrical propulsion power is that more power must be generated by the diesel generators with a higher sfc curve and higher electrical losses. In this scenario configuration 1 becomes more attractive with its larger and more efficient generator sets. A default value of around 0,25 proved to be the best in this case. Having one value as a distinct best means that this parameter is sensitive.

Next to changes in the more uncertain parameters, variations were made to the given data in the operational profile. A practical example might be that the vessel has less transit hours and DP hours, but more port time and standby times.
If the vessel would operate with more low-loading demand, there is a shift to more hybrid power. With lower loads, a PTI will become more beneficial. However, a shift to more electric power means that a diesel electric system becomes more attractive. The difference between configuration 1 and 3 somewhat decreased, but for configuration 1 to become the best the profile hours had to be altered more than 100%. A reasonable accuracy with the stated input profile can be assumed, so configuration 3 remained the best.

However, it must be noted that an offshore vessel such as the one described here are often have a full diesel electric system. This is because of the DP requirements. Electric motors have a faster response time in delivering torque to the propeller than diesel engines. Next to this, high manoeuvrable podded propulsors generally only make sense in a full electric vessel. Therefore, out of practical reasons not considered in the algorithm, a diesel electric vessel might be the best for an AHTS vessel.

5.3 Navy OPV

Figure 5.12: Navy OPV reference vessel Hr. Ms. Holland

Offshore patrol vessels often have two requirements regarding speed: slow sailing for general patrolling and a high speed/interception speed. The required power to obtain this speed must be available, but that might mean that the engine is running on lower load and lower efficiency when slow sailing. Whether a PTI function is beneficial will be investigated in this paragraph. Also, the electric motor might have a PTO function. The main question is how to divide the electric power and the mechanical power.

5.3.1 Operational profile

Unfortunately a detailed description of the expected operational profile was not available. The reference vessel did have a description of the installed main engine power and the speed requirements. This is converted to the assumed P_D. No actual conclusions can be derived from this data; it only serves as an example to validate the functionality of the design algorithms developed in this thesis.

The auxiliary power is based on [2.6] and [2.8]. The hotel load is based on a conventional cargo vessel with an average number of crew members. This vessel might have a larger crew and more electrical equipment, so it is expected that the auxiliary load is a little too low.

The port times were also not specified. Since data is missing a port time of 40% is assumed based on similar vessels in other countries.

The operational profile is presented in Table 5.17 and Figure 5.13.

		P_D [kW]	P_{aux} [kW]	Time [hr/year]
Mode 1	**Port**	0	150	3504
Mode 2	**< 11 kn**	730	650	1577
Mode 3	**< 15 kn**	2900	1050	2102
Mode 4	**20 kn**	10400	3050	1577

Table 5.17: Operational profile of the Navy OPV

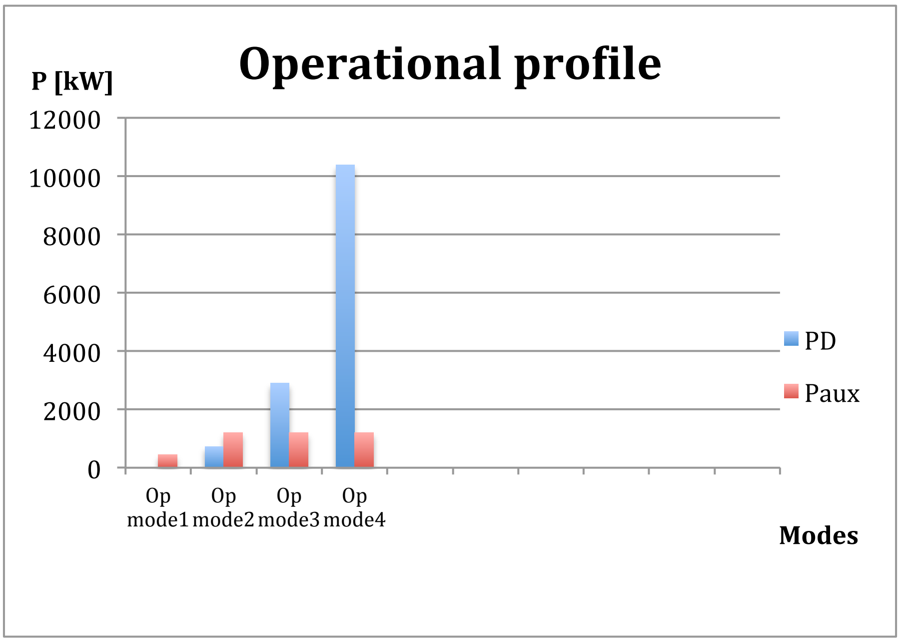

Figure 5.13: Operational profile of the Navy OPV

5.3.2 Suggested configurations

The V28/33 engine comes with two different loading profiles. One is called the navy load profile and has an increased mean effective pressure and a higher rotational speed, so more power can be generated from the same engine dimensions. A higher power density makes it suitable for these type of vessels. With a sequential turbocharger the operating envelope can be extended for the lower load regions.

The interesting part of the investigation of this vessel is to see whether a PTI function is beneficial and whether this can be combined with a PTO function for other operating modes.

Configuration 1

As a reference, a diesel mechanic system will be investigated first. All propulsion power will come from the main engines and all auxiliary power from the diesel generator sets. Two main engines are selected, one per shaft. The auxiliary power will be provided by two DG's while a third one is installed for redundancy reasons. This input data is shown in Table 5.18.

This is the cheapest configuration with a single stage gearbox and no electric machine and converter. Again, the indicated cost prices are for the two shafts combined:

Main engines 1	4,080	k€	
Diesel generators	1,056	k€	
Gearboxes	363	k€	
CPP + shaftlines	1,783	k€	+
Total	7,282	k€	

The results are presented in Table 5.19.

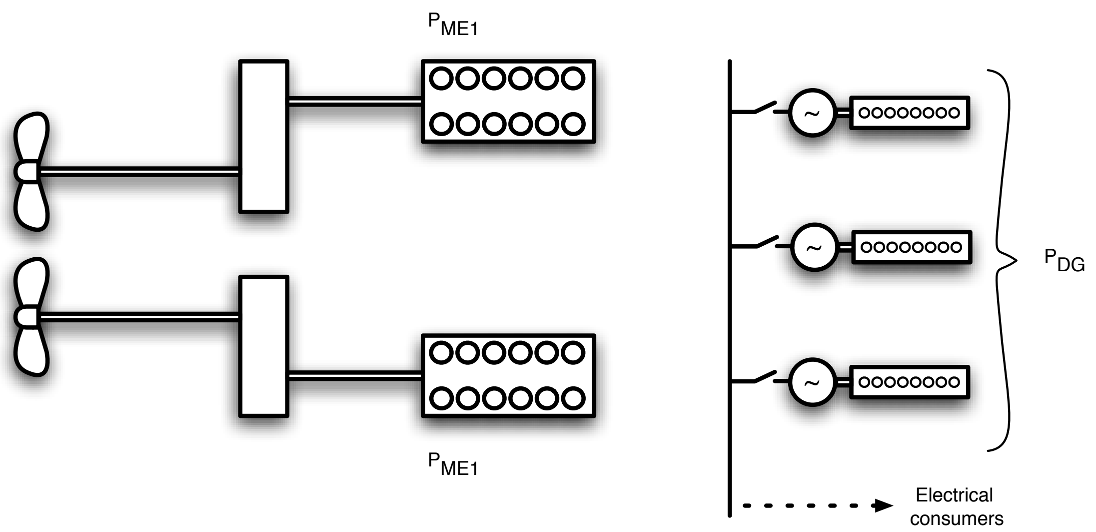

Figure 5.14: OPV configuration 1 layout

Main engine 1		
Number	2	
Type	V28_33D_STC_Navy	
Cylinders	12	
Cyl power	500	kW
Max load	92%	
Pb engine	6000	kW
Pb total	12000	kW
n nom	1032	rpm

Diesel generator sets		
No. DG	3	
Type	L16_24_G	
Cylinders	8	
Cyl power	110	kW
Max load	100%	
Pbrake (1)	880	kW
Pbrake (2)	1760	kW
Pbrake (3)	2640	kW
Pb total	2640	kW
Eta gen	0,95	
n nom	1200	rpm

Other input		
PTI selected if PD <	-	PB ME
EM nominal speed	-	
Include FC	No	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
trm	0,96	
Maximum EM power	0	kW

Table 5.18: Input data for OPV configuration 1

Best result Configuration 1

		Best Delivered kW	= 4 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	54,4%	192,7	11,5	3504	323,4	1200	1
	EM	0	0						0	
Op mode2	ME 1	730	780	6,5%	308,4	20,1	1577	379,6	619	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	1577	380,4	1200	2
	EM	0	0						0	
Op mode3	ME 1	2900	3017	25,1%	208,8	52,5	2102	1324,2	651	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	2102	507,0	1200	2
	EM	0	0						0	
Op mode4	ME 1	10400	10712	89,3%	190,1	169,7	1577	3211,2	1032	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	1577	380,4	1200	2
	EM	0	0						0	

Total fuel consumption 6506,1 t/yr

Table 5.19: Results for OPV configuration 1

This diesel mechanic configuration serves as a reference for the other tested configurations, so there are no real surprises here. The power of the main engines goes up to the maximum allowable loading point. Two DG's are running just as was expected in the selection of the components. This configuration has the highest gearbox efficiency, as can be seen in appendix D.

