Delft University of Technology

Faculty Mechanical, Maritime and Materials Engineering

Designing and evaluating propulsion concepts of surface combatants

MSc thesis

Report nr:
SDPO.11.019.m

LTZT3 G.F. van Es
Student nr: 1218301

September 16, 2011

Defensie Materieel Organisatie
Ministerie van Defensie

TU Delft

Mechanical Engineering – Transportation Engineering – Mechanical Systems and Integration

MSc graduation project:

Designing and evaluating propulsion and power generation concepts of surface combatants

Delft University of Technology

Faculty of Mechanical, Maritime and Materials Engineering (3ME)
Mekelweg 2
2628 CD Delft

Professor:
Prof. ir. D. Stapersma

Supervisors:
Dr. ir. H.T. Grimmelius (DUT)
Ir. E. van Dijk (DMO)
Ir. M.W. Jansen (DMO)
KLTZT J.P. Spruit (DMO)

Delft, 2011

Abstract

Sometime – probably quite at the beginning – in any ship design process the question raises about what propulsion and power generation configuration should be installed on the ship. A methodology is developed to design and evaluate different configurations, and finally come to the 'best' solution. Different components, with all different characteristics, have to be put together on the platform, in order to meet the requirements and perform its tasks succesfully. The performance of the propulsion and power generation configuration can be assessed on a number of criteria. A multiple criteria analysis will reveal the 'best' solution.

Contents

Preface

In order to complete the Master of Science (MSc) programme in Mechanical Engineering at the Delft University of Technology (DUT), I started my graduation project in September 2010. After completing my Bachelor of Science (BSc) equivalent studies on the Netherlands Defence Academy (NLDA) on Military Platformsystems, I got an offer from the NLDA I couldn't refuse. They offered me the opportunity to get my MSc degree in Mechanical Engineering on the DUT. This MSc programme takes two year, of which one year is the graduation project. Because I work for the Ministry of Defence (MoD), I thought it would be nice to do my graduation project for this organisation. A very interesting part of the MoD is the Defence Materiel Organisation (DMO). The DMO is responsible for all materiel within the Defence organisation: from procurement and major maintenance to disposal. I came into contact with Martin Jansen of the Platform Technology division, which is part of the Directorate of Weapon Systems, Sea Systems branch. He came up with an interesting subject for my graduation project. He told me about a design study on a future surface combatant that had recently started, and brought me into contact with the lead designer of the design study, Erik Takken. Together we agreed on the plan that I would investigate possible propulsion and power generation configurations for this design study. So it happened, and the result is in front of you.

I would like to use this preface to give some special thanks to some people that have been of great help to me during my graduation project. In random order: Edwin van Dijk, Kees Posthumus, Ab Blokland, Martin Jansen, John-Paul Spruit, Erik Takken, Joos Bongartz, Thomas Hetherington, Martin Uitdenbogerd, Joop van Son, Ron Straks and Henk Post of the DMO, Douwe Stapersma, Hugo Grimmelius and Henk Polinder of the DUT. Everyone has been of great help in collecting the required data. I would also like to thank Tom Stuivenberg and Mark Verzeilberg for their assistance in the use of LaTeX, because this was my first time using this typesetting program. Finally, I would like to thank Marieke, my girlfriend, for her great support during this tough, and seemingly everlasting year.

Chapter 1

Introduction

Within the Royal Netherlands Navy (RNLN) a design study has recently been start up for a replacement platform for the, already 20 years old, Karel Doorman-class multipurpose frigates. The design study is hidden under the name: Surface Combatant (SFC). As with every design study of a naval platform, at sometime in the design process – probably quite at the beginning – the question raises:

What propulsion and power generation configuration is best for the ship?

After all, the ship needs propulsion and electrical power to perform its intended tasks. This question is the main question of this study. It is a question that is asked in every ship design study, but it is worked out in this thesis on the basis of the SFC design study. The purpose is a deliberate choice for a propulsion and power generation concept for the future surface combatant. But, the seemingly simple question is much more complex than it initially seems.

The propulsion and power generation configuration significantly contributes to the capabilities of the ship. So, the problem starts with the first subquestion:

What are the requirements for the ship?

This subquestion is covered in Chapter 2. The requirements for the ship are described in outline in the Operational Concept (OC), but it doesn't put clear demands on the technical details of these requirements. The requirements in the OC need to be translated into technical details. For example, the required maximum speed determines the installed propulsion power and the requirements on sensor-, weapons- and communication systems (SEWACO) affect the amount of generated auxiliary power. In military ships there are often the special requirements on shock resistance and signature profile of the ship.

Once the requirements in the OC are translated into technical requirements, a propulsion and power generation configuration can be put together which meets the requirements. Every configuration comprises of an energy source and a number of energy converters to convert the energy from the source ultimately into movement of the ship. Selection can be made from a rather limited number of energy sources. Though, the number of energy converters that can be selected from is much larger. Every component has its own characteristics. Before the engineer can decide on the components to use, he needs to know the characteristics. So, the next subquestion is:

What are the component characteristics?

There are a lot of components, that are all contributing to the propulsion and power generation of the ship, but in this thesis is only focussed on the main components. Chapter 3 gives an outline of the most common main components and their characteristics, as far as it is possible to say something about them in an early design stage. The relevant characteristics are the available power, dimensions, weight, operating speed, efficiency, signature profile, maintainability, reliability and purchase costs. The characteristics of components are different per component and per type and manufacturer. It would be useful for the engineer if he could estimate on the characteristics in an early design stage in order to compare components and propulsion concepts. In this study, it is tried to develop models, as accurate as possible, that do estimations on the component characteristics. It has to be kept in mind that the models serve as estimation models in the early design stage, so they can't be too detailed.

If the engineer knows the characteristics of the main components, he can put them together in order to come to a propulsion and power generation configuration for the future ship that meets the requirements. The combinations of components are almost endless. The type of components, the power level, the number and the layout can all be varied, which results in different propulsion concepts. In order to compare the different concepts, every concept needs to be assessed on a number of criteria that describe the characteristics of that concept. The next subquestion is:

What are the characteristics of the propulsion and power generation concept?

The assessment criteria should be related to the requirements in the OC in order to easily determine the suitability of each concept. The relevant concept characteristics are the maneuverability, susceptability, survivability, number of main components, space consumption, weight, required fuel capacity, reliability, maintainability, vulnerability, purchase costs, annual fuel costs and maintenance costs. Together these characteristics give an idea on the overall performance of a propulsion and power generation concept. In Chapter 4 a number of possible concepts for the SFC is worked out, and the method of assessment is illustrated.

If all propulsion concepts are assessed on the criteria, it has to be made up which one is the best. This brings back to the main question of this study:

What propulsion and power generation configuration is best for the ship?

A methodology is needed to weigh the scores on the assessment criteria of all concepts. This can be done in a Multiple Criteria Analysis (MCA). Each assessment criterion gets a weight factor that indicates the importance of that criterion. Combining all results should lead to the 'best' solution. The word 'best' is emphasized, because this is a very subjective word which can have a different meaning to everyone. In Chapter 5 the methodology of performing a MCA is discussed. From the MCA a 'best' solution should be found, according to the definition of 'best' of the author. But because the definition of 'best' is so subjective and differs per person, it is interesting to see how robust the outcome of the MCA is with respect to a changing definition of 'best'. In a sensitivity analysis the stability of the solution has to be determined to see how valuable the solution is.

The final conclusion of this study will be a deliberate advise for a particular propulsion and power generation concept for the future Surface Combatant, and recommendations for improving the selection process.

Chapter 2

About project: Surface Combatant

In this chapter, the project *Surface Combatant* (SFC) within the Defence Materiel Organisation (DMO) of the Netherlands Ministry of Defence (MoD) will be explained and described. Formally, it is not yet called a project, but still a design study because the parlement has not yet agreed on it. For the sake of convenience, in this thesis it will be called a project. The project is also called *M-frigate replacement* and it forms the basis for this thesis. Some background information about the formation of the project and the design process will be explained and an insight in the initial requirements that are described in the Operational Concept (OC) will be given. These initial requirements determine to great extent what the propulsion and power generation systems will look like.

2.1 Design process

Moss & Thomson (1994) explain the design process of warships very clear:

> *"The starting point of any warship design has to be an operational requirement, usually defined in terms of operational role, weapons fit, speed and range etc. With these targets the designer sets down a preliminary idea of the outline ship design. Once the designer established a first estimate for size and shape of the ship, the powering requirement can be made. This in turn sets the requirements for machinery space and weight and for fuel, to achieve the desired range. The process continues to consider other issues such as arrangement, manning, vulnerability, signatures etc., before the first design iteration is complete. This cycle is then repeated to better accuracy until the final balanced and compliant design is achieved."*

The realization of the design process within the NLMOD will be explained briefly. The Directorate of Operational Policy, Requirements and Plans (DOBBP)[1] within the Netherlands MoD places a need for a new ship, in the form of an Operational Concept (OC). The OC describes in terms of operational needs what the ship should look like and what it should be capable of. This OC is deposited at DMO, section *Concept Analysis*. The naval engineers of this section translate the OC into multiple varying conceptual designs in cooperation with a.o. the sections: *Sensor, Weapons and Communication systems* (SEWACO), *Propulsion and platform systems*, *Signatures*, *Vulnerability*, *Construction* and *Hydromechanics*. Together with DOBBP the concepts are analysed on vessel capability and costs and the level of ambition might be adjusted if the initial level seems to be too high. In an iterative process with DOBBP, DMO will change the concepts until all players agree on a concept. Then a shipyard, and producers of subsystems (e.g. weapon and sensor systems), must be found that will design and build the ship.

[1]Dutch abbreviation for: Directie Operationeel Beleid, Behoeftestellingen en interne Plannen

A number of these design cycli are carried out, which return different designs with other optimization strategies: a design-to-requirements, multiple designs-to-costs, a couple of designs-to-manning and some unconventional out-of-the-box designs to widen the scope.

2.2 Formation of the project

The Royal Netherlands Navy (RNLN) came to the conclusion that there is need for new ships that will replace the current Multipurpose frigate of the Karel Doorman class. The M-frigate replacement is initially planned for the period 2019-2021, but this date may very well be postponed due to shrinking budgets. The first ship of the Karel Doorman class was commissioned in 1991. The M-frigate is designed for a service life of about 25-30 years, so these frigates are getting at the end of their service life. Besides that, the M-frigates no longer completely serve the needs of the operational and strategical path that the RNLN wants to follow in the future. The ship was designed for the threats and the warfare of the Cold War. Nowadays, threats are much different from that time. The RNLN has to deal with, so called, asymmetric warfare more and more. This means that the enemy is not always another warship, submarine or jetfighter, but can as well be a small group of terrorists in a rubber boat with rocket-propelled-grenades (RPG) as a weapon. This puts other requirements on the ships design and the design of sensor and weapon systems.

Ships will nowadays be used for much more tasks than only warfare. They must be operable in Maritime Security Operations (MSO) and be able to secure merchant ships against pirates and terrorists. Humanitarian operations and emergency relief are among the possible operations as well as Maritime Interdiction Operations (MIO). The operational area of the ship has moved from blue water (open seas) more to brown water (coastal waters). This has impact on a.o. the design of power systems, but also on for example the sensors and weapons. For example, ships will sail less at top speeds or transit speeds and more at lower operational speeds, or because they operate more in coastal waters, where emission controlled areas (ECA) are located, clean engines/fuel should be chosen.

Next to the 'modern' tasks of the ship, it still has to be able to do the 'classical' tasks:

Anti-Submarine Warfare (ASW)

Anti-Surface Warfare (ASuW)

Anti-Air Warfare (AAW)

These specified tasks also have a certain impact on the design of the power systems. For example, for ASW operations the sonar installation is needed which is used at certain speeds. At those speeds, the ship should be very quiet (engines and propeller) in order not to disturb the sonar operation and not giving away its own position.

2.3 Requirements

The OC describes in a *Capability Statement* on a very high level how the operational needs affect the design of the ship. It describes some requirements for the sensor and weapon systems, and for the hull design. Those requirements are not very relevant for this thesis, apart from the fact that it dictates the available space for power systems and it gives an idea of the required electrical power for the SEWACO systems.

Design requirements are selected from the SFC Capability Statement and listed below. The selected requirements are the ones that have direct impact on the design of the power systems.

All requirements are under discussion in the iterative process between DMO and DOBBP, but it is not within the scope of this research to question these. These are operational, technical and financial decisions. In this study the design of propulsion and power generation systems based on these initial requirements is worked out.

The following relevant requirements on the design of the propulsion system are filtered from the capability statement and are the starting point for this study:

Availability

- SFC should have a maximum speed of approximately 30 knots up to and including Sea-State 3, minimal 26 knots
- SFC shall have transit speed of 18 knots up to and including Sea-State 3
- SFC shall have a range of 5000 nm at transit speed up to and including Sea-State 3
- SFC should have a lifetime of 35-40 years
- To improve reliability, power systems must not be designed on the limits, but a certain margin must be used

Effective intelligence

- SFC shall use UAV/USV/USSV/UUV/AUV[2] for collecting data
- SFC shall have long range sonar (active and passive) for ASW picture compilation

Mobility and deployability

- SFC shall operate in blue and littoral waters
- SFC shall operate from pole-to-pole in open waters (incl. tropics). No ice class required
- SFC should have unmanned launch and recovery systems for USV/USSV/RHIB[3]/UAV/UUV/AUV
- SFC could have Dynamic Positioning (if required for UxV handling)
- SFC should have short machinery reaction time (fast power delivery and short start-up times)
- SFC shall be able to sail slow for long periods

[2] Unmanned Aerial Vehicle/Unmanned Surface Vehicle/Unmanned Sea Surface Vehicle/Unmanned Underwater Vehicle/Autonomous Underwater Vehicle

[3] Rigid Hull Inflatable Boat

Logistic sustainability

- SFC shall be capable of receiving F76 and F44 from supply ships at sea
- SFC shall have low energy consumption
- SFC should employ proven COTS[4] platform technology (*"Civil products where possible, military where necessary"*)
- SFC should incorporate common equipment with relation to other RNLN ships
- SFC should have long maintenance intervals and short maintenance periods
- SFC should have high quality equipment

Survivability and force protection

- SFC should have a reduced infrared (IR) signature
- SFC shall have a reduced underwater acoustic signature to enable ASW operations and reduce detection (0-15 kts high priority)
- SFC shall remain safe sailing after a stand-off underwater explosion (shock resistance)
- SFC should apply redundancy for mission essential systems
- The engine room concept should be redundant minimal up to cruise speed
- Redundancy for 1 engine room failure

From the translation of the capability statement and the current state of technique, it seems that there will be two different towed sonar systems onboard Low Frequency Active Sonar (LFAS) and a Towed Array Sonar (TAS) to meet the requirement stated in "Effective intelligence". The impact of these systems on the platform layout is significant. Both are sonar systems that are towed behind the ship and need large storing spaces at the aft of the ship. Description of these different sonar equipment and operations lays not within the scope of this thesis, but three different important operational speeds are distinguished based on different sonar operations:

→ Operational speed 1 (15 kts): operational speed for LFAS operation

→ Operational speed 2 (12 kts): operational speed for TAS operation

→ Operational speed 3 (10 kts): optimal speed for TAS operation

These operational speeds are important in designing the propulsion concept.

A requirement in the list above that does not seem relevant at first sight is that the ship should be capable of carrying, launching and recovering all kinds of unmanned vehicles, collected under the name UxV. This requirement has impact on the hull design, because some kind of launch and recovery installation should be present for those unmanned vehicles. For the unmanned water vehicles this will logically (not necessarily) be a ramp at the aft of the ship. The presence of such a ramp for example would make the use of a waterjet very difficult.

[4] Commercial Of The Shelf

2.4 Conceptual Design

Up to this date, the Concept Analysis section has made a number of conceptual designs-to-requirements. One of the conceptual designs is picked for this study. This conceptual design is coded **VIIIa** and described in Takken (2010). An artist impression is found in figure 2.1.

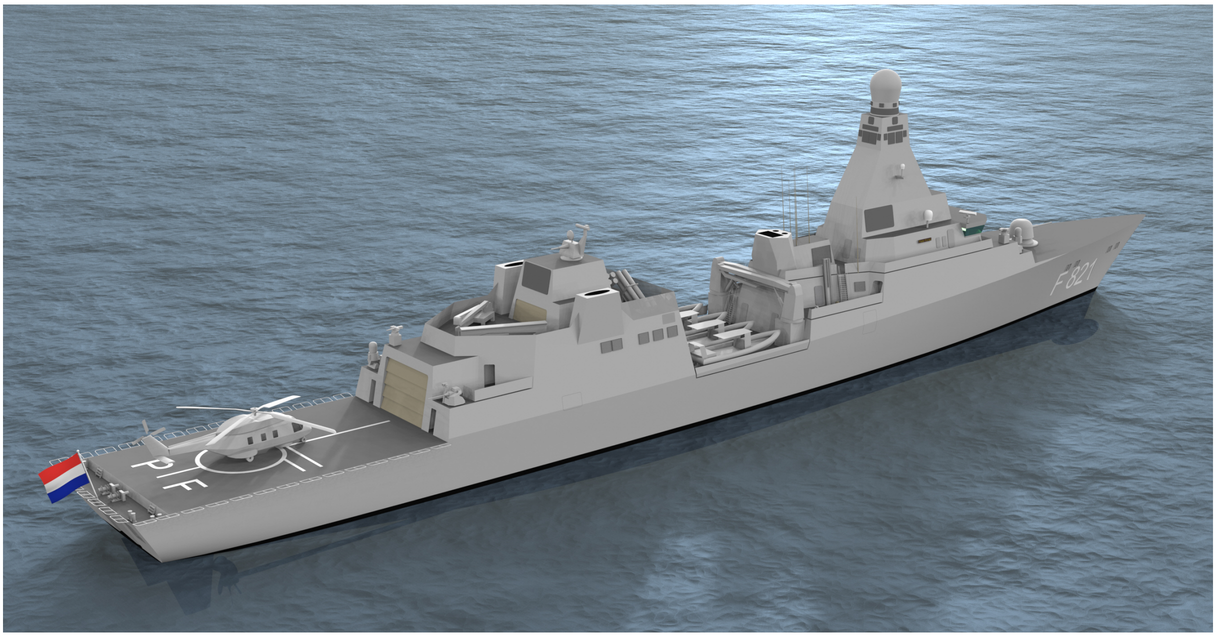


Figure 2.1: Artist impression of the concept version VIIIa revision 3

(source: Takken (2010))

Ship resistance

Concept VIIIa is a conceptual design with a displacement of approximately 5200 ton. A resistance curve of this conceptual design is determined by the Concept Analysis section with a corrected Holtrop & Mennen method. The resistance curve is plotted in figure 2.2.

The Holtrop & Mennen method is widely used at the initial design stage of ships for estimating the resistance of a ship based on data of other ships. Holtrop & Mennen did a statistical evaluation of model test results, selected from the archive of the Netherlands Ship Model Basin. The evaluation was carried out using multiple regression analysis methods (Holtrop & Mennen, 1982). At the DMO a corrected method is used, that is based on data of naval ships.

The Holtrop & Mennen method returns a.o. approximations of ship resistance, thrust deduction factor (t) and wake fraction (w) at every ship speed with a number of parameters describing hull shape as an input. The resistance curve is based on full displacement conditions, seastate 3, 6 months out of dock. Semi-planing or planing is not taken into account, so for the high speeds the resistance curve might be a little to pessimistic.

Figure 2.2 also shows the ships resistance in trailing shaft conditions. A trailing shaft adds to the appendage drag of the hull, thus increasing the resistance of the ship. Also, the rudder increases the total resistance because it has to be at an angle to compensate for the asymmetric thrust. The additional resistance is assumed to be 15% of the resistance overcome by that propeller, according to assumptions made in KleinWoud & Stapersma (2003). In a 2 propeller concept, the assumed total resistance that has now to be overcome by the remaining propeller is calculated with:

$$R = \frac{R_0}{2} + 1.15 \cdot \frac{R_0}{2} = 1.075 \cdot R_0 \tag{2.1}$$

With R is ship resistance, and index 0 indicating initial condition: no trailing shaft.

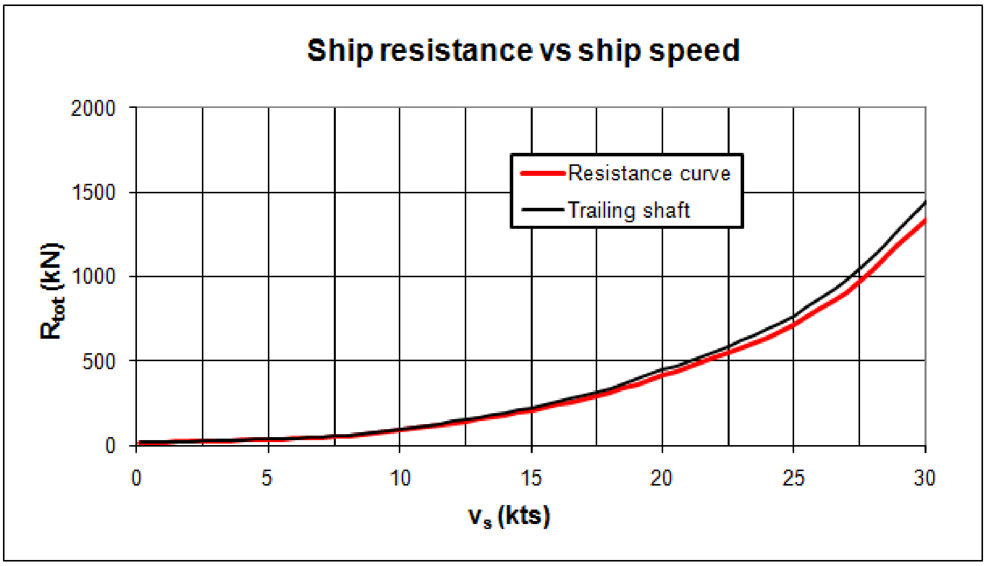


Figure 2.2: Total ship resistance (incl. appendages) in (kN) vs ships speed (kts) of the concept version VIIIa revision 3

Propulsion power

The corrected Holtrop & Mennen method also calculates the brake propulsion power (P_B) that has to be delivered by the engines. To calculate the P_B from the ship resistance curve, some assumptions and approximations are made about the losses along the line from propeller to flange.

Losses occur at the hull (hull efficiency: η_H) as a result of thrust deduction (t) and wake factor (w), at the propeller (open water propeller efficiency: η_O) and as a result of non-uniform velocity field (relative rotative efficiency: η_R), at the gearbox (gearbox efficiency: η_{GB}) and other transmission losses (η_{TRM}). On top of that a certain seamargin (SM) is applied. This margin is derived from the difference between theoretical methods and measurements on other ships. For operation at sea there also has to be accounted for some additional power, due to seastate ($P_{add,ss}$) and wind ($P_{add,wind}$). Now P_B can be estimated from the resistance curve, taking into account all mentioned losses, see equation 2.2. Figure 2.3 presents the result.

$$P_B = \underbrace{\underbrace{\underbrace{\underbrace{\underbrace{\underbrace{\underbrace{\frac{R}{k_p \cdot (1-t)}}_{T} \cdot v_s \cdot (1-w) \cdot}_{P_T} \frac{1}{\eta_O}}_{P_O} \cdot \frac{1}{\eta_R}}_{P_p} \cdot k_p}_{P_D} \cdot SM}_{P_{D,sea}} \cdot \frac{1}{\eta_{GB} \cdot \eta_{TRM}} + P_{add,wind} + P_{add,ss}}_{P_B} \tag{2.2}$$

With v_s is ship speed and k_p the number of propellers.

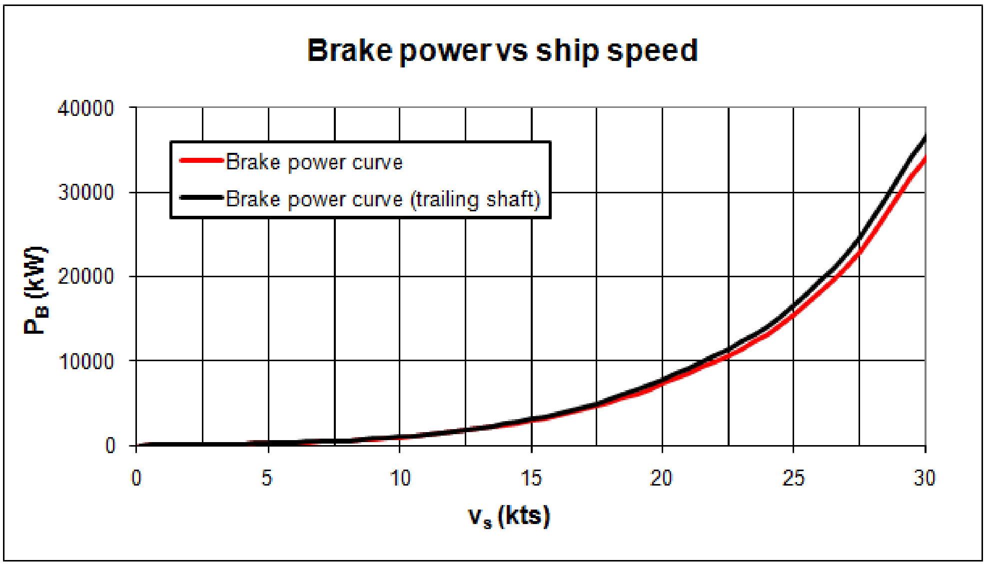


Figure 2.3: Brake propulsion power (incl. wind and seastate addition) in (MW) vs ships speed (kts) of the concept version VIIIa revision 3

In the corrected Holtrop & Mennen method, the losses are calculated with:

$$\eta_H \stackrel{\text{def}}{=} \frac{1-t}{1-w} \tag{2.3}$$

$$\eta_O = \frac{2}{1+\sqrt{1+C_T}} - 0.175 \tag{2.4}$$

$$\eta_R = 0.948 \tag{2.5}$$

$$\eta_{GB} \cdot \eta_{TRM} = 0.932 \tag{2.6}$$

$$SM = 1.15 \tag{2.7}$$

With C_T is thrust loading coefficient (see equation 3.113 on page 111), -0.175 is a correction factor on ideal axial propeller efficiency. η_R is a function of the prismatic coefficient of the underwater body of the ship (C_P), the longitudinal position of the center of buoyancy (L_{CB}) and the pitch/diameter ratio of the propeller (P/D).

The additional wind power ($P_{add,wind}$) is a function of ship speed and ship's breadth over all. For *this* ship it is calculated with:

$$P_{add,wind} = 14.85 + 1.37 \cdot v_s \tag{2.8}$$

The additional seastate power ($P_{add,ss}$) is a function of ship speed, ship's length between perpendiculars and ship's breadth on the waterine, and is calculated for *this* ship with:

$$P_{add,ss} = -1.275 \cdot 10^{-3} \cdot {v_s}^2 + 22.24 \cdot v_s \tag{2.9}$$

In equation 2.8 and 2.9, v_s is in knots.

Auxiliary power

The auxiliary power of Concept VIIIa is estimated based on comparable ship types (M-frigate, Holland class OPV and Sigma class Indonesian corvette).

Bowthruster

There is discussion whether a bowthruster is needed. This would be the case if there are requirements for dynamic positioning (DP) or autonomous mooring without tugboats. Probably these requirements are not relevant and a bowthruster is not needed, but if it does the power of the bowthruster is estimated around 1 MW. From here on, the bowthruster is not taken into account anymore.

Summary on Conceptual Design

Relevant data on the Concept VIIIa is summarized in table 2.1, in which the three operational speeds (see page 6), transit speed and required maximum speed are recognized. For electrical power demand three different conditions are distinguished: harbour, transit and operation. Operational mode is a situation in which all SEWACO systems are in use or standby and the ship has its full capability standby. This operational mode represents the maximum power demand. During the design process the conceptual design will change, and so do the propulsion power and auxiliary power demand, but the data given in Takken (2010), as listed in table 2.1, is used in this thesis.

Table 2.1 also mentions the Froude number (Fn) at every speed. The Froude number is a dimensionless number that is used to determine the resistance of a ship (including wave making resistance), and permits the comparison of ships of different sizes. It is defined as:

$$\mathrm{Fn} = \frac{v_s}{\sqrt{g \cdot L}} \tag{2.10}$$

With v_s is ship's speed, g is the gravitational acceleration and L is the length of the ship at the water line.

Propulsion data

	Max. speed	Transit	Operational 1	Operational 2	Operational 3
v_s (kts)	30.0	18.0	15.0	12.0	10.0
P_B (kW)	36800	5600	3200	1700	1100
Fn (-)	0.43	0.26	0.22	0.17	0.14

Electrical data

	Harbour	Transit	Operation
Power demand (kW)	800	1100	1350

Hull data

Thrust deduction factor t (-)	0.063
Wake factor w (-)	$\approx$ 0.045
Draught T (m)	$\approx$ 5
Max. propeller diameter D_p (m)	$\approx$ 4.5

Table 2.1: Summary of ship data as described in conceptual design in Takken (2010)

Part I

Components

Chapter 3

Main components

This thesis is about designing and evaluating propulsion and power generation concepts for a particular purpose. Propulsion on a warship asks for a different approach than on other ships. There are a lot of requirements to meet. But there are also a lot of components and machinery on the market, and still developed every day, that can meet these requirements and fulfill the tasks. In the design phase it is the question: *What components to use?*. All components have their own characteristics. The purpose of this chapter is to get insight in the properties and characteristics of the main components. The list of components is not complete, because it is simply too much, and not all components are relevant for this study. Non-proven techniques are left out of scope, only components that are reasonable and mature for immediate transition to naval warships are considered. The choice of the components that are described in this study is based on the essential choice of fuel to be used. Only prime movers running on marine diesel fuel are considered. The choice of fuel is explained in the first section. The following list of main components of interest will be described:

- Diesel engine
- Gasturbine
- Fuel cell
- Electrical machines (motor + generator)
- Gearbox
- Electrical auxiliaries (switchboard + converter)
- Cooling system
- Propeller
- Waterjet
- Podded propulsor

In the design of propulsion concepts it will always be a trade-off between capabilities and costs. Capability in this sense is the contribution of the component to performing the task of the ship, i.e. the **effectiveness**. Costs are the initial purchase but also the through-life costs for fuel and maintenance, i.e. the **(cost)efficiency** (not to confuse with fuel efficiency). As a matter of fact it is a trade-off between effectiveness and efficiency. Efficiency can pretty well be described by numbers, just adding all the costs, but it is difficult to give numbers to effectiveness. Below are listed the component characteristics that are described in this study because they either influence the effectiveness or the efficiency of the ship:

- *Power:* the amount of power determines the ability to accelerate up to a certain maximum speed (maneuverability)
- *Dimensions:* determines the space consumption of the component, thus the space that is left for task-related equipment
- *Weight:* total weight determines the buoyancy volume, thus dimensions of the hull, thus ship resistance
- *Operating speeds:* determine the need for speed conversion (e.g. gearbox or electrical converter)
- *Efficiency:* determines the fuel usage of the ship, thus fuel costs and range
- *Signatures:* determines susceptability of the ship, which is very important for warships to be able to perform tasks undetected
- *Maintainability:* determines the number of maintenance personnel on board, the maintenance costs and the operational readiness (maintenance down-time)
- *Reliability:* determines the chance of failure but also the number of spare parts and unscheduled maintenance tasks
- *Initial purchase costs:* determine the initial cost of the ship

Methodology

The goal is to describe all components in terms of the above characteristics. The structure of power, dimensions, weight etc. will for that reasons be found for every component. The information is collected from manufacturers information, available data at DMO from RNLN vessels, expert experience, literature and from GES[1].

GES is a software tool, developed by national research organization TNO[2] with which energy concepts can be analyzed and designed. Preperatory to this software tool, a similar study like in this thesis has been carried out within the projectgroup AES[3]. This preperatory study is described in van Dijk *et al.* (1998). The AES project has been carried out in 1998/1999 under supervision of the Netherlands Institute for Maritime research (NIM). In this study it was tried to describe main components in terms of dimensions, weight, efficiency, life span, Mean Time Between Failure (MTBF), Mean Time To Repair (MTTR), initial purchase costs (IPC) as a function of the rated power. These models have been implemented in GES. The implementation of the models is described in van Vugt *et al.* (1998). In this thesis, the GES models are tested with known data from RNLN or manufacturer's information, and corrected where necessary. The initial plan at the start of this thesis was to use the relations from the AES study, but during research it was found that a lot of those relations do not fit recent data and had to be redefined.

Note: *The relations, as determined in this study, serve as indication. For exact numbers on dimensions, weight, efficiency etc. manufacturers need to be consulted.*

The used data is collected from manufacturers information (techspecs, projectguides, websites, brochures) and from machinery currently in service at the RNLN. All collected data was put into databases in Microsoft Excel. Some bigger, some smaller, depending on the available

[1]Dutch abbreviation for "Geïntegreerde Energie Systemen" meaning Integrated Energy Systems
[2]Dutch abbreviation for "Toegepast Natuurwetenschappelijk Onderzoek" meaning Applied Science Research
[3]All Electric Ship

information. Excel was used because the data can easily be presented in figures and relations between variables can easily be determined with the 'fit trendline' function. This function also gives information about the accuracy of the fitted trendline with the coefficient of determination, also called R^2. R^2 is a value between -1 and +1, with -1 perfect negative correlation and +1 perfect positive correlation and 0 is no correlation. It gives a measure of how certain one can be in making predictions based on the data, it represents the percentage of the data that is the closest to the line of best fit. For example, if $R^2 = 0.85$ it means that 85% of the total variation in y can be explained by the given equation, and the other 15% remains unexplained.

Besides just putting a trendline through the data, in some cases physical analysis based on fundamental input parameters is done. For example, there is a physical relation between the delivered torque and the rough dimensions of a diesel engine. With these physical analysis it is better possible to interpret the trendlines as constructed by Excel.

Signatures

A special topic in designing propulsion concepts for warships, is signatures. A ship always has a certain signature profile, that can be detected by others. Signatures can be used by the enemy to locate the ship's position, to guide missiles or to denonate an underwater mine. For a surface combatant it is crucial to have a low signature profile. Underwater and above water signatures are distinguished. Underwater signatures are underwater noise, electric and magnetic signature profile and pressure signature. Above water are infrared, radar-cross section, laser and visual signature.

- *Underwater noise:* structure-borne (transmits via the foundation into the ships hull into the water) or air-borne (transmits via the air into the ships hull into the water). Caused by vibrations of the machinery, and easily transmits via the foundation or auxiliary piping and hosing into the ships hull. Resilient mounting of the machinery is in some cases an option to damp the noise.

- *Magnetic:* the distortion of the earth magnetic field generated by a ferro-magnetic object. Typical non-ferrous materials generate no distortion. Permanent or induced magnetic field. Permanent field is independent from the magnetic earthfield and can be removed by *deperming* treatment. Deperming treatment is performed by passing the ferro-magnetic components through a proper de-magnetizing coil. The induced signature is generated by ferro-magnetic materials subjected to the earthfield and can only be compensated by means of *degaussing* coils. A solution to prevent magnetic signature is to use anti-magnetic materials (e.g. glass reinforced plastics, stainless steel), but these are in many cases unaffordable and would induce severe technical risks.

- *Electric:* the induced electric field in the seawater (electrolyte) causes ionic currents which can be detected. Static or alternating electric fields are generated. Causes for static fields are corrosion and corrosion-protection currents. Alternating fields are caused by for instance variations of the contact resistance in the propulsion shaft bearings (shaft rate/blade rate) and stray fields of electric components inside the ship.

- *Pressure:* the pressure wave caused by the movement of the ship through the water. Depending on the hull shape and the speed of the vessel.

- *Infrared:* the temperature difference of an object and its background. The infrared signature is caused by the hull heated up by solar heat, the plume, but the most important is the funnel. Heat-isolation is important. Hotspots need to be prevented. The exhaustgases can be cooled by watercooling. Deck wetting system can decrease hull temperature. An option to lower IR signature is to bring the exhaust gas exits to a position low above the water line, preferably with the option to switch between starboard and port side in order to deflect the exhaust gases away from the threat side of the ship. Exhaust gases will remain longer below the horizon when emitted low above the water line.

- *Radar cross-section:* the parameter to assess the ability to locate an object against a background with a radar. The radar cross-section (RCS) (in m^2 of dBm^2) depends on the shape and size of the topside, the angle at which the radar beam meets the ship and the frequency and polarisation of the radar wave. RCS can be decreased by a.o. preventing reflecting corners, inclined plane (smooth) outer surface without openings (e.g. davits) and appendages (e.g. cranes).

- *Laser:* the reflection of laser signals of an object. To decrease reflection of laser to the originator, diffraction should be realized by means of high relecting surfaces or high absorbing coating should be applied.

- *Visual:* visibility of an object on its background caused by difference in e.g. color, shape, reflection. Measures to decrease visibility are applying background colors, breakings lines of visibility (dazzling) and avoiding reflecting surfaces.

The design of the propulsion concept is very important in the signature profile of the ship, in particular for the underwater noise, electric, magnetic and infrared signature. From analysis in Hendriks *et al.* (2011) it follows that the basic solution for meeting noise requirements is to be found in the propulsion concept, though much attention should be given to the drive and design of auxiliary systems (lubrication oil pumps, cooling water pumps, fire fighting pumps, hydraulically driven pumps), because especially at low speeds these systems are dominant in producing underwater noise.

In this chapter, typical signature profiles of the components will be described. Some numbers are given about frequencies and power levels to explain the signatures. These numbers are all indicative and no rights reserved, because test results related to signature profiles are all confidential. Information about signature profiles can give away valuable information for the enemy. Hendriks *et al.* (2011) is an example of a signature analysis, which is carried out to support the M-frigate replacement programme.

Reliability

A way to quantify reliability is with numbers for Mean Time Between Failure (MTBF) for repairable products and Mean Time To Failure (MTTF) for non-repairable products. These terms are often used when reliability issues are discussed. Information on reliability is scarce, and very unreliable in itself. The trouble with the determination of numbers for MTBF is the fact that it can only be based on historical statistical data which strictly spoken is not anymore applicable on the new equipment and secondly is often not available at all. To give indication on reliabiality the numbers for MTBF that were determined in the AES study are used. It should be noted that these are old numbers.

Initial purchase costs

Initial purchase costs of components are very difficult to determine on the naval market. Often it are special products, with very special requirements, which are only produced for military purpose. This is a rather small market, which means that development and production costs are divided over a small number of products. Besides that, purchase costs are highly variable because they are dependent on all kinds of side effects, like the conclusion of maintenance contracts next to the purchase or the intention to buy products in the future.

Ideally, a price per kW or per kg could be given of all components. For some components such numbers are found in literature. The AES study also mentions such numbers, but this is

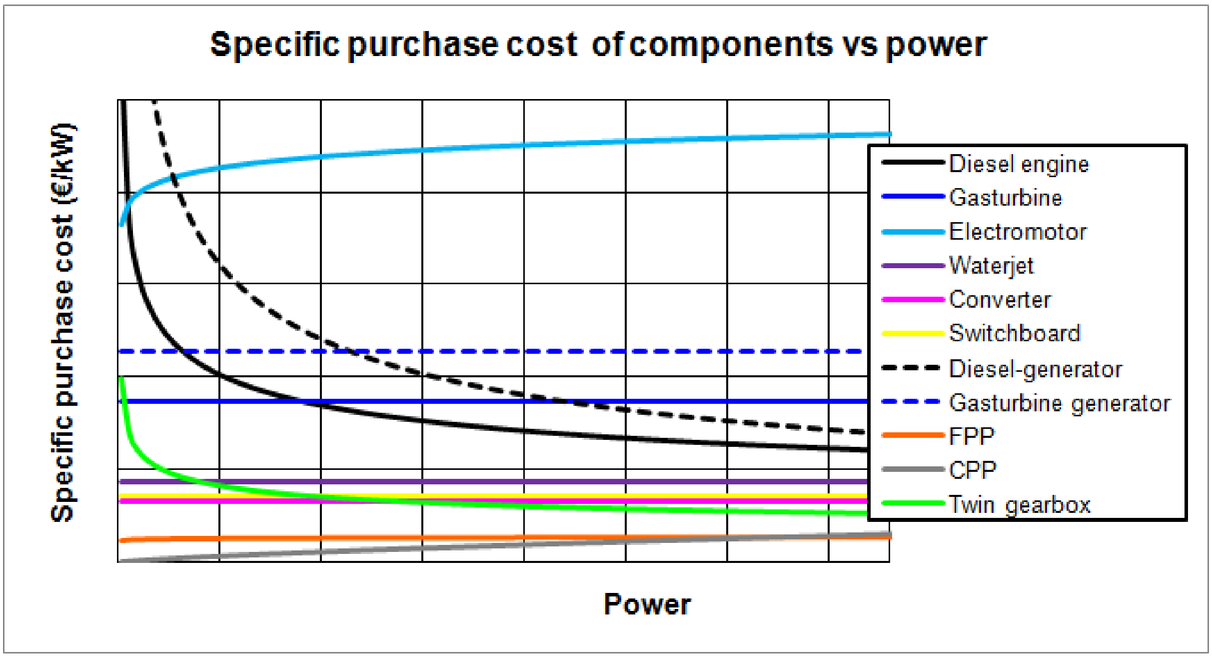


Figure 3.1: Specific purchase costs (€/kW) of relevant components vs rated power

very old information, thus expected to be no longer applicable. The old numbers are corrected for inflation at an assumed mean rate of 2% per year. Another, more reliable source, is the Cost Analysis section at DMO. This section keeps a record of costs from previous purchases and quotations. From this data the Cost Analysis section has determined Cost Estimating Relationships (CER) of the relevant components. These relationships are used in this study to do estimates on purchase costs. This information is commercially confidential and for that reason is not explicitly mentioned in this study. A confidential appendix holds the exact information. Figure 3.1 shows the relative relations of the specific purchase costs of the relevant components. The CER's of propellers and gearbox are a function of weight, so some assumptions had to be made to show the specific purchase cost as a function of power. For the propellers weight is estimated on diameter, and diameter is estimated on power. For the gearbox a twin gearbox with 200 rpm output speed is assumed, in order to estimate weight.

3.1 Fuels and emission regulations

The essential choice, that has to be made at the start of a propulsion study, is which fuel will be used. The fuel is the prime energy source that is converted in the power and propulsion machinery to finally deliver useful work. The fuel choice is very much influenced by logistical aspects, fuel costs, availability, safety issues and nowadays more often by emission regulations. A list of available energy sources or energy carriers is briefly described to point out the possibilities and to motivate the fuel choice. Also a short outline of the most important International Maritime Organization (IMO) emission regulations is given and the most important reduction measures.

3.1.1 Fossil fuels

Until early 20^{th} century coal was used as a fuel, but then the change was made to liquid fuel because that didn't need man handling, was more power dense and propulsion systems got greater flexibility in control, Plumb (1987). For liquid fuels, a distinction is made between distillate fuels and residual fuels. Distillate fuels consists of the lighter fractions from the distillation of the crude oil. Residual fuels (HFO) are about 35% cheaper then distillate fuels[4], but contain longer hydrocarbon chains that are more difficult to combust. The RNLN uses distillate fuel F76 (NATO fuel) or DMA for their engines. This is because of a.o. logistical benefits of international cooperation with other navies.

The problem with fossil fuels is the polluting emissions they cause, especially the heavy fuel oils. These emissions become more important everyday, because they are the cause of environmental problems. Another very important problem is the depletion of the oil resources. In short time the world may run out of oil, between 2075 and 2125 according to Webster *et al.* (2007b). So, the world is looking for suitable and cheap alternatives to replace fuel oil as an energy source, completely or partly. This will also affect naval ship building.

LNG

Recent developments make it possible to run marine diesel engines on natural gas (predominantly methane, CH_4). For ease of storage and transport it is liquefied, called liquefied natural gas LNG. This development is primarily driven by emission regulations, but it is competitively priced and expected to become cheaper than fuel oil. LNG has great potential. It gives significantly lower polluting emissions than fuel oil: 86% reduction in NOx, 98-100% reduction in SOx, 98-100% reduction in particulates and close to 30% reduction in CO_2 according to Flusund (2011). Disadvantage is the lack of a well established LNG infrastructure, and it has a lower power density than fuel oil, about 60% of the power density of diesel fuel. This means that a larger tank capacity is needed to have the same range, about 4 times the volume of a diesel fuel tank (including double walls, cooling etc.). Besides, there are some safety issues which need to be investigated: what happens with a missile hit, and what if cooling of the tanks fails? Another very important issue is refuelling at sea. As far as known, the possibilities are not investigated and difficulties might be expected. For these reasons, application of LNG as a fuel is not (yet) considered ready for transition to naval warships and is not taken into account in this study.

3.1.2 'Green' energy sources

The world gives us a number of other energy sources that are more sustainable and environmental friendlier than oil. Solar, wind and wave power are such 'green', inexhaustible energy sources. But the collection and conversion of these energy sources are still infeasible for surface combatants due to conflict with primary ship missions, Webster *et al.* (2007b). The topside impacts of, for example, wind energy conversion systems and the low reliaility of wind are ill

[4]http://www.bunkerworld.com/prices/

suited to surface combatants. But these energy sources might become more important in the near future as secondary systems for auxiliary power.

3.1.3 Nuclear energy

An energy source that is suitable for use on warships and has already been used many times, is nuclear energy. The general thought on nuclear energy is the potential danger it carries for human health, and the issue of radio-active rest products. This negative idea on nuclear energy will, at least in the near future, keep the naval ship designers from implementing nuclear power conversion onboard ships. Probably, if technique further develops, and safety increases this will be an option in the future.

The great advantage of nuclear energy is the extreme power density compared to oil, and the low fuel costs. Refueling is not necessary during service life of the ship. Disadvantage, besides the safety issues, is the bulky machinery needed for energy conversion. Nuclear reactors with all their safety are very heavy. In Webster *et al.* (2007a) an extensive research is described, in which the possibilities of nuclear energy on small and medium surface combatants were investigated. One of the conclusions is that nuclear propulsion systems are technically feasible for these ship types using existing reactor designs. Small surface combatant is defined as surface combatant with displacement between 7500 and 12000 tons. The surface combatant we are talking about in this thesis is much smaller (approximately 5000 tons). The technical feasibility on such a small ship is not investigated but is doubtful. For now, only ships with high energy demands benefit from nuclear power.

3.1.4 Energy carriers

Besides the different fuels, there are other options to provide energy. Via energy carriers, instead of energy sources. Batteries are an example of an energy carrier. Battery technology is rapidly developing, but current state of technology is still not satisfying enough for application on naval warships. Power density is too low, which means an enormous battery pack is needed for only a small ship operating range. For this reason batteries are not considered an option as primary energy supply for ship propulsion.

Another energy carrier that gains interest nowadays, is hydrogen. Hydrogen is not an energy resource (only when it is used in nuclear fusion) because it requires more energy to make it than is obtained by burning it, so hydrogen functions as an energy carrier, like a battery. Hydrogen may be obtained from fossil sources like methane. Hydrogen can be used in fuel cells to generate electric power. The advantage of using fuel cells to generate electric power is the clean conversion and high theoretic efficiency. Hydrogen needs to be produced, which means that energy has to be put into it, which can later be extracted in a fuel cell. Hydrogen can be produced from a.o. diesel fuel by steam reforming, but this is a rather slow and complex process. Another option could be to store hydrogen in tanks (compressed or liquefied), but because hydrogen is highly flammable and forms explosive mixture with air, this is not a very safe option on a naval warship. Though, this is done on German 212 submarine; the hydrogen is stored in the form of a metal hydride. There are a lot of other storage methods for hydrogen.

3.1.5 Emission regulations and reduction measures

Governing bodies like the IMO are introducing retroactive legislations that are of great influence on the ship power systems. The sulphur oxide (SOx) emissions are bound by limits in MARPOL 73/78 Annex VI regulation 14. From May 2005 maximum sulphur content in fuels was limited to 4.5%. This changes to 3.5% after 1 January 2012, and to 0.5% after 2020 (or 2025, depending on the outcome of a review in 2018). The SOx emissions are primarily solved by using low sulphur fuel. The SOx can also be washed out by water-scrubbers. The RNLN normally uses low-sulphur fuel with an average sulphur content of 0.1%, so the sulphur regulations are not the problem. Though, sulphur emissions can be a problem for the crew. If the concentration SOx in the plume is too high this is harmful to the personnel that is working on the open decks. This also holds for the NOx emissions.

The Marpol Annex VI regulation 13 mentions very stringent limits to the nitrous oxide (NOx) emissions which apply to marine diesel engines with power output > 130 kW installed on a ship constructed after 1 January 2000. The regulations are introduced in three steps: Tier I (1-1-2000), Tier II (1-1-2011) and Tier III (1-1-2016). The values for NOx emissions in (g/kWh) in the different Tier steps are pointed out in table 3.1 and visualized in figure 3.2.

	Tier I (2000)	Tier II (2011)	Tier III (2016)
$N < 130$ rpm	17	14.4	3.4
$130 \leq N \leq 2000$ rpm	$45 \cdot N^{-0.2}$	$44 \cdot N^{-0.23}$	$9 \cdot N^{-0.2}$
$N > 2000$ rpm	9.8	7.7	2.0

Table 3.1: NOx emission limitis in (g/kWh) according to IMO Marpol Annex VI regulation 13

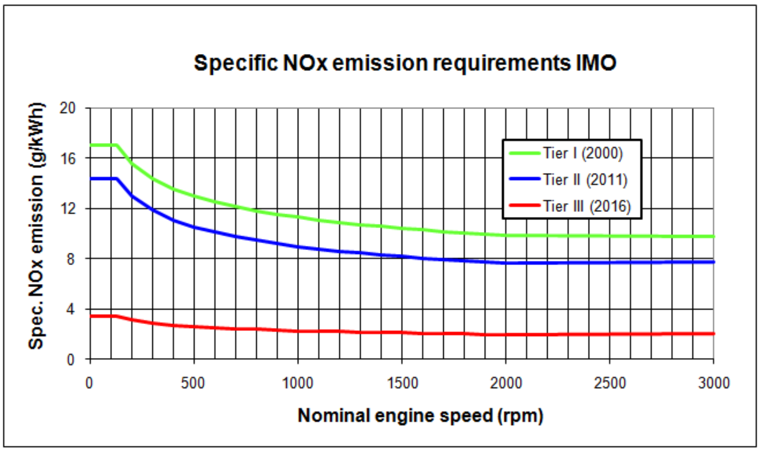


Figure 3.2: NOx emission requirements for marine diesel engines according to IMO Marpol Annex VI

The NOx are primarily so-called thermal NOx that are formed at high temperature spots in the cylinder. There are several ways to decrease the NOx emissions of the engine, but the two most promising technologies that will meet the Tier III limits are: switching to LNG as a fuel, or using exhaust gas aftertreatment.

An example of aftertreatment is a Selective Catalytic Reduction (SCR) reactor to convert NOx molecules back to harmless H_2O and N_2 by means of urea. Urea consumption typically is 15 l/MWh for a 40wt-% urea solution. Such a SCR unit consists of reactor with a catalyst inside, which is placed at a hot point in the exhaust pipe ($T \approx 300 - 450°C$), a reagent pumping unit, a reagent dosing unit, a control unit and an injection unit. Limiting factor for this system is the maximum allowable backpressure of the engine. For example Wärtsilä offers such a system, which they call NOR (Nitrogen Oxide Reducer).

Another example of exhaust gas aftertreatment is the so-called Corona reactor. This reactor generates a plasma, exhaust gas is lead through the plasma and OH-radicals arise from the humid air which react with the NOx to nitric acid (HNO_3). Nitric acid easily dissolves in water, and the NOx are removed from the exhaust gas.

3.2 Diesel engine

The first component is the diesel engine. This is a prime mover, which converts chemical energy stored in fuel into mechanical energy at the output shaft. The diesel engine is an example of a reciprocating internal combustion piston engine. The power cycle can be completed in two or four strokes of the piston. The power cycle consist of four stages:

1. Inlet and compression of combustion air
2. Combustion of injected fuel vapour ignited by high temperature of combustion air
3. Expansion of the combustion gas delivering work to the moving piston
4. Blow-down of the exhaust gas

The work delivered to the piston during the expansion stroke is translated into a rotational speed of the crankshaft. A diesel engine onboard a ship is used as propulsion engine to drive the propeller or in a diesel generator set to drive a generator for generation of electrical power. The electrical power can be used to drive electrical motors (which can drive the propeller) or other electrical equipment.

> Some well-known marine diesel engine manufacturers are: Wärtsilä, MAN B&W, MTU, Isotta Fraschini, Caterpillar, Bergen Rolls Royce, Cummins, Deutz, SEMT Pielstick, Volvo Penta.

Fuel

Diesel engines normally run on liquid fuel, it is also possible to run diesel engines on liquefied natural gas, but gas-fuelled diesel engines are not considered in this study for reasons that were pointed out in the previous section. The RNLN uses distillate fuel F76 (NATO fuel) or DMA for their engines, because of a.o. logistical benefits of international cooperation with other navies.

Auxiliary systems

To operate a diesel engine, some auxiliary system are needed. The most important auxiliary systems are now explained.

To start a diesel engine, a starting system and a pre-heating system is needed. Marine diesel engines are normally started with HP air. Small engines can be started with electric motors. Compared to a gasoline ('petrol') engine, diesel engines have very high compression ratios to provide for reliable and complete ignition of the fuel without spark plugs. An electric starter powerful enough to turn a large diesel engine would itself be so large as to be impractical. Thus there is the need for an alternative system. When starting the engine, compressed air is admitted to whichever cylinder has a piston just over top dead centre TDC, forcing it downward. As the engine starts to turn the air start valve on the next cylinder in line opens to continue the rotation. Another option to start the diesel engine is with a pneumatic starter motor which can be coupled to the flywheel.

Besides the starting system other auxiliary systems are necessary to operate a diesel engine: an engine cooling system, fuel supply and lubrication oil supply. Marine diesel engines are water-cooled. More about cooling in section 3.8. The fuel supply consists of fuel feed pumps, filters and sometimes separators. A separator separates the water and other undesirable constituents from the fuel in a high speed centrifuge. When heavy fuel is used, a fuel heating system is required.

Heavy fuel needs to be heated to make it viscous enough for transportation and injection in the cylinders.

In the future, with the more stringent emission regulations, it might become inevitable to use auxiliary systems for exhaust gas after treatment to reduce polluting emissions. Another important reason for emission reduction might be the well-being of the personnel. If the concentration of harmful emissions (NOx, SOx, particles) in the plume is too high this is harmful to the personnel that is working on the open decks. There are a number of different techniques to reduce emissions. Sulphur emissions are normally not a problem with the RNLN because low-sulphur fuel is used, but with a scrubber the SOx emissions could be washed out of the exhaust gas. NOx emission reduction techniques comprise primary methods and secondary methods. Primary methods try to prevent formation of NOx and secondary methods try to get formed NOx out of the exhaust gas, also called 'end-of-pipe' technology. Primary methods focus on lowering peak temperatures and are modifications to the engine. Primary methods don't affect the ship's layout very much, most promising technique seems to be exhaust gas recirculation. Secondary methods have a much bigger impact on ship layout but also have more promising effect. Examples of secondary methods are selective catalytic reduction (SCR) with ammonia or urea, wet scrubbers and non-thermal plasma. With current state of technology the SCR installations are most practical. The emission of particles or soot can be reduced by filters. Stapersma (2010) gives a good description of the available emission reduction techniques.

Data analysis

For analysis of this chapter a database of diesel engines (Stapersma, 2009a) is used, which includes all kinds of marine diesel engines from around 1997 up to now. This database is updated and expanded with some newer engine data from mini programs from engine manufacturers and with data from Henderson (2010). The engines in the database are divided into three groups:

1. High-speed engines, containing 54 engines (1 MAN, 3 CAT, 50 MTU) with nominal operating speeds higher than 1000 rpm, all V-engines

2. Medium-speed engines, containing 116 engines (53 MAN, 4 CAT, 16 MaK, 26 Wärtsilä, 4 Ruston, 4 SWD, 5 Pielstick, 4 Sulzer) with nominal operating speeds in the range 300-1000 rpm, subdivided in line and V-engines (62 vs 54)

3. Low-speed engines, containing 296 engines (216 MAN, 80 Sulzer) with operating speeds lower than 300 rpm, all line engines

3.2.1 Available power

High-speed diesel engines, >1000 rpm, are available in a power range from very small outboard engines of 7.5 kW and even lower, up to approximately 9 MW. High-speed diesel engines are also used in the automotive industry. Medium-speed diesel engines, 300-1000 rpm, are available in a wider power range of approximately 0.5 MW up to 35 MW. Low-speed diesel engines, <300 rpm, are available from 1 MW up to a power of 84 MW.

Diesel engines can not deliver full power at all speeds. The diesel engine is approximately a constant torque machine, but the maximum torque limit is dramatically narrowed by the limits of the turbocharger, which means it can not deliver full torque at low speeds. Sequential turbocharging solves great part of this problem.

The diesel engine also has a minimum speed (idling speed, normally 25-35% of nominal speed) and a minimum torque (25-40% of nominal torque) below which it will not run smoothly and causes damage to the engine. The engine starts fouling because of incomplete combustion. This means that the lower power limit for continuous operation normally lies at approximately 20%.

For lower loads there are restrictions to the running time of the engine. To give an example: in the project guide from a medium speed Wärtsilä engine the following restrictions are mentioned to the use of the engine:

- $> 20\%$ no restrictions
- $5 - 20\%$ max. 100 hours
- $0 - 5\%$ max. 6 hours

These are typical values for a diesel engine. The percentage means the percentage of the maximum torque at that specific engine speed. Experience learns that high speed engines suffer more from engine damage due to low loading, so high speed engines have a higher low-load-limit.

3.2.2 Dimensions

In this subsection it is tried to relate the dimensions of a diesel engine to the nominal output power of that engine. Engine brake power P_B is dependent on mean effective pressure p_{me}, engine speed n, number of cylinders i, number of shaft revolutions per power stroke k ($k = 1$ for 2-stroke, $k = 2$ for 4-stroke) and stroke volume (V_S).

$$P_B = p_{me} \cdot \frac{n \cdot i}{k} \cdot V_S \tag{3.1}$$

Stroke volume and number of cylinders are key parameters in the dimensions of an engine. Stroke volume is determined by stroke length (L_S) and bore diameter (D_B).

$$V_S = L_S \cdot \frac{\pi}{4} \cdot {D_B}^2 \tag{3.2}$$

The ratio between stroke length and bore diameter λ_S is a more or less fixed parameter for reasons of proper combustion. For high and medium speed engines this parameter varies between 1 and 1.5, for slow speed engines it is higher, in the range of approximately 2 to 4. This means that increasing the stroke length results in an increased bore diameter, and vice versa.

$$\lambda_S = \frac{L_S}{D_B} \tag{3.3}$$

Engine speed is also a key parameter in the sizing of engines, because it is closely related to the stroke length, thus bore diameter. Stroke length and engine speed are related to each other by the mean piston speed c_m. The mean piston speed is more or less constant for all engines because it is limited for reasons of wear and lubrication issues. Normally c_m lies between 8 and 12 m/s.

$$c_m = 2 \cdot L_S \cdot n \tag{3.4}$$

Combining the above equations proofs that P_B is proportional to ${D_B}^2$.

$$\left.\begin{aligned} P_B &= p_{me} \cdot \frac{n \cdot i}{k} \cdot V_S \\ &= p_{me} \cdot \frac{n \cdot i}{k} \cdot L_S \cdot \frac{\pi}{4} \cdot {D_B}^2 \\ &= p_{me} \cdot \frac{i}{k} \cdot \frac{c_m}{2} \cdot \frac{\pi}{4} \cdot {D_B}^2 \end{aligned}\right\} \quad P_B \propto {D_B}^2 \tag{3.5}$$

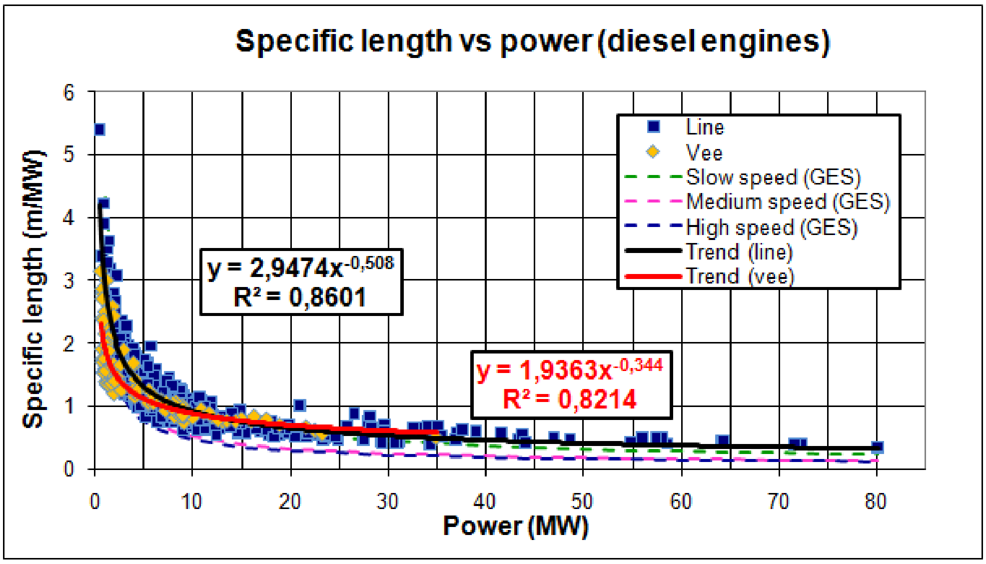


Figure 3.3: Specific length (m/MW) of database diesel engines and from GES vs rated power (MW)

The length of the engine L is determined by the number of cylinders and the bore diameter of the cylinder. With the proportionality shown in equation 3.5 the length is related to the power.

$$L \propto D_B \propto \sqrt{P_B} \tag{3.6}$$

The length of the database engines is presented as specific length $\frac{L}{P_B}$ in figure 3.3, and a trendline is fitted through the data. According to equation 3.6 the relation between power and specific length should be:

$$\frac{L}{P_B} \propto \frac{\sqrt{P_B}}{P_B} = \frac{1}{\sqrt{P_B}} \tag{3.7}$$

This relation is recognized in the data. Distinction is made between line and V-engines. The following general relations are made based on the data, with length in meter and P_B in megawatt:

Line engines:

$$\boxed{\text{Length} = (2.95 \cdot {P_B}^{-0.51}) \cdot P_B} \tag{3.8}$$

V-engines:

$$\boxed{\text{Length} = (1.94 \cdot {P_B}^{-0.34}) \cdot P_B} \tag{3.9}$$

Specific length of V-engines is smaller, which is logical because, roughly seen, in a V-engine 2 cylinders are placed next to each other. This means length could be half, theoretically. From the trend the difference between V- and line engines is factor 1.5 because in practice the cylinders are not perfectly next to each other, but have some overlap. This might also clarify the higher power of the relation.

The spread on the data can be imputed to the spread on p_{me} (min. 10.9 bar, max. 28.6 bar). The stroke bore ratio (λ_S) is also not constant for every engine. The goal of determining the trends is to give an indication of the dimensions of a diesel engine in an early design stage. In this early stage the designer should not be too specific about such details.

The standard deviation σ of the specific length of engines in the database from the trend is:

- $\sigma = 0.31$ for V-engines, with a max. deviation of 39%

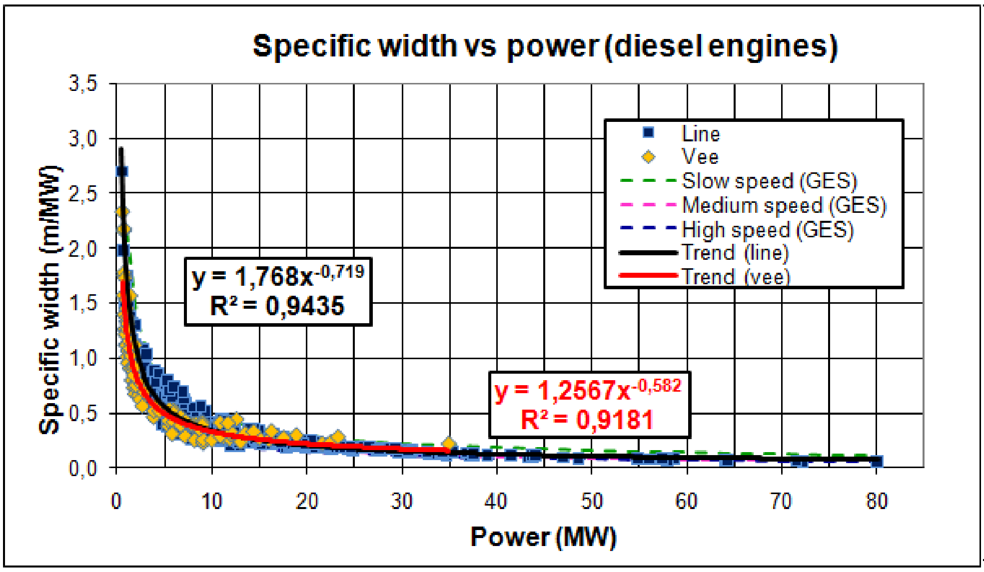


Figure 3.4: Specific width (m/MW) of database diesel engines and from GES vs rated power (MW)

- $\sigma = 0.26$ for line-engines, with a max. deviation of 56%

The relations from GES are also presented in figure 3.3. GES uses a fixed value for specific volume (m^3/MW) and fixed proportions between length, width and height. The values for specific volume in GES are 30 m^3/MW for slow speed engines, 5 m^3/MW for medium speed and 3.6 m^3/MW for high speed. The proportions length:width:height are fixed at 4:2:3. With this, a relation between length and cube root power is defined, where a square root is expected according to equation 3.6.

The GES relations show a worse correlation with the data, with higher standard deviation values and higher maximum deviations:

- $\sigma = 0.31$ for slow speed engines, with a max. deviation of 76%
- $\sigma = 0.74$ for medium speed engines, with a max. deviation of 57%
- $\sigma = 0.36$ for high speed engines, with a max. deviation of 50%

For the width of the engine the same analysis is done as with the length. Assumed is that engine width W is proportional to the bore diameter, like the engine length.

$$W \propto D_B \propto \sqrt{P_B} \tag{3.10a}$$

$$\frac{W}{P_B} \propto \frac{\sqrt{P_B}}{P_B} = \frac{1}{\sqrt{P_B}} \tag{3.10b}$$

The width of the database diesel engines is presented in figure 3.4 as specific width. In this figure are also presented the relations as implemented in GES by the dotted lines.

From the data the following trendlines are found, with width in meter and P_B in megawatt:

Line engines:

$$\boxed{\text{Width} = (1.77 \cdot {P_B}^{-0.72}) \cdot P_B} \tag{3.11}$$

V engines:

$$\boxed{\text{Width} = (1.26 \cdot {P_B}^{-0.58}) \cdot P_B} \tag{3.12}$$

The standard deviation σ of the specific width of engines in the database from the trend is:

- $\sigma = 0.17$ for V-engines, with a max. deviation of 55%
- $\sigma = 0.09$ for line-engines, with a max. deviation of 59%

Remarkable is that the specific width of the V-engines is less than of the line engines. In a V-engine two cylinder banks are next to each other so wider engines could be expected. In practice, V-engines have smaller bore diameter, and a higher number of cylinders, when compared to equal power line engines. Obviously this results in less wide engines.

The trendlines, as they are found from the data, differ from the theoretical relation in equation 3.10b. The power of the relations is lower than expected, more in the range of $-\frac{2}{3}$. This means a cube root relation between width and power, which is given by the GES relation. Still, the GES relations show a worse correlation with the data, with higher standard deviation values, but lower maximum deviations:

- $\sigma = 0.14$ for slow speed engines, with a max. deviation of 116%
- $\sigma = 0.17$ for medium speed engines, with a max. deviation of 50%
- $\sigma = 0.32$ for high speed engines, with a max. deviation of 46%

Finally, the height of the diesel engine is related to the engine power. Engine height is proportional to stroke length, and according to equation 3.3 stroke length is proportional to bore diameter under the assumption that λ_S is (more or less) constant. Earlier it was found that bore diameter is proportional to the square root of brake power, so for the height of the engine H the following holds:

$$H \propto D_B \propto \sqrt{P_B} \tag{3.13a}$$

$$\frac{H}{P_B} \propto \frac{\sqrt{P_B}}{P_B} = \frac{1}{\sqrt{P_B}} \tag{3.13b}$$

Figure 3.5 shows the results for the specific height of the database engines. A very obvious difference between 2- and 4-stroke engines is distinguished. This difference in height is caused by the difference in building technique between these engine types and higher values for λ_S. 2-Stroke engines are normally crosshead engines, while 4-stroke engines are normally trunk piston engines. The trendline of the 4-stroke (trunk piston) engines follows the relation as expected according to equation 3.13b. The 2-stroke (crosshead) engines show a weaker (lower power) relation with P_B, more towards the cube relation as implemented in GES. The GES relations are also presented in figure 3.5 by the dotted lines.