Configuration 2
The same engines are selected for the main engines and as the generator sets. But in this configuration, an electric machine is added. The goal is to find the best value for α. With a value of 0,15 a PTI will be selected for mode 2 in the initial results. Running the optimisation algorithm will determine whether this is also the optimum changeover point. The input data is shown in Table 5.20 and leads to the following investment costs:

Main engines 1	4,080	k€	
Diesel generators	1,056	k€	
Electric machines	65	k€	
Gearboxes	417	k€	
PWM Converters	175	k€	
CPP + shaftlines	1,783	k€	+
Total	7,576	k€	

The results are shown in Table 5.21.

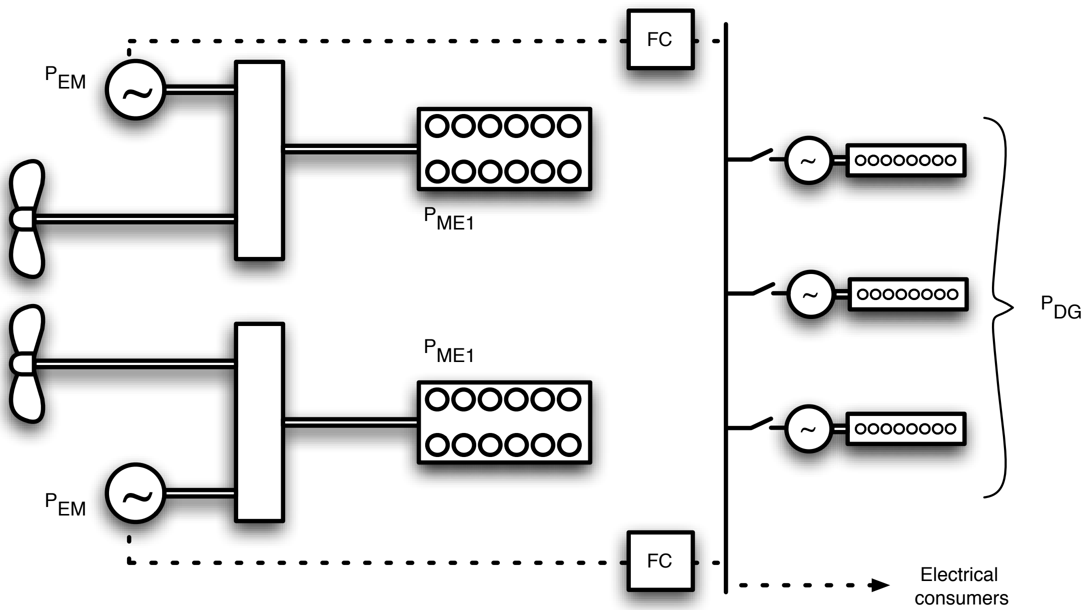

Figure 5.15: OPV configuration 2 layout

Main engine 1		
Number	2	
Type	V28_33D_STC_Navy	
Cylinders	12	
Cyl power	500	kW
Max load	92%	
Pb engine	6000	kW
Pb total	12000	kW
n nom	1032	rpm

Diesel generator sets		
No. DG	3	
Type	L16_24_G	
Cylinders	8	
Cyl power	110	kW
Max load	100%	
Pbrake (1)	880	kW
Pbrake (2)	1760	kW
Pbrake (3)	2640	kW
Pb total	2640	kW
Eta gen	0,95	
n nom	1200	rpm

Other input		
PTI selected if PD <	0,15	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	1300	kW

Table 5.20: Input data for OPV configuration 2

Best result Configuration 2

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	54,4%	192,7	11,5	3504	323,4	1200	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1577	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1940	2191	83,0%	189,7	17,3	1577	655,6	1200	3
	EM	-730	-872						720	
Op mode3	ME 1	4110	4383	36,5%	199,5	72,9	2102	1837,9	738	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	0	0	0,0%	0,0	0,0	2102	0,0	0	0
	EM	1210	1286						515	
Op mode4	ME 1	10400	10811	90,1%	190,2	171,3	1577	3242,4	1032	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	1577	380,4	1200	2
	EM	0	0						0	

Total fuel consumption 6439,7 t/yr

Table 5.21: Results for OPV configuration 2

As expected, in mode 2 a PTI will be selected for slow sailing. This utilises all three generators that are loaded at their optimal loading point. The difference with configuration 1 is about 66 t/yr, which is 1%. In mode 3 a PTO will be selected. The total fuel consumption in this mode is slightly higher than in configuration 1. This is due to the higher losses with this gearbox type. Running this mode with the DG's producing the auxiliary power instead of the PTO generator would increase the total fuel consumption with about 10 t/yr, which is not a large amount.

At full speed in mode 4, the selected engines are not large enough to provide for both the propulsion power and auxiliary power. Increasing the main engine power has an overall negative effect on the other operating modes and also increases the investment costs.

Configuration 3

In this configuration the diesel generator power is increased so they can provide the propulsion power for both mode 2 and mode 3. This would obviously also require a larger electric machine. Compared to configuration 2 this will determine what the best value for α is. This configuration requires larger DG's and electric equipment, resulting in a higher investment cost:

Main engines 1	4,080	k€	
Diesel generators	2,112	k€	
Electric machines	160	k€	
Gearboxes	417	k€	
PWM Converters	432	k€	
CPP + shaftlines	1,783	k€	+
Total	8,984	k€	

The input data is shown in Table 5.22 and the results in Table 5.23. The layout is the same as shown in Figure 5.15.

Main engine 1		
Number	2	
Type	V28_33D_STC_Navy	
Cylinders	12	
Cyl power	500	kW
Max load	100%	
Pb engine	6000	kW
Pb total	12000	kW
n nom	1032	rpm

Diesel generator sets		
No. DG	3	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pb total	5280	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,3	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	3200	kW

Table 5.22: Input data for OPV configuration 3

Best result Configuration 3		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	27,2%	204,6	12,3	3504	343,4	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1577	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1940	2208	62,7%	190,5	26,3	1577	663,1	900	2
	EM	-730	-887						720	
Op mode3	ME 1	0	0	0,0%	0,0	0,0	2102	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	4110	4697	89,0%	189,5	37,1	2102	1871,1	900	3
	EM	-2900	-3253						720	
Op mode4	ME 1	10400	10811	90,1%	190,2	171,3	1577	3242,4	1032	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,3	30,1	1577	380,1	900	1
	EM	0	0						0	
						Total fuel consumption		6500,3	t/yr	

Table 5.23: Results for OPV configuration 3

The main effect in this configuration is that the propulsion power in mode 3 can also be delivered by the electric motor. Although the three DG's are running close to their optimal sfc point, the extra electric losses require a higher brake power than when this power is provided by just the main engines. So a PTI for mode 3 is actually not beneficial.

Slightly larger diesel generators also mean that they run on a lower relative load in mode 2, resulting in a higher sfc and therefore a slightly higher fuel consumption.

Configuration 4

Since sailing at 20 knots does not have the highest timeshare of the profile, it might be beneficial to run the electric motor as a booster. This means that the installed main engine power can be reduced. A smaller engine bore type is not available for this power, but a different loading profile might be more efficient. The V28/33 with the ferry loading profile will be selected here.

Main engines 1	3,713	k€	
Diesel generators	2,112	k€	
Electric machines	65	k€	
Gearboxes	417	k€	
PWM Converters	175	k€	
CPP + shaftlines	1,783	k€	+
Total	8,265	k€	

Table 5.24 shows the input data for this configuration and Table 5.25 the results. This configuration also has the same configuration as shown in Figure 5.15.

Main engine 1		
Number	2	
Type	V28_33D_STC_Ferry	
Cylinders	12	
Cyl power	455	kW
Max load	100%	
Pb engine	5460	kW
Pb total	10920	kW
n nom	1000	rpm

Diesel generator sets		
No. DG	3	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pb total	5280	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	1300	kW

Table 5.24: Input data for OPV configuration 4

Best result Configuration 4

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	27,2%	204,6	12,3	3504	343,4	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1577	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1940	2191	62,3%	190,5	26,1	1577	658,4	900	2
	EM	-730	-872						720	
Op mode3	ME 1	4110	4382	40,1%	195,4	71,4	2102	1800,1	738	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	0	0	0,0%	0,0	0,0	2102	0,0	0	0
	EM	1210	1286						531	
Op mode4	ME 1	10400	10811	99,0%	188,4	169,7	1577	3212,3	1000	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,3	30,1	1577	380,1	900	1
	EM	0	0						0	

Total fuel consumption $\boxed{6394,4}$ t/yr

Table 5.25: Results for OPV configuration 4

The first difference that can be noted occurs in mode 2 where a smaller electric machine (see Table 5.24) now operates with a higher relative load leading to a slightly better efficiency.
More remarkable changes are in mode 3, where a different operation mode is chosen compared to configuration 3. In this mode, a PTO is selected instead of a PTI in configuration 3, making it resemble configuration 2 more. While the negative effects of a PTI have been discussed briefly in configuration 3, it is interesting to see that a different value of α again determines the choice of operation that leads to a local minimum. Therefore this parameter is a critical one and without further sensitivity analysis it is clear that α must be altered manually.

As in configuration 3, the algorithm selects a PTO for mode 4. But now the engines have to be loaded to their maximum. The idea behind this configuration is to operate with a PTI booster in mode 4, so it is interesting that this is not the case. Still a diesel mechanical operation is the most fuel-efficient. Shifting more power from the larger engines to the smaller generator sets does not improve fuel consumption. The difference is around 60 t/yr.

The main difference is however the overall lower sfc of the V28/33 engine with a ferry loading profile.

5.3.3 Sensitivity analysis

The main difference between the tested configurations is when to operate with a PTI and when not. This means α is already determined and does not need to be tested for sensitivity.

The gearbox efficiency seems to be the deciding factor for some modes compared to a diesel mechanic system. Therefore, variations were made to $\eta_{GB,0}$.
When decreasing $\eta_{GB,0}$, the losses become more dominant for more complex gearbox types. After decreasing the gearbox efficiency by a few per cent, the diesel mechanic concept tested in configuration 1 becomes even more advantageous than the hybrid concepts in configuration 2 and 4. These results are displayed in Table 5.26 and Table 5.27. Appendix D also shows the determining gearbox efficiency. It must be noted that the NCR point of the main engines is at 93% MCR in mode 4 for configuration 4. This is actually not allowed. Next to this, the engines cannot deliver enough propulsion power so the EM must operate as a PTI booster. This means more running wheels and shafts at the gearbox, lowering the efficiency even more.
Altogether this is a fairly large change in $\eta_{GB,0}$, so it can be stated that configuration 4 is fairly robust. Increasing $\eta_{GB,0}$ slightly would lead to configuration 4 to become even more stable.

Variations to the nominal efficiency of the electric machine were also made. Lowering $\eta_{EM,0}$ only had a small proportional effect. Only when it was lowered beyond realistic values did configuration 1 become more favourable.

Another assumption was made in determining the auxiliary power demand. It is based on a cargo vessel and not a navy vessel. Therefore it might be interesting to increase the P_{aux} demand. Increasing P_{aux} makes a PTO in mode 3 more efficient, just with more absolute power. The main engines in mode 4 are already at max, so it doesn't change the choice of operation of the components. The P_{aux} was increased stepwise to 150%. Configuration 4 remains the best with a PTI in mode 2, PTO in mode 3 and mechanically in mode 4. While the second best configuration 2 had similar results, the smaller engines with better loading points make configuration 4 even stronger with increasing P_{aux} over configuration 2. Therefore it can be concluded that this configuration is has stable solution.

Best result Configuration 1

		Best Delivered kW	= 4 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	54,4%	192,7	11,5	3504	323,4	1200	1
	EM	0	0						0	
Op mode2	ME 1	730	818	6,8%	302,2	20,6	1577	389,6	619	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	1577	380,4	1200	2
	EM	0	0						0	
Op mode3	ME 1	2900	3092	25,8%	208,0	53,6	2102	1352,3	657	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	2102	507,0	1200	2
	EM	0	0						0	
Op mode4	ME 1	10400	10889	90,7%	190,3	172,7	1577	3267,4	1032	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1210	1274	72,4%	189,4	15,1	1577	380,4	1200	2
	EM	0	0						0	

Total fuel consumption 6600,4 t/yr

Table 5.26: Results for OPV configuration 1 after $\eta_{GB,0}$ variations

Best result Configuration 4

		Best Delivered kW	= 4 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	3504	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	3504	0,0	0	
	DG	455	479	27,2%	204,6	12,3	3504	343,4	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	1577	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1940	2286	64,9%	190,1	27,2	1577	685,5	900	2
	EM	-730	-943						720	
Op mode3	ME 1	2900	3192	29,2%	202,1	53,8	2102	1356,1	664	
	ME 2	0	0	0,0%	0,0	0,0	2102	0,0	0	
	DG	1210	1274	72,4%	189,3	30,1	2102	506,7	900	1
	EM	0	0						0	
Op mode4	ME 1	10064	10840	99,3%	188,6	170,3	1577	3223,3	1000	
	ME 2	0	0	0,0%	0,0	0,0	1577	0,0	0	
	DG	1546	1686	95,8%	190,9	40,2	1577	507,6	900	1
	EM	-336	-384						720	

Total fuel consumption 6622,7 t/yr

Table 5.27: Results for OPV configuration 4 after $\eta_{GB,0}$ variations

5.4 RoPax

Figure 5.16: RoPax reference vessel Blue Star Naxos

RoPax vessels have varying loads in both propulsion demand and auxiliary demand. At certain times there may be a large number of passengers that will increase the hotel load. At other times the vessel is loading cars or trucks with less auxiliaries running. This particular vessel will sail between islands in Greece. There are large portions of slow sailing and manoeuvring close to shore, where more auxiliary power is required. The total sailing route is summarised here to a manoeuvring mode and a cruising mode.

5.4.1 Operational profile

A detailed description of the various required ship speeds and hotel loads was unfortunately not available. However, a general description with four distinct operating modes with the required propulsion and auxiliary demand was specified and will suffice as an input for the operational profile. It is presented in Table 5.28 and Figure 5.17.
These port times are split into two modes. In mode 1 the engine auxiliaries are switched off and in mode 2 the engine room is standby for departure.

		P_D [kW]	P_{aux} [kW]	Time [hr/year]
Mode 1	**Port**	0	500	2936
Mode 2	**Port standby**	0	900	559
Mode 3	**Manoeuvring**	15000	2600	1188
Mode 4	**Cruising 24 kn**	27400	1000	4077

Table 5.28: Operational profile of the RoPax vessel

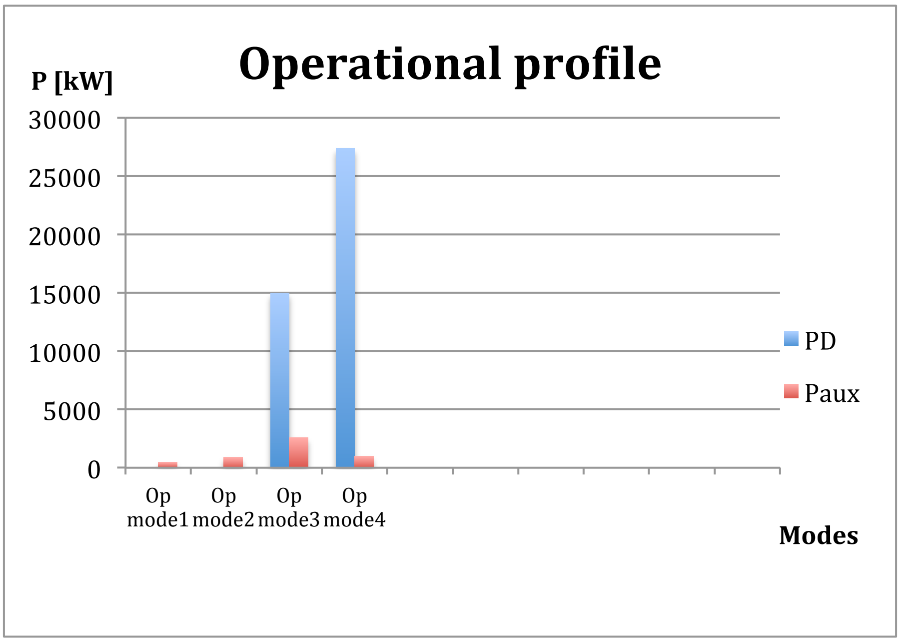

Figure 5.17: Operational profile of the RoPax vessel

5.4.2 Suggested configurations

This vessel has a high manoeuvring load, which is almost half of the total propulsion power. The several configurations tested in this paragraph are to determine the selection between a PTI for manoeuvring or a PTI for a boost. A comparison between a diesel mechanic system will also be made.