The following general relations are made based on the data, with height in meter and P_B in megawatt:

2-stroke engines:

$$\boxed{\text{Height} = (5.64 \cdot {P_B}^{-0.74}) \cdot P_B} \tag{3.14}$$

4-stroke engines:

$$\boxed{\text{Height} = (1.70 \cdot {P_B}^{-0.54}) \cdot P_B} \tag{3.15}$$

The standard deviation σ of the specific height of engines in the database from the trend is:

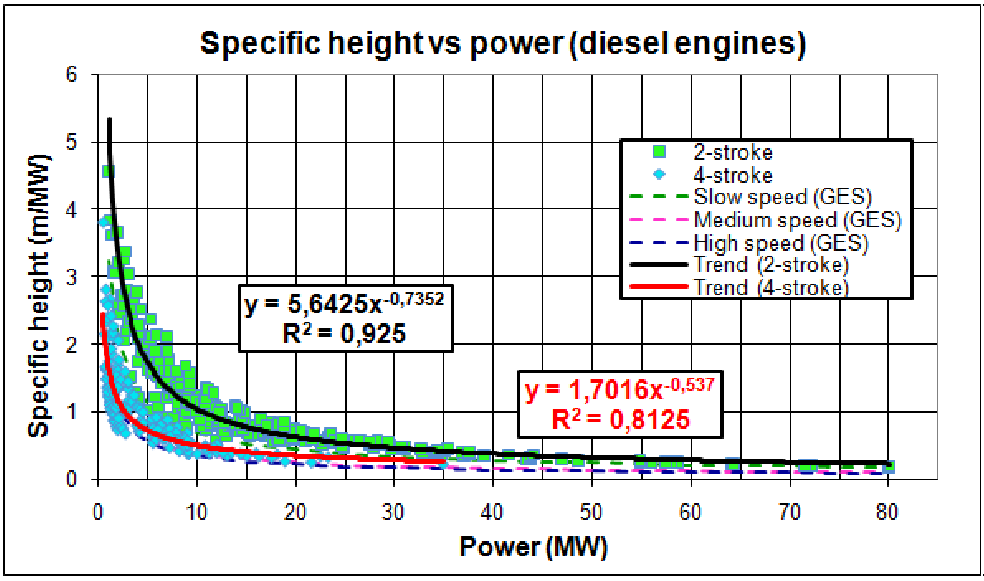


Figure 3.5: Specific height (m/MW) of database diesel engines and from GES vs rated power (MW)

- $\sigma = 0.29$ for 2-stroke engines, with a max. deviation of 81%
- $\sigma = 0.34$ for 4-stroke engines, with a max. deviation of 64%

The GES relations show a worse correlation with the data, with higher standard deviation values, but lower maximum deviations:

- $\sigma = 0.52$ for slow speed engines, with a max. deviation of 58%
- $\sigma = 0.34$ for medium speed engines, with a max. deviation of 45%
- $\sigma = 0.28$ for high speed engines, with a max. deviation of 41%

Note: *Dimensions are very dependent on engine speed. The direct effect of the engine speed is not shown because dimensions are presented as function of power. The speed dependency could be ruled out by presenting the dimension as function of engine torque, but for ease of usage it is presented as function of power. Besides that, power and speed are proportional through the 'fixed' piston speed: power is proportional to stroke volume, stroke volume to bore diameter and stroke length, and because piston speed has a limit value, the stroke length is proportional to engine speed. So indirectly the speed dependency is taken into account.*

Inlet and outlet

In calculating the space consumption of a diesel engine, the space for in- and outlet ducts should not be neglected, since these consume a significant amount of space. The in- and outlet ducts have the function to transport the mass flow of air and exhaust gas to and from the engine. Besides that they also serve for filtering and conditioning the air and muffling the noise of the combustion. The size of ducts is proportional to the amount of air that the engine consumes which is proportional to the engine power. To generate a certain power at the shaft P_B (MW), a fuel flow $\dot{m}_f$ (kg/s) needs to be combusted. This requires a certain massflow of air $\dot{m}_{air}$ (kg/s). $\dot{m}_{air}$ is determined by the fuel flow, the stoichiometric air/fuel ratio σ (-) and the air excess ratio

Variable	Assumed value	Unit	Condition
λ	2	$(-)$	
σ	14.5	$(-)$	
η_e	0.44	$(-)$	
LHV_F	42.700	(MJ/kg)	
ρ_{air}	1.27	(kg/m^3)	at $20°C$, 1 atm
ρ_{gas}	0.60[a]	(kg/m^3)	at $350°C$[b], 1 atm
$v_{max,in}$	15	(m/s)	
$v_{max,out}$	35[c]	(m/s)	

[a]: Data of air at given conditions

[b]: Based on moderate exhaust gas temperature of Wärtsilä W20

[c]: Based on assumptions in project guide MAN L21/31

Table 3.2: Assumptions in determining dimensions of in- and outlet duct of a diesel engine

λ (-). The required fuel flow for a certain amount of power depends on the engine efficiency η_e (-) and the lower heating value of the fuel LHV_F (MJ/kg).

$$\dot{m}_{air} = \lambda \cdot \sigma \cdot \frac{P_B}{\eta_e \cdot LHV_F} \tag{3.16}$$

With a known maximum airflow and the maximum allowable air velocity in the duct $v_{a,max}$ (m/s), the duct's minimum area A_{duct} (m^2) is calculated. The air velocity has a maximum value, for reasons of noise and to prevent underpressure in the duct.

$$A_{duct,in,min} = \frac{\dot{m}_{air}}{\rho_{air} \cdot v_{air,max}} = \lambda \cdot \sigma \cdot \frac{P_B}{\eta_e \cdot LHV_F \cdot \rho_{air} \cdot v_{air,max}} \tag{3.17}$$

With equation 3.17 the minimum duct area of the inlet duct is calculated. For the size of the outlet duct, $\dot{m}_{air}$ is multiplied with the so-called 'fuel addition factor' (δ), because the massflow has increased after combustion due to the addition of fuel. The fuel addition factor is only a few percent differing from unity (≈ 1.03). Further, the density of air is replaced by the density of the exhaust gas which depends on the temperature. For the outlet duct a higher value for maximum velocity is used, because there is no risk of underpressure.

$$A_{duct,out,min} = \frac{\dot{m}_{air} \cdot \delta}{\rho_{gas} \cdot v_{gas,max}} = \lambda \cdot \sigma \cdot \frac{P_B}{\eta_e \cdot LHV_F \cdot \rho_{gas} \cdot v_{gas,max}} \tag{3.18}$$

The assumed values in equation 3.17 and 3.18 are listed in table 3.2. Inserting these values results in the following numbers for specific duct area:

inlet duct: 0.081 m^2/MW
outlet duct: 0.076 m^2/MW

In GES, fixed values for inlet and outlet ducts are used, not being a function of the power. For slow speed engines a diameter of 1 m is assumed, for medium and high speed diameters between 0.5-0.8 m are assumed.

SCR

As explained before, with the upcoming legislations exhaust gas treatment might become inevitable in the future for diesel engines. The major difficulties with emission legislations on future RNLN vessels are expected with the NOx emission. As long as LNG is not used as a fuel, the most promising technology for meeting the NOx emission regulations is selective catalytic reduction. Wärtsilä offers a range of SCR units, the so-called NOR (Nitrogen Oxide Reducer). The dimensions of this product range is added here to give an idea of the size of such an unit. Mind that this only includes the dimensions of the reactor with the catalyst inside, and not the dimensions of pumping, dosing, controlling and injection units. The reactor is placed in the funnel.

Engine power (kW)	Reactor width (m)	Reactor height (m)
$\leq$ 1260	1.57	4.51
1261-2240	1.88	4.78
2241-3500	2.20	5.05
3501-5040	2.15	5.22
5041-6860	2.83	5.41
6861-8960	3.14	5.78
8961-11340	3.46	5.95
11341-14010	3.97	6.31
14011-16950	4.46	6.71
16951-20170	4.83	6.79

Table 3.3: Dimensions of SCR reactor for NOx reduction from the Wärtsilä NOR range

3.2.3 Weight

As seen in the previous subsection, the dimensions of the diesel engine show convenient relations with the rated power. Dimensions, volume and weight are closely related. It might be expected that the weight of the engine also shows such convenient relations with the power. Figure 3.6 shows the specific weight of the database engines in relation with the power. It shows a cloud in which hardly any relation can be recognized. This figure also shows the fixed values that GES uses for specific weight of diesel engines. The spread on these mean values is very wide and most of the measured values are far above the mean values from GES. A more accurate estimation method for the weight of diesel engines is preferred.

A general trend of slow speed 2-stroke engines being heavier is recognized in figure 3.6. This gives rise to the idea of the specific weight being proportional to the speed. In Stapersma (2001) it is described that there is a relation between the weight of the engine and the swept volume. Engines with bigger swept volume run slower, for mechanical and combustional reasons. Indirectly there is a relation between weight and engine speed, because dimensions are also very dependent on speed as in explained the note on page 27. The relation between weight and speed is shown in figure 3.7, together with again the values from GES.

The following general relation is made based on the data, with weight in ton and N in rpm:

$$\boxed{\text{Weight} = (1281.2 \cdot N^{-0.76}) \cdot P_B} \tag{3.19}$$

The standard deviation of the specific weight of engines in the database from the trend is:

- $\sigma = 8.19$, with a max. deviation of 94%

The relations that are used in GES are a function of power, and have the following values for standard deviation and maximum deviation:

- $\sigma = 16.60$ for slow speed engines, with a max. deviation of 85%
- $\sigma = 8.20$ for medium speed engines, with a max. deviation of 79%

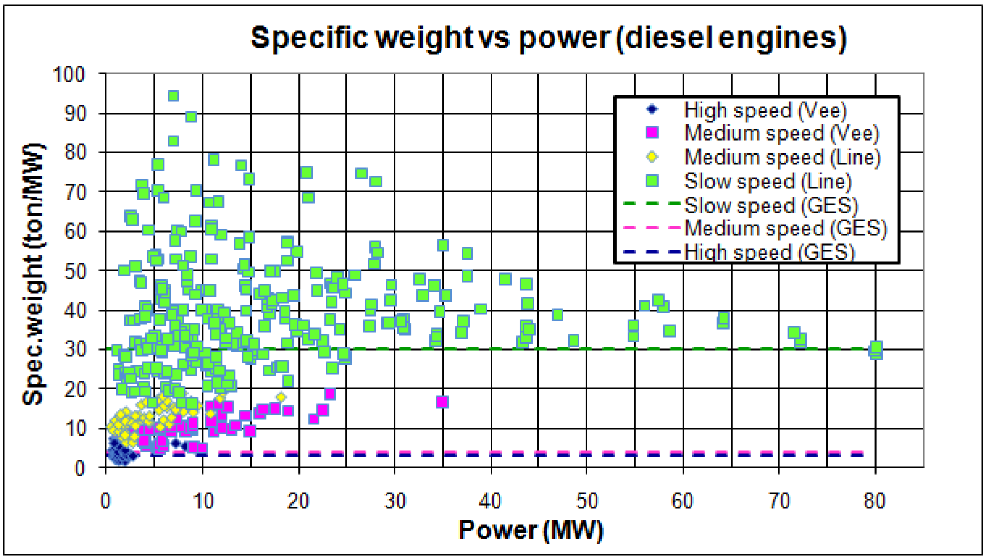


Figure 3.6: Specific weight (ton/MW) of database diesel engines and from GES vs rated power (MW)

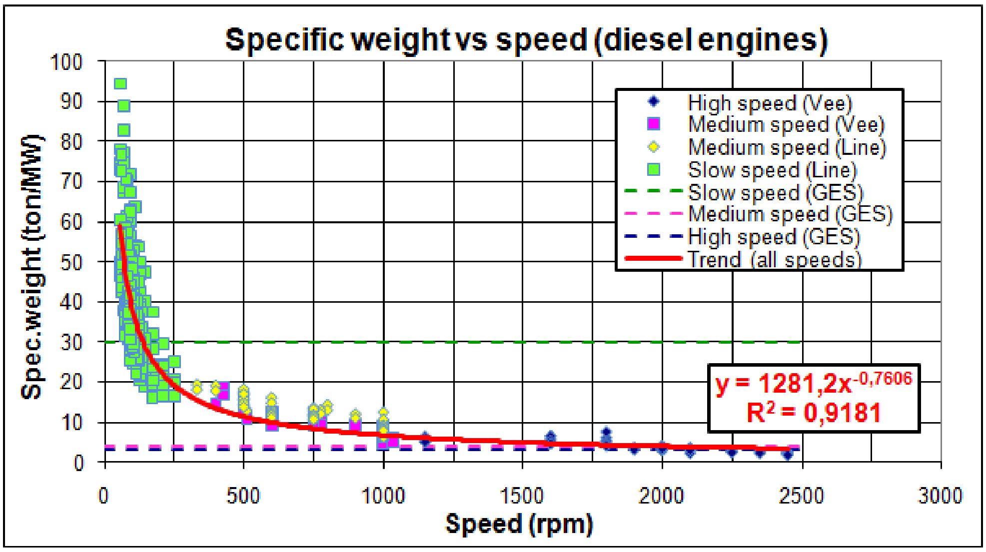


Figure 3.7: Specific weight (ton/MW) of database engines vs rated speed (rpm)

- $\sigma = 1.52$ for high speed engines, with a max. deviation of 72%

Especially in the slow speed category this method is very inaccurate.

The higher speed engines are relatively lighter than lower speed engines; which means those are more power dense. These higher speed engines are made of lighter materials, necessary to come to higher speeds, and higher power density.

Inlet and outlet

GES also takes into account the weight of inlet and outlet. The weight is depending on the length, and the length is determined by the design of the ship and the placing of the engines in the ship. In this study this is not taken into account, but the values that are mentioned in GES are: slow speed (Ø 1m) 300 kg/m, medium speed and high speed (Ø 0.5-0.8m) 225 kg/m.

SCR

Weight of SCR reactors from the Wärtsilä NOR range are listed in table 3.4.

Engine power (kW)	Reactor weight (kg)
≤ 1260	3100
1261-2240	4100
2241-3500	5500
3501-5040	7050
5041-6860	9050
6861-8960	12050
8961-11340	14350
11341-14010	18000
14011-16950	23050
16951-20170	29050

Table 3.4: Weights of SCR reactor for NOx reduction from the Wärtsilä NOR range

3.2.4 Operating speeds

Rated operating speeds are varying from ±55 rpm for very big, slow speed engines, to up to ±2500 rpm, for high speed engines. A diesel engine can operate at varying operating speeds with a minimum rotational speed (idling speed) of typically about 30% of the rated speed.

3.2.5 Efficiency

Engine efficiency is a key parameter to determine fuel usage of the engine. Efficiency is determined by the friction losses, heat losses, combustion losses and rest heat in the exhaust gas, see figure 3.8 based on KleinWoud & Stapersma (2003).

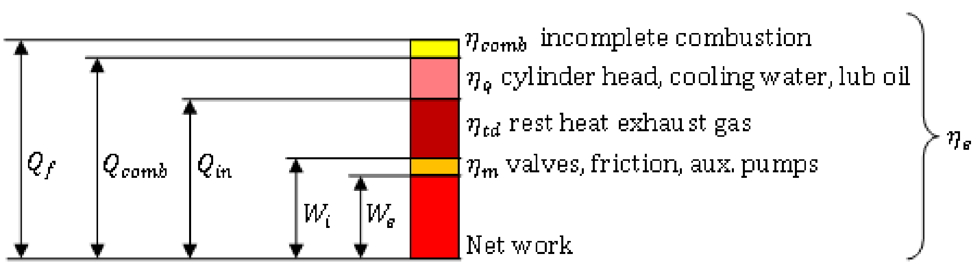


Figure 3.8: Schematical overview of losses in diesel engine

Typical nominal efficiency for slow-speed 2-stroke engines is around 50% at nominal point. For 4-stroke engines this is somewhat lower. There is no clear relationship between nominal power and nominal efficiency. It is more convenient to relate nominal efficiency to nominal speed. In Stapersma (2001) it was already shown that a relation between nominal speed and nominal efficiency can be used. Figure 3.9 presents the nominal efficiency of the database engines versus the nominal speed.

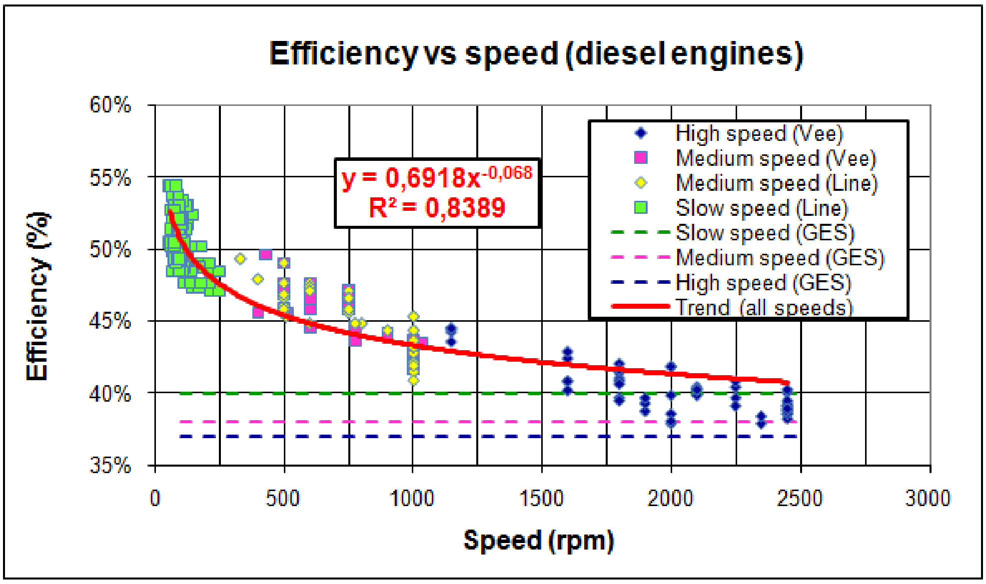


Figure 3.9: Nominal engine efficiency (%) of database diesel engines and from GES vs rated speed (rpm)

From the data in figure 3.9 an obvious relation between efficiency and speed is recognized. The general trend for nominal overall engine efficiency ($\eta_{e,nom}$) as a function of nominal engine speed (N_{nom}) in rpm is described by:

$$\boxed{\eta_{e,nom} = 0.69 \cdot N_{nom}^{-0.07}} \tag{3.20}$$

In GES a mean value approach is used to estimate the nominal engine efficiency. For slow speed engines a fixed value of 40% is used, for medium speed engines 38%, and 37% for high speed engines. These values are also presented in figure 3.9 by dotted lines. It can be seen that the GES approach shows large deviations from the data.

Slower engines are more efficient. The fuel used in diesel engines burns relatively slow, in slower engines the combustion and heat release are more efficient. Disadvantage of the slower engines is that they have a lower power density (see figure 3.7). The trade-off for the engineer will be between efficiency and power density, this is shown in figure C.1 in Appendix C.1.

Part load efficiency

Engine efficiency of a diesel engine decreases at part load operation. Normally, efficiency has a maximum at approximately 85% part load. This optimal point can be moved a little bit depending on the tuning of the engine and turbocharger. In Frouws (2008) a formula developed by prof. D. Stapersma is mentioned that describes the partload fuel efficiency fc^* of a diesel engine as a function of engine speed and torque.

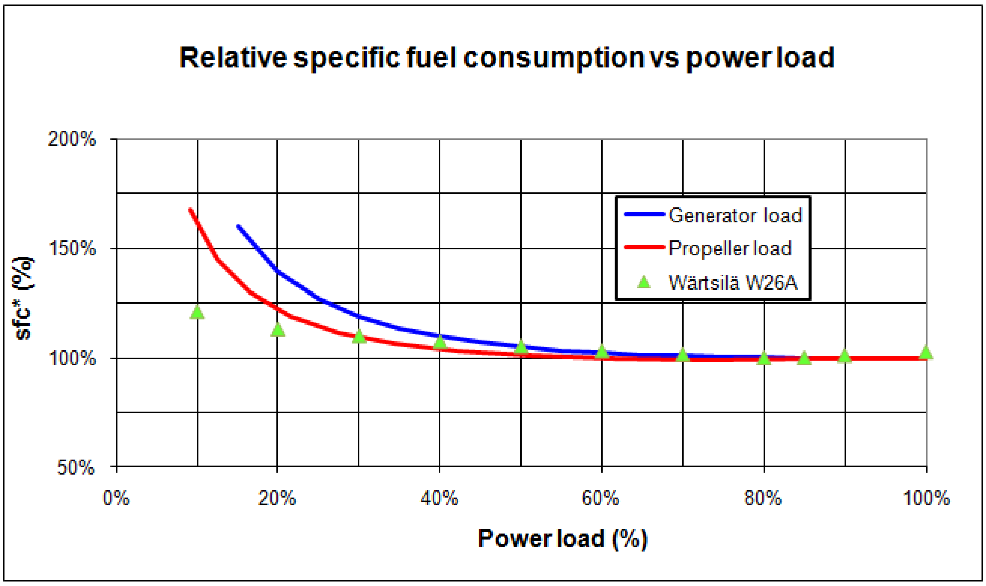


Figure 3.10: Relative specific fuel consumption compared to nominal value (%) vs powerload (%) for a typical diesel engine according to model from D. Stapersma and from Wärtsilä W26A (FPP propeller curve) project guide

(source: Wärtsilä (2003))

$$\frac{fc^*}{N^*} = 1 - a(1-N^*) + b(1-N^*)^2 - c(1-M^*) + d(1-M^*)^2 + 2e(1-N^*)(1-M^*) \quad (3.21)$$

fc^* is fuel consumption as a fraction of nominal fuel consumption, N^* is engine speed as a fraction of nominal engine speed, M^* is engine torque as a fraction of nominal engine torque, a, b, c, d and e are shaping parameters that are chosen as:

$a = 0.1 \quad b = 0.26 \quad c = 1.03 \quad d = 0.16 \quad e = 0.04$

Equation 3.21 was rewritten to a formula that gives specific fuel consumption as a fraction of nominal, sfc^*.

$$sfc^* = \frac{1 - a + aN^* + b - 2bN^* + bN^{*2} - c + cM^* + d - 2dM^* + dM^{*2} + 2e - 2eM^* - 2eN^* + 2eM^*N^*}{M^*} \quad (3.22)$$

When the diesel is used at constant speed (generator load), then $N^* = 1$ and torque is proportional to power. Equation 3.22 is now rewritten to equation 3.23.

$$sfc^* = \frac{1 - c\cdot(1-P^*) + d\cdot(1-P^*)^2}{P^*} \quad (3.23)$$

The results of this equation are presented in figure 3.10. In this figure are also presented some datapoints from the Wärtsilä W26A project guide. The datapoints at low loads (< 30%) are probably somewhat optimistic in the project guide.

Specific fuel consumption is a quantity that describes engine efficiency η_e. Engine efficiency is calculated from the specific fuel consumption (in g/kWh) with:

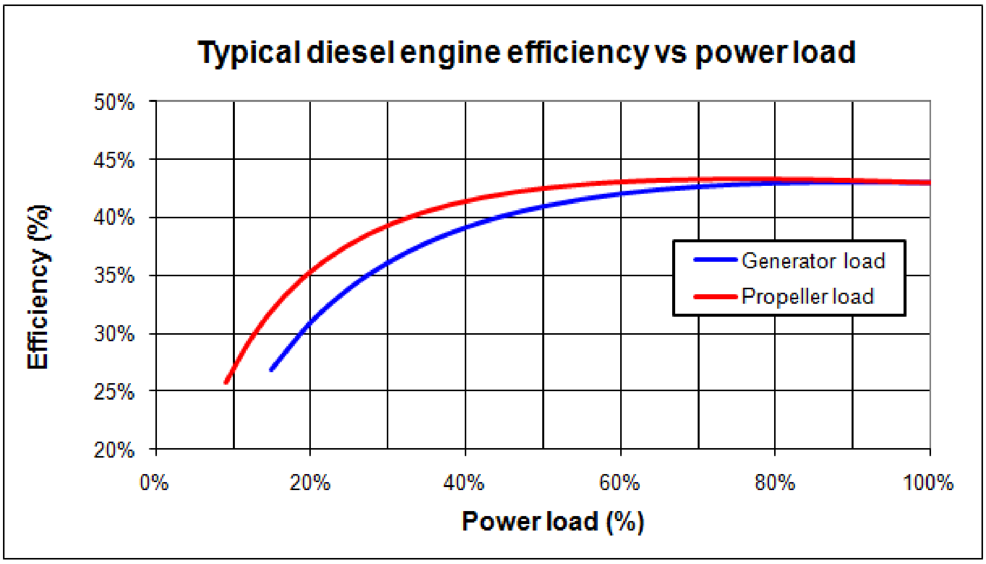


Figure 3.11: Typical efficiency curve of a diesel engine with $\eta_{nom} = 0.43$

$$\eta_e = \frac{3600 \cdot 1000}{sfc \cdot LHV_F} \tag{3.24}$$

In which LHV_F is the lower heating value of the fuel. For marine diesel oil a typical value is 42700 kJ/kg. Figure 3.11 presents a typical efficiency plot of a diesel engine based on equation 3.22 with $sfc_{nom} = 196$ g/kWh.

3.2.6 Signatures

Underwater noise

The diesel engine is a reciprocating machine, so the piston speed is not constant. It changes direction each revolution. Extreme forces are introduced in the structure of the engine with a certain frequency. These cause vibration of the structure and noise is produced. The metal structure conducts these vibrations perfectly. Noise transmits also via 'flanking paths' (e.g. cooling water pipes, exhaust system, electrical wiring). In particular this structure-borne noise is harmfull to underwater noise. Solutions to reduce structure-borne noise produced and transmitted by diesel engines and/or diesel generator sets are:

- Resilient mounting (springs), single or double, to prevent/decrease the structure-borne sound into the ships hull
- Flexible coupling, single or double, between engine and gearbox or generator
- Flexible couplings for cooling water pipes, exhaust system etc. to avoid acoustic short-cuts

Resilient mountings are common practice on medium and high speed diesel engines. For high speed diesel engines double elastic mounting is often applied to further reduce structure-borne noise. In most cases, medium speed diesels are too heavy for double mounting. Resilient mounting is often not possible at all for low speed diesels, because they are too heavy and this might cause problems with shaft alignment.

Air-borne noise can be reduced by putting the engine in an acoustic enclosure. Especially high speed engines have severe air-borne noise.

The bigger the engine gets the bigger the forces and thus the power of the vibrations. The main vibrations are caused by the moving piston. The rotational speed and the number of cylinders of the engine determine the frequency of the vibrations. From TNO measurements (confidential report) on a medium speed diesel engine it follows that underwater noise of the diesel engine is a low frequency vibration (40-125 Hz) with a lot of power ($\approx$ 130 dB). The low frequency noise is more harmful to the warship than high frequency noise, because it carries a longer distance through the water than high frequency sound and higher frequencies can more easily be damped by resilient mountings.

Other, less powerful, vibrations are also introduced by other moving parts off the engine, like valves, tappets, turbocharger, auxiliary pumps, etc. Another important source of noise are torsional vibrations on the shaft caused by a badly balanced engine or off-design conditions of the engine. Off-design conditions are for example: wrong injection timing, not proper operation of the turbocharging device, incorrect valve timing, excessive wear of piston rings and/or piston liners etc. This can be prevented, to great extents, by good maintenance.

The noise production of diesel engines, especially in diesel generator sets, are of great concern. From tests it is found that at low ship speeds (up to 10-12 knots) the auxiliary machinery is dominant for the underwater noise levels in the low to medium frequency range. The diesel generators are the most critical components. The combination of a diesel type (rpm) and noise reduction measures (single/double mounts and enclosure) has to be matched carefully.

Infrared

A diesel engine produces hot exhaust gases which cause an infrared signature. The temperature of the exhaust gas leaving the engine is typically around $350 - 400°C$ depending on the load and the work delivered in the turbocharger. The engine structure and chimneys are heated up and transmit infrared light. Infrared light is not visual to the human eye but with an infrared telescope the enemy is able to detect it and track the ship. The hot engine itself is not a problem for the infrared signature, because the engine is situated within the ship's hull.

Transmission of infrared light by the funnels can be decreased heat-isolators and a façade around the funnel. The exhaust gas itself can not be isolated, because it is blown into the environment. To reduce the infrared transmission of the gas, it needs to be cooled as much to the environmental temperature as possible. This can be achieved by leading the exhaust gas through water-cooled heat exchangers. In that case the heat can even be used profitable in heating of compartments for example. Another option to lower infrared signature of the plume is to bring the exhaust gas exits to a posiotion low above the water line. Preferably with the option to switch between starboard and port side in order to deflect the exhaust gases away from the threat side of the ship. Exhaust gases will disappear faster behind the horizon when emitted low above the water line. A problem with this exhaust system is the dynamic back pressure of the sea water on the exhaust, and the fact that the exhaust gases will stay close to the ship and cause severe nuisance to the personnel.

Electro-magnetic

Ferro-magnetic components in a diesel engine can cause a magnetic signature. By deperming these components this permanent magnetic field can be removed. The induced magnetic field can only be removed by degaussing coils. Specialized engines can be treated with such means to reduce the magnetic signature of the engine.

3.2.7 Shock resistance

Diesel engines are robust machines and therefore show good shock resistance. To improve shock resistance, the engine can be placed on flexible mountings (springs), which is also beneficial for signatures. But, the heavier the machine, the more difficult it becomes to place it on flexible mountings, besides there might be problems with shaft alignment. Engines are preferably constructed of ductile metal. For example cast iron is very brittle and there is the risk of snapping engine supports in case of a shockwave.

3.2.8 Maintainability

A diesel engine engine is exposed to wear and requires maintenance to keep it employable. The pistons move up and down in the cyliners which causes them to wear. Wear is higher for higher speed engines and larger number of cylinders. Maintenance varies from relative small maintenance that is done by the crew of the ship to major maintenance tasks and overhauls that are normally performed by personell from the manufacturer. Smaller maintenance tasks are for instance changing filters and oil, changing fuel injectors, valve tuning. The manufacturer gives some maintenance schedule with prescribed maintenance tasks after a certain number of running hours. An overview of maintenance tasks and intervals is given by table 3.5.

Maintenance job	**Interval** (running hours)
Check tightening of screws	50
Change lub oil + filters	1000
Valve inspection	2000
Check + clean turbocharger	4000
Inspect aux. pumps	8000
Check bearings + liners	16000
Overhaul cyl. head + injection pumps	16000
Replace cyl. liners + bearings + piston crowns + piston skirts	48000

Table 3.5: Typical maintenance tasks with intervals on a diesel engine

(based on: Wärtsilä W20)

Maintenance costs money, depending on the type of engine and the running hours. In GES there is implemented a mean value for maintenance costs as function of nominal engine power and running hours. This value is called specific unit maintenance cost (sumc) which is presented in €/MWh. For slow speed engines a value of 6 *fl.*/MWh is assumed and for medium and high speed engines 20 *fl.*/MWh. These are numbers from 1998, if they are corrected for inflation at a rate of 2% per year and presented in €, this gives:

- Slow speed = 3.52 €/MWh
- Medium/High speed = 11.74 €/MWh

In Stapersma (2001) chapter 6.2, based on findings in Kok (1997), it is also tried to do estimations on maintenance costs of diesel engines. It is described that maintenance costs can be related to the rated power and the time the engine is used. The relation dates from 2001 and is based on an engine with the following data, probably a Wärtsilä W38:

- sumc = 3 €/MWh
- $\lambda_S = 1.25$
- $D_B = 380$ mm
- $c_m = 9.5$ m/s
- $n = 10\ \mathrm{s}^{-1} \rightarrow 600$ rpm

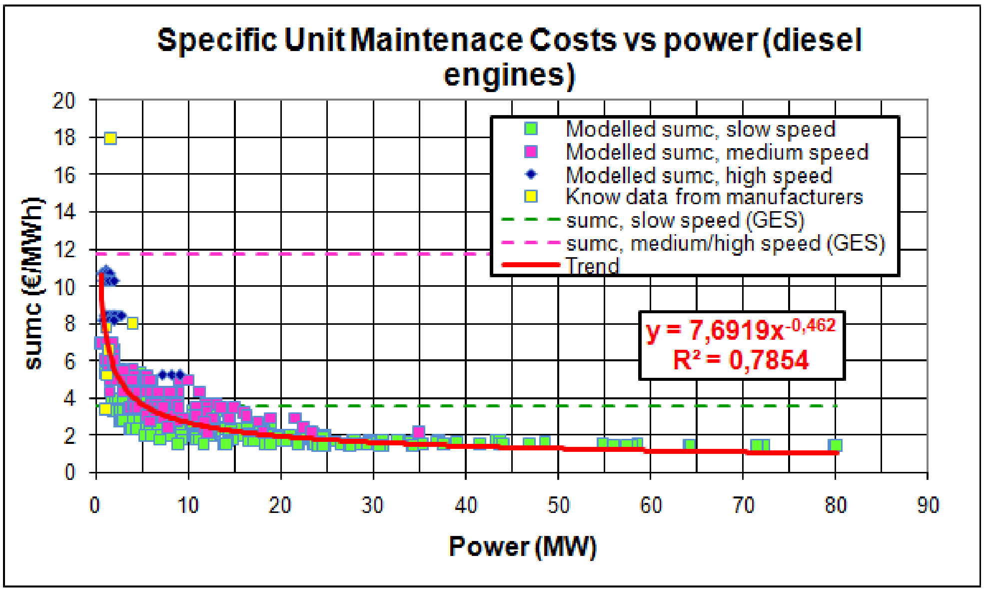


Figure 3.12: Specific Unit Maintenance Costs (according to model in Stapersma (2001)) in €/MWh of database diesel engines and some known manufacturers data vs power (MW)

The sumc value in Stapersma (2001) already includes an assumed additional value of 20% for unscheduled maintenance. The value is corrected for inflation at a rate of 2% per year, because this number dates from 2001. The corrected sumc value is 3.65 €/MWh. In Stapersma (2001) the maintenance costs are related to mean piston speed c_m, stroke/bore ratio λ_S and nominal speed n.

$$\text{sumc} = 3.65 \cdot \frac{9.5}{c_m} \cdot \frac{\lambda_S}{1.25} \cdot \frac{n}{10} \tag{3.25}$$

For all database diesel engines a specific unit maintenance cost is calculated according to formula 3.25. The results are plotted in figure 3.12 together with some recently collected data from different diesel engine manufacturers. The red line in figure 3.12 indicates the trend of the modelled sumc as function of the nominal power. The manufacturers data does not follow this trend. Maintenance cost in €/h is given by formula 3.26.

$$\boxed{\text{Maintenance cost}_{\ (€/h)} = (7.70 \cdot {P_B}^{-0.45}) \cdot P_B} \tag{3.26}$$

Kok (1997) also mentions a relation between spare parts consumption and a certain wear rate. This wear rate (wr) is dependent on number of cylinders (i), bore diameter (D_B), mean effective pressure (p_{me}) and piston speed (c_m), see formula 3.27. The higher the wear rate, the higher the consumption of spare parts, the higher the sumc should be.

$$wr = i \cdot D_B \cdot p_{me} \cdot c_m \tag{3.27}$$

The recently collected data about maintenance cost is tested with the wear rate hypothesis, but no trend was recognized. The results are shown in figure C.2 in appendix C.1. In this appendix, in figure C.3 is also shown the relation between sumc and nominal power. These are values given by different manufacturers for scheduled maintenance. Unscheduled maintenance is not included, but an addition of 20% is normally assumed. Wear rate is in this case calculated with D_B in mm, p_{me} in bar and c_m in m/s. From the results it seems to be true that sumc is higher for higher wear rate, but there are large differences between manufacturers. Besides that, specification of the maintenance costs are poor, so it is hard to check if comparison is fair. Further research is needed to get better insight in these costs. All variables in equation 3.27 have the same influence on the wear parameter, perhaps different powers for each variable should be used.