Configuration 1

This will be a diesel mechanic mode with a larger bore medium speed engine. The V48/60 CR engines would fulfil the propulsion demand including a PTO for auxiliary demand. Not a large amount of auxiliary power is required, but a with a passenger vessel one wants to prevent a blackout at all costs, so extra redundancy is installed for the diesel generators. Two 8-cylinder L21/31 engines with a total brake power of 3520 kW would be sufficient, but four DG's will be installed. The input data of this configuration is shown in Table 5.30.

The estimated investment costs are:

Main engines 1	10,752	k€	
Diesel generators	2,816	k€	
Electric machines	145	k€	
Gearboxes	1,021	k€	
PWM Converters	392	k€	
CPP + shaftlines	4,697	k€	+
Total	19,823	k€	

Table 5.31 shows the results for this configuration.

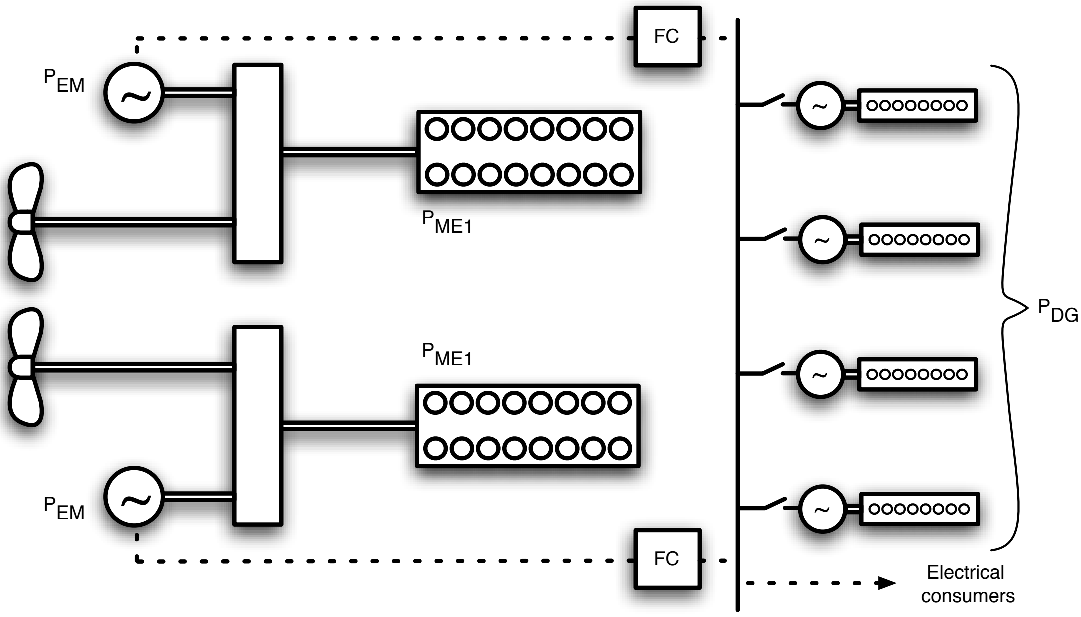

Table 5.29: RoPax configuration 1 layout

Main engine 1		
Number	2	
Type	V48_60CR	
Cylinders	16	
Cyl power	1200	kW
Max load	90%	
Pb engine	19200	kW
Pb total	38400	kW
n nom	514	rpm

Diesel generator sets		
No. DG	4	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pbrake (4)	7040	kW
Pb total	7040	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	2900	kW

Table 5.30: Input data for RoPax configuration 1

Best result Configuration 1

		Best	= 3							
		Delivered kW	Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	2936	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2936	0,0	0	
	DG	500	526	29,9%	202,3	13,3	2936	312,6	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	559	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	559	0,0	0	
	DG	900	947	53,8%	192,1	22,7	559	101,7	900	1
	EM	0	0						0	
Op mode3	ME 1	17599	18560	48,3%	183,4	212,7	1188	4043,0	403	
	ME 2	0	0	0,0%	0,0	0,0	1188	0,0	0	
	DG	1	0	0,0%	0,0	0,0	1188	0,0	0	0
	EM	2599	2763						565	
Op mode4	ME 1	28400	29709	77,4%	177,9	330,3	4077	21543,3	472	
	ME 2	0	0	0,0%	0,0	0,0	4077	0,0	0	
	DG	0	0	0,0%	0,0	0,0	4077	0,0	0	0
	EM	1000	1076						661	

Total fuel consumption $\boxed{26000,6}$ t/yr

Table 5.31: Results for RoPax configuration 1 (16-cylinder engine)

There are no major changes in the results from what was initially expected for this diesel mechanic configuration. In operating modes 1 and 2 just the diesel generators provide the power. Mode 3 results in the main engines being half loaded and in mode 4 they are loaded to their design point. This is a good reference configuration with low losses.

The main engines operate with a maximum loading of 77%. Therefore, less brake power can be installed. It is interesting to see that a smaller 14-cylinder main engine instead of the 16-cylinder engine would result in a PTO not being selected (see Table 5.32). The benefits of a better sfc point of the engine are counteracted by the higher losses in the gearbox, similar as in configuration 2 of the OPV described in paragraph 5.3.2.

Best result Configuration 1

		Best Delivered kW	= 3 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	2936	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2936	0,0	0	
	DG	500	526	29,9%	202,3	13,3	2936	312,6	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	559	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	559	0,0	0	
	DG	900	947	53,8%	192,1	22,7	559	101,7	900	1
	EM	0	0						0	
Op mode3	ME 1	15000	15506	46,1%	183,9	203,7	1188	3387,9	397	
	ME 2	0	0	0,0%	0,0	0,0	1188	0,0	0	
	DG	2600	2737	77,8%	188,8	32,3	1188	613,8	900	2
	EM	0	0						0	
Op mode4	ME 1	27400	28214	84,0%	173,2	349,1	4077	19927,3	514	
	ME 2	0	0	0,0%	0,0	0,0	4077	0,0	0	
	DG	1000	1053	59,8%	190,9	25,1	4077	819,4	900	1
	EM	0	0						0	

Total fuel consumption 25162,7 t/yr

Table 5.32: Results for RoPax configuration 1 (14-cylinder engine)

Main engine 1		
Number	2	
Type	V48_60CR	
Cylinders	14	
Cyl power	1200	kW
Max load	90%	
Pb engine	16800	kW
Pb total	33600	kW
n nom	514	rpm

Table 5.33: Alternative main engine RoPax configuration 1

With the 14-cylinder engine, the total investment costs can be reduced. No electric machine and converter need to be installed and the gearbox can be cheaper. This saves a total of 691 k€, resulting in a total cost price of 19,132 k€.

Configuration 2

For more flexibility in the main engine power, two main engines per shaft can be considered with roughly the same brake power. A total of four 16-cylinder V32/44 CR engines are installed. The installed DG power can remain the same. This leads to the following costs:

Main engines 1	5,018	k€	
Main engines 2	5,018	k€	
Diesel generators	2,816	k€	
Electric machines	145	k€	
Gearboxes	1,074	k€	
PWM Converters	392	k€	
CPP + shaftlines	4,697	k€	+
Total	19,160	k€	

Table 5.34 shows the full input data for this configuration and Table 5.35 the results.

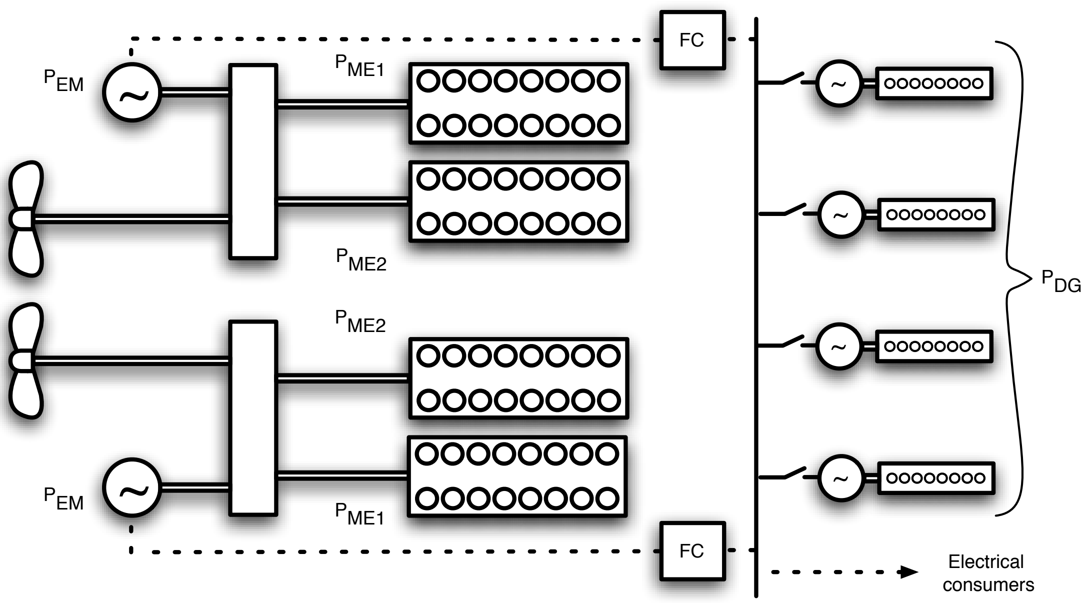

Figure 5.18: RoPax configuration 2 layout

Main engine 1		
Number	2	
Type	V32_44CR	
Cylinders	16	
Cyl power	560	kW
Max load	90%	
Pb engine	8960	kW
Pb total	17920	kW
n nom	750	rpm

Main engine 2		
Number	2	
Type	V32_44CR	
Cylinders	16	
Cyl power	560	kW
Max load	90%	
Pb engine	8960	kW
Pb total	17920	kW
n nom	750	rpm

Diesel generator sets		
No. DG	4	
Type	L21_31_G	
Cylinders	8	
Cyl power	220	kW
Max load	100%	
Pbrake (1)	1760	kW
Pbrake (2)	3520	kW
Pbrake (3)	5280	kW
Pbrake (4)	7040	kW
Pb total	7040	kW
Eta gen	0,95	
n nom	900	rpm

Other input		
PTI selected if PD <	0	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	2900	kW

Table 5.34: Input data for RoPax configuration 2

Best result Configuration 2

		Best Delivered kW	= 5 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	2936	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2936	0,0	0	
	DG	500	526	29,9%	202,3	13,3	2936	312,6	900	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	559	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	559	0,0	0	
	DG	900	947	53,8%	192,1	22,7	559	101,7	900	1
	EM	0	0						0	
Op mode3	ME 1	15000	15533	86,7%	175,8	170,7	1188	3244,2	750	
	ME 2	0	0	0,0%	0,0	0,0	1188	0,0	0	
	DG	2600	2737	77,8%	188,8	32,3	1188	613,8	900	2
	EM	0	0						0	
Op mode4	ME 1	13700	14286	79,7%	177,6	158,6	4077	10344,3	695	
	ME 2	13700	14286	79,7%	177,6	158,6	4077	10344,3	695	
	DG	1000	1053	59,8%	190,9	25,1	4077	819,4	900	1
	EM	0	0						0	

Total fuel consumption 25780,3 t/yr

Table 5.35: Results for RoPax configuration 2

This configuration allows two engines for mode 3 and requires all four engines for mode 4. This has a clear effect. Using a PTO here would have a negative effect. It seems that the relative small auxiliary power demand has only a small positive effect on the performance of the engines, while at the same time has a larger negative effect on the gearbox efficiency. With this power demand and these configurations, there is a delicate balance.

It turns out that the electric machine does not necessarily have to be installed, so the EM and converter can be left out. This also means that the gearbox can be cheaper. A total of 671 k€ can be saved, resulting in a total cost price of 18,489 k€.

Configuration 3

This configuration is used to investigate whether a PTI will be beneficial for manoeuvring in mode 3. The same main engine power is required for cruising speed in mode 4, but larger DG's should be installed for the propulsion power requirement in mode 3. A total of three DG's would need to run to provide this power, while the fourth one is available for standby.
The estimated investment costs are:

Main engines 1	10,752	k€
Diesel generators	9,139	k€
Electric machines	140	k€
Gearboxes	1,021	k€
PWM Converters	378	k€
CPP + shaftlines	4,697	k€ +
Total	26,127	k€

Table 5.36 shows the input data for this configuration and Table 5.37 the results.

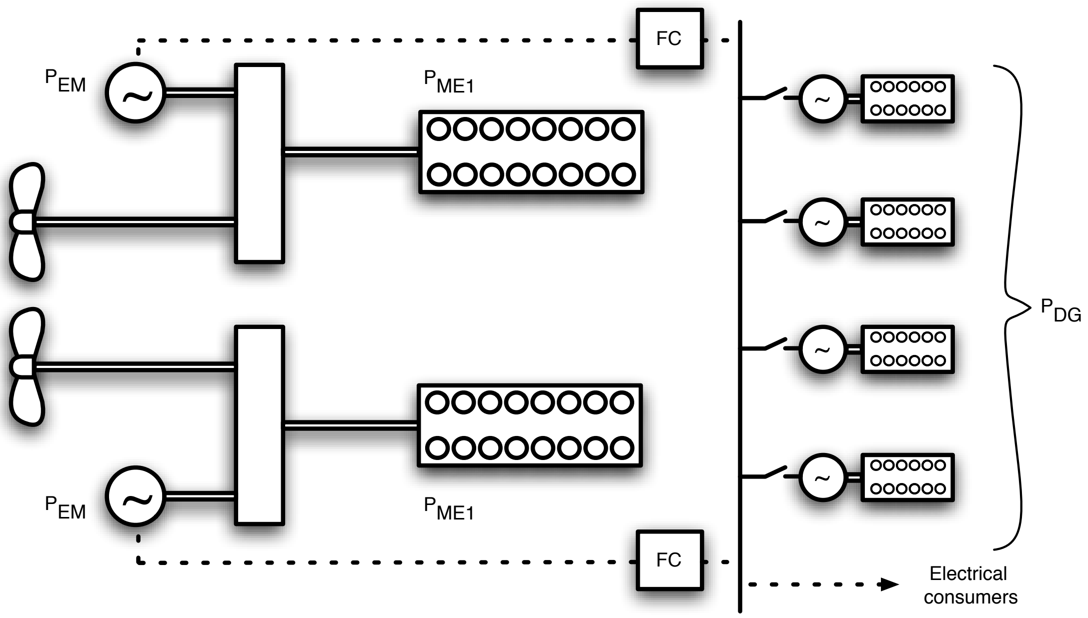

Figure 5.19: RoPax configuration 3 layout

Main engine 1		
Number	2	
Type	V48_60CR	
Cylinders	16	
Cyl power	1200	kW
Max load	90%	
Pb engine	19200	kW
Pb total	38400	kW
n nom	514	rpm

Diesel generator sets		
No. DG	4	
Type	V32_44CR_G	
Cylinders	12	
Cyl power	560	kW
Max load	100%	
Pbrake (1)	6720	kW
Pbrake (2)	13440	kW
Pbrake (3)	20160	kW
Pbrake (4)	26880	kW
Pb total	26880	kW
Eta gen	0,97	
n nom	720	rpm

Other input		
PTI selected if PD <	0,5	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	2800	kW

Table 5.36: Input data for RoPax configuration 3

Best result Configuration 3

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	2936	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2936	0,0	0	
	DG	500	515	7,7%	320,0	13,7	2936	484,2	720	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	559	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	559	0,0	0	
	DG	900	928	13,8%	245,2	19,0	559	127,2	720	1
	EM	0	0						0	
Op mode3	ME 1	17600	18561	48,3%	183,4	212,7	1188	4043,2	403	
	ME 2	0	0	0,0%	0,0	0,0	1188	0,0	0	
	DG	0	0	0,0%	0,0	0,0	1188	0,0	0	0
	EM	2600	2764						565	
Op mode4	ME 1	28400	29709	77,4%	177,9	330,3	4077	21543,1	472	
	ME 2	0	0	0,0%	0,0	0,0	4077	0,0	0	
	DG	0	0	0,0%	0,0	0,0	4077	0,0	0	0
	EM	1000	1075						661	

Total fuel consumption 26197,7 t/yr

Table 5.37: Results for RoPax configuration 3

Even with an engine room configuration capable of running a PTI, the algorithm chose not to do this. The main engines are running similar to configuration 1. Utilising a PTI in mode 3 would result in a higher overall fuel consumption than two efficient main engines on part load. Another effect is that the larger diesel generators than configuration 2 will have to run on a lower load in port. This means that in mode 1 and 2 this engine selection has a negative effect compared to configuration 2. The difference is however small.