3.2.9 Reliability

Reliability is always a difficult topic. It is difficult to put numbers to it, and there is not much data available. The AES study mentions MTBF numbers for medium and high speed diesel engines of 10000 hours and a MTTR of 5 hours. It does not mention numbers for slow speed engines. Probably, the MTBF of a medium speed engine should be higher than the stated number, and for a slow speed engine even higher. For medium speed diesel engines probably a MTBF value of 12000 hours and for slow speed engines 14000 hours would be better.

3.2.10 Initial purchase costs

To be able to do estimates on purchase costs of diesel engines the Cost Estimating Relationship (CER) of the Cost Analysis section is used. This CER is based on historical data of purchases and quotations. The CER is commercially confidential information which is not explicitly presented here. It can be found in a confidential appendix of this thesis. In figure 3.13 the CER from DMO is compared to CER's from other sources.

In a preliminary propulsion and power study by Rolls Royce, Vrijdag (2011), some ball-park figures to estimate the costs of a diesel engine and a diesel-generator set are mentioned. The figures are converted from GBP to €, at a rate of 1.15 GBP/€.

- High-speed: 230-285 €/kW
- High-speed (DG-set): 316 €/kW
- 1000 rpm: 340 €/kW
- 900 & 1200 rpm (DG-set): 460 €/kW
- Medium-speed: 400 €/kW
- 720 rpm (DG-set): 518 €/kW
- 600 rpm (DG-set): 575 €/kW

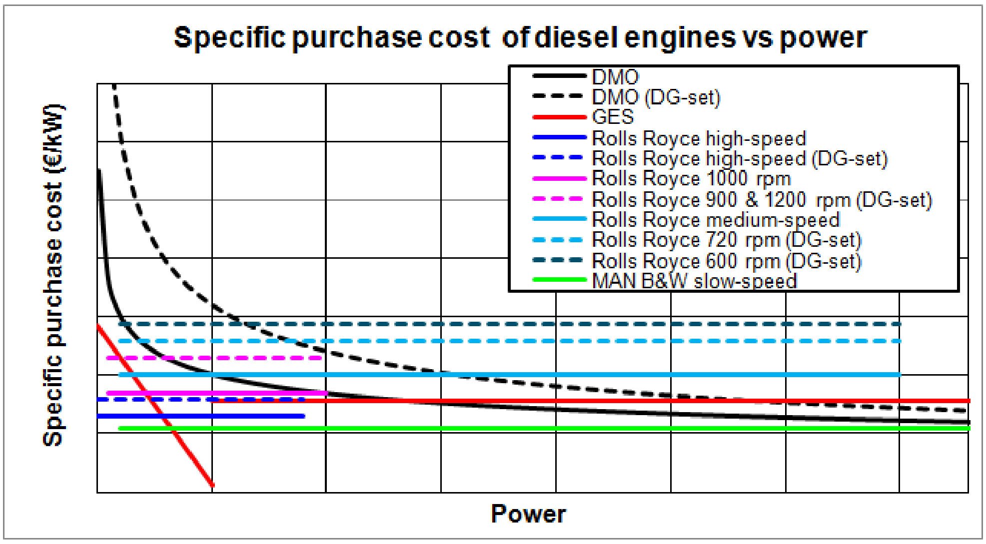


Figure 3.13: Specific purchase costs (€/kW) of diesel engines vs rated power, according to different sources. Cell height is 200 €/kW

For the price level of slow speed diesel engines MAN B&W is consulted, Nijsen (2010).

- Slow-speed: 220 €/kW

In figure 3.13 are also presented the relations as mentioned in the AES study. These figures date from 1998. They are converted to €, and increased at an inflation rate of 2% per year.

- < 5 MW: $\left(570 - 109 \cdot P_{[kW]} + 0.012 \cdot P^2_{[kW]}\right)$ €/kW
- > 5 MW: 311 €/kW

In the component study in Frouws (2008), the specific purchase costs are related to the bore diameter of the diesel engine. This approach is less practical for early design estimations, because then the designer has to determine the bore diameter in early stage. Bore diameter is related to the power according to equation 3.5, but still a CER as function of power is more practical.

- $\left(270 \cdot \left(\frac{D_b}{0.38}\right)^{0.7}\right)$ €/kW

3.3 Gasturbine

A gasturbine converts chemical energy from a fuel into mechanical energy of a rotating shaft. It is a rotary internal combustion engine that extracts energy from a flow of combustion gas. Unlike the diesel engine the gasturbine has a continuous combustion in the combustion chamber. Temperatures are more constant in the machine, and can, for material stress reasons, not be as high as peak temperatures in a diesel engine. A gasturbine comprises of a compressor, combustor and a turbine. Air is used as a working medium. Marine gasturbines are often derived from aerospace, so-called Aero Derivative GasTurbines (ADGT). These engines have high fuel requirements. For that reason, marine gasturbines normally run on marine diesel fuel. Industrial gasturbines, for example Siemens SGT-500, can run on HFO or even on coal.

In the compressor the air is compressed to a higher pressure and temperature. In the combustion chamber the fuel is added and combusted which further increases the temperature. In the turbine the combustion gas is expanded to a lower pressure and temperature. During the expansion power is extracted from the hot gas. A part of the power is used to drive the compressor and the rest can be used to drive a generator or a shaft with a propeller. A marine gasturbine normally comprises of two parts: the gas engine and the power turbine. The gas engine is a module containing the compressor(s), combustion chamber and the turbine(s) to drive the compressor(s). In case of ADGT's, this formerly was the airplane engine. In marine application the gasturbine should drive a shaft, that's why the power turbine is positioned after the gas engine. The power turbine converts the output power of the gas engine into mechanical energy of a rotating shaft.

> Some well-known marine gasturbine brands are: General Electric, Rolls Royce, Siemens, Pratt&Whitney.

Auxiliary systems

A gasturbine is started with an air start system (pneumatic starter motor). Starting of a gasturbine takes some time because the turbine has to deliver enough power to be able to drive the compressor so it can keep itself running. Further, a gasturbine needs a lubrication system and fuel supply. Lubrication oil system is normally included in the module. Fuel supply consists of feed pumps and fuel treatment. The combustion in an ADGT requires pure and clean fuel to prevent pollution and corrosion in the gasturbine. For this reason seperators are normally used in the fuel supply line of gasturbines to take out the water and other undesirable constituents.

Data analysis

For analysis of this chapter a database of gasturbines is used. This database is made with manufacturers data from the internet or technical specifications and information from the GES database and with data from Henderson (2010). The database holds 59 gasturbines, of which also non-marine gasturbines. For analysis of the data only a part of is considered. Only the marine gasturbines of which sufficient data is available are considered. This leaves a number of 13 gasturbines, 12 simple cycle and 1 Inter-Cooled Regenerative cycle (ICR). These are all gas engines plus a free power turbine.

- General Electric: LM500, LM1600, LM2500, LM2500+, LM2500+G4, LM5000, LM6000
- Rolls Royce: Olympus TM3B, Tyne RM1C, Spey SM1A, Spey SM1C (military version), Marine Trent 30, WR21 (*ICR*)

3.3.1 Available power

Other than with marine diesel engines, the number of marine gasturbines on the market is only limited. Gasturbines are available in a range of approximately 3.5 to 45 MW. There are also smaller gasturbines available, so-called microturbines ($30 - 250$ kW). These could for example be used in combination with an alternator in an all electric ship concept.

Gasturbines are very common in aviation because of their high power density, but in the marine sector relatively few gasturbines are used. Most of the marine gasturbines started as airplane engines, and on special request are further developed for marine use. These are so-called aeroderivative gasturbines (ADGT). Development of marine gasturbines is an expensive job, so manufacturers develop engines for specific goals. That's the reason why the choice on the marine gasturbine market is very limited, compared to diesel engines.

3.3.2 Dimensions

As with the diesel engine it is tried to relate the dimensions of a gasturbine to brake power. Delivered power of a gasturbine is proportional to the massflow through the engine $\dot{m}$ and the temperature change ΔT.

$$P_B = \dot{m} \cdot \Delta H = \dot{m} \cdot c_p \cdot \Delta T \tag{3.28}$$

A bigger massflow requires a larger diameter D of the gasturbine, if equal velocity is assumed.

$$\dot{m} = \rho \cdot \frac{\pi}{4} \cdot D^2 \cdot v \tag{3.29}$$

With ρ is density and v is flow velocity. Engine diameter is proportional to the width W and height H of the engine. Combining this with the above equations results in the proportionality of width and height with P_B.

$$W \text{ and } H \propto D \propto \sqrt{P_B} \tag{3.30}$$

The length of the gasturbine is proportional to the number of compressor stages. The number of stages is determined by the pressure ratio ϵ and compressor technology level (pressure ratio achieved per compressor stage). The pressure ratio is proportional to the temperature ratio τ.

$$\epsilon^{\frac{\kappa-1}{\kappa}} = \tau \tag{3.31}$$

Because engine power is dependent on temperature differences and temperature ratio is dependent on pressure ratio and pressure ratio determines the number of compressor stages and the number of compressor stages determines the length of the engine, a relation between engine length L and engine power P_B is expected.

The specific dimensions of the database gasturbines are presented in figure 3.14. These are the dimensions of the complete gasturbine modules. Obvious relations between dimensions and P_B are recognized, but these are not consistent with the theoretically expected relations. The dependency on P_B is much weaker than expected. Apparently, there is another dependency of diameter with power that mitigates the square-root dependency in formula 3.30. The relations between dimensions in meters and P_B in megawatt are described with:

$$\boxed{\text{Length} = (3.25 \cdot {P_B}^{-0.72}) \cdot P_B} \tag{3.32}$$

$$\boxed{\text{Width} = (1.50 \cdot {P_B}^{-0.82}) \cdot P_B} \tag{3.33}$$

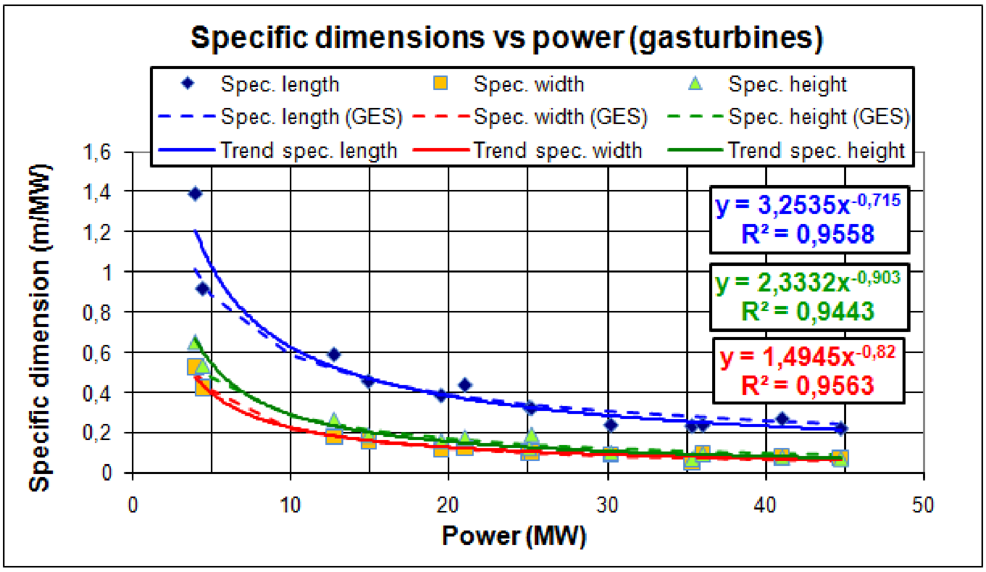


Figure 3.14: Specific dimensions (m/MW) of database gasturbines vs rated power (MW)

$$\boxed{\text{Height} = (2.33 \cdot {P_B}^{-0.90}) \cdot P_B} \tag{3.34}$$

The standard deviations of the specific dimensions of engines in the database from the trends are:

- $\sigma = 0.081$ for specific length, with a max. deviation of 22%
- $\sigma = 0.018$ for specific width, with a max. deviation of 42%
- $\sigma = 0.030$ for specific height, with a max. deviation of 37%

In figure 3.14 are also presented the models as used in GES (dotted lines). The GES models are (almost) similar to the trendlines that Excel produced. This is logical, because almost all gasturbine data comes from the GES database.

A remarkable deviation from the trend in figure 3.14 is the height of the ICR cycle gasturbine. This is logical, because the ICR gasturbine has the heat exchanger on top of the engine. From the data it follows that the ICR machine is 1.5 times as high as a comparable simple cycle.

Inlet and outlet

In calculating the space consumption of a gas turbine, the space for in- and outlet ducts should also be taken in to account, since these consume a significant amount of space. The inlet and outlet duct transport the large mass flow of air and exhaust gas to and from the engine. The in- and outlet also have the function of filtering and conditioning (humidity) the air or exhaust gasses and muffling the noise of the combustion and the rotary equipment. The mass flow of air through a gasturbine is very high. A gasturbine normally operates with an air excess ratio (λ) of 3 to 5. This large air excess is necessary to cool the turbine blades. The massflow of air through the engine and the required surface area of the duct can be calculated with equation 3.16, and the minimum duct area of in- and outlet duct with equation 3.17 and 3.18. The assumed values for this formula are presented in table 3.6. Inserting these assumed mean values in formula 3.17 and 3.18 gives the following values for specific duct area:

inlet duct:	0.193 m^2/MW
outlet duct:	0.165 m^2/MW

Variable	Assumed value	Unit	Condition
λ	4	(-)	
σ	14.5	(-)	
η_e	0.37	(-)	
LHV_F	42.700	(MJ/kg)	
ρ_{air}	1.27	(kg/m^3)	at 20°C, 1 atm
ρ_{gas}	0.45	(kg/m^3)	at 550°C, 1 atm
$v_{max,in}$	15[a]	(m/s)	
$v_{max,out}$	50[a]	(m/s)	

[a]: Based on assumptions from Rolls Royce

Table 3.6: Assumptions in determining dimensions of in- and outlet duct of a gasturbine

In GES a fixed diameter value for the ducts is mentioned per power range. Three power ranges are distinguished: low powers (<2 MW), medium powers (2-10 MW) and high powers (>10 MW). For low powers GES uses a duct diameter value of 550 mm for both in- and outlet duct, which corresponds with a duct area of 0.24 m^2. For medium powers 1000 mm, which is 0.79 m^2, and for high powers 1500 mm corresponding with 1.77 m^2.

3.3.3 Weight

The data of the gasturbine is presented in figure 3.15. The relation between specific weight and power of a gasturbine shows a little more convenient relation than the diesel engine, but still not very satisfying. The figure also shows the weight model that is used in GES. This model shows a growing specific weight with higher power, while the data shows a decreasing specific weight. It seems that the GES model is highly inaccurate. The weight of the complete modules is used, this includes the baseplate. The ICR cycle is put separately, because this obviously deviates from the trend of the other engines. An ICR gasturbine comes with some additional heavy engine parts; the heat-exchangers for the intercooler and the recuperator. Based on the one ICR datapoint we might say that an ICR cycle gasturbine is twice as heavy as a same power simple cycle gasturbine.

The following general relation for the simple cycle gasturbines is made based on the data, with weight in ton and P_B in megawatt:

$$\boxed{\text{Weight} = (5.97 \cdot {P_B}^{-0.59}) \cdot P_B} \tag{3.35}$$

The standard deviation of the specific weight of engines in the database from the trend is:

- $\sigma = 0.46$, with a max. deviation of 55%

Where the GES model has much worse deviations:

- $\sigma = 0.99$, with a max. deviation of 208%

Based on data from Rolls Royce Spey SM1C propulsion unit Rolls-Royce (1997) a weight breakdown is made:

Gasturbine engine: 7%

Acoustic enclosure: 13%

Power turbine: 10%

Base plate: 27%

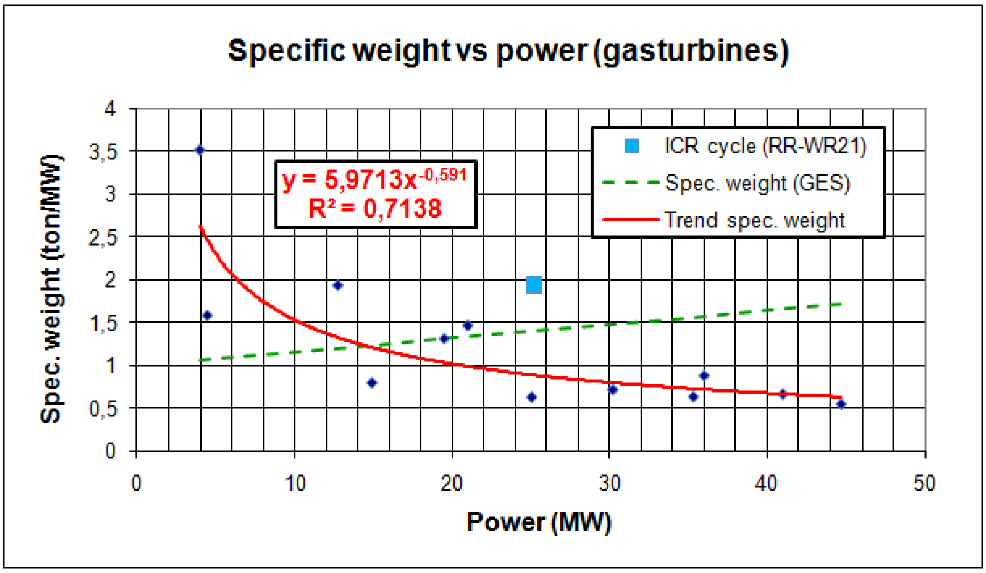


Figure 3.15: Specific weight (ton/MW) of database gasturbines and from GES vs rated power (MW)

The rest of the weight is of ducts, bellows, volutes and structure in which the engine is mounted. The weight of the gas engine itselfs is only a small part of the weight of the complete propulsion unit.

When figure 3.15 is compared to figure 3.6 it is obvious that a gasturbine is much more power dense in terms of weight than a diesel engine. This is exactly the strong point of the gasturbine as a prime mover compared to its competitors (diesel engine, steam plant, nuclear plant).

3.3.4 Operating speeds

The operating speed of a gasturbine can be described in two ways. There is the rotating speed of the gasgenerator and the speed of the powerturbine, which are not physically coupled. The powerturbine is coupled to the outgoing shaft, so that is the speed of interest. Three nominal powerturbine speeds are distinguished from the gasturbines in the database: 3600 rpm, 5600 rpm and 7000 rpm. From the data one can roughly conclude:

Nominal power (MW)	Nominal speed (rpm)
< 15	7000
$15 - 25$	5600
> 25	3600

See also figure C.4 in Appendix C.2. The output speed of the powerturbine is not constant, but can be adjusted by reducing the massflow through the turbine. Basically there is no minimum speed of the outgoing shaft as long as the gasgenerator has a speed above self-sustaining speed. Though, to operate the powerturbine a little efficient the minimum operating speed is about 30% of nominal, for lower output speeds the efficiency decreases dramatically.

3.3.5 Efficiency

The gasturbine is not known for its sublime efficiency, in particular at part load, while nominal efficiency of a gasturbine *can* get close to diesel engine efficiency. Efficiency of a diesel engine is higher because heat addition takes place at higher temperatures. Peak temperatures in a diesel engine are in the order of magnitude $1500 - 2000°C$, where peak temperature in gasturbine is around $1000 - 1250°C$, Stapersma (2009b). This temperature can not be as high because of the continuous combustion in a gasturbine. There are no materials that can withstand these temperatures and forces continuously. But with heat-regeneration techniques the thermodynamic efficiency can be increased. This is called recuperation. There are more improvements that can be done to the simple cycle, which make it an advanced cycle. Improvements either increase thermodynamic efficiency (η_{th}) or specific power ($\dot{w}$) of the gasturbine.

- Recuperation: pre-heating of the compressed air by the hot exhaust gases in a heat exchanger to lower the heat input by fuel ($\eta_{th} \uparrow$, $\dot{w} \downarrow$)
- Inter-cooling: cooling of the air between compression stages to lower compressor work ($\eta_{th} \downarrow$, $\dot{w} \uparrow$)
- Reheating: additional fuel heat input between expansion stages to increase output power ($\eta_{th} \downarrow$, $\dot{w} \uparrow$)

A good example of such an advanced cycle gasturbine is Rolls Royce WR21, which combines inter-cooling and recuperation. Efficiency of this engine is higher than comparable simple cycle gasturbines, especially at partload, see figure 3.18. Disadvantages of applying these techniques is that the gasturbine will become heavier, see figure 3.15, react slower to load steps, due to the time lag associated with a heat exchanger, but also the reliability decreases due to added complexity and maintenance effort increases.

The efficiency data from the database gasturbines is plotted versus the rated power in figure 3.16. The ICR cycle engine is plotted separately. Based on the one ICR datapoint, the assumption is made that ICR efficiency is 5% higher than a comparable simple cycle machine. The general trend for nominal overall engine efficiency ($\eta_{e,nom}$) of the simple cycle gasturbines as function of nominal brake power ($P_{B,nom}$) in MW is given by formula 3.36:

$$\boxed{\eta_{e,nom} = 0.27 \cdot {P_{B,nom}}^{0.11}} \tag{3.36}$$

In GES a constant nominal efficiency of 32% for all gasturbines is modeled, which is also plotted in figure 3.16.

The main efficiency problem of a gasturbine is the rapid efficiency drop at part load operation. Part load efficiency of gasturbines is approximated with the quadratic parametric formula from equation 3.23.

$$sfc^* = \frac{1 - c \cdot (1 - P^*) + d \cdot (1 - P^*)^2}{P^*}$$

The parameters are matched with known data from the Rolls Royce Spey, in Rolls-Royce (1997), and turn out to be $c = 0.902$ and $d = 0.045$. For the ICR cycle Rolls Royce WR21, the parameters are $c = 1.08$ and $d = 0.16$. Results are shown in figure 3.17

Figure 3.18 shows the typical efficiency plot of both gasturbines. To increase part load efficiency of simple cycle gasturbines, variable geometry blades can be applied in the compressor and turbine, but this increases complexity of the system.

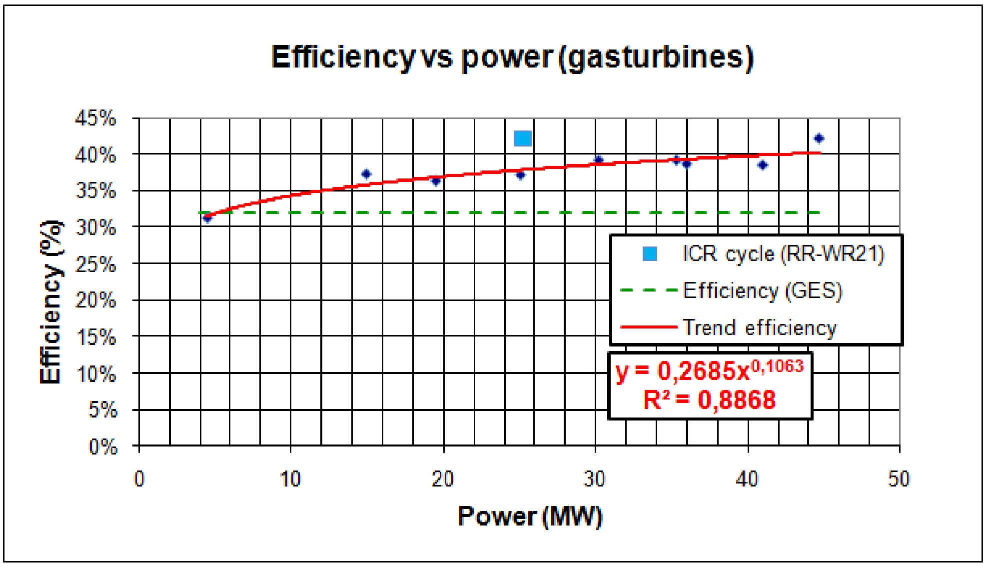


Figure 3.16: Nominal engine efficiency (%) of database gasturbines and from GES vs rated power (MW)

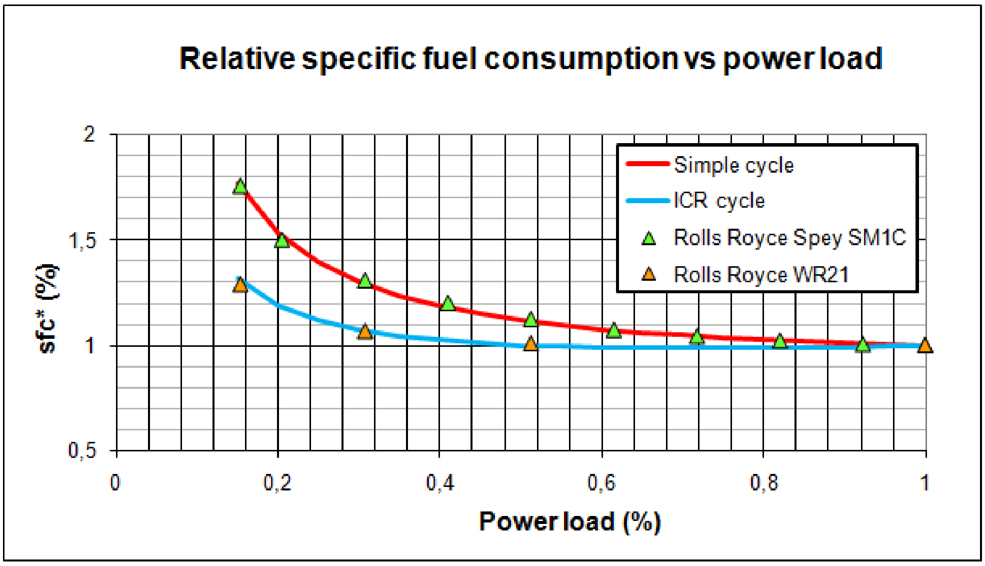


Figure 3.17: Relative specific fuel consumption compared to nominal value (%) vs powerload (%) for a simple cycle gasturbine and an ICR cycle gasturbine

(source: Rolls-Royce (1997) and Rolls-Royce (1999))

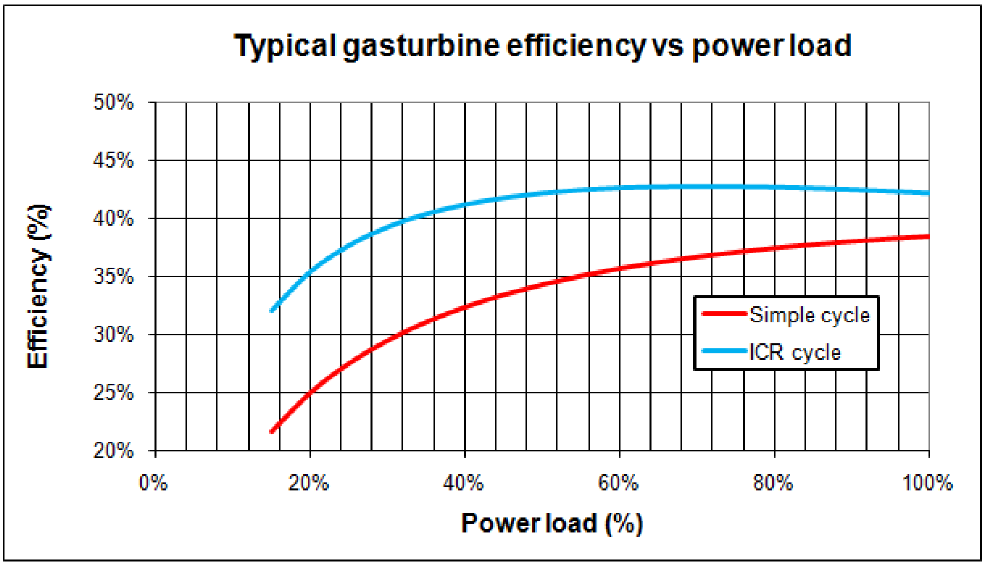


Figure 3.18: Typical efficiency curve of a simple cycle gasturbine (based on Rolls Royce Spey SM1C) and an ICR cycle gasturbine (based on Rolls Royce WR21)

3.3.6 Signatures

Underwater noise

The gasturbine is a rotating machine with a continuous combustion proces. This has advantages for the signature profile when compared to a reciprocating diesel engine with a discontinuous combustion proces. Because of the continuous combustion proces in the gasturbine there are no pulsating forces exerted on the shaft. This gives a flat torque output.

Sound is produced by gases flowing through the machine at high velocities. The flow interacts with solid bodies in the engine which start vibrating. Turbulence in the flow, thus noise, is enhanced if rotor blades or stator vanes are damaged. The interaction of rotor blades and stator vanes also introduce noise. The number of interacting blades and vanes determines the frequency. By increasing the distance between rotor and stator this effect can be decreased. Unsteadiness of the combustion is another source of noise which can be prevented by proper design of the combustor. The so-called 'bleeding valve', which opens at low speeds to bleed off compressor air, produces very sharp tonals in the high frequency bands (related to compressor blade frequency), when opened. The airborne noise a gasturbine produces is primarily at higher frequencies. From measurements on Olympus gasturbine it follows that this is in the range 125 Hz - 25 kHz at levels around 120 dB. The human ear is very sensitive for these high frequencies. For that reason marine gasturbines are in an acoustic enclosure. Structureborne noise of a gasturbine is low. Resilient mounting decreases structureborne noise to even lower levels.

Compared to the diesel engine, the gasturbine has much less harmfull underwater noise. Extra advantage is the lower weight of the gasturbine, which makes it possible to place it higher up in the ship. The higher the machinery is situated in the ship, the less underwater noise, because the ship's structure acts as damper.

Infrared

The exhaust gas temperature of a gasturbine is higher than that of a diesel engine. Infrared signature will be of greater concern with a gasturbine. The temperature of the exhaust gas leaving the engine is typically around $550°C$ depending on the load. The chimneys are heated up and transmit infrared light. Infrared light is not visual to the human eye but with an infrared telescope the enemy is able to detect it and track the ship. Transmission of the chimneys can be decreased by isolating the chimney with heat-isolators and a façade around the chimney. The exhaust gas itself can not be isolated, because it is blown into the environment. To reduce the infrared transmission of the gas, it needs to be cooled as much to the environmental temperature as possible. This can be achieved by leading the exhaust gas through water-cooled heat exchangers. In that case the heat can even be used profitable in heating of compartments for example. With an advanced cycle gasturbine, with recuperator, the exhaust gas temperature will be lower. Part of the heat is regenerated to the combustion air, which results in lower IR signature.

3.3.7 Shock resistance

Gasturbines have very good shock resistance. Onboard ships they are placed in a housing which serves as a shockdamper. The housing might be placed on flexible mounting, which improves shock resistance and is also advantageous for signatures.

3.3.8 Maintainability

Maintenance on a gasturbine will be described by the maintenance plan of the Rolls Royce Spey SM1C, in service with the RNLN. The manufacturer prescribes a certain maintenance plan, which in practice seems to be very conservative and inefficient. Within RNLN, research is done on doing more efficient maintenance, a.o. described in Oudenaller (2011). The ship's crew maintains the engine according to the prescribed plan, which includes checking the engine status regularly and changing of fuel pump and air starter motor every 4000 resp. 2000 hours. When an engine breaks and needs repair, it is lifted out of the ship and sent to the manufacturer (Rolls Royce) for repair. The Spey engine needs an overhaul (of at least one major part) every 5000 running hours. In practice, this number of running hours fits within the service life of the ship with the current operating profile.

Repair of a gasturbine is a specialistic and expensive job. For that reason, the RNLN has put together a Memorandum Of Understanding (MOU) with the Royal Navy (RN) and the Belgian Navy (BN) for repair and overhaul of the Spey SM1A engine, which is an earlier version of the SM1C. Together, those navies have a common pool of spare engines from which one can be picked in case of a repair of an engine. The costs of repair and overhaul by the manufacturer are divided by the three parties. These costs are recorded over a period from 1991 untill now, so an insight in maintenance costs of the SM1A engine per runninghour can be given based on these costs. The results are presented in figure 3.19. The figure shows (1) the costs per year divided by the number of runninghours in that year (blue line) and (2) the cumulative costs from 1991 untill a certain year divided by the cumulative number of runninghours from 1991 untill that year (red line). The cumulative cost per runninghour lies around 80€. The Spey SM1A has a power of 12.75 MW, so specific unit maintenance cost (sumc) is about 6.50 €/MWh. If this number is generalized for all gasturbines this means a maintenance cost in €/h:

$$\boxed{\text{Maintenance cost}_{(€/h)} = 6.50 \cdot P_B} \tag{3.37}$$

For comparison, a diesel engine of the same power as the SM1A has a sumc of approximately 2.40€/MWh, according to the used model.