Configuration 4

This configuration has less installed main engine power, so a PTI boost function would be required in mode 4. The same DG power is installed as in configuration 3, so a PTI would be possible for mode 3. It would be interesting to investigate whether two main engines in mode 3 with a PTI boost in mode 4 is better than fully loaded DG's in mode 3 and fully loaded main engines in mode 4. The input data is shown in Table 5.38 and leads to these investment costs:

Main engines 1	7,776	k€	
Diesel generators	9,139	k€	
Electric machines	480	k€	
Gearboxes	1,021	k€	
PWM Converters	1,296	k€	
CPP + shaftlines	4,697	k€	+
Total	24,409	k€	

The results are presented in Table 5.39.

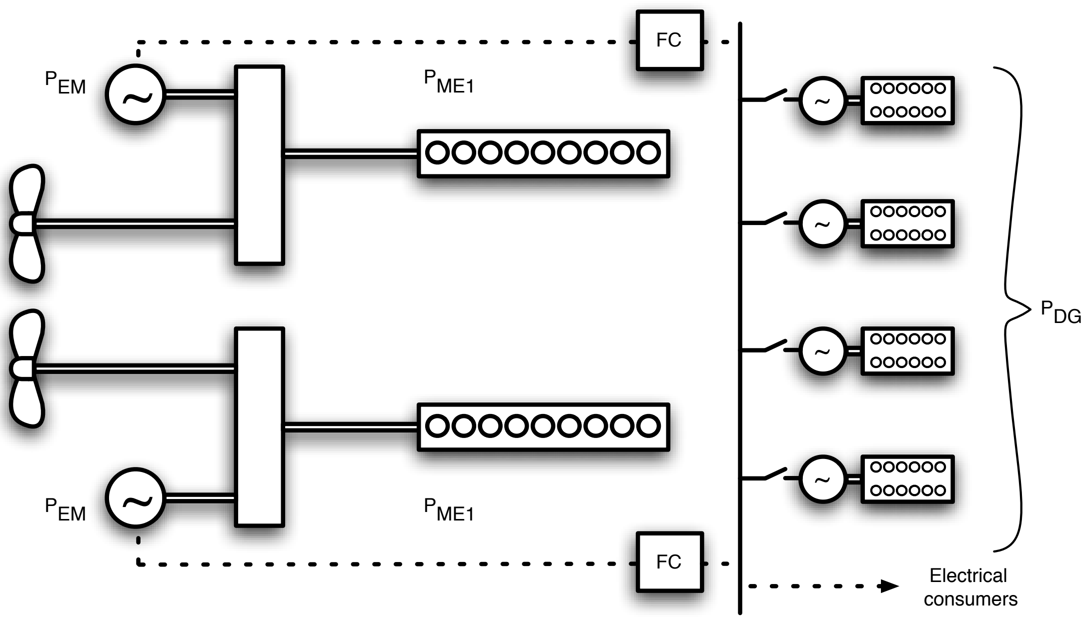

Figure 5.20: RoPax configuration 4 layout

Main engine 1		
Number	2	
Type	L48_60CR	
Cylinders	9	
Cyl power	1200	kW
Max load	90%	
Pb engine	10800	kW
Pb total	21600	kW
n nom	514	rpm

Diesel generator sets		
No. DG	4	
Type	V32_44CR_G	
Cylinders	12	
Cyl power	560	kW
Max load	100%	
Pbrake (1)	6720	kW
Pbrake (2)	13440	kW
Pbrake (3)	20160	kW
Pbrake (4)	26880	kW
Pb total	26880	kW
Eta gen	0,97	
n nom	720	rpm

Other input		
PTI selected if PD <	0,25	PB ME
EM nominal speed	720	
Include FC	Yes	
Combinator curve		
Constant n below	60%	ne
Constant n above	80%	load
Nominal efficiencies		
Shaft	0,99	
EM	0,96	
FC	0,98	
trm	0,96	
Maximum EM power	9600	kW

Table 5.38: Input data for RoPax configuration 4

Best result Configuration 4

		Best Delivered kW	= 1 Brake kW	load %	sfc g/kWh	Mf cyl (kg/hr)	time hr/yr	fuel t/yr	speed rpm	DG's running
Op mode1	ME 1	0	0	0,0%	0,0	0,0	2936	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	2936	0,0	0	
	DG	500	515	7,7%	320,0	13,7	2936	484,2	720	1
	EM	0	0						0	
Op mode2	ME 1	0	0	0,0%	0,0	0,0	559	0,0	0	
	ME 2	0	0	0,0%	0,0	0,0	559	0,0	0	
	DG	900	928	13,8%	245,2	19,0	559	127,2	720	1
	EM	0	0						0	
Op mode3	ME 1	17600	18562	85,9%	174,9	360,7	1188	3856,6	514	
	ME 2	0	0	0,0%	0,0	0,0	1188	0,0	0	
	DG	0	0	0,0%	0,0	0,0	1188	0,0	0	0
	EM	2600	2816						720	
Op mode4	ME 1	18662	19487	90,2%	175,5	380,0	4077	13944,4	514	
	ME 2	0	0	0,0%	0,0	0,0	4077	0,0	0	
	DG	9738	11029	82,1%	176,7	81,2	4077	7945,1	720	2
	EM	-8738	-9698						720	

Total fuel consumption 26357,6 t/yr

Table 5.39: Results for RoPax configuration 4

It seems that two high loaded main engines in mode 3 and including two high loaded DG's in mode 4 is the most beneficial. In spite of this, the V48/60 CR engine is quite an efficient engine with a relative flat $\dot{m}_f$ curve shape. Shifting power to smaller engines with increased transmission losses simply results in a higher overall fuel consumption. Like in configuration 3, the negative effect of the larger diesel generators is also seen here in mode 1 and 2.

5.4.3 Sensitivity analysis

The positive effect between the selected configurations again seems to be very dependent on the gearbox efficiency. Therefore variations were made to the $\eta_{GB,0}$ to see whether it affects the choice of a PTO or not. With lower $\eta_{GB,0}$, the losses with a more complex gearbox become more significant, so the effects are greater. This would still suggest a diesel mechanic system to be the most efficient. A 14-cylinder engine would suffice in this case. With a higher $\eta_{GB,0}$ a PTO might become favourable. However, increasing $\eta_{GB,0}$ would only lead to a slight improvement. Only without gearbox losses a PTO would be beneficial. This is of course not a realistic value, so it can be concluded that the gearbox efficiency is not as sensitive as initially expected.

Variations were also made to the electric machine efficiency. This had similar effect on the choice of configuration as variations to the gearbox efficiency, but to a lesser degree. Varying $\eta_{EM,0}$ would only have a proportional effect, while the gearbox efficiency has a stepwise negative effect.

Since many possible load variations are not included in the operational profile, the effects of a hybrid concept cannot be fully utilised. Based on this particular profile, a diesel mechanic configuration with a 14-cylinder V48/60 CR engine is the most fuel-efficient.

6 Conclusions and recommendations

This thesis provides a tool to compare the total annual fuel consumption of various ship propulsion plants and has the capability of optimising the power generation among the different installed diesel engines.
In the introduction chapter of this thesis, some fundamental questions were asked to structure to research steps. After the development of the design methodology and the validation of the used algorithms, this chapter will provide an answer to these questions. Next to this, recommendations for further research or improvement are given.

6.1 Conclusions

After validation of the developed tool and algorithms some conclusions can be drawn concerning the functionality. Running many different test cases gave clues as to which parameters are sensitive. The changes in the efficiencies are reflected in the changes in the decision variables proportionally. A different efficiency does alter the results, but does not alter the configuration itself. This is backed up by the fact that the changes in the assumed efficiencies will be small, in the range of a few per cent at maximum.
One parameter that did prove to be sensitive was the PTI-PTO changeover parameter α. At first it was thought that this was only to set a difference in the hybrid operation mode in the initial results. The optimisation algorithm will then find the true optimal changeover point. However, setting the changeover at a certain level (e.g. 20% P_B) meant that in the end result the changeover would occur at this point. When the changeover point was increased to e.g. 30% P_B, the changeover in the end result would occur at this new point. This resulted in a lower value of M_{fuel}, suggesting that the first changeover at 20% P_B resulted in an initial solution near a local minimum instead of the global one.
It was also noted that often the best end result originated from a hybrid initial result from the array, or at least an initial result with several hybrid modes in it. This is not that strange, since this approach only takes into account the annual fuel costs as a target and disregards the installation costs. This changeover parameter is only applicable to the hybrid mode. Therefore the sensitivity of α influences the end result often.

As with the dual variables there is no systematic method to show the sensitivity of α. This parameter is only introduced to have a distinction in the hybrid mode in the initial results. It does not occur in the objective function or in the decision variables. It only contributes to the starting values of the GRG method.
Again, the problem with the different starting values arises. They seem to be of high influence since the objective function has many local minima. This can be attributed to

the shape of the sfc data that make up the mf curves. These are all relatively flat; changing the load will change the resulting absolute fuel consumption, but apparently not that much as operating a different engine type. Because that is what this parameter α influences clearly: whether the power should come from a larger main engine on lower load or a smaller diesel generator set with a higher load.

It is therefore advised to set the changeover parameter α at a reasonable assumption based on the operational profile and then run the solver for several configurations. The configuration with the lowest fuel consumption can then be tested for several values of α. If the end result varies a lot, this process can be repeated based on the best scoring α, but then with smaller variations of α. The great variety in the input operational profile makes it rather difficult to set one method that always tests for different variations of α. Sometimes this parameter is not sensitive, so one does not always want to increase the number of solvers to run because this increases the calculation time considerably.

Finally there is one more important assumption that can change the outcome considerably: the input data in the operational profile. These are often based on expectations rather than measured data. The more accurate the given input data is, the more accurate the results will be.

In short the conclusions can be summarised as followed:

- In combination with the orthogonal arrays, the tool provides plausible results. A clear finding is that the optimisation result of the objective function is very dependent on the input, i.e. the initial values. The actual search algorithm only provides small improvements over the initial results from the orthogonal arrays. This suggests that the best combinations from the orthogonal arrays itself are already quite good results.

- The implementation of the optimisation algorithm in Excel gives some practical problems. On major issue is that the solver constraints cannot handle logic functions. This means that all decision variables have to be defined as the required delivered power, not the actual brake power of the components. This does not only introduce complexity in the worksheets, but also problems with algebraic loops.

- The relative flat shapes of the sfc curves and therefore the $\dot{m}_f$ curves of several engines, might lead to a less straightforward search for the global minimum in the optimisation in this accuracy range. Still, the results seem plausible since the tested configurations for the four reference vessels often result in engine loadings around the 85% MCR loading point. This is the loading point where the engines in the MDT portfolio have their lowest sfc.

Making these curves non-linear with a second order polynomial adds some complexity to the algorithm, but the results do not differ from a linear system very much. This suggests the algorithm handles the non-linearity well.

- The user still has to make the decision to which configuration and engine type would be most suitable for a particular demand. The engine selection seems to have more influence on the fuel efficiency than the operation of a particular engine. This again can be contributed to the relatively flat shape of the sfc curves in the range between 50% and 100% MCR loading; the individual value of the sfc at 85% for every engine seems to be more dominant than the actual loading point.

To answer the questions in the introduction:

What are the determining factors to decide whether to use a hybrid propulsion concept?
Operational profile seems to be the most dominant input in the decision to go for a hybrid propulsion system. The shape of the fuel consumption curve of the main engines and diesel generators is what determines the potential for fuel savings. This study focussed on fuel efficiency and left redundancy and flexibility out of the algorithm, so this was expected.

What components are used within the concept and what are their characteristics?
The main components in a hybrid propulsion system are described in chapter 3. The main parameters in the design are the main engine power P_{ME1} and P_{ME2}, the total diesel generator power P_{DG} and the electric machine power P_{EM}. The diesel engines must provide all the required power. The electric machine can transfer power in both directions. Another main component that is important for the losses in the system in the gearbox. Also the frequency converter is required for a variable frequency and the ability to operate with a variable propeller speed, introducing electrical losses.

What determines the layout and the parameters within the design?
In general the required propulsion power determines the brake power of the main engines. The electric auxiliary power demand determines the brake power of the diesel generators.
The fuel saving potential of a hybrid system comes from the operating point of the engines and their respective specific fuel consumption at that loading point. If the extra electric losses are higher than the better sfc point of the engines, the use of a PTO or PTI is not recommended. Since the sfc curve shape is flatter at higher loads, PTO power is more likely to be beneficial. This is especially the case for vessels that have a high timeshare with high load e.g. in transit. Of course the main engines should be able to provide this extra power and need to be larger than the initial required propulsion power.

The positive effect of a PTI occurs when the main engine loading becomes lower than around 25%. At this point the sfc increase is around 20 g/kWh which corresponds to a decrease of efficiency of around 10%. This percentage is in the order of the efficiency loss when operating with an electric motor. So an engine loading below 25% brake power would indicate that a PTI would be beneficial. In this case the diesel generators will have to be larger than the initial required auxiliary power. Operating below 25% load must also be prevented due to possible fouling of the engine.
Operating with a PTI booster function has the main advantage that the installed main engine power can be smaller, but this has to be compensated with larger diesel generators.

Are the results plausible?
To test the plausibility of the results, the design algorithm was validated with several real life examples of ship operational profiles. In conclusion these results were as followed:

- The result of the tanker with one main engine including a PTO for transit and a PTI for the low loads proved to be the best. The actual vessel operates with the same configuration, although the PTI is only used as a redundancy option in the case the main engine fails.
- With the AHTS vessel configuration 3 is the best, with two main engines and a PTI and PTO. The actual design has a full diesel electric system. With this design method a diesel electric system proved to be the second best. However, important factors for a DP system such as manoeuvrability and redundancy are not taken in account here.
- The actual OPV has two main engines and a PTI installed for the slow sailing modes. The best option based on fuel consumption as calculated here is to include both a PTI and a PTO for the transit mode. However this might introduce some extra investment costs for the gearbox, electric machine and other electrical equipment. These are not considered here.
- The fourth tested vessel was a RoPax vessel. The rather rough estimate of the operational profile makes it harder to determine the validity of the results. The presented calculation clearly favours a conventional diesel mechanical system with its higher overall efficiency. The actual vessel has a full diesel electric system. Unclear load variations and possible higher auxiliary loads not specified in this input data might be the cause for this difference.

Which parameters have a greater influence on the concept and which less?
After testing the validity of the results, variations to the various parameters were made to determine their sensitivity. The first changes were made to the assumed nominal efficiencies of the installed equipment. The efficiency of the diesel engines were given, so these can be treated as not sensitive, although they have a large influence. The electric machine efficiency did change the outcome of some of the results, but not the layout of

the configuration. Larger gearbox losses however did change the outcome in some cases. This means that the assumed gearbox efficiency is a sensitive parameter. This can be explained by the fact that the power always has to flow through the gearbox, in both PTO and PTI modes.

Another sensitive parameter is the introduced changeover parameter α that determines the use of a PTO or PTI in the initial starting value of optimisation algorithm. The default was set at 25% of the installed main engine brake power. Below this value a PTI is selected and above this value a PTO will be selected. Changing this value leads to different initial values and this also resulted in different end results. This shows that the initial starting point of the optimisation algorithm is particularly important and the danger of solving for a local minimum is always present.

6.2 Recommendations

The main goal of this thesis was to develop a design methodology to determine the layout and dimensioning of hybrid propulsion systems. The focus was on efficiency and fuel consumption. In order to validate the developed methodology it had to be implemented in MS Excel. This led to some boundary conditions and limitations to the research. One important one is the number of degrees of freedom in the system.

A larger number of degrees of freedom as opposed to the defined four in paragraph 2.4, would require larger orthogonal arrays. For example, disregarding the assumption that a two-shaft vessel would operate symmetrically for all operating modes would introduce three more variables: ME3, ME4 and EM2. The diesel generators are still considered to be one variable.

The number of possible ways to fulfill the power demand would now increase from 3 to 9. This means that the L27 array is no longer applicable. Instead an L81 array is now required, with a minimum of 81 distinct possibilities to be calculated. This causes a problem that would be impractically large to incorporate in Excel.

Another assumption was that the generators always operate with equal load sharing. It is however also possible to run several generators to their maximum allowable loading and let the last generator with fluctuating power. This would mean that the variable DG is now also split into two possibilities, increasing the number of possible ways to fulfill the power demand to 18. This would create so many local minima, that even the orthogonal array method would become impractical.

Other limitations to this research that provide room to further improvement or expansion could be summarized as follows:

- For a full integration of all the components in the engine room, the propeller has to be included. MDT should be in the design process at an earlier stage. This way the potential for the optimum efficiency of the propeller is included. This is more in the region of 5-6 per cent as de Ruyck [2011] and Rolls Royce stated.

- Perhaps this method can be implemented in other software, more stable in algebraic loops and with a more practical solver tool. This could implement more decision variables, such as the PTI/PTO changeover parameter α or the number of cylinders. It would also be useful to make the engine selection automatic. This would mean many extra iteration steps in the solver algorithm, making the Excel platform unstable.

- The number of possible engines could be expanded. For a full span of all the possible propulsion layouts, not only the MDT 4-stroke portfolio should be considered. For example also smaller 2-stroke engines in this power range might be suitable.