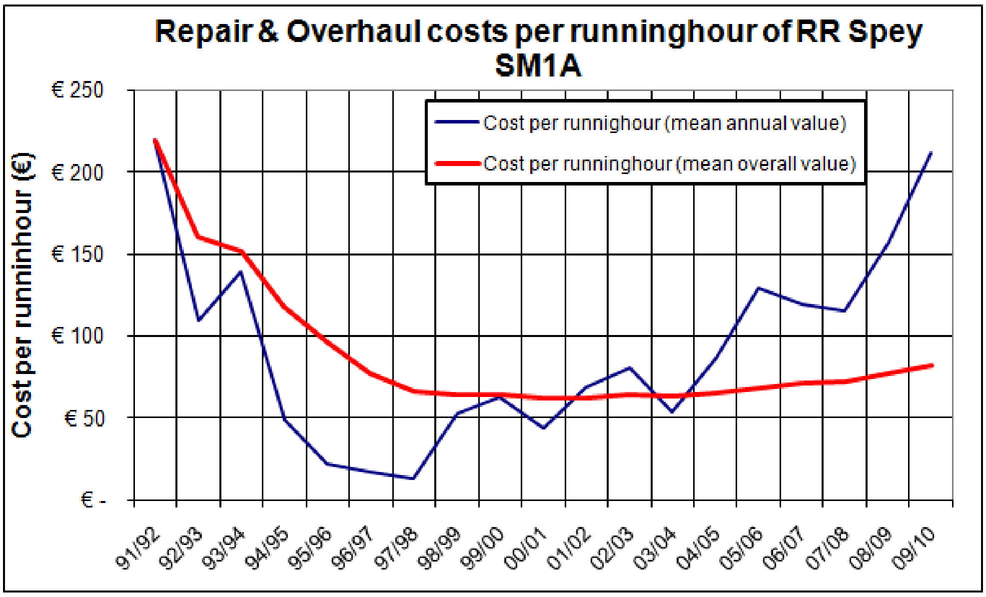


Figure 3.19: Overview of repair and overhaulcosts per runninghour of Rolls Royce Spey SM1A (12.75 MW) per year

3.3.9 Reliability

In the AES study the MTBF value for simple cycle gasturbines is 4500 hours and for intercooled recuperated (ICR) cycle 20000 hours. These numbers are probably not representative for current gasturbines. These MTBF numbers are very old, when the ICR gasturbine wasn't even in service yet. MTBF numbers of the ICR gasturbine are based on prospects for the WR21. In practice the ICR gasturbine is less reliable and shows more failures than a simple cycle gasturbine, caused by the recuperator. The MTBF value for simple cycle gastubines is probably a bit on the low side, based on experience within RNLN. A suggestion for correction of these numbers would be in the order of 8000 hours for simple cycle gasturbines, and in the order of 5000 hours for ICR gasturbines. For both types a MTTR of 8 hours was assumed in the AES study, which sounds reasonable.

3.3.10 Initial purchase costs

The Cost Estimating Relationship (CER) that the Cost Analysis section at DMO uses to estimate the costs of gasturbines is not explicitly mentioned here (see confidential appendix), but is compared to the CER that is mentioned in the AES study in figure 3.20. The CER in the AES study dates from 1998. It is converted to €, and increased at an inflation rate of 2% per year. Two cost relations are distinguished: for simple cycle gasturbines and intercooled recuperated (ICR) cycle gasturbines. The cost relation of ICR cycle is only based on the Rolls Royce WR-21 in a time before the WR-21 was even on the market, so this relation will probably be far from correct. The CER's in the AES study are given by:

- Simple cycle: $\left(2918 \cdot P_{[kW]}^{-0.23}\right)$ €/kW
- ICR: $\left(2300 \cdot P_{[kW]}^{-0.16}\right)$ €/kW

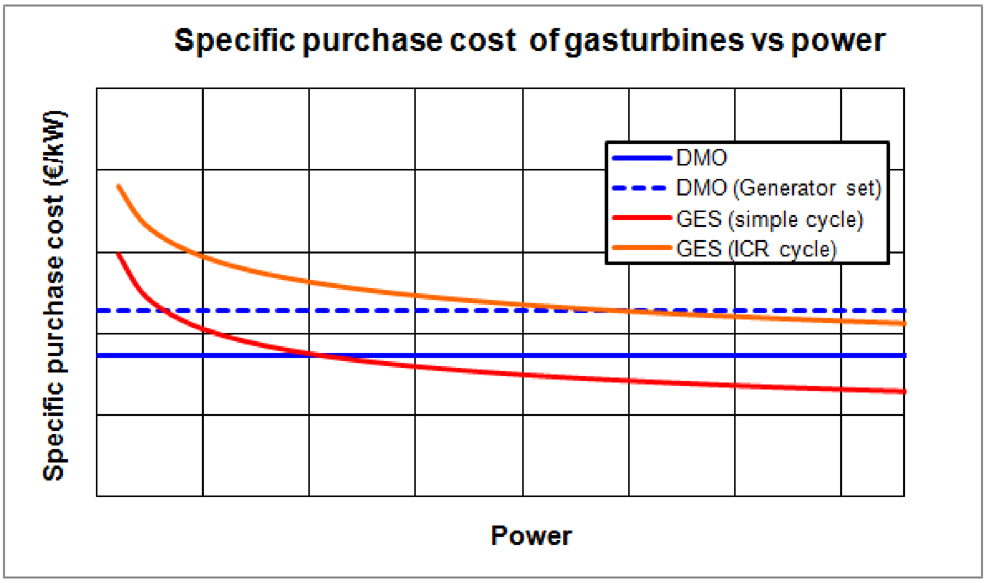


Figure 3.20: Specific purchase costs (€/kW) of gasturbines vs rated power, according to different sources. Cell height is 200 €/kW

3.4 Fuel cell

A fuel cell is an electrochemical cell that converts chemical energy from a fuel into electric energy. A fuel cell consist of an anode, which is fed with a fuel flow, a cathode, which is fed with the oxidizer, and an electrolyte in between, which makes the chemical reactions possible. There are many different types of fuel cells, but they are all based on the same principle:

> at the anode a catalyst oxidizes the fuel (usually hydrogen), turning it into a positively charged ion and a negatively charged electron → the ions pass through the electrolyte to the cathode side, and the electron via a wire creating an electric current → at the cathode side they reunite and react with the oxidizer (normally oxygen) and create water or carbon dioxide.

The fuel cell is a very interesting piece of technology with high potential for clean, emission free, energy conversion with high efficiencies. Because fuel cells convert chemical energy directly into electricity without combustion, it is not governed by Carnot's law, and has higher theoretical efficiency. Especially when the produced (waste) heat is utilised. Fuel cells have high power density. Another advantage for use in naval vessels is the quiet operation, because there are no moving parts in a fuel cell. The absence of moving parts makes it potentially a highly reliable system.

As with everything, there are also disadvantages, which make that a fuel cell is not a reasonable option for this design study. Main reason is costs! The goal is to reduce the cost in order to compete with current market technologies including internal combustion engines, but for now fuel cells are very expensive. Other issues with fuel cells that need further development are the durability (service-life) and the power level. Currently, fuel cells are only developed for limited power levels, order of magnitude 1 MW. Development is focussed on the so-called Proton Exchange Membrane Fuel Cell (PEMFC) which is a low temperature (30-100 ^{circ}C) fuel cell. Disadvantage is the required fuel purity of this fuel cell type. If diesel fuel is used as a fuel this first goes through a large reformer plant before it enters the fuel cell. This consumes a lot of space and reacts rather slow.

In short, the fuel cell is a very promising technology, but not yet mature for transition to naval warships. At least not as provider of propulsive power, auxiliary power supply might be an option, but is not investigated in this study.

3.5 Electrical machines

Electric machines are used to convert electrical energy into mechanical energy or vice versa. There is the *user* of electrical energy, the motor, and the *provider* of electrical energy, the generator. The application is opposite, but the technique is the same.

Electric motor: Electric energy $\longrightarrow$ mechanical energy

Electric generator: Mechanical energy $\longrightarrow$ electric energy

Both are rotating machines. Explanation of the working principles is based on KleinWoud & Stapersma (2003), chapter 2.3.5. The working of the motor is based on the Lorentz force (F_L) principle; a force will act on a current-carrying conductor when it is placed in a magnetic field. A torque is created because the conductor turns and the forces act on both sides in opposite direction. The torque on the rotor is given by:

$$M = K_M \cdot \Phi \cdot I \tag{3.38}$$

Where M is torque in (Nm), K_M is a constant for a certain motor which depends on size and number of windings an the flux density variations in the motor, Φ is the magnetic flux of the magnetic field on the stator in (Wb) and I is the current through the conductor on the rotor in (A).

The working of the generator is based on Faraday's law for magnetic induction. It states that an induction voltage (E), also called electromotive force (EMF), is generated over a conductor when it is moving in a magnetic field. In a generator, the conductor on the rotor is coupled to, for example, the outgoing shaft of a diesel engine and is rotated in a magnetic field on the stator. The induced voltage, or electromotive force is given by:

$$E = K_E \cdot \Phi \cdot n \tag{3.39}$$

Where E is induced voltage in (V), K_E is a constant for a certain generator which depends on size and number of windings an the flux density variations in the coil, Φ is the magnetic flux of the magnetic field in (Wb) and n is the rotational speed of the coil in the magnetic field in (s^{-1}).

In an electrical machine two sets of windings can be distinguished:

- Field windings to create a magnetic field (not for permanent magnet machines)
- Armature windings to provide current-carrying conductors

Two main types of electrical machines can be distinguished: synchronous and asynchronous machines. Based on the type of supply to the windings (direct current DC, alternating current AC or permanent magnet PM), another subdivision can be made. See table 3.7. The working principles will be explained, but from the *'motor point of view'*. The principles are the same for generators, only opposite. In general, generators are always of the 2^{nd} type: AC synchronous. A schematic view of the machine types is presented in figure 3.21.

1. The synchronous DC motor has the stator and the rotor fed with direct current. The stator can also be a permanent magnet. To be able to feed the rotor and to switch the direction of the current, the rotor needs a special connection to the DC source, with commutator brushes and split rings. The brushes and split rings wear and need to be replaced from

Synchronous			Asynchronous
DC	*AC*	*PM*	*AC*
(**1**) Stator and rotor both excited by DC	(**2**) Armature on stator fed with AC and field windings on rotor with DC	(**3**) No field windings, armature fed with AC or DC	(**4**) Stator supplied with AC, rotor too via induction

Table 3.7: Distinction between electrical machine types

time to time. It is also a possibility to induce an alternating current on the rotor and then rectify this current; this makes it possible to have a brushless DC motor. Main characteristic of a DC motor is that it has a constant torque limit up to a certain base speed. The maximum torque is determined by maximum current through the armature or rotor. Base speed is the speed of the motor at maximum torque and maximum voltage of the armature. Speed is controlled by the armature voltage, up to base speed. For higher speeds than base speed the armature voltage is set to maximum and the magnetic flux is reduced by field weakening of the stator electromagnet. In case of a permanent magnet on the stator this is not possible. Above base speed the DC motor has a constant power limit. With modern power electronics, the voltage of the stator and rotor can be controlled separately with DC-DC choppers. Advantages of a brushed DC motor include low initial cost, high reliability, and simple control of motor speed. Disadvantages are high maintenance and low life-span for high intensity uses. DC machines are rarely used as generator

2. The synchronous AC motor has the rotor excited by direct current or has a permanent magnet rotor. The rotor exactly follows the frequency of the stator windings, thus no slip. This also means that the motor has slip-rings to feed the rotor or a brushless rotor with rectifier when the current on the rotor is induced, like in a brushless DC motor. Generators onboard are normally brushless AC synchronous machines. The rotor of an synchronous machine can also be a permanent magnet. The speed of this motor can directly be controlled by adjusting the AC frequency on the stator. The torque can be controlled by the DC voltage on the rotor. This type of motor is often used in constant speed applications (pumps) because speed is independent of load, as is the case in the asynchronous (slip increases with increasing load).

3. The PM machine. As mentioned above, the DC machine and the AC synchronous machine can be equiped with permanent magnets. Permanent magnets have a very high torque-density, and therefore are suitable if weight and space are important. According to Trouwborst (1998), typical power density of a conventional electrical motor is 225 kW/m^3 and 2500 kW/m^3 for PM motors. In a PM machine, there are no losses in the rotor, so less heat development and less cooling is needed. Other advantage is the absence of brushes. Experience learns that magnetic material is very expensive, which makes permanent magnet machines about twice as expensive. Another disadvantage is that the magnetic field of a permanent magnet can not be controlled.

4. The AC asynchronous induction motor is the most widely used type of electric motor because of its simple construction. It doesn't have slip-rings because the rotor isn't powered externally. The rotor winding current is induced. This makes the rotor follow the rotating magnetic field of the stator. There is some slip between the rotation of the magnetic field and the rotation of the rotor, which is why this motor is called asynchronous motor. This slip is necessary to create electro-magnetic force, but makes it impossible, even in theory,

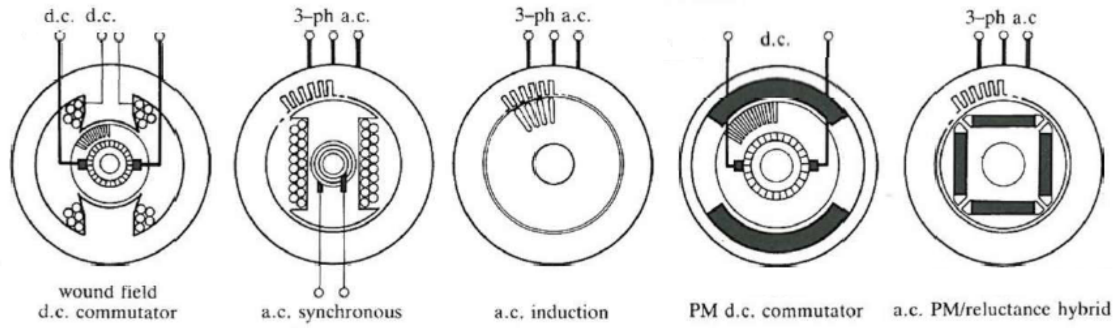


Figure 3.21: Electrical machine types as normally found onboard of ships

(source: Miller (1989))

to achieve zero rotor losses. This is one of the limitations of the induction motor. There are two types the (squirrel) cage rotor motor and wound rotor motor. The cage rotor motor is normally controlled by adjusting the frequency (with pulse width modulation or pulse frequency modulation). The wound rotor motor can be controlled by adjustable resistors that are in series with the rotor windings; downside is that the rotor needs to be connected to these resistors by a slip-ring. Both types can also be controlled in a limited range by adjusting the supply voltage. Disadvantage of the AC asynchronous motor in warships is the small airgap between rotor and stator it requires. A small airgap is bad for shock resistance.

Note: *The choice between AC or DC machines can be driven by the choice between AC or DC power distribution. DC power distribution has some advantages over AC because a.o. it can save installation costs, has higher distribution efficiencies and is easier to control. Disadvantage of DC power distribution traditionally is the expense and mass of air circuit breakers, but these converge towards power electronic converters necessary in AC distribution. Some references state that DC is the distribution medium of the future, Hodge & Mattick (2001).*

A great advantage of using an electrical machine as propulsion motor is the capability of so-called *four quadrant* operation. This ability depends on the type of converter that is driving the motor. Four quadrant operation means that the machine is able to deliver positive torque both at positive and negative rotational speed, and negative torque (braking) both at positive and negative rotational speed. This offers great maneuverability. Especially when compared to diesel engines and gasturbines which have a very narrow operating enveloppe when compared to electrical motors. When the electrical motor 'delivers' negative torque (braking) it is able to absorb the electric energy and deliver it back to the net. In this way energy can be won back.

New development in the world of electrical machines is high temperature superconducting (HTS) AC synchronous machines. The windings are made of superconducting material that has no electrical resistance at a temperature of $-179°C$. A HTS wire can carry more than 150 times the current of similar-sized conventional copper wire. This technique is fully in development. According to a producer of such machines, American Superconductor Corporation (AMSC), the machines can be half the size and 35% to 50% of the weight of comparably alternatives. Figure 3.22 gives a good example of the difference in size. Other advantages that are stated are lower losses especially at part load, despite there is a significant cooling power needed, this results in three to four times higher efficiency at part loads, the machines are acoustically quiet, highly reliable and need less maintenance. The fact that these machines are more compact makes it a good option to put in pods. A pod, or podded propulsor, is a streamlined gondola under the ship which holds the electric motor and propeller. This gives the benefits of saving space inside the hull, high manoevrability with steerable pods, better efficiency and less cavitation thus noise and vibration due to the absence of rudders, steering gear and thruster tunnels, Trouwborst (1998).

> Some well-known manufacturers of electric machines are: ABB, Holec, Siemens, Converteam, Alconza, VEM, Hyundai, Magnet Motor, Jeumont Electric.

Auxiliary systems

To operate electrical machines, some auxiliary machinery is needed, like cooling and lubrication. A lot of heat is generated in the windings caused by the resistance losses of the metal, also called 'copper losses' because normally copper is used. The resistance losses, thus cooling power, are proportional to I^2. Cooling can be forced air cooling, forced air-to-air cooling, forced air-to-water cooling or water cooling. When temperature reaches maximum value, overheating and burning of the insulation may occur. Electrical machines also need lubrication in the bearings. Further, an electric motor needs power supply, this can come from batteries or from an electric network, that needs to be fed with for example diesel-generators. Depending on the voltage of the supply, transformers might be needed. Transformers can have significant dimensions. Motors also need a converter to control the motor. These converters adapt the voltage and frequency of the power supply to the electric motor as required for the desired motor speed. The converter is described in chapter 3.7, even as the switchboard.

Data analysis

For the analysis of the electrical machine in terms of dimensions, weight etc., at first the AES study, van Dijk *et al.* (1998), was used as a reference. The relations found in this study form the base for the software tool GES. Some data from currently in service machines, or machines in development, is used to validate the GES relations. Data is collected for the following machines:

- 3 DC motors (Type 23 (UK), Walrus class, Celtic Explorer (IRL))
- 7 AC asynchronous motors (LPD-1[5] (+bowthruster), HOV[6], JSS[7], OPV[8], Type 45 (UK) and LPD-2 bowthruster)
- 1 AC synchronous generator (LCF[9])
- 2 PM transverse flux propulsion motors
- 1 PM generator
- 2 HTS synchronous motors

The two HTS motors (5MW and 36.5MW) from AMSC are added, to see the difference with conventional machines, which can also be seen in figure 3.22. The JSS and Type 45 motors are more advanced types, which are smaller and lighter. Alstom developed a family of 15 phase Advanced Induction Machines (AIM). The Type 45 motor is from the AIM family. Permanent magnet motors are gaining interest for their high power density. Three PM machines are added. One 20 MW, 180 rpm transverse flux Permanent Magnet Propulsion Motor (PMPM) from the United Kingdom Ministry of Defence (UKMOD) is found in literature, Newell *et al.* (1998). Further, a 1.04 MW PM motor and a 1.7 MW PM generator developed by MTG are added.

[5]Landing Platform Dock
[6]Hydrografic research vessel
[7]Joint Support Ship
[8]Oceangoing Patrol Vessel
[9]Airdefence and Command Frigate

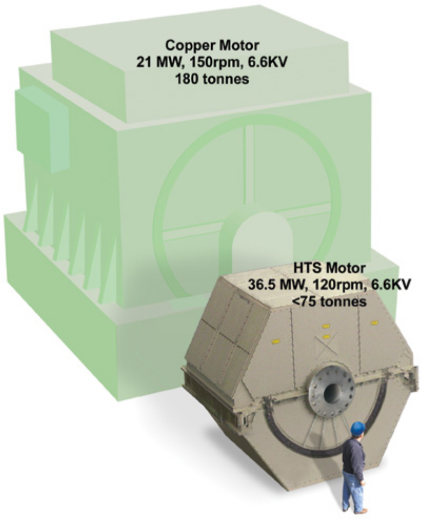


Figure 3.22: Example of difference in size of HTS motor and conventional motor

(source: http://www.amsc.com/products/motorsgenerators/shipPropulsion.html (february 2011))

3.5.1 Available power

In electrical engineering two main definitions of power are distinguished: effective power P and apparent power S. Effective power is the power that can actually be used, and apparent power is the power that the machine has to be designed for. The difference is caused by a phase shift ϕ between voltage U and current I. This phase shift makes that the product of voltage and current (apparent power) is not actually the power that is usable (effective power), so a 'power loss' occurs. The ratio between effective power and apparent power is called the power factor and is equal to $\cos(\phi)$. Direct current (DC) machines have no phase shift thus effective and apparent power are the same.

$$S = U \cdot I \tag{3.40}$$

$$P = U \cdot I \cdot \cos(\phi) \tag{3.41}$$

The power that is of interest to the designer of the propulsion and power generation machinery is the effective power. Electrical machines are available in a wide power range, although the selection from available motors in the higher power range is rather limited. Electrical machines in the higher power range normally operate with high voltage to limit thermal loading of the machine. With higher voltage, current can be lower, thus 'copper losses' and winding temperatures are lower.

Induction machines are available up to about 40 MW with supply voltages up to 14 kV (Converteam), though usually onboard ships maximum voltage is 6.6 kV with maximum power 25 MW. AC synchronous machines are in particular suitable for the higher powers (>8 MW) and are available up to 100 MW (Converteam), maximum found on a ship is approximately 50 MW. DC machines have limited power and are available up to about 5 MW, with supply voltages up to 1000 V. DC machines have limited power, because the commutator brushes have to carry the full motor current. The motor commutation voltage and the current density of the brushes set these power limitations to DC machines.

High temperature superconducting machines are currently undergoing great development. American Superconductor and Northrup Grumman created and demonstrated a 36.5 MW (120 rpm) ceramic superconductor ship propulsion motor. With respect to permanent magnet machines, the largest known is the 20 MW (180 rpm) Permanent Magnet Propulsion Motor.

3.5.2 Dimensions

The GES relations are used as a starting point to estimate electrical machine dimensions. In GES, relations are available for conventional AC machines, DC machines and PM machines. These relations are listed below:

$$\left.\begin{aligned} T_{DC,AC} &= 55 \cdot 10^3 \cdot {D_i}^2 \cdot l_i \\ T_{PM} &= 50 \cdot 10^3 \cdot {D_i}^2 \cdot l_i \\ D_{e,DC} &= D_i + 0.5 \\ l_{e,DC} &= l_i + 1 \\ D_{e,AC} &= D_i + 0.17 + \frac{\pi D_i}{4p} \\ l_{e,AC} &= l_i + 0.3 + \frac{\pi D_i}{2p} \\ D_{e,PM} &= D_i + 0.006 \\ l_{e,PM} &= l_i + 0.12 \end{aligned}\right\} \text{Relations in GES} \tag{3.42}$$

The internal dimensions (index i), in other words the rotor dimensions, are determined by the torque T with a certain ratio between length and diameter. In GES this ratio is assumed 1. The external dimensions (index e) of DC- and PM-machines are obtained by adding a fixed value to the rotor dimensions, and for AC-machines the added value depends on the number of pole pairs p. The volume of the machine can now be calculated by multiplying the external diameters, and a certain cooling volume factor (CVF) is used to correct the volume for the extra cooling volume.

From analysis on this relations with the data, it was found that the GES relations are not representative for the electrical machines in the database. This can also be seen later on in figures 3.25-3.29. So, another approach is required.

Estimating dimensions of an electrical machine is very difficult. Of course there are physical principles and limitations that dictate some dimensions, but there are quite some degrees of freedom for the designer. Single or double wounded windings, single or double armature, transverse, radial or axial flux; all these design details influence the final dimensions. Besides that, developments go on and material characteristics are improved. This makes it difficult to compare the machines from the database with each other and with the theoretic background. Theoretic background is based on Miller (1989), chapter 2.

Two main parameters of an electrical machine in determining the dimensions are the rotational speed n and power P, to be able to calculate the torque T.

$$P = T \cdot 2\pi n \tag{3.43}$$

A torque is given by a certain force times the lever arm. The force is the electro-magnetic force EMF at the rotor surface A_{rotor}. The rotor radius r_{rotor} is the lever arm.

$$T = EMF \cdot r_{rotor} \tag{3.44}$$

$$A_{rotor} = 2\pi \cdot r_{rotor} \cdot l_{rotor} \tag{3.45}$$

EMF causes an average shear stress σ in N/m^2 at the rotor surface, the so-called airgap shear stress.

$$EMF = \sigma \cdot A_{rotor} \tag{3.46}$$

The value of this shear stress is determined by the magnetic and electric loading of the rotor.

$$\sigma = B \cdot A \tag{3.47}$$

The magnetic loading of the rotor is in terms of flux density B in T or Wb/m^2. The electric loading of the rotor, A in A/m, is defined as the current density in the windings in the orthogonal direction. Both B and A have certain limits. Flux density B has a maximum value, depending on the saturation value of the core material. As an example, for ferrite this is around 0.2 (T) and for specialized grain-oriented silicon steel around 1.7 (T), Lee (n.d.). Permanent magnets can have flux densities around 10 (T). The electric loading A also has a limitation, which is dictated by thermal factors. The better the cooling, the higher the electric loading of the windings can be.

So, the maximum value of the average airgap shear stress, thus torque, is very much determined by the technique of the electrical machine. At this point, one could expect the torque capability or 'specific output' to be determined by the surface area of the rotor. But it is primarily the rotor volume that determines torque capability. As the diameter is increased, both the current and the flux increase (with equal electric and magnetic loadings). Hence the diameter appears squared in any expression for specific output. If, on the other hand, the length is increased, only flux increases. Therefore length is linearly proportional to the specific output. Together this gives rise to a rotorvolume (V_{rotor}) dependent torque capability of the rotor *Torque per Rotor Volume*, TRV, in (Nm/m^3):

$$TRV = \frac{T}{V_{rotor}} = \frac{T}{\pi \cdot r_{rotor}^2 \cdot l_{rotor}} = 2 \cdot \sigma \tag{3.48}$$

In literature, values are found for σ or TRV. Table 3.8 gives some typical values for TRV and σ. Values for TRV of DC- and conventional AC-machines are from the AES study, see equation 3.42 and correct with $\pi/4$. Hodge & Mattick (2001) gives a description of the advanced AC machines, the AIM family and mentions values for TRV. In Newell *et al.* (1998) the Permanent Magnet Propulsion Motor (PMPM) is described and an airgap shear stress value is mentioned. For HTS machines, no value was found in literature but a value is assumed. Because TRV values increase for better cooled machines, a higher value is assumed for HTS machines. For bigger and higher loaded machines TRV values also increase, because for bigger machines the rotor cooling can be more efficient, which gives higher TRV values. So, actually TRV values should be dependent on rated power of the machine, but this effect is not taken into account.

With a certain value of TRV, which depends on the technique, the rotor volume can be calculated with equation 3.48.

$$V_{rotor} = \frac{T}{TRV}$$

To estimate the stator volume, a fixed value or pole pair dependent value can be added to the rotor dimensions as is done in the AES study, but also a so-called 'split ratio' (s) can be used as explained in Miller (1989):

$$s = \sqrt{\frac{V_{rotor}}{V_{stator}}} \tag{3.49}$$

Machine type	σ (kN/m^2)	TRV (kN/m^3)
DC machines	35	70
Conventional AC machines	35	70
Advanced AC machines	$100-120$	$200-240$
HTS machines	125	250
PM machines	100	200

Table 3.8: Typical TRV and σ values for common motor types

(sources: van Dijk et al. *(1998), Hodge & Mattick (2001), Newell* et al. *(1998))*

The 'split ratio' is a rotor/stator diameter ratio. In Miller (1989), a typical value for an AC machine is mentioned in the range of 0.55-0.65, but from the data it seems that a lower value suits better. For DC machines no value is given, but it is assumed to be lower because stator pack of a DC machine is normally bigger and the commutator consumes significant space. For advanced, HTS and PM machines somewhat higher values are assumed, because the compactness of these machines. The data from the database does not contain information about all rotor diameters, so a 'split ratio' can not be calculated from the data. Assumptions are made for the 'split ratio' of the different machine types, based on feeling and fitting with the data, see table 3.9.

Machine type	Assumed mean s value
DC machines	0.4
Conventional AC machines	0.45
Advanced AC machines	0.5
HTS machines	0.5
PM machines	0.5

Table 3.9: Assumed mean 'split ratio' values

$$V_{stator} = \frac{V_{rotor}}{s^2} = \frac{T}{TRV \cdot s^2} \tag{3.50}$$

Length and width of the electrical machine are dependent on the ratio between length and diameter of the stator L/D. In the AES study a fixed value of 1 for all machine types is assumed. In reality this is not a fixed value. The engineer has some degree of freedom in chosing L/D of the rotor with which the L/D of the stator is directly related. Typical mean values of L/D for different machine types are given by table 3.10, based on the machines from the database.

Machine type	Typical mean L/D value
DC machines	1.2
Conventional AC machines	1.6
Advanced AC machines	1.2
HTS machines	1
PM machines	1

Table 3.10: Typical L/D values based on the electrical machines from the database

As said before, the designer has a degree of freedom in choosing L/D. So, the values can as well

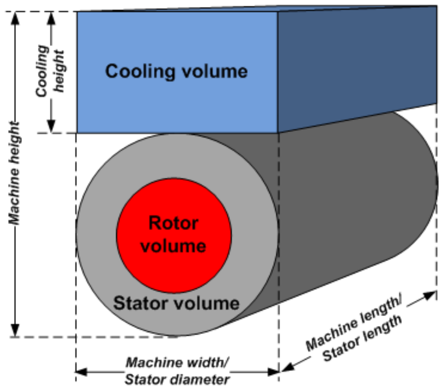


Figure 3.23: Schematic overview of different volumes and dimensions in an electrical machine

be 0.5 or 2. The trend for slower (lower rpm) machines is towards lower values of L/D, and for faster machines higher. The reason for this is that machine efficiency is closely related to circumferential speed. This has to do with voltage drop over the airgap. When circumferential speed of the rotor is higher (bigger diameter), the electric field is higher, and the voltage drop forms a smaller part of the electric field, which means lower losses. So in principal, the designer will go for as large as possible diameter to get the highest efficiency. But fast machines can not go for too large diameters for mechanical reasons.

With an L/D value the diameter and length of the stator can be calculated from the stator volume. The width and length of the machine are equal to the diameter and length of the stator, D_{stator} resp. L_{stator}, see figure 3.23.

$$D_{stator} = \left(\frac{V_{stator}}{\frac{\pi}{4} \cdot L/D}\right)^{\frac{1}{3}} = \left(\frac{T}{TRV \cdot s^2 \cdot \frac{\pi}{4} \cdot L/D}\right)^{\frac{1}{3}} \tag{3.51}$$

$$\boxed{\text{Width} = D_{stator} = \left(\frac{T}{TRV \cdot s^2 \cdot \frac{\pi}{4} \cdot L/D}\right)^{\frac{1}{3}}} \tag{3.52}$$

$$L_{stator} = D_{stator} \cdot L/D \tag{3.53}$$

$$\boxed{\text{Length} = L_{stator} = L/D \cdot \left(\frac{T}{TRV \cdot s^2 \cdot \frac{\pi}{4} \cdot L/D}\right)^{\frac{1}{3}}} \tag{3.54}$$

The height of the machine can not directly be obtained from the diameter, because there has to be accounted for the cooling equipment volume $V_{cooling}$, which is normally on top of the machine, see figure 3.23. In the AES study the concept of a certain Cooling Volume Factor CVF is mentioned. The definition is not well described, but with the known data in the database, the concept is implemented as the ratio between the total installation volume $V_{install}$ and the stator volume:

$$CVF = \frac{V_{install}}{V_{stator}} = \frac{\text{Length} \cdot \text{Width} \cdot \text{Height}}{\frac{\pi}{4} \cdot \text{Width}^2 \cdot \text{Length}} = \frac{\text{Height}}{\frac{\pi}{4} \cdot \text{Width}} \tag{3.55}$$

The type of cooling that is applied determines the value of CVF. For natural air cooling no extra volume is accounted, but adding an air-to-air cooling system can double the total volume of the motor, according to van Dijk *et al.* (1998). CVF values are determined per machine type, based on the data and listed in table 3.11. Almost all types have internal-air-to-water cooling. The value that is mentioned in van Dijk *et al.* (1998) for this type of cooling is 1.5, but this value is too low when compared to the data. Other CVF values in van Dijk *et al.* (1998) are: forced-air cooling 1.25, forced-air-to-air cooling 2 and water cooling 1.1.

Machine type	Typical mean CVF value
DC machines	2
Conventional AC machines	2.5
Advanced AC machines	1.7
HTS machines	1.5
PM machines	1.5

Table 3.11: Typical Cooling Volume Factors (CVF) based on the electrical machines from the database

Note: *The values in table 3.8-3.11 for TRV, s, L/D and CVF are based on very scarce data. More data is needed for more reliable numbers.*

The height of the electrical machine can now be extracted from equation 3.55. Combining equations 3.55, 3.50, 3.52 and 3.54 leads to the following derivation for machine height.

$$\begin{aligned}
\text{Height} &= \frac{V_{install}}{\text{Length} \cdot \text{Width}} \\
&= \frac{V_{stator} \cdot CVF}{\text{Length} \cdot \text{Width}} \\
&= \frac{\frac{V_{rotor}}{s^2} \cdot CVF}{\text{Length} \cdot \text{Width}} \\
&= \frac{\frac{T}{TRV \cdot s^2} \cdot CVF}{L/D \cdot \left(\frac{T}{TRV \cdot s^2 \cdot \frac{\pi}{4} \cdot L/D}\right)^{\frac{2}{3}}} \\
&= \frac{\left(\frac{T}{TRV \cdot s^2}\right)^{\frac{1}{3}} \cdot CVF}{(L/D)^{\frac{1}{3}} \cdot \left(\frac{4}{\pi}\right)^{\frac{2}{3}}}
\end{aligned}$$

$$\boxed{\text{Height} = \left(\frac{T}{TRV \cdot s^2}\right)^{\frac{1}{3}} \cdot CVF \cdot (L/D)^{-\frac{1}{3}} \cdot \left(\frac{\pi}{4}\right)^{\frac{2}{3}}} \tag{3.56}$$

Results of the analysis of dimensions and volumes, with the assumptions from table 3.8-3.11 are presented in figure 3.25-3.29. It should be noted that the deviations in figure 3.27 and 3.28 are mainly caused by the fact that mean values of L/D are used, and in reality this varies much between machines. The torque in the figures is put on a logarithmic scale because of the wide spread. The legend for these figures is presented separately in figure 3.24. The dotted lines represent the relations from GES.

The standard deviation of the volume of electrical machines in the database from the model (as presented in figure 3.25 and 3.26) is:

- DC machines: $\sigma = 2.1$ on V_{stator} and $\sigma = 3.9$ on $V_{install}$, with max. deviations of 41% resp. 16%

- Conventional AC machines: $\sigma = 0.79$ on V_{stator} and $\sigma = 2.7$ on $V_{install}$, with max. deviations of 53% resp. 67%
- Advanced AC machines: $\sigma = 2.6$ on V_{stator} and $\sigma = 4.9$ on $V_{install}$, with max. deviations of 21% resp. 23%
- HTS machines: $\sigma = 3.0$ on V_{stator} and $\sigma = 4.9$ on $V_{install}$, with max. deviations of 48% resp. 45%
- PM machines: $\sigma = 4.2$ on V_{stator} and $\sigma = 5.1$ on $V_{install}$, with max. deviations of 75% resp. 86%

Deviations of the data from the models is in some cases very large. This once more shows how difficult it is to catch all electrical machines in a simple sizing model, since there are so many different types and different designer choices. Because of the scarce data, results might also be unreliable. But this gives an outline on how to approach on these problems.

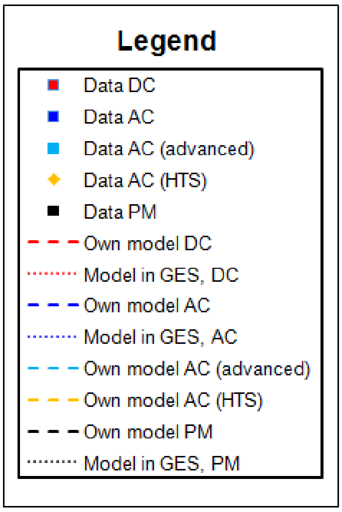


Figure 3.24: Legend for figures 3.25-3.31

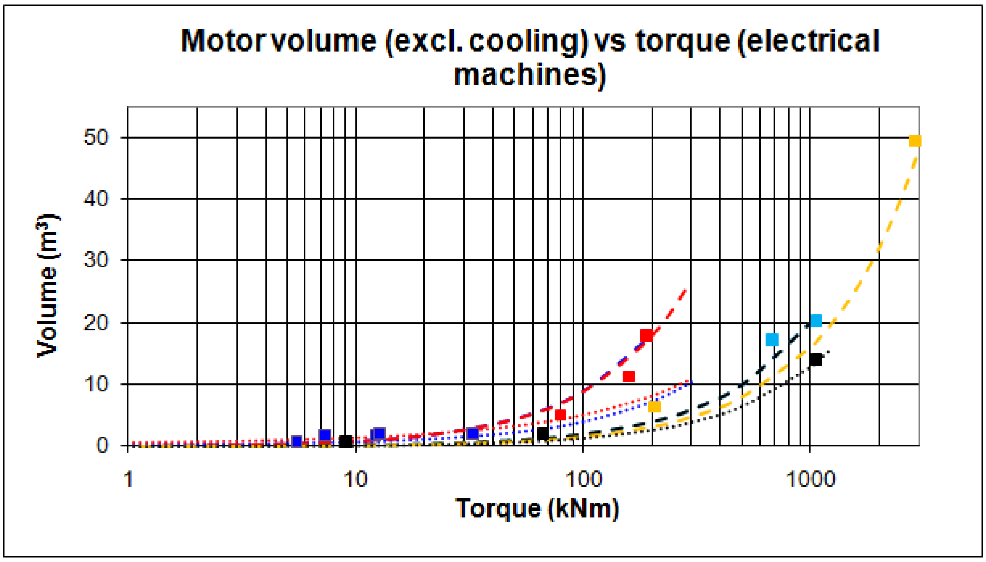


Figure 3.25: Stator volume (m^3) vs torque (kNm) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

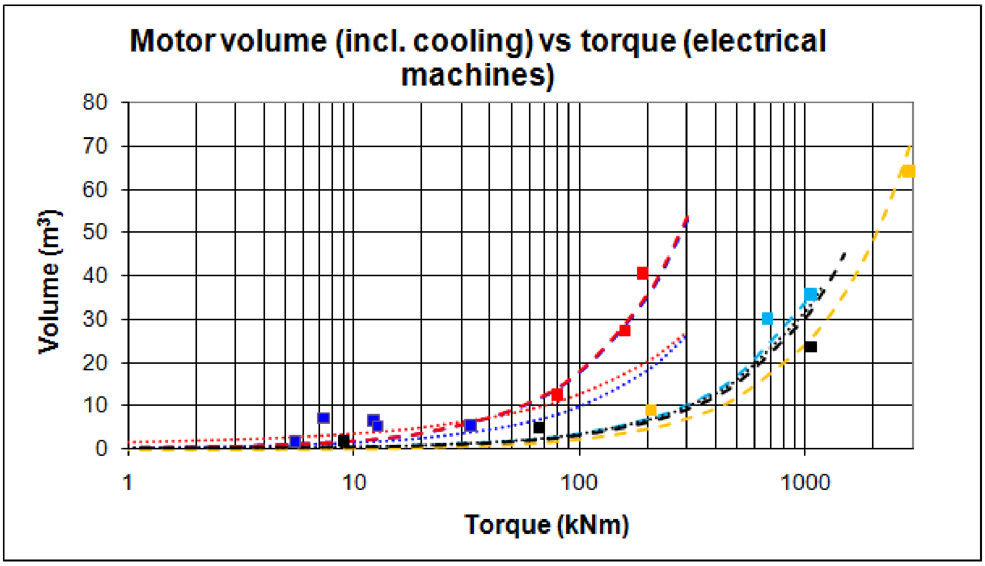


Figure 3.26: Installation volume incl. cooling (m^3) vs torque (kNm) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

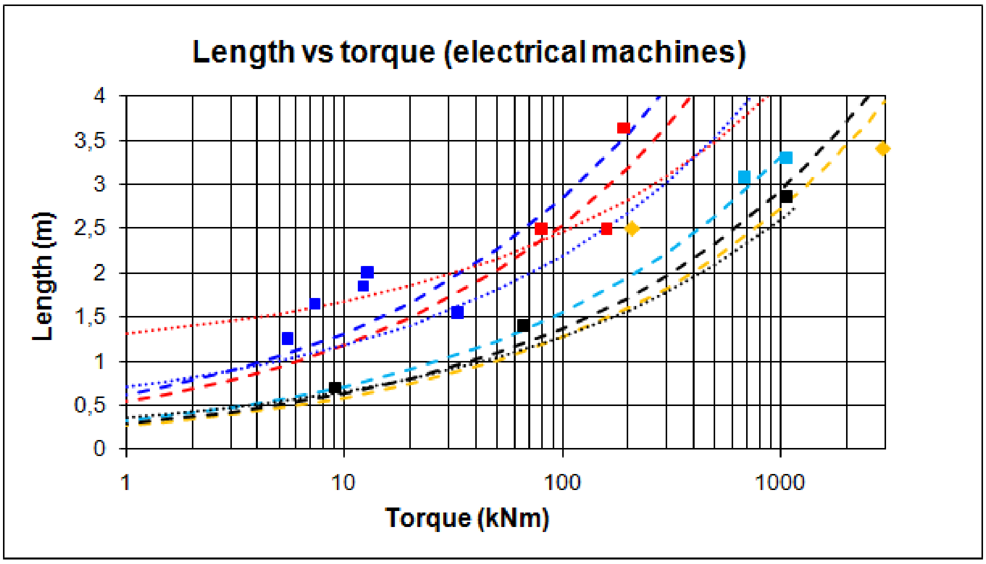


Figure 3.27: Length (m) vs torque (kNm) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

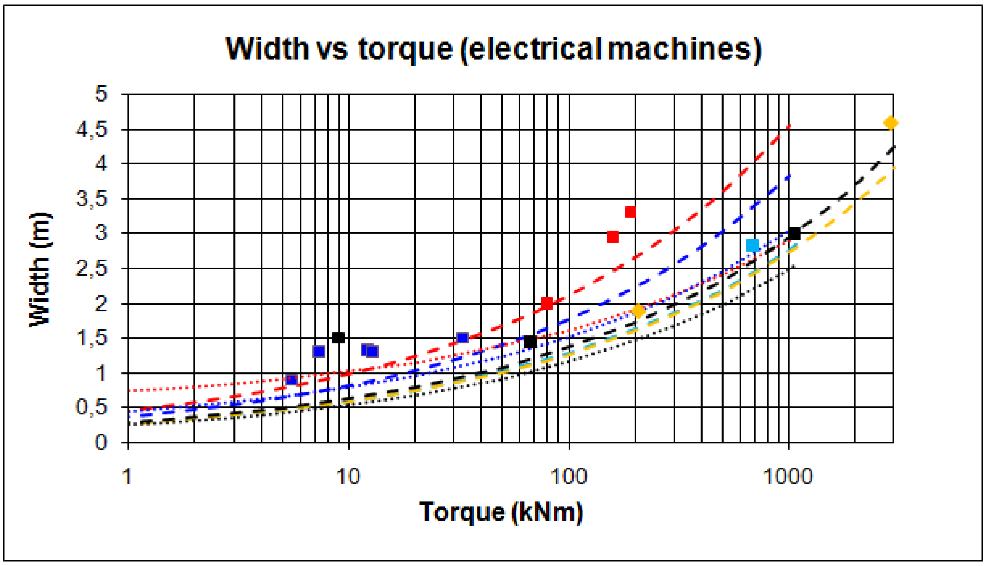


Figure 3.28: Width (m) vs torque (kNm) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

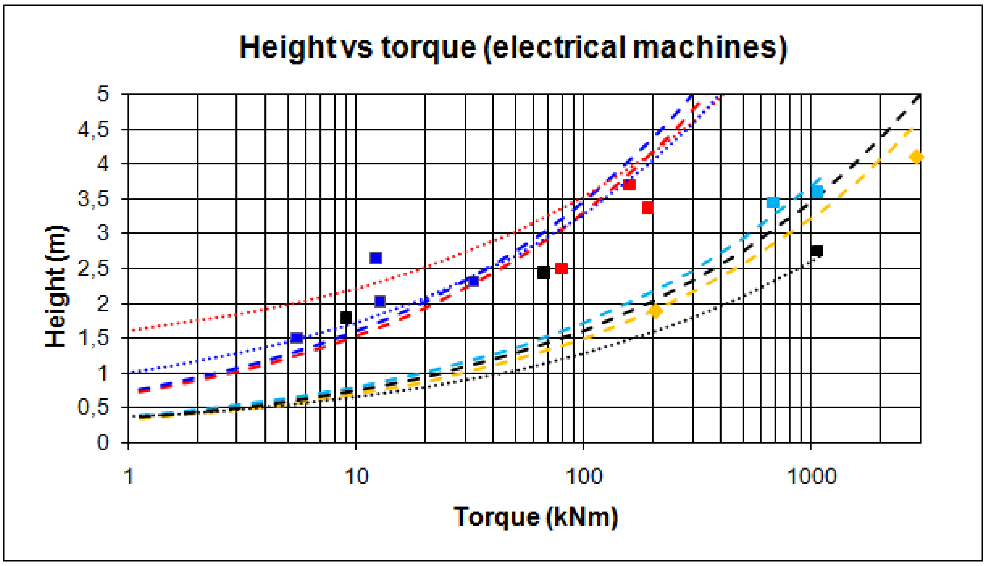


Figure 3.29: Installation height incl. cooling (m) vs torque (kNm) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

3.5.3 Weight

In the previous subsection is described how the volume of an electrical machine can be estimated based on torque requirement. Volume and weight are closely related, so similar relations between weight and torque are expected. In the AES study, van Dijk *et al.* (1998), weight is related to the squared diameter and the length of the machine, with other words a volume. These relations are available for conventional AC and DC machines and PM machines and are put in figure 3.30, whereby the volume for the GES model is calculated with formulas 3.42 and $CVF = 2.5$ and $p = 4$. Tested with the known data from the small database, this relation from the AES study gives too high weights for conventional AC and DC machines, and too low weights for PM machines. So there is need for a better relation.

The data, together with linear trendlines (forced through zero), are plotted in figure 3.30. From this figure can be concluded that the PM machines and advanced type AC machines have the largest inclination, with other words the highest mean density. One could expect very high weights of these machine types because of their high volume specific weight, but their compactness (see figure 3.25 and 3.26) mitigates this effect. The DC machines have a somewhat lower specific weight, but still will probably have higher weight because of larger volumes. The determination of a trend for the conventional AC machines is very difficult because of their wide spread. The datapoints show a trend of lower weight for higher volume, but this against expectations. Still a linear trend forced through zero is used to describe the weight vs volume, which gives an expected lower specific weight than DC machines, with a value comparable to the specific weight of HTS machines. The following values are found, with weight in (kg) and volume in (m^3):

DC machines:

$$\boxed{\text{Weight} \approx 1500 \cdot V_{install}} \tag{3.57}$$

Conventional AC + HTS machines:

$$\boxed{\text{Weight} \approx 1200 \cdot V_{install}} \tag{3.58}$$

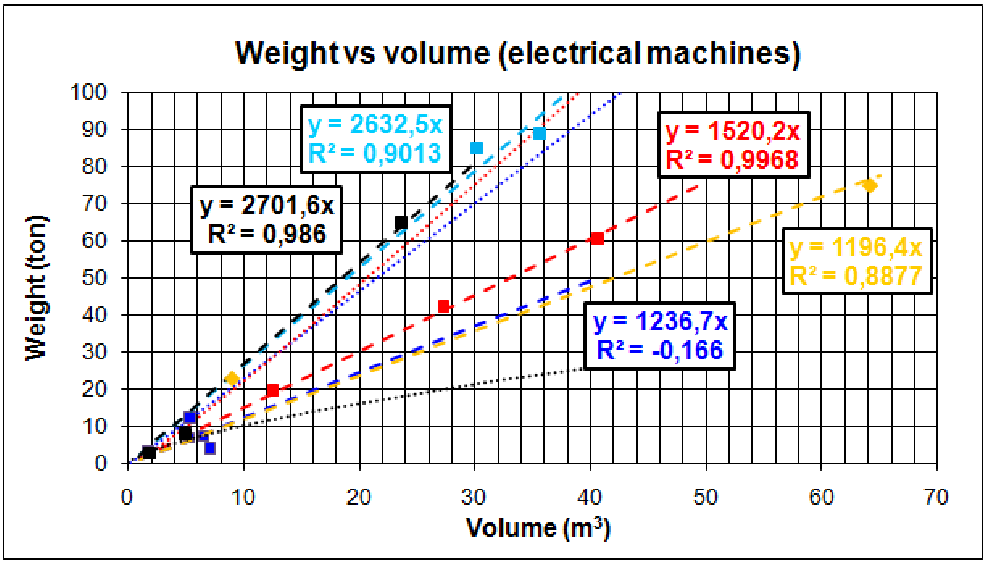


Figure 3.30: Installation weight incl. cooling (ton) vs installation volume incl. cooling (m^3) of database electrical machines and according to GES models and according to own models (legend: see figure 3.24)

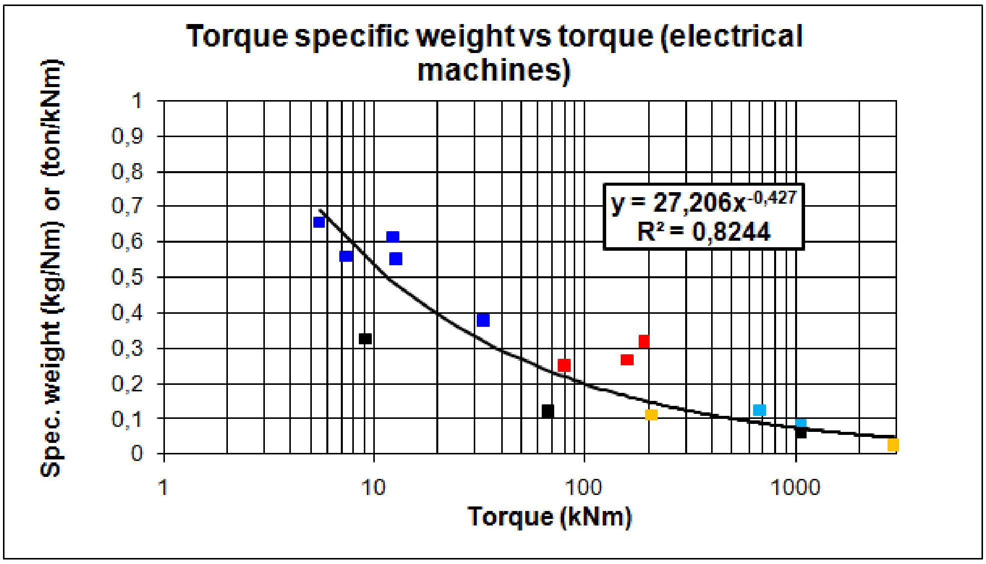


Figure 3.31: Torque specific weight (kg/Nm) or (ton/kNm) vs torque (kNm) of database electrical machines (legend: see figure 3.24)

Advanced AC + PM machines:

$$\boxed{\text{Weight} \approx 2650 \cdot V_{install}} \tag{3.59}$$

Note: *The values for specific weight are based on very limited data. More data is needed for more reliable numbers*

Another way to express weight, is as a function of the output. This is how it is done with all the other components. In case of electrical machines it is best to use torque as the output. The results are shown in figure 3.31, where weight is presented as torque specific weight. The differences are a little more difficult to recognize, but this figure shows the results as expected. For all machine types there is the scaling effect (lower specific weight for higher torque output). For the PM and HTS machines a significantly lower weight per unit torque is recognized, compared to the conventional types. Between the DC and AC machines it seems that DC machines have a slightly higher specific weight. To generate a general relation for weight of an electrical machine regardless of the type of machine, a trend is created through the datapoints in figure 3.31. This trendline gives the following relation between torque specific weight in (kg/Nm) or (ton/kNm) and torque in (Nm)! :

General trend, all machine types:

$$\boxed{\text{Weight} = (27.2 \cdot T^{-0.43}) \cdot T} \tag{3.60}$$

3.5.4 Operating speeds

Electrical machines are available in a large range of operating speeds. The big advantage of electric motors is that they can deliver maximum torque at all speeds, so an electric motor can be used within its entire speed range, different from gasturbine and diesel engine. The operating speed determines the dimensions and weight of the motor. Normally, the available space onboard determines the design speed of the motor. For a generator normally fixed speeds are used, because the rotational speed (n) determines the frequency (f) of the generated current, depending on the number of pole pairs (p). See formula 3.61.

$$f = \frac{p}{2} \cdot n \tag{3.61}$$

With n in s^{-1}, f in Hz. The same formula can be used for calculating the speed of a synchronous AC motor, with f the frequency of the supply current and p the number of pole pairs in the motor. When a generator is coupled to a high speed driving engine (like a gasturbine), it should als be capable of running very high speeds. Nowadays this is possible, and generators are available for speeds even higher than 10,000 rpm.

It is hard to distinguish different speed ranges for the different machine types. All machine types are at least available for usual purpose low speeds (100-300 rpm) and high speeds (2000-5000 rpm). And some types are even available at lower or higher speeds for special purposes, like the induction machine and AC synchronous.

3.5.5 Efficiency

Electric motors normally have high efficiencies (above 90%), also in part loads. Nominal efficiency of higher power machines tends to be higher than of lower power machines. In the AES study, van Dijk *et al.* (1998), a general relation between nominal power and nominal efficiency is mentioned. In Bosklopper (2009), this relation was refined to more state-of-the-art standards. This relation is given by formula 3.62 and is based on AC asynchronous motors. Because there is no data available in the database on efficiencies of electrical machines, this formula is adopted. Note that nominal power is given in (MW) in this formula.

$$\boxed{\eta_{nom} = \frac{1}{1.026 + \frac{0.003162}{\sqrt{P_{nom}}}}} \tag{3.62}$$

Nominal efficiency depends a little bit on technique of the machine. DC motors are normally somewhat more efficient than AC motors. Especially the asynchronous machines have a somewhat lower efficiency. HTS machines have higher efficiencies, especially at part load. But this has to be corrected for the extra cooling power that HTS machines require. The differences are normally small and differ from machine to machine, so for all machines the same model is used for nominal efficiency shown in figure 3.32.

Part load efficiency is considerably higher for HTS machines than for conventional machines. Efficiency of a permanent magnet machine is comparable to conventional machines. The part load losses of the advanced induction machines seem to be worse compared to other machine types, figure 3.33. Losses in an electrical machine can be divided into iron losses ($P_{loss,fe}$) and copper losses ($P_{loss,cu}$). Iron losses are more or less constant at all loads, but copper losses are quadratically proportional to the current. Part load efficiency of an electric motor is not dependent of speed. At nominal power the iron losses and copper losses are approximately equal. The following formulas are used to determine the losses in a conventional electrical machine, according to van Dijk *et al.* (1998).

$$P_{loss,nom} = (1 - \eta_{nom}) \cdot P_{nom} \tag{3.63}$$

$$P_{loss,fe} = \frac{1}{2} \cdot P_{loss,nom} \tag{3.64}$$

$$P_{loss,cu} = c \cdot I^2 \tag{3.65}$$

In which I is current in (A) and c is a constant which can be determined at nominal load. Current depends on power and voltage. Under the assumption that voltage is constant during static

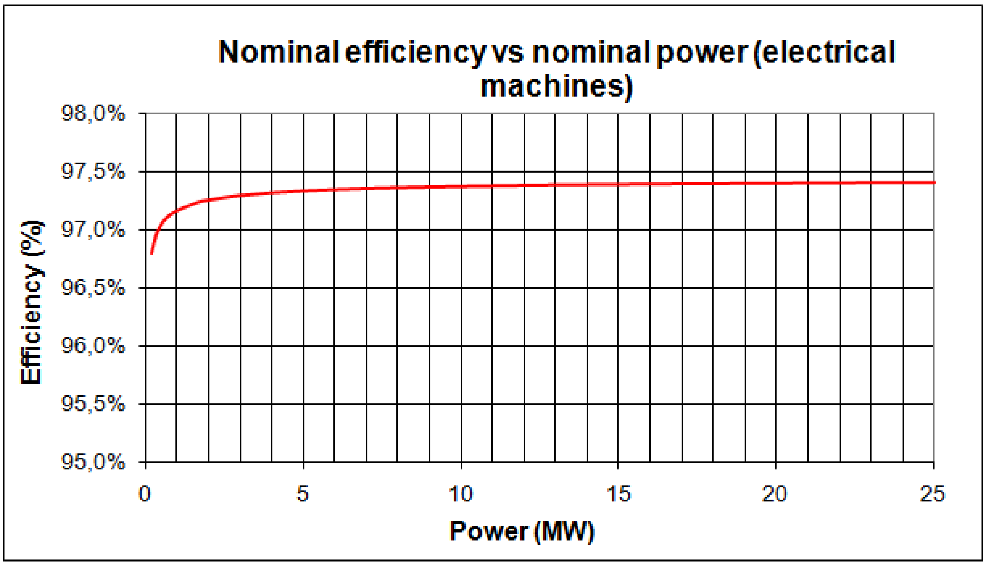


Figure 3.32: Estimation of nominal efficiency of electrical machines based on nominal power

conditions, figure 3.33 is made and shows a typical efficiency plot of a conventional electrical machine. A practical formula to determine partload efficiency of conventional electrical machine is given by equation 3.66, with $a = 0.016$, $b = -0.01$ and $c = 0.3$. The result is also plotted in figure 3.33. This model can be used down to 2% partload, where efficiency is approximately 35%, for lower loads this model should not be used because it gives negative values.

$$\eta = \left[\frac{P^* - (a + b \cdot P^{*c}}{P^*} \right] \cdot \eta_{nom} \tag{3.66}$$

Figure 3.33 also shows a typical efficiency plot of an AIM machine, PMPM machine and a HTS machine. The data of the AIM and PM efficiency come from Hodge & Mattick (2001). The losses of a HTS machine are modeled as described in Bosklopper (2009), see formulas 3.63 and 3.67.

$$\frac{P_{loss}}{P_{loss,nom}} = 0.85 \cdot \frac{P}{P_{nom}} + 0.15 \tag{3.67}$$

This means that 15% of the nominal loss is constant and 85% is dependent on the powerload. This gives the HTS machines a significant benefit at part load as seen in figure 3.33.

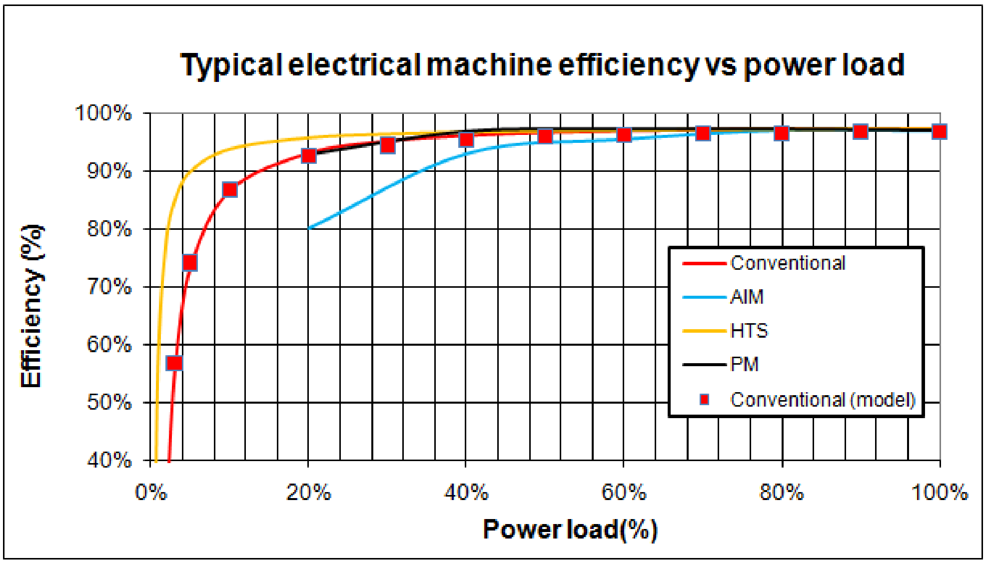


Figure 3.33: Typical efficiency curves of different types of electrical machines

3.5.6 Signatures

Underwater noise

Electrical machines emit noise, like all other machines. For motors, the way of controlling is very determining in the noise production of the machine. In most cases the voltage of an electrical motor is supplied by a power converter, which causes additional noise next to the noise by the machine itself. A brief description of noise sources is given, based on de Jong & Hoeijmakers (2002).

An electrical machine itself generates noise because the magnetic field along the circumference of the air gap is not perfectly sinusoidal in space and time. The stator is responsible for one magnetic field in the airgap and the rotor is responsible for a second magnetic field in the same airgap. These two waves interact with each other. This results in an actual magnetic flux which is a combination of all the space and time harmonics of both the stator and rotor fields. The noise being produced is a function of the square of this new magnetic flux wave and therefor changes again, enlarging the noise spectrum even more. Not all entries in this spectrum are radiated at the same efficiency, therefor a few frequencies are then heard. The stator structure, if not stiff enough, emits the vibrations. Further, there is the noise caused by bearings, rotor unbalance and aerodynamics. In a synchronous machine, the magnetic field around the rotor is not varying in time - because the rotor is excited with DC - but only varying in space - because of rotor and stator slots. In an asynchronous machine also time varying harmonics are introduced, but it has the benefit of not having slip rings. The slip rings also produce some noise.

As said before, the controller of the machine has a very large part in the noise. Higher harmonics are introduced and cause oscillations on the voltage supply, which causes torque ripple. The noise sources of converters is described in section *Electrical auxiliaries*, but an overview of the effect on the machines is briefly described here.

A DC motor can have a relative simple controller, a so-called chopper. In comparison with other converters, only small pulsations on the torque are caused. This has advantages for noise. No pulsations, no noise. Experience learns that DC motors can be very quiet with the right design.

AC synchronous motors are frequency controlled on the stator; this controls the speed of the motor. The DC voltage on the rotor determines the delivered torque, this can be controlled by a chopper. The AC synchronous machine is also very quiet. Still, for extra quiet operations often DC motors are used.

AC asynchronous motors are frequency controlled by pulsewidth or pulsefrequency modulation. These converters introduce higher harmonics in the output voltage. These harmonics cause stress waves and vibrations. All vibrations are transferred to the ships hull and radiated as underwater noise, contributing to the signature of the ship. Another way of controlling this motor is with resistances on the cage, but still this is not very quiet in comparison to the other types. The produced noise is dependent on the rotor slip.

Producers of HTS machines state that they have lower sound emissions than conventional motors and generators. Because there is little or no iron in the magnetic path of the motor or generator, there is very little distortion in the power supply and little noise feedback from the motor. The lower weight of the machine rotor also reduces noise, Unknown (2002).

Installing an electric motor in a pod *can* have some benefits for underwater noise of the ship, because of better streamline under the ship, there is less cavitation. Cavitation inception speed can go up, Trouwborst (1998). Though, it requires measures to take this benefit. In RNLN's LPD-2, the electric motors are mounted against the walls of the pods for better cooling. This results in more underwater noise instead of less. But with proper design, pods have the potential to produce less underwater noise.

Infrared

Infrared signature is not an issue for electrical machines. Of course an electrical machine produces heat, but it is cooled with air or water, and the contribution to the infrared signature of the ship is nihil.

Electro-magnetic

Electro-magnetic signature is an increasing point of concern when more electrical machines are onboard. The magnetic fields that are induced in the electrical machines and in conducting cables are damped a little bit by the steel hull of the ship, but are outside the ship very well detectable and contribute to the signature of the ship. The magnitude and frequency of the magnetic field depends on the current in the electrical machines. Frequency of the alternating current in electrical machines onboard are in the range 0-100 Hz. The presence of converters introduces higher harmonics. Harmonics in the magnetic field are also caused by inherent motor 'imperfections': stator slots, rotor slots, rotor eccentricity and iron saturation. Higher frequency magnetic fields can very well be damped by enclosing the electrical machine with conducting material, creating the so-called Faraday cage. The magnetic signature of a ship consist of the static and dynamic magnetic field. The static field can very well be reduced by means of demagnetization coils (DEMAG). When the dynamic field can be predicted, active measures can be taken. Research is done on prediction models for magnetic signature of ships, Dill & Evenblij (2003). The dynamic field of a frequency controlled AC machine is more complex than of a voltage controlled DC machine.

3.5.7 Shock resistance

The greatest issue with respect to shock resistance of electrical machines is the airgap freespace. The space between rotor and stator is called the airgap. In case of a shock, the rotor will move. The bigger the airgap, the better the shock resistance. The induction machines (AC asynchronous) have a serious drawback compared to other types for this reason. The induction machine requires a comparatively small airgap with low tolerance figures for concentricity.

As with the other components, the electrical machine can be placed on flexible mountings to improve the shock resistance but also the underwater noise signature. Electrical motor mountings can not be too flexible for reasons of shaft alignment. This is not the case for generators, because they are on a stiff baseplate together with the driving engine. Electric motors can also be placed in so-called pods under the ship, see section 3.11.

3.5.8 Maintainability

Generally spoken, an electrical machine does not need much maintenance. Electrical machines are designed for service-life of the ship, which means that they normally don't need overhauling. Regular maintenance differs a little between the different electrical machine types. All machines need regular checking of bearings, replacement of cooling filters, in case of watercooling checking and cleaning of the cooling tubes, checking vibration levels, cleaning of the windings, check and change bearing lubricating grease/oil, checking insulation resistance. Maintenance of DC machines and non-brushless synchronous machines involves regularly replacing the brushes and springs which carry the electric current, as well as cleaning or replacing the commutator. The interval for this is approximately once every year. The maintenance on the brushes is seen as a significant disadvantage of the brushed machines.

3.5.9 Reliability

It is difficult to put reliable numbers to machine reliability. In the AES study an indication of MTBF and MTTR is given for DC-, AC synchronous, asynchronous and PM machines. The mean time between failure in electrical machines is primarily determined by the auxiliary systems, because the chance of failure of the insulation or the windings is nihil. With brushed machines, the brush or slip rings introduce some unreliability, but generally spoken electrical machines are very reliable. The AES study mentions some MTBF numbers, 17 years for DC machines, 23 years for AC synchronous machines and 28 years for AC asynchronous machines and permanent magnet (PM) machines. For all machine types a MTTR of 120 hours was assumed. Nowadays there are also the HTS machines, manufacturers state that these have higher reliability because HTS machines have better power quality because of harmonic-free voltages. There are no MTBF or MTTR numbers known to the author.

3.5.10 Initial purchase costs

The initial purchase costs of an electrical machine are estimated with the Cost Estimating Relationship (CER) as developed by the DMO. This cost relationship only holds for electrical motors. The costs of generators are estimated with either the CER of diesel-generators or the CER of gasturbine-generators.

With DMO's CER the purchase costs are estimated as a function of the power. Another option could be as a function of weight, because then the influence of motorspeed is ruled out. Slower motors are much bigger and heavier, so are expected to be more expensive. In practice it seems that the CER as a function of power is more accurate. The AES study (GES) also mentions a CER for electromotors which is a combination of both; the specific price of an electric motor in €/ton is defined as a function of power:

- $\left(8805 + 1174 \cdot \ln(P_{[kW]})\right)$ €/ton

This relationship is converted from Dutch guilders to €and increased for inflation at a rate of 2% per year from 1998 on. To compare the relation from the AES study with either the power relation or the weight relation of the DMO, some assumptions have to be made. If a certain motor speed is assumed, the torque can be calculated as a function of power. With the torque the motor weight can be estimated, according to equation 3.60, although this is a very rough estimation. With the weight the specific price in €/kW can be calculated for the CER in GES. This is done for two motor speeds, 200 and 2000 rpm, and presented in figure 3.34. It can be seen that the difference is very large. The specific price with DMO's CER is much higher than in GES.