- Sfc data is presented as a function of % load. It would be more interesting if this data was specified as a function of both torque and speed, using a so-called mussel diagram. This way the actual fuel consumption based on the assumed load curve (combinator curve) would be more accurate. Unfortunately full information was not available for all engines.

- For more reliable results, the input operational profiles must be more detailed. In this thesis they only served as a test to check the function of the tool. Often the provided operational profiles are specified as estimates, so they cannot be very detailed. In this case the potential for a hybrid system might be uncertain; the results can only be as reliable as the provided input.

- In these results, all fuel consumption was based on the assumption that all engines run on the same fuel. Although most MDT engines run on HFO, it is possible that for whatever reason a second fuel type should be made available for another engine type. This does not necessarily have an influence on mass of the fuel consumption, but it does influence the fuel costs. However, fuel costs are not included in this study. For a full cost analysis, next to the investment costs, the fuel costs have to be included.

- The decision for a certain propulsion plant is based on fuel efficiency. Although these are the main cost driver in the operational costs, it does not provide the full cost picture. Some information regarding investment costs is presented, but to have a good cost comparison a full life cycle costs (LCC) analysis has to be carried out. In such an analysis more important cost factors such as maintenance costs based on running hours and lubrication oil costs play an important role.

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Nomenclature

Symbol	Description	Unit
c	boundary constraint	-
D	diameter	m
E	induction voltage	V
f	frequency	Hz
g(**X**)	constraint function	-
H(**X**)	Hessian matrix	-
h^L	lower heat value	J/kg
i	number of cylinders	-
i	gear ratio	-
I	current	A
J	advance ratio	-
k	number of revolutions per cycle (1 or 2)	-
k_e	number of engines on shaft	-
K_E	coil constant	-
K_M	motor constant	-
k_p	number of propellers	-
K_Q	torque coefficient	-
K_T	thrust coefficient	-
m	number of variables in **X**	-
m	mass	kg
$\dot{m}_f$	mass flow of fuel	kg/s
M	torque	Nm
M*	normalised torque	-
M_B	engine brake torque	Nm
M_{flange}	torque at output flange	Nm
M_p	propeller torque	Nm
M_S	shaft torque	Nm
M_{fuel}	total annual fuel consumption	t/year
$\nabla M_{fuel}(\mathbf{X})$	gradient at point **X**	-
n	number of time intervals in operational profile	-
n	rotational speed	rev/s

n^*	normalised rotational speed	-
n_e	engine rotational speed	rev/s
n_p	propeller rotational speed	rev/s
n_s	synchronous rotational speed	rev/s
N_{DG}	number of running diesel generators	-
p	number of poles	-
P	power	kW
P	pitch	m
P_{aux}	auxiliary power	kW
P^*	normalised power	-
P/D	pitch ratio	-
P_B	brake power	kW
P_{cu}	copper losses	kW
P_D	delivered propulsion power	kW
P_{DG}	diesel generators effective power	kW
P_E	effective towing power	kW
P_{aux}	effective electric power demand	kW
P_{EM}	electric machine effective power	kW
P_{Fe}	iron losses	kW
$P_{friction}$	friction losses	kW
p_{me}	mean effective pressure	Pa
P_{ME1}	main engine 1 effective power	kW
P_{ME2}	main engine 2 effective power	kW
P_O	open water propeller power	kW
P_p	propeller power	kW
P_S	shaft power	kW
P_T	thrust power	kW
Q	torque	Nm
$\dot{Q}_f$	heat flow of fuel	J/s
R	ship resistance	N
R	electrical resistance	Ω
R_{Fe}	electrical resistance in iron	Ω
s	number of test cases in orthogonal array	-
s	slip	-
S	set (range of engines)	-
sfc	specific fuel consumption	g/kWh
t	thrust deduction factor	-

T	thrust	T
t_n	time per mode n	hr/year
U	voltage	V
v_A	advance velocity	m/s
v_R	relative water velocity	m/s
v_s	ship speed	m/s
V_s	stroke volume	m3
w	wake factor	-
W_e	effective work	J
X	variables vector (P_{ME1}, P_{ME2}, P_{DG}, P_{EM})	-
X'	intermediate position of **X**	-

Greek Symbol	Description	Unit
α	crank angle	rad
α	changeover parameter between PTO and PTI	-
∂	partial derivative	-
η	efficiency	-
η^*	part load efficiency	-
η_b	bearing efficiency	-
η_D	propulsive efficiency	-
η_e	engine efficiency	-
η_{FC}	frequency converter efficiency	-
η_{GB}	gearbox efficiency	-
η_{gen}	generator efficiency	-
η_H	hull efficiency	-
η_{motor}	electric motor efficiency	-
η_O	open water efficiency	-
η_P	combined plant efficiency	-
η_R	relative rotative efficiency	-
η_s	shaft seal efficiency	-
η_S	shaft efficiency	-
η_{swb}	switchboard efficiency	-
η_t	teeth efficiency	-
η_{trm}	transmission efficiency	-
θ	pitch angle	rad
λ	Lagrange multiplier	-

Λ	Lagrangian function	-
ρ	density	kg/m3
φ	phase shift	rad
Φ	magnetic flux density	Wb
ω	rotational speed	rad/s

List of abbreviations

Abbreviation	Description
AC	Alternating current
AHTS	Anchor handling, tug & supply vessel
APM	Alternative propulsion motor
B	Breadth
CODLAD	COmbined Diesel eLectric And Diesel
CODLAG	COmbined Diesel eLectric And Gasturbine
CPP	Controllable pitch propeller
CR	Common rail
CSI	Current source inverter
CSR	Continuous service rating
DC	Direct current
DG	Diesel generator set
DP	Dynamic positioning
Dwt	Deadweight
EFD	Energy flow diagram
EM	Electric machine
EM	Engine margin
EMF	Electromotive force
ES	Energy source
FC	Frequency converter
FPP	Fixed pitch propeller
GRG	Generalised reduced gradient
GT	Gross Tonnage
HFO	Heavy fuel oil
HSVR	High Speed Vessel Rules
IGBT	Integrated gate-commutated thyristor
IMO	International Maritime Organisation
KKT	Karush-Kuhn-Tucker
LCI	Load-commutated inverter
Loa	Length over all
MCR	Maximum continuous rating
MDO	Marine diesel oil
MDT	MAN Diesel & Turbo SE
ME	Main engine
MGO	Marine gas oil

MSB	Main switchboard
MT	Motor tanker
OPV	Offshore patrol vessel
PM	Particulate matter
PTH	Power Take Home
PTI	Power Take In
PTO	Power Take Off
PWM	Pulse width modulation
RoPax	Roll-on Roll-off and passenger vessel
sfc	Specific fuel consumption
SM	Service margin
STC	Sequential turbocharging
T	Draught
VSI	Voltage source inverter

Appendix A

A brief explanation about the efficiency between the effective towing power P_E and the required installed brake engine power P_B of the engines will be given in this appendix.

The propulsive efficiency η_D contains the open water efficiency of the propeller η_O, the relative rotative efficiency η_R and the hull efficiency η_H:

$$\eta_D = \eta_O \cdot \eta_H \cdot \eta_R \tag{A.1}$$

The open water efficiency η_O is the efficiency of the propeller in open water. It is a measure of how effective the propeller ideally is in transforming the open water (i.e. homogeneous inflow) torque power P_O to actual thrust power P_T:

$$\eta_O = \frac{P_T}{P_O} \tag{A.2}$$

The hull efficiency is the ratio between the effective towing power P_E and the thrust P_T that the propeller delivers to the water:

$$\eta_H = \frac{P_E}{k_p \cdot P_T} = \frac{1-t}{1-w} \tag{A.3}$$

Where k_p is the number of propellers. [A.3] also shows the thrust deduction factor t and the wake factor w. The thrust deduction factor holds effects of the propeller sucking in water from behind the hull and thus creating an added resistance. The wake factor is the difference of the ships speed and advance velocity in front of the propeller, as ratio of ships speed.

$$t = \frac{k_p \cdot T - R}{k_p \cdot T} \rightarrow R = (1-t) \cdot k_p \cdot T \tag{A.4}$$

$$w = \frac{v_S - v_A}{v_S} \rightarrow v_A = (1-w) \cdot v_S \tag{A.5}$$

The relative rotative efficiency takes into account the difference of a uniform flow into the propeller (open water flow) and when there is a hull in front of it. It is the ratio between the open water power P_O and actual delivered power P_P but does not differ much from 1:

$$\eta_R = \frac{P_O}{P_P} \tag{A.6}$$

The delivered power P_D to the propellers is then:

$$P_D = \frac{P_E}{\eta_D} = k_p \cdot P_p \tag{A.7}$$