Both CER's make no distinction between different motortypes. A more detailed cost model is needed to compare different motor types, for example permanent magnet motor versus conventional technology. Some opinions on different motor types are found in literature, but no

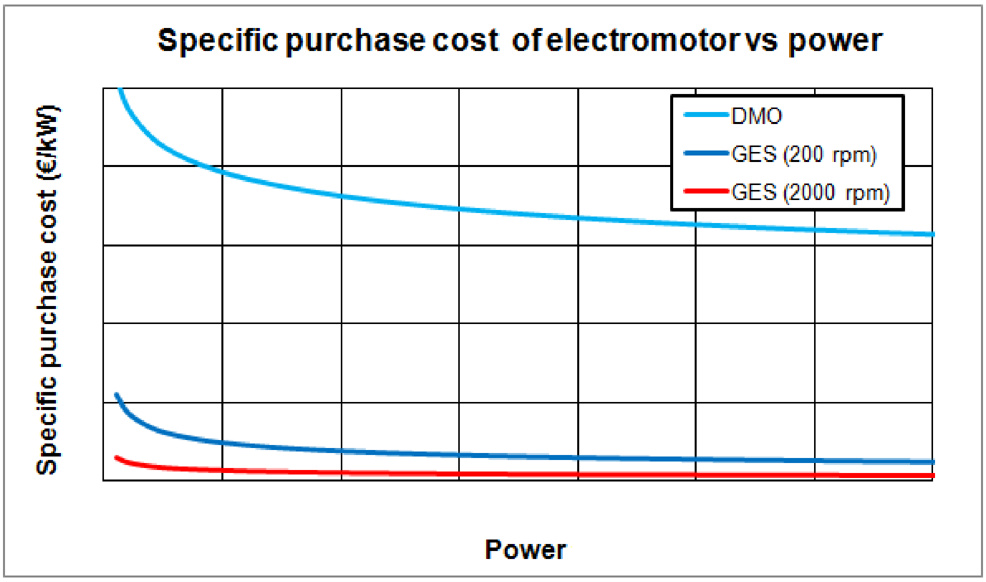


Figure 3.34: Specific purchase costs (€/kW) of electromotors vs rated power, according to different sources. Cell height is 250 €/kW

numbers to prove it. It is said that PM motors are twice as expensive as conventional motors. In a preliminary study to the propulsion configuration of the JSS, it was found that asynchronous motors are the cheapest solution for machines upto 5 MW. About HTS motors it is said in Unknown (2002) that they should be commercially available at prices equivalent to conventional machines. But all these are statements which can't be proved by numbers at this time.

3.6 Gearbox

Gearboxes are used when output speed of the driving machine is higher than the operating speed of the propeller. Gearboxes can also be used to couple multiple movers to one shaft, or to couple one mover to multiple shafts. Sometimes a gearbox only has the function to reverse the direction of rotation in case the ship needs to go astern. Three main types are distinguished within this thesis:

- Single gears (Single Input Single Output, SISO)
- Twin gears (Double Input Single Output, MISO)
- Cross-connect gears (Single/Multiple Input Multiple Output, MIMO)

For single gears, another distinction can be made between vertical and horizontal offset or coaxial gears, see figure 3.35. The multiple input gears are normally horizontally offset. The distance between the in- and outgoing shaft is called the offset distance. Cross-connect gears are also horizontally offset, and are used to couple two gearboxes to each other when, for example, two main shaftlines share one high speed gasturbine, see figure 3.36. In the configuration as seen in figure 3.36 it can be possible (depending on complexity of the cross-connect gear) to drive two shafts by 1 diesel engine, 2 diesel engines, gasturbine, 1 diesel engine + gasturbine or 2 diesel engines + gasturbine.

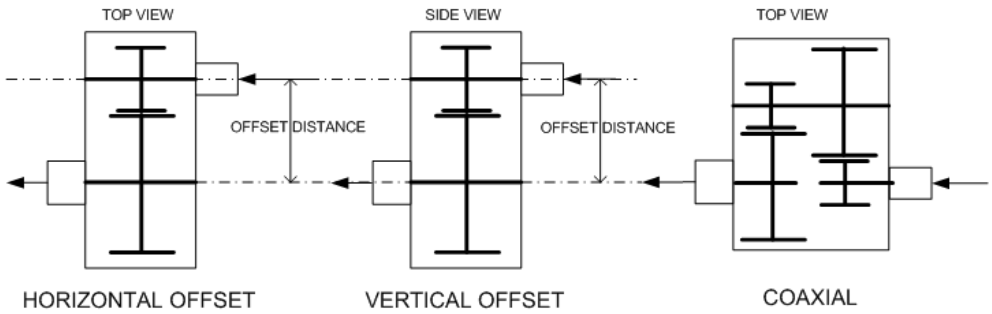


Figure 3.35: Schematic view of horizontally, vertically and coaxial offset gears

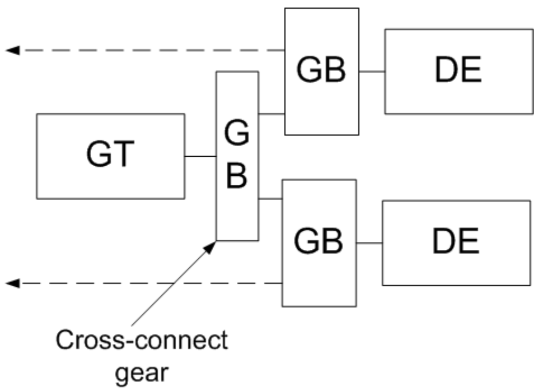


Figure 3.36: Schematic example of a cross-connect

Basically, marine gearboxes consist of meshing teeth on pinions and wheels, which transfer power from a drive shaft (primary) to a driven shaft (secondary) and reduce speed. A certain gear ratio i, equation 3.69, can be obtained by single stage reduction (one set of pinions and wheels)

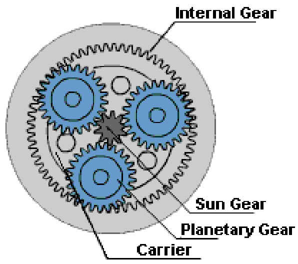


Figure 3.37: Example of planetary or epicyclic gearing

(source: http://www.campingatv.net/2010/07/selecting-an-atv-winch/ (february 2011))

or by multiple stage reduction (two or more sets of pinions and wheels). According to van Dijk *et al.* (1998), maximum gear ratio for single stage reduction is 10. The working principle of a gearbox is based on coupling two wheels by means of interlocking teeth, so both wheels have the same circumferential velocity but different angular velocity because the radius differs. Normally, rotational speed needs to be decreased in ships; this is done by driving a big wheel with a small one in a single stage or in multiple stages. Reducing the speed increases the torque on the output shaft, equation 3.68.

$$M_{shaft} = \eta_{gb} \cdot i \cdot M_{engine} \tag{3.68}$$

$$i = \frac{N_{in}}{N_{out}} = \frac{n_{in}}{n_{out}} = \frac{\omega_{in}}{\omega_{out}} \tag{3.69}$$

Where M in (Nm), N in (rpm), n in (s^{-1}) and ω in (rad/s).

A speed ratio can be achieved by wheels next to each other (parallel axis gears) as normally applied, or with planetary/epicyclic gears where the wheels are inside each other. An example is shown in figure 3.37, the high speed shaft is connected to the sun gear and the low speed shaft to the carrier of the planet gears. Planetary gears have the advantage of being more compact, having lower weight and better shock resistance. Still, this type of gearing is not often used on naval vessels, probably because of the complexity and lower reliability. Another reason could be that this type of gearing is not suitable for higher powers. An example from the field: the German navy had planetary gears on the K130 corvette, but these gearboxes were completely damaged during trial run.

For better and more homogeneous distribution of the forces between the wheels in a gearbox they can have helical teeth. This also introduces an axial force component with axial vibrations, thus noise as a result. When extra quiet operation is required, the wheels can be double-helical, to cancel out the axial forces, see figure 3.38.

Depending on the propulsion configuration (CODAD, CODOG, CODAG, CODLAG etc.) a gearbox needs a number of clutches to be able to connect and disconnect machines and shafts to the gearbox. Three types of clutches are distinguished:

- Friction plate clutch (pneumatically or hydraulically actuated)
- Fluid couplings
- Self-shifting-synchronous clutch

Figure 3.38: Example of double-helical gearing

(source: http://www.gearmanufacturers.net/double-helical-gear.html (february 2011))

The type of clutch that is used has impact on vibrations, thus noise, and the costs. A fluid coupling has a vibration attenuation function besides its clutch function and no wear will take place because it has no friction plates. This type of clutch is much more expensive, besides that it has significant loss (1-2%) through slip. The plate clutch and SSS-clutch have a 'hard' mechanical connection, so vibrations are very well transmitted. Flexible torsion-elastic couplings with rubber elements can be applied to damp the vibrations from the driving machine, but these couplings can only be used at the 'low torque side', thus high-speed side, of the gearbox. An advantage of the SSS-clutch over the plate clutch is the ability to smoothly overtake the driving of a shaft from one machine to another.

The main design parameters of a gearbox are the offset distance, the reduction ratio and the input power. Those parameters determine the size of the wheels, weight and costs of the gearbox.

> Well-known manufacturers of gearboxes are for example: Renk, Reintjes, Masson, Flender, Lohmann & Stolterfoht and ZF.

Auxiliary systems

Auxiliary systems on a gearbox are a lubrication oil system, hydraulic or pneumatic supply to clutches and a turning gear, normally electrically driven. Normally the gearbox is delivered as a complete system, with all the auxiliary systems mounted on it.

Data analysis

The analysis of gearboxes is based on gearboxes from the GES database and six gearboxes in service with the RNLN. Starting point are the relations in GES. But GES only holds relations for dimensions and weight of CODOG gearboxes, SISO gasturbine and SISO diesel engine gearboxes. This range is too narrow to estimate all gearbox types. Still, the GES relations will be tested on the SISO diesel gears of HOV and LPD-1, and on the CODOG gears of LCF and M-frigate. Another source is the study by Frouws (2008). In this study some relations were found for dimensions and weight of gearboxes. These relations are checked with the collected data. The GES database contains around 800 standard catalogue gearboxes, Commercially-Of-The-Shelf (COTS). For a lot of those gears, information is missing about the type (single or twin) and the offset (horizontal, vertical or coaxial), which makes good comparison impossible. Only the horizontal offset gearboxes are filtered out, because normally horizontal offset gears are used in naval ships. The gears that are considered in the analysis are:

- 26 COTS single-stage horizontally offset single gears (Lohmann & Stolterfoht GCH-series)

- Horizontally offset single gear of the HOV and LPD-1
- Horizontally offset twin (CODOG) gear of the LCF and M-frigate
- Horizontally offset twin (CODAD) gear of the AOR[10]
- Horizontally offset twin (CODLOD) gear of the OPV

3.6.1 Available power

Gear systems are used in all kinds of applications. In steamturbine powerplants, gearboxes for powers up to 140 MW are used. In the maritime sector, gear systems up to about 40 MW are found. Especially in the naval ship application it is common that a gearbox is especially designed for a certain application, so as a matter of fact (almost) every desired gearbox can be delivered.

3.6.2 Dimensions

In Frouws (2008) it is described that a relation exists between input power, outgoing speed and gear ratio on the one hand and the offset distance of the in and outgoing shaft on the other. In Aalbers (Unknown date) the relation between offset distance and length, width and height was shown before. Put together this gives relations for dimensions as a function of input power, outgoing speed and gear ratio. These relations were checked with the database gears. Concluded is that the model for calculation of the offset distance in Frouws (2008) is too inadequate for this purpose. The calculated values for offset distance show large deviations from the real values. Still using this formula would introduce large uncertainties in the modelling.

As an alternative, relations are searched between power/speed ratio and dimensions instead of offset and dimensions. The ratio of ingoing power and outgoing shaftspeed is a number that is normally given for commercial gearboxes and expressed in (kW/rpm). It describes the two main design parameters of a gearbox: maximum torque, M_{shaft} which is on outgoing shaft in a reduction gear, and the speed ratio i. To proof, rewrite equation 3.68 and 3.69. Leaving the constant factor $60/2\pi$ out, gives the power/speed ratio.

$$M_{shaft} = M_{engine} \cdot i = M_{engine} \cdot \frac{\omega_{engine}}{\omega_{shaft}} = \frac{P_{in}}{N_{out}} \cdot \frac{60}{2\pi} \tag{3.70}$$

Dimensions are including auxiliary pumps etc. and defined as:

- Length is measured along the ships longitudinal
- Width is measured in the breadth direction of the ship
- Height is measured vertically

The results are shown in figure 3.40, 3.41 and 3.42. Trendlines are constructed based on the Commercially-Of-The-Shelf (COTS) available gears and extrapolated (dotted line).

The two single gears of the HOV and LPD-1 are close to the trend of the COTS gears in all three dimensions. The others only in height, equation 3.73. Still there is some deviation from the trend, so a new trend is determined, equation 3.73. Length comes close, but it seems that length structurally is a little longer for the RNLN gears. For the length a new trend is given by the red line in figure 3.40, equation 3.71. The extra length of the RNLN gears is probably

[10]Auxiliary Oiler and Replenishment ship

caused by clutches and the double-helical wheels (for silent operation). Single- or multiple stage reduction is not taken into account. Except for the AOR all twin gears have two different input machines (CODOG or CODLOD) with different input powers and output speeds. In principal these are two gears with different power/speed ratios next to each other, with a common output shaft. To estimate width it is tried to add the dimensions of two separate gears based on the trendline of the commercial gears, equation 3.72. This hypothesis is tested and results are shown in figure 3.41 with dots:

> LCF has a power/speed ratio of: 39.8 (kW/rpm) in diesel drive and 111.6 (kW/rpm) in gasturbine drive. This means according to the model a gearbox width of: 2.3 (m) resp. 3.0 (m). Together 5.3 (m). Measured value is 4.8 (m). Deviation is +10%.
>
> M-frigate has a power/speed ratio of: 26.7 (kW/rpm) in diesel drive and 78.3 (kW/rpm) in gasturbine drive. This means according to the model a gearbox width of: 2.1 (m) resp. 2.8 (m). Together 4.9 (m). Measured value is 4.3 (m). Deviation is +14%.
>
> OPV has a power/speed ratio of: 4.0 (kW/rpm) in electric motor drive and 23.5 (kW/rpm) in diesel drive. This means according to the model a gearbox width of: 1.2 (m) resp. 2.0 (m). Together 3.2 (m). Measured value is 2.8 (m). Deviation is +14%.
>
> AOR has a power/speed ratio of: 71.7 (kW/rpm) in both diesel drives. This means according to the model a gearbox width of: $2 \cdot 2.7$ (m). Together 5.4 (m). Measured value is 5.5 (m). Deviation is -2%.

The following trends are concluded from the data, with P_{in} in (kW) and N_{out} in (rpm). For twin gears, length and height are calculated with the maximum value of power/speed ratio and the width is calculated by adding two separate gears, as described above. So $n = 1$ for single, $n = 2$ for twin gears.

$$\boxed{\text{Length} = 0.54 \cdot \left(\frac{P_{in}}{N_{out}}\right)_{max}^{0.37}} \tag{3.71}$$

$$\boxed{\text{Width} = \sum_{i=1}^{n} 0.84 \cdot \left(\frac{P_{in}}{N_{out}}\right)_{i}^{0.27}} \tag{3.72}$$

$$\boxed{\text{Height} = 0.98 \cdot \left(\frac{P_{in}}{N_{out}}\right)_{max}^{0.27}} \tag{3.73}$$

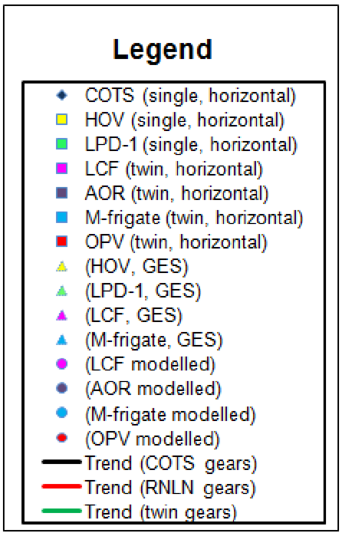


Figure 3.39: Legend for figures 3.40-3.43

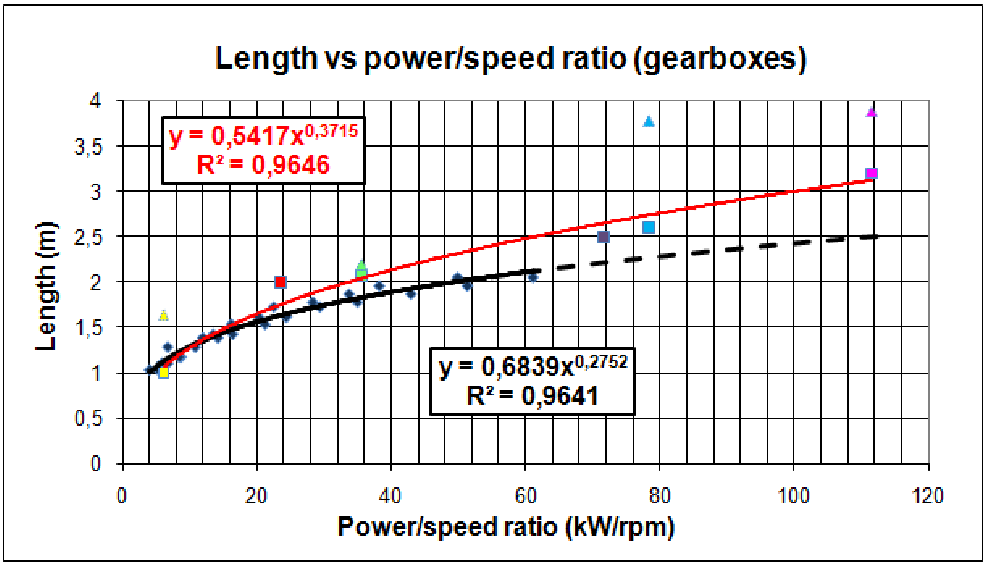


Figure 3.40: Length (m) vs power/speed ratio (kW/rpm) of database gearboxes (legend: see figure 3.39)

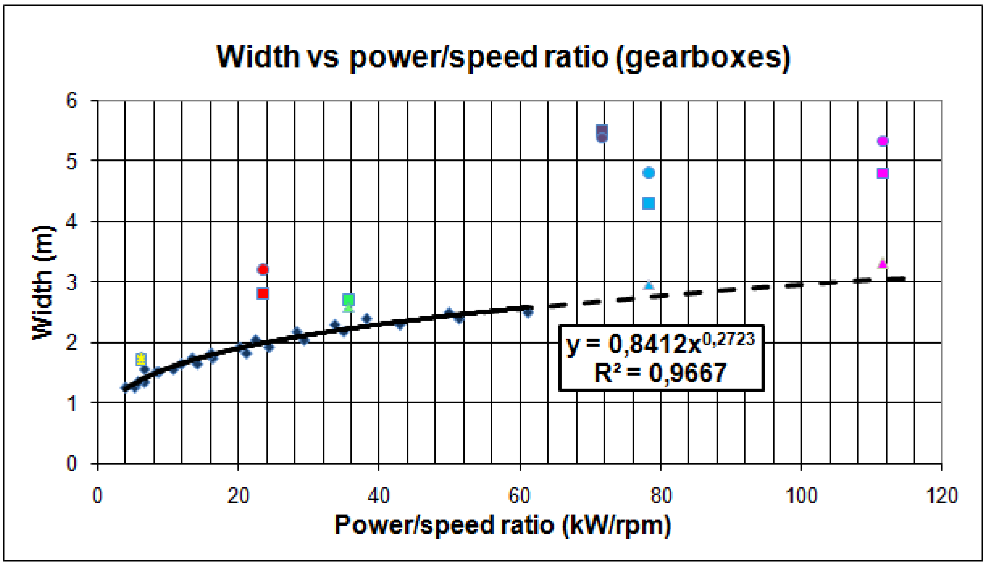


Figure 3.41: Width (m) vs power/speed ratio (kW/rpm) of database gearboxes (legend: see figure 3.39)

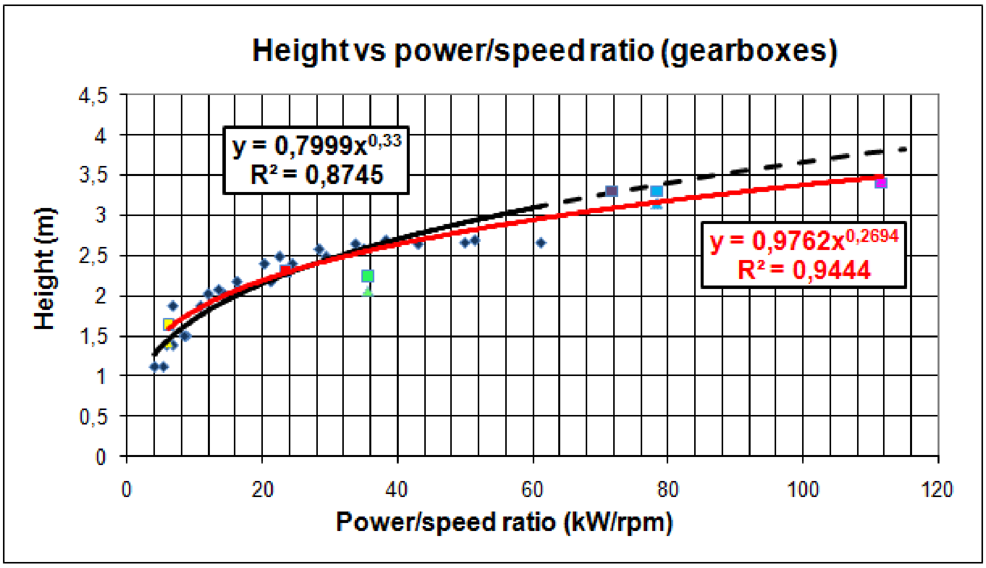


Figure 3.42: Height (m) vs power/speed ratio (kW/rpm) of database gearboxes (legend: see figure 3.39)

3.6.3 Weight

Trend is determined for the commercial single gears by a linear trend forced through zero, see equation 3.74 in (ton). The RNLN single gears show good correlation with this trend. For the twin gears the masses of two separate gears was added, like was done with the width of twin gears. This gives too low weights for the twin gears. The weight of the LCF is estimated factor 1.7 too low in this way, for M-frigate factor 1.5 too low, for OPV factor 1.6 too low and for AOR factor 1.2. Mean value is factor 1.5 too low for twin gears. This factor was used to correct the weight of twin gears. The results are presented in figure 3.43 by the circular datapoints. Deviations after this correction are: LCF $\rightarrow$ 15%, M-frigate $\rightarrow$ 0%, OPV $\rightarrow$ 6.3% and AOR $\rightarrow$ 18.8%.

Single input gears:

$$\boxed{\text{Weight}_{\text{single}} = 0.3 \cdot \left(\frac{P_{in}}{N_{out}}\right)} \tag{3.74}$$

Next it is tried to formulate a new relation especially for twin gears which can produce more accurate answers than the previously explained approach. The new trendline for twin gears is given by the red line in figure 3.43 and described by the following equation, with weight in (ton), P_{in} in (kW) and N_{out} in (rpm), the highest power/speed ratio of the twin gear should be used:

Twin input gears:

$$\boxed{\text{Weight}_{\text{twin}} = 0.6 \cdot \left(\frac{P_{in}}{N_{out}}\right)} \tag{3.75}$$

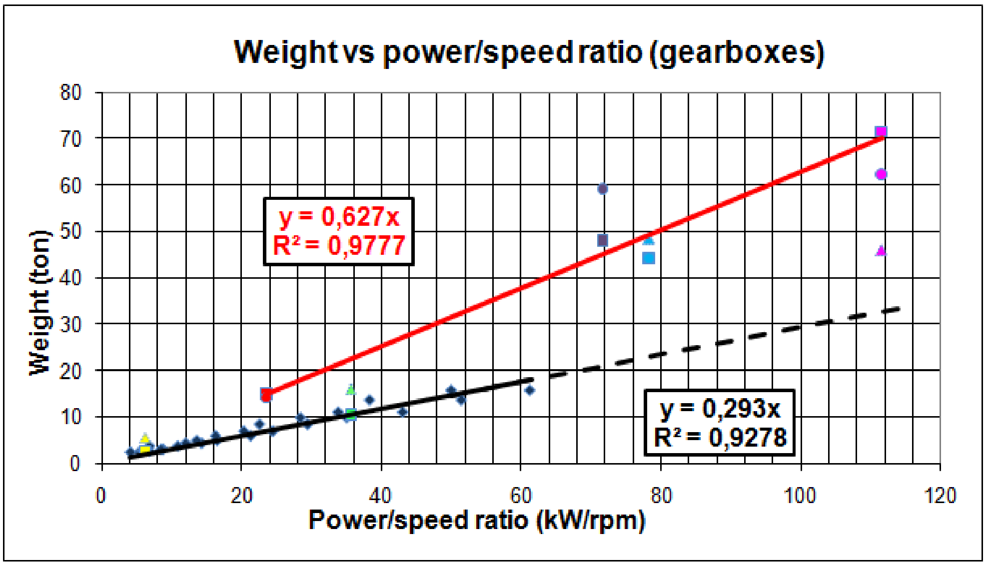


Figure 3.43: Weight (ton) vs power/speed ratio (kW/rpm) of database gearboxes (legend: see figure 3.39)

3.6.4 Operating speeds

Gearboxes can be produced for all speeds and all reduction ratios. According to van Dijk *et al.* (1998), maximum gear ratio for single stage reduction is 10. For higher gear ratio it needs to go to multiple stage reduction. Gear ratio is obtained by coupling two wheels with a different number of teeth. In practice this means that not every gear ratio is possible, but it goes with discrete steps.

3.6.5 Efficiency

Losses in a gearbox are caused by friction between the gearing wheels and by liquid friction. These losses are power-dependent, but there is also a part of the losses that is proportional to the rotational speed: the power delivered to the adhered hydraulic pump (fixed displacement and constant pressure). According to measurements at the LCF, this speed-dependent loss is about 0.1% of the nominal power through the gearbox. Compared to the other losses this powerloss is only a very small part and can be considered constant. From the measurements on the LCF an efficiency plot is made, see figure 3.44. This figure excludes the speed-dependent powerloss to the hydraulic pump.

In Stapersma (1994), a gearbox powerloss model is described. This model describes the powerloss as a function of power loading, torque loading and rotational speed, see equation 3.76. In which P^*_{loss} is powerloss in the gearbox as a fraction of the powerloss at nominal power, P^* is power load as a fraction of nominal power, M^* is torque load as a fraction of nominal torque and N^* is rotational speed as a fraction of nominal speed.

$$P^*_{loss} = aP^* + bM^* + cN^* \tag{3.76}$$

Based on experience, Stapersma (1994) advises to use $a = 0.4$, $b = 0.4$, $c = 0.2$. After some rewriting, it follows that partload efficiency can now be written as:

$$\begin{aligned} \eta &= 1 - \left(\frac{\frac{P_{loss}}{P_{nom}}}{P^*} \right) \\ &= 1 - \left(\frac{P^*_{loss} \cdot (1 - \eta_{nom})}{P^*} \right) \\ &= 1 - \left(\frac{(aP^* + bM^* + cN^*) \cdot (1 - \eta_{nom})}{M^* \cdot N^*} \right) \end{aligned} \tag{3.77}$$

If speed dependency is left out of scope ($N^* = 1$) and the values for a, b and c are filled in we get a more practical equation for partload efficiency of a gearbox:

$$\eta = 1 - \left(\frac{(0.8 \cdot P^* + 0.2) \cdot (1 - \eta_{nom})}{P^*} \right) \tag{3.78}$$

Down to approximately 3% partload this simplified efficiency model is pretty accurate as can be seen in figure 3.44. For lower partload this model can no longer be used, because outcome might become negative.

Nominal efficiency of a gearbox can not be derived from the database gearboxes, because there is not enough data available on efficiency. No model was found in literature, to estimate nominal gearbox efficiency as function of nominal power. According to Stapersma (1994) single stage reduction gears have a nominal efficiency between 98-99/%, and multiple stage gears between 95-98%. The AES study mentions values of 96-98% for single stage and 94-97% for multiple stage.

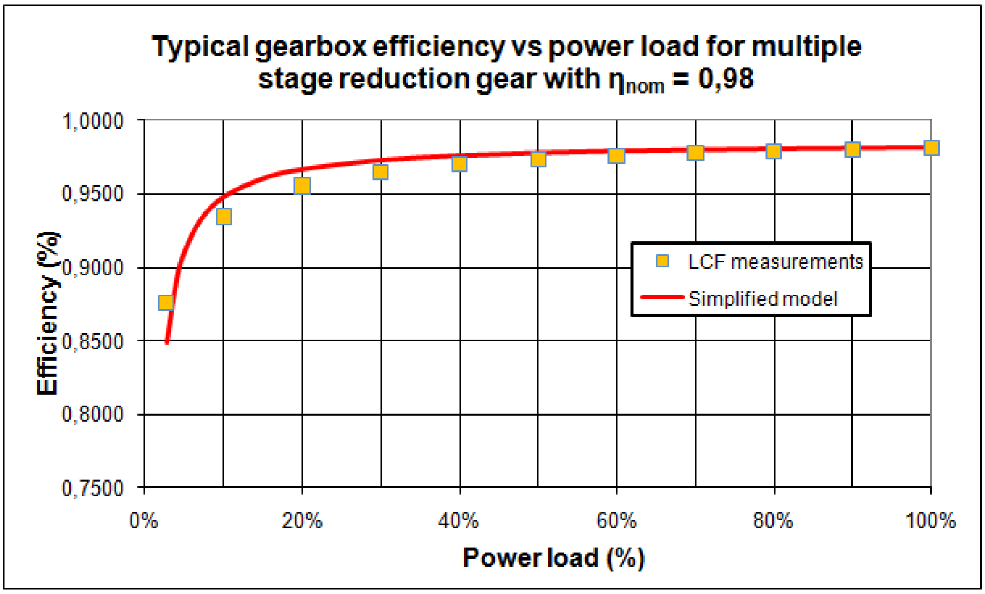


Figure 3.44: Typical efficiency curve of a gearbox (based on horizontal offset, multiple stage twin gear of the LCF) and partload efficiency calculated with model from Stapersma (1994), and partload efficiency according to simplified model of equation 3.78

Single stage reduction gears:

$$\boxed{\eta_{gb,nom} = 0.98 \text{ to } 0.99} \tag{3.79}$$

Multiple stage reduction gears:

$$\boxed{\eta_{gb,nom} = 0.95 \text{ to } 0.98} \tag{3.80}$$

Cross-connect gears add some extra losses. Gearbox manufacturer RENK states that total loss over a cross-connect gearsystem with one engine driving two shafts is about 6%. From this statement it is assumed that loss over a cross-connect gear is 3%.

Cross-connect gear:

$$\boxed{\eta_{gb,nom} = 0.97} \tag{3.81}$$

In figure 3.44 a plot of measurements on LCF gearbox with $\eta_{nom} = 0.98$ is presented, together with the results from the simplified model in equation 3.78. It is seen that the simplified model fits the measurements very well.

Other transmission losses

The gearbox losses are part of the total transmission losses. There is also some heat production, thus loss, in the bearings and seals of the shaftline. Because the shaftline is not further described in one of the sections, the losses are described in this part and called η_S. Shaft losses are low, typically 0.5 to 1% at nominal power. Shaft losses vary with shaft speed, since they represent a constant torque.

$$\boxed{\eta_{S,nom} = 0.99 \text{ to } 0.995} \tag{3.82}$$

3.6.6 Signatures

Underwater noise

The gearbox is known to be an important producer of underwater noise. Depending on the speed of the driving motor, the teeth impact frequency cause dominant vibrations. Frequency of the underwater noise is typically in the medium frequency domain (10^2 to 10^3 Hz) and of high power (160-170 dB). These frequencies can be reduced with a hard resiliently mounting system, but this might cause troubles with outlining of the shafts. Great part of the noise that is produced by the gearbox comes from the torque ripple on the incoming shaft that is connected to a diesel engine, gasturbine or electric motor. To decrease the noise the torque ripple should be minimized. Gasturbine has probably the lowest ripple because it has constant combustion, the diesel engine produces a ripple with the ignition frequency, and electric motors with the frequency of the converter higher harmonics. Putting a flexible coupling between the motor and the gearbox decreases the torque ripple, thus the noise of the gearbox.

In general, applying gearboxes is a seriuous design risk, especially when having stringent requirements, because they are a dominating noise source, according to Hendriks *et al.* (2011).

Infrared and electro-magnetic

The influence of a gearbox on infrared signature and electro-magnetic signature, is not considered very large.

3.6.7 Shock resistance

The gearbox is a heavy and robust piece of machinery which gives it good shock resistance. The housing and gearwheels have to be designed for shockwaves. The gearwheels have a very precise alignment with low tolerances which makes it sensible to shockwaves. The double helical teeth improve the shock resistance, because the wheels are more or less fixed in relation to others. Planetary gears have better shock proofness compared to 'normal' gearboxes, because of the compact design. Sometimes, gearboxes are placed on resilient mountings, but these are very stiff with maximum deflection of only millimeters (for shaft alignment) which is worthless for catching shockwaves.

3.6.8 Maintainability

Maintenance on a gearbox is nihil. They are designed for ship lifetime, normally no overhaul is required if cooling and lubrication works well. Of course lubrication oil has to be replaced from time to time.

3.6.9 Reliability

Gearboxes have very high reliability. No numbers are known to the author, but the AES study mentions a MTBF of 100000 hours and a MTTR of 5 hours, which sounds reasonable.

3.6.10 Initial purchase costs

To estimate the purchase costs of a gearbox, a Cost Estimating Relationship (CER) is used. The Cost Analysis section of DMO determined a CER based on purchases and quotations from the past. DMO's CER is a function of the weight of the gearbox. In the AES study is also mentioned a CER, which is a function of power and/or gear ratio. A relation based on weight seems more suitable because that rules out the gear ratio and differences between single and twin gears. In the AES study, distinction has to be made between CODOG gearbox and SISO gearbox (gasturbine or diesel). The CER from the AES study is converted to €, and increased at an inflation rate of 2% per year (1998):

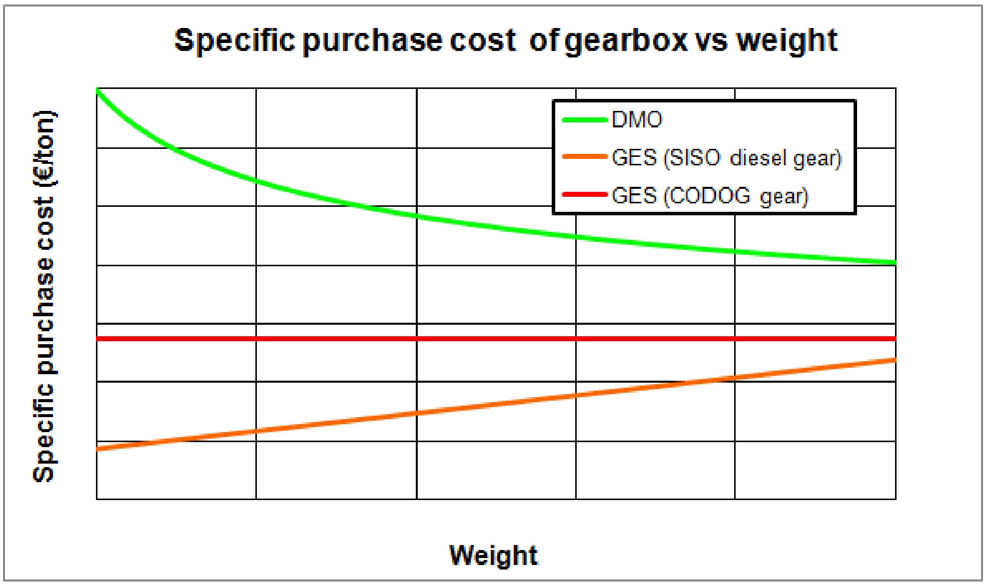


Figure 3.45: Specific purchase costs (€/ton) of gearboxes vs weight, according to different sources. Cell height is 10000 €/ton

- CODOG: 82 €/kW (gasturbine power)
- SISO: $\left(2.17 \cdot i + 0.135 \cdot 10^{-3} \cdot i \cdot P_{[kW]}\right)$

In this form the CER's from the AES study can't be compared with DMO's CER. The AES relationships are converted to relations as a function of weight with equations 3.74 and 3.75. A fixed outgoing speed N_{out} of 200 rpm is assumed. In the SISO relation a gear ratio $i = 5$ is assumed, which is representative for a diesel drive. Figure 3.45 shows the results of the comparison.

3.7 Electrical auxiliaries

This section describes the electrical auxiliary systems: switchboard and converter. These are not main components, but are consequential auxiliaries of chosing electrical main components. They need to be discussed because dimensions, weight and costs are significant and might influence the choice for one propulsion concept or another.

Switchboard

On board practically every ship electrical power generation is found, mostly by diesel-generator sets. Other options are gasturbine-generator or fuel cells or electrical power is stored on board in batteries. The electrical power needs to be distributed throughout the ship to all the users. A number of electrical power sources and users are connected to a switchboard and the switchboard is the device that directs the electric power from a source to a user by means of switches. Every connection of a source or a user to a switchboard is called a field and has its own switch. A user can be a large single user (like a bowthruster), or a group of smaller users on a lower level distribution board. In this analysis only the main switchboard is considered, and for high-voltage applications also the low-voltage distribution switchboard. Low voltages are $<$1 kV and high voltages $>$ 1 kV. High voltage is only used when very powerful users are present ($\geq$ 1 MW) like propulsion motors or bowthrusters. When high voltage is generated onboard there are two switchboards: the high-voltage main switchboard (typical value 6.6kV) and the low-voltage distribution switchboard (typical value 440V) with a transformer in between. For redundancy reasons all boards are doubled (starboardside and portside) with an interconnection, so-called bus-coupler. The amount of power going into a switchboard must equal the power going out to the loads. The switchboard also has controls for the supply machinery (diesel-generator etc.) to ensure this balance. It has frequency control for AC power distribution and load sharing controls. Inside a switchboard there is a bank of busbars (wide copper or aluminium strips) to which switchgear is connected. Two main types of switches can be be distinguished for both high- and low-voltage applications:

Switch-disconnector: also called contactor, is a relatively simple switch with a fixed-value fuse at every phase.

Circuit breaker: is a more complex switch with built-in control. The fuse value can be adjusted. In case the fuse value is crossed the circuit breaker can be reset and still be used unlike the switch-disconnector.

Onboard naval ships both types of switches are used, but the circuit breaker has logistical advantages. All fields in the switchboard can use the same switch, but with different fuse values. Advantages of switch-disconnectors are the lower costs, the smaller dimensions and normally longer lifetime.

> Some well-known manufacturers of switches are: ABB, Siemens, Converteam, Schneider Electric.

Auxiliary systems

Switchboards dissipate heat, so need cooling. Normally, this is natural-air cooling.

Data analysis

The relations for dimensions, weight etc. that are found in the AES study are used in this section. Drawings of the switchboards of LPD-2 and JSS are used to validate these relations.

3.7.1 Available power

Switches are available at all power levels that are relevant for this study. A distinction between low voltage (<1 kV) and high voltage (>1 kV) switches can be made. In practice, high voltage is 6.6kV. Low voltage is within NATO standardized to 440V.

3.7.2 Dimensions

Switchboards are built up by a number of modules with standard dimensions (depending on the manufacturer). The number of modules depends on the number of fields (n_f). Every field has its own switch. The dimensions of a switch depend on the voltage and current. For smaller switches, more than one can be vertically stacked in a module, but for high powers every switch has its own module. At the back of the switchboard is the busbar that connects all fields. The dimensions of the busbar also depends on the current.

In the AES study, van Dijk *et al.* (1998), dimensions of switchboards are described. Distinction is made between low voltage LV (<1 kV) and medium voltage MV (>1 kV) switchboards. This comes from times when there was distinction between low, medium and high voltage. Nowadays distinction is only made between low and high voltage: low voltage<1kV<high voltage, this distinction is adopted. van Dijk *et al.* (1998) gives standard module dimensions (width x depth x height) 0.4x1.7x2.2 m, assuming one field per module, and for high voltage 0.6x1.7x2.2m, also assuming one field per module.

The dimensions of the AES study are checked with some switchboards known to the RNLN. Dimensions and layout of switchboard modules differ per manufacturer. The module dimensions given by van Dijk *et al.* (1998) for high voltage switchboards correspond to the dimensions of the Converteam modules onboard the JSS (0.6x1.7x2.2m). The modules of ABB onboard the LPD-2 have different size: 0.65m wide, 1.35m deep and 2.60m high. And the modules onboard the LCF are also 0.65m wide, but only 1.30m deep and 2.20m high. In this it is assumed that all fields have circuit breakers. A switch-disconnector can have a smaller module (half the width onboard LPD-2), but in early stage design it is best to account for maximum dimensions.

For high voltage switchboards the following maximum dimensions can be calculated, with n_f the number of fields (in+out):

High voltage:

$$\boxed{\text{Width} = 0.65 \cdot n_f} \tag{3.83}$$

$$\boxed{\text{Depth} = 1.70} \tag{3.84}$$

$$\boxed{\text{Height} = 2.6} \tag{3.85}$$

Height is set to 2.6m, but can also be 2.2m. Generally speaking, the height of the switchboard does not exceed the height of the room. Around the module an extra space (approximately at the value of the depth of the module) must be free for changing switching gear.

If the number of fields (in+out) n_f is not known, it must be estimated. The following assumptions can be made: 2 switchboards, 2 incoming generator fields per switchboard, 1 bus coupler field per switchboard, 1 outgoing field per bowthruster per switchboard, 1 or 2 outgoing fields per electrical propulsion motor per switchboard depending on the power, 2 outgoing fields to the distribution transformers, 1 spare outgoing field and possibly 1 field for high voltage shore connection.

The modules used in a low voltage switchboard are smaller. The dimensions of a switch are depending on the current. For this case, distinction is made between low current (<630A)and high current (>630A). Normally, the inputs of the switchboard (dieselgenerator, shore connection, transformer connection, bus-coupler) are high current connections which need bigger switches. This means only one switch per module of 0.65m. The outgoing fields of a low voltage switchboard are normally lower current connections with smaller switches. This means that more switches are stacked in a module, 4 on average. To estimate the width, a distinction between number of input fields ($n_{f,in}$) and output fields ($n_{f,out}$) is made. Depth of a low voltage switchboard is smaller than of a high voltage board, around 1m (0.9m on LCF and LPD-2). Height will normally also be smaller (2.20m on LCF incl. shock mountings and 2m on LPD-2), but is not of much interest because it will not exceed the height of a room. Around the module an extra space (approximately at the value of the depth of the module) must be free for changing switching gear.

Low voltage:

$$\boxed{\text{Width} = 0.65 \cdot \left(n_{f,in} + \frac{1}{4} \cdot n_{f,out}\right)} \tag{3.86}$$

$$\boxed{\text{Depth} = 1} \tag{3.87}$$

$$\boxed{\text{Height} = 2.20} \tag{3.88}$$

If the number of fields is not known an assumption must be made. Normally the low voltage switchboards on board have: 2 incoming fields from diesel generators or from high voltage switchboard, 1 bus coupler field, 1 shore connection, 1 emergency generator. The number of outgoing fields can be estimated by dividing the electrical load of the ship by the voltage (440V) and dividing that number by an average field current of 630A. This is not completely true because some users on the switchboard have their own field, like fire pumps and chilled water makers. To correct for this, add 2 to the outcome.

The way the fields are put together in a switchboard can be adjusted to the available space onboard. Normally all fields are placed next to each other, but they can also be placed back-to-back or even in U-form. The designer has some degree of freedom.

The dimensions as described above are including cooling volume.

3.7.3 Weight

The weight of a switchboard consists of the weight of the modules, the copper work and the switches. In the AES study the following 'all-in' weights were determined: 500 kg per low voltage module and 800 kg per high voltage module. According to the data of the LCF ($\approx$ 900 kg/module), JSS ($\approx$ 1000 kg/module) and LPD-2 ($\approx$ 1000 kg/module), the weight of high voltage module is set to 1000 kg per module including the bedplate. In this it is assumed that all fields have circuit breakers. Modules with contactors are smaller and lighter, 550 kg/module onboard of the LPD-2. The low voltage modules are lighter than presented in the AES study. Based on the data from the LCF it follows that the modules with high-current incoming fields (one field per module) have a weight of 500 kg/module and the modules with low-current outgoing fields (four fields per module) have a weight of 450 kg/module.

High voltage:

$$\boxed{\text{Weight} = 1000 \cdot n_f} \tag{3.89}$$

Low voltage:

$$\boxed{\text{Weight} = 500 \cdot n_{f,in} + 450 \cdot \frac{1}{4} \cdot n_{f,out}} \tag{3.90}$$

3.7.4 Operating speeds

The operating speed of a switchboard can be defined as the switching speed. Breaking and closing time are distinguished, both in the order of magnitude 50 ms.

3.7.5 Efficiency

Losses in a switchboard are low compared to the transmitted power. Efficiency of a switchboard is assumed to be:

$$\boxed{\eta_{nom} = 0.995} \tag{3.91}$$

3.7.6 Signatures

The signature profile of a switchboard does not contribute significantly to the total profile. The switching makes noise, but it does not occur at a certain rate or frequency and the soundlevels are relatively low. The contribution of a switchboard to the electro magnetic profile can also be neglected.

3.7.7 Shock resistance

Switchboards hold delicate elements which are sensitive to shock. In case of a shockwave the switches could open or close unwanted with dramatic consequences. For ships with shock requirements it is common to place the switchboards on springs.

3.7.8 Maintainability

The maintenance on a switchboard consist of changing the switching connectors or complete switches after a number of switching actions. Relative to other components, maintenance on a switchboard is very low.

3.7.9 Reliability

Reliability of a switchboard is very high. As with all components the determination of a reliability number is always difficult. In the AES study 100% reliability is assumed. MTBF = ∞ and MTTR = 0, which results in a 100% availability.

3.7.10 Initial purchase costs

The initial purchase costs of switchboards are estimated with a Cost Estimating Relationship (CER). The Cost Analysis section at DMO has determined such a CER based on historical data from purchases and quotations. This CER is commercially confidential. The AES study also mentions some relationships. Distinction is made between high voltage and low voltage switchboards. The costs are a function of the number of fields. The number of fields can be estimated with the input power, as explained earlier on page 87. This is used to present the AES CER as a function of power. This is presented in figure 3.46 together with the CER from DMO.

The CER's in the AES study are:

- Low voltage: 25 k€/field
- High voltage: 35 k€/field

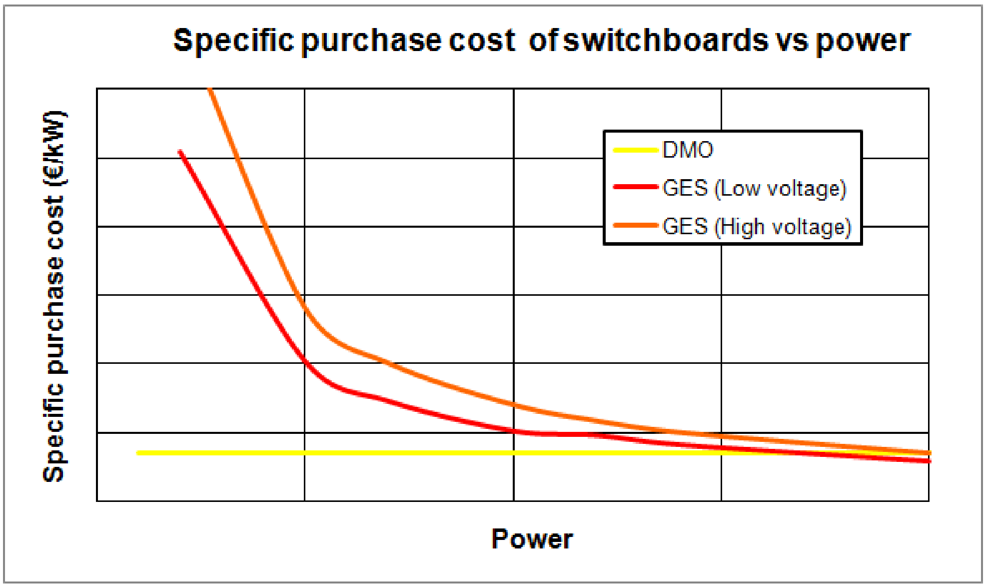


Figure 3.46: Specific purchase costs (€/kW) of switchboards vs power, according to different sources. Cell height is 200 €/kW

Converter

Converters are used to adapt the voltage and/or the frequency of an electric power supply to the need of a secondary power supply or the need of an electric drive system (propulsion motor, pump, fan etc.). By adapting the voltage and frequency, the speed and delivered torque of an electric motor can be controlled. A converter actually is the electrical equivalent of a gearbox. A converter generally consists of coils, capacitors, diodes, transistors, Insulated-gate Bipolar Transistor (IGBT), ordinary thyristors, Gate Turn-Off (GTO) thyristors or Integrated Gate-Commutated Thyristors (IGCT). When a converter is used to control an electric motor, the speed and torque can be controlled by adapting the voltage and/or the frequency of the supply. Depending on the type of motor a certain type of converter can be chosen:

- Controlled rectifier (AC-DC)
- Inverter (DC-AC)
- Chopper (DC-DC)
- Synchro-converter (AC-DC-AC)
- Cyclo-converter (AC-AC)
- Pulse Width Modulation (PWM) converter (AC-DC-AC)

Description of the different types of converters and switching elements is based on chapter 9 in KleinWoud & Stapersma (2003).

Controlled rectifier: Generates variable output voltage (DC) between zero and maximum (of the AC input). Output voltage is not perfectly constant. Are used to control speed of DC-motors, or as DC-link in synchro- or PWM-converters. Uses ordinary thyristors as active elements.

Inverter: Generate AC with varying frequency from the DC input. Output voltage can not be controlled, only by variation of DC supply. Inverters using Pulse Width Modulation (PWM) can control both frequency and voltage of the output. Inverters are used to supply and control speed of asynchronous and synchronous AC motors.

Chopper: Generate a varying DC output from a DC source. Output voltage is more or less constant. Are used to control speed of DC-motors. Uses transistors or GTO thyristors as active elements. Two possible switching strategies: Pulse Width Modulation (PWM) and Pulse Frequency Modulation (PFM). PWM uses constant switching frequency of the transistor but variable 'on'-time (pulse width). PFM uses constant 'on'-time but variable switching frequency.

Cyclo-converter: Generate a varying frequency output between 0-35% of a constant frequency without a DC link. Uses ordinary thyristors as active elements, 12 per motor winding. The large number of components decreases the reliability of this converter type and gives it a relative low power density in weight and volume. Are used to power and to control the speed of AC synchronous and asynchronous motors, especially high power (order of magnitude 100 MW), low speed applications. Advantage of this type is the rather smooth sine it produces. Has a variable power factor, normally varying between 0.3-0.75.

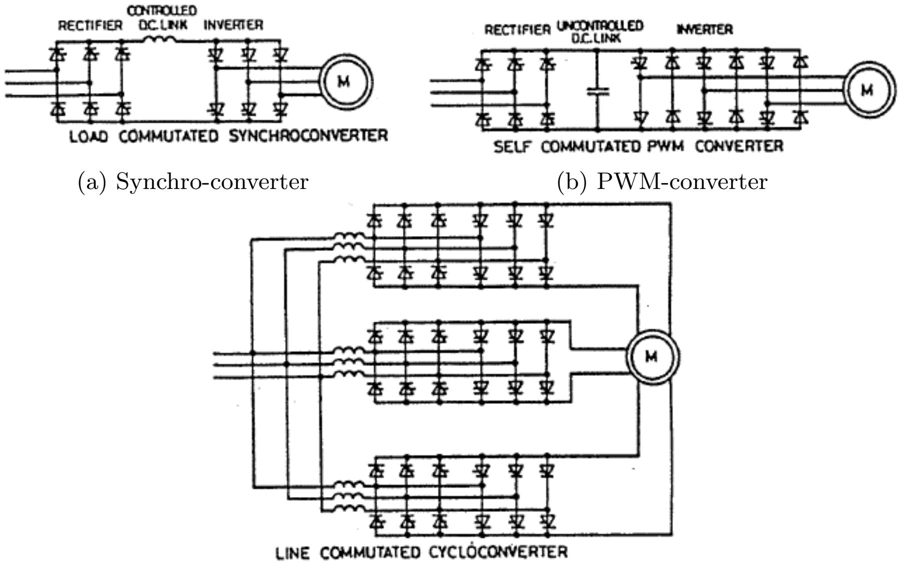
Synchro-converter: Also called Current Source Inverter (CSI). Generate a varying frequency and varying voltage output from a constant frequency and constant voltage input via an in-between DC link including a smoothing reactor (coil). Practically the same as a controllable rectifier + an inverter. Can only be used to power and to control speed of AC synchronous motors. Uses ordinary thyristors as switching elements and can therefore drive high powers (up to 100 MW). Synchro-converters have higher power density than PWM and cyclo, because it uses less semiconductors, only 4 per motorwinding, and for that reason has a high reliability. The converter is commutated by the load but at low speeds (0-10%), motor cannot commutate the bridge and it is forced-commutated, which introduces higher torque ripples. The power factor varies with speed (0.3-0.85) because the rectifier input circuit is controlled, unlike the PWM which has uncontrolled DC link.

PWM-converter: Also called Voltage Source Inverter (VSI). Generate a varying frequency and varying voltage output from a constant frequency and constant voltage input via an in-between DC link including a bank of capacitors. Practically the same as a uncontrollable (diode) rectifier + a PWM inverter, with transistors, IGBT's, IGCT's or GTO thyristors as switching elements, 6 per motorwinding. Are used to power and to control speed of asynhronous motors and synchronous motors (together with a chopper for the rotor supply). With IGBT's as semiconductors the converter can power motors up to approximately 10 MW, with IGCT's up to approximately 25 MW. It has a constant high power factor (> 0.95) and fixed harmonic frequencies because of uncontrolled DC link.

Figure 3.47 shows the principle schemes of the three main converter types for AC machines.

In practice, on board naval ships only PWM-converters are found for AC motors, and choppers for DC motors. The cyclo- and synchro-converters are not much found nowadays, because the PWM-converter has advantages over these types in power density and controllability. PWM-converters use other switching elements which have higher power density and can be controlled better, because of switching-off capabilities. Some main characteristics of the switching elements, or semiconductors that are used in converters are mentioned below:

Transistor: Switch on (while base-current >treshold) and off (while base-current = 0), voltage drop around 1 Volt, maximum switching rate approximately 20 kHz.



(a) Synchro-converter

(b) PWM-converter

(c) Cyclo-converter

Figure 3.47: Principle schemes of AC converters, excluding the motor exciter installations that are necessary for AC synchronous motors

Ordinary thyristor: Switch on (with current-pulse from gate to cathode) but not off (only when anode current = 0), voltage drop around 1 Volt, cheaper than transistors and suitable for high powers but relative low switching frequency.

Gate turn-off thyristor: (GTO) Switch on (with current-pulse from gate to cathode) and off (with negative current-pulse from gate to cathode), voltage drop typically twice of ordinary thyristor, suitable for high voltages and currents, disadvantage is the limited switching frequency (max. ±1 kHz) compared to IGCT and the rather high control current that is needed.

Integrated gate-commutated thyristor: (IGCT) Switch on and off, like the GTO, but have lower conduction loss and can operate at higher frequencies than GTO (max. several kHz)

Insulated-gate bipolar transistor: (IGBT) Switch on and off, noted for high efficiency and fast switching, voltage drop around 2 Volt, suitable for high current (hundreds of Ampères) and high blocking voltages, high power applications.

Some well-known manufacturers of converters are: ABB, Siemens, Converteam, ASI-Ansaldo.

Auxiliary systems

Converters dissipate a lot of heat in the switching elements, so cooling equipment is needed. This can be forced-air cooling or water cooling (possibly with internal air cooling). Normally water cooled converters are used. For low powers (<200 kW) air cooling may be used.

Data analysis

The relations from the AES study serve as a starting point for analysis of dimensions and weight. Data is gathered of converters onboard RNLN ships to check on the relations from the

AES study: LPD-1, LPD-2, HOV, JSS and Walrus class submarine. The converters that are considered are all driving AC asynchronous machines and are all water-cooled PWM-converters, except for the air-cooled chopper onboard the Walrus class. There is no data about cyclo- or synchro-converters available within RNLN, so no comparison between different converter types can be made, unfortunately. The converter on the HOV is a special type from ABB which uses the Direct Torque Control (DTC) principle. DTC is a somewhat different control strategy. It is no further explained in this thesis because it is not relevant for this study. The Walrus class submarine is driven by a DC motor. A chopper controls the DC input to the shuntfield of the motor.

3.7.11 Available power

There are limitations to the voltage and current through the semiconductors in converters. Each type of semiconductor has its own characteristics and limitations. To know more about power of a converter, one should go more in detail to the principles of the semiconductors, which is not within the scope of this thesis.

In general the following judgements exist about the different types of converters: DC choppers have limited power output (transistors), cyclo-converters are able to drive high torque motors at low speeds (ordinary thyristors), synchro-converters have a low power density (ordinary thyristors), PWM-converters have a high power density (IGCT/IGBT/GTO) but limited power output, up to approximately 10 MW with IGBT's and up to approximately 25 MW with IGCT's.

The capability of a converter is given in terms of apparent power S, because the real delivered power P depends on the 'power factor' of the load, the so-called $\cos(\phi)$.

$$S = U \cdot I \tag{3.92}$$

3.7.12 Dimensions

Dimensions are only known from the AES study and from the RNLN converters, and these are all PWM-converters, so relations can only be derived for PWM-converters. Cyclo-converters have more semiconductors and have for that reason a lower power density, thus larger dimensions. Synchro-converters have less semiconductors and have a higher power density, thus smaller dimensions.

A converter is, like a switchboard, built up by a number of modules or cabinets. For a PWM-converter, as normally used, these are:

- Rectifier cabinet
- Inverter cabinet
- Cooling unit cabinet
- Control system cabinet
- In- and output line cabinets

A chopper, for driving DC machines is smaller, because less components are needed. The control system, cooling and in- and output lines are of course still needed, but the chopper itself is smaller than a rectifier + an inverter.

The cabinets that are used to built up a converter unit have more or less standardized dimensions. The height will vary per manufacturer, but almost always be around 2.30 m (the height of a room). The depth of the cabinets depends a bit on the switching elements that are used and the

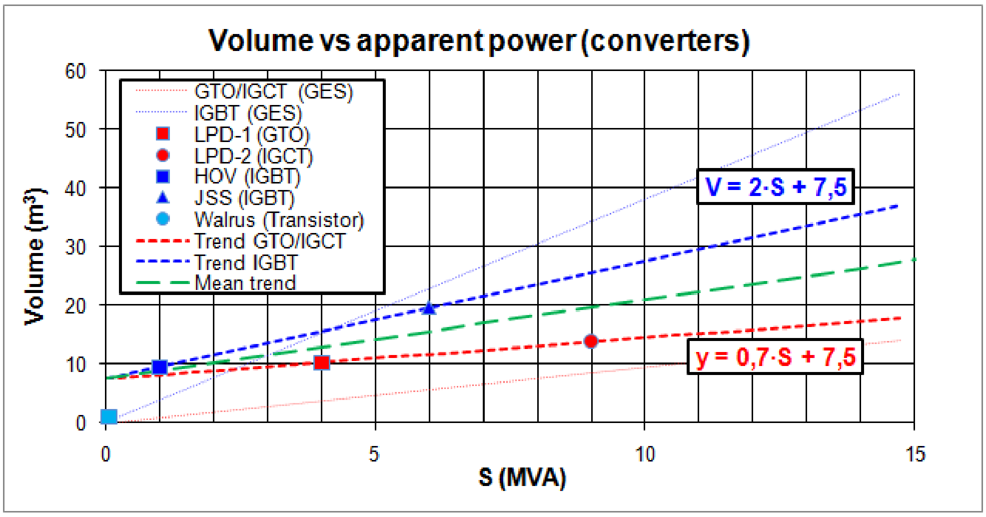


Figure 3.48: Volume (m^3) vs apparent power (MVA) of database converters and according to model in GES

apparent power. In the AES study van Dijk *et al.* (1998) distinction is made between IGBT, GTO/IGCT. The results from the AES study give a relation for converter volume as a function of real power in (MW). With a mean value of the power factor (0.95) this is calculated in a volume per apparent power in (MVA). These relations are presented in figure 3.48, together with the data from RNLN converters. Converters with IGBT's are calculated for 3.6 m^3 per MVA and GTO and IGCT converters for 0.9 m^3 per MVA in the AES study. Cooling in a converter with IGBT's consumes more space. In the converter database there are 2 converters with IGBT switching elements (HOV and JSS), 1 with IGCT's (LPD-2) and 1 with GTO thyristors (LPD-1). The chopper onboard the Walrus-class uses transistors. Since this is the only chopper in the dataset, and choppers are not considered in the AES study, an analysis is impossible. It is seen in figure 3.48 that both models, from the AES study, for volume of converters are not compliant with the data. In the power range we are talking about, a certain minimum volume is set. From the data follows 7.5 m^3. The IGBT converters increase volume with 2 m^3 per MVA, and the GTO and IGCT converters with 0.7 m^3 per MVA. But because the designer should not concern about the switching elements in a converter in an early design stage a mean value for volume is taken for the PWM converter.

Mean value PWM converters:

$$\boxed{\text{Volume} = 7.5 + 1.35 \cdot S} \tag{3.93}$$

The volume of a converter is determined by the power and the type of converter. The height of power converters will (almost) always be about the height of a room, except for special products that have to fit in small places, like in a submarine. For the converters in the database the mean height is 2.30 m.

Mean value PWM converters:

$$\boxed{\text{Height} = 2.3} \tag{3.94}$$

The depth is dependent on power. The components are smaller in less powerfull converters, which results in shorter modules. From the converters in the database a trend between power and depth was found, see figure 3.49. Both trendlines are forced through a certain value of minimal depth of a module. Based on the data, 0.70 m seems to be best fitting. It is seen

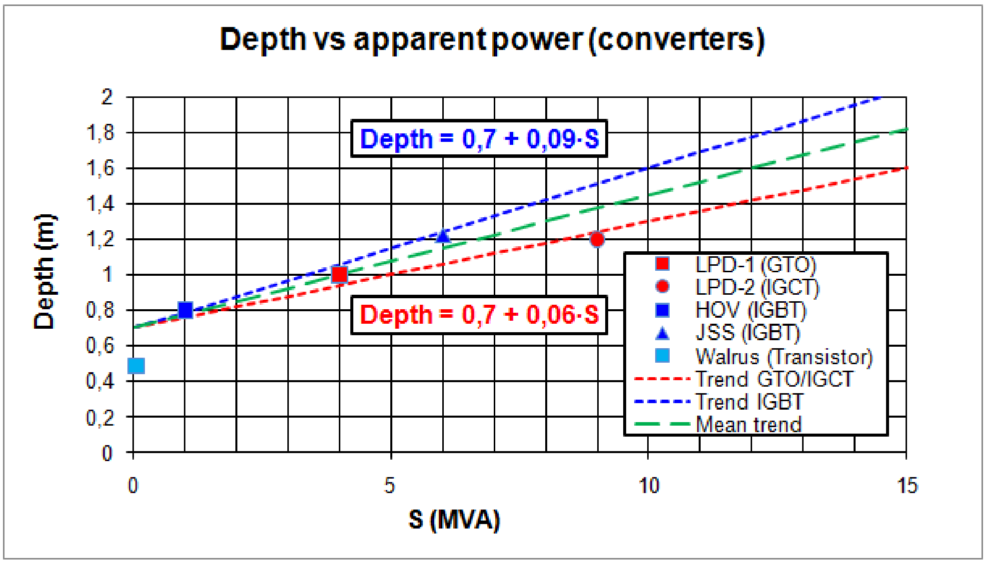


Figure 3.49: Depth (m) vs apparent power (MVA) of database converters

that the converters with IGBT's increase a little faster in depth than converters with IGCT's or GTO's. In an early design stage it should not be of concern what type of semiconductors are used in the converters, so a mean trend is concluded:

Mean value PWM converters:

$$\boxed{\text{Depth} = 0.7 + 0.075 \cdot S} \tag{3.95}$$

From the trends and found relations for mean volume, mean depth and mean height, a mean relation for the width of a converter can be calculated with $\frac{\text{Mean volume}}{\text{Mean height}\cdot\text{Mean depth}}$:

Calculated mean value PWM converters:

$$\boxed{\text{Width} = \frac{7.5 + 1.35 \cdot S}{2.3 \cdot (0.7 + 0.075 \cdot S)}} \tag{3.96}$$

Like with the switchboard, a certain maintenance space around the converter should be taken into account. Normally a space at the value of the depth of the converter is enough.

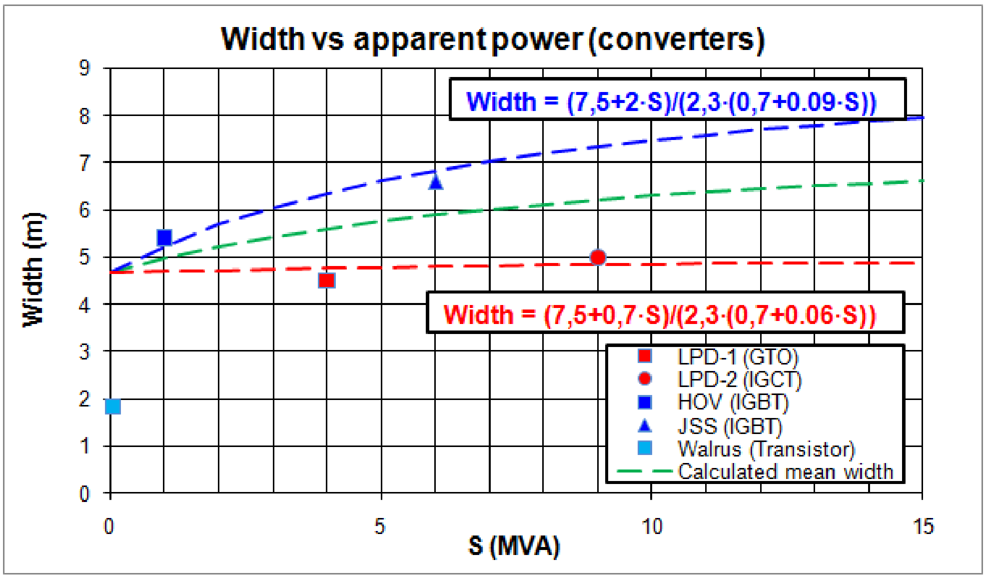


Figure 3.50: Width (m) vs apparent power (MVA) of database converters and calculated mean value

3.7.13 Weight

For the weight of converters the same approach as with the dimensions is made. The relations from the AES study, as implemented in GES, serve as a reference and the found data is tested with these relations. Relations were only derived for PWM-converters, because the lack of information about cyclo, synchro and chopper. But cyclo-converters are heavier than PWM, and synchro-converters and chopppers have lower weights.

In the AES study a linear relation for the weight of IGBT converters was found and a logaritmic relation for the GTO and IGCT converters. Both relations and the data from the small database are presented in figure 3.51. As can be seen in this figure the weight of all types are very much on a straight line. The purpose of this study is giving a general trend for estimating dimensions and weight in an early design phase. The data gives the idea that a general linear trend for all types of converters makes sense in this case. Of course it is only based on very limited datapoints. The weight of the chopper onboard the Walrus class clearly deviates from the trend. Although it is an older converter, which expects to be heavier, it is lighter because it is the only air-cooled and is relatively a lot smaller than the PWM converters. For determining a trend for choppers, more data would be needed.

Mean value PWM converters:

$$\boxed{\text{Weight} = 1500 + 750 \cdot S} \tag{3.97}$$

In the weight is included the complete converter: cabinets with bedplates and shockmounting, semiconductors and cooling equipment.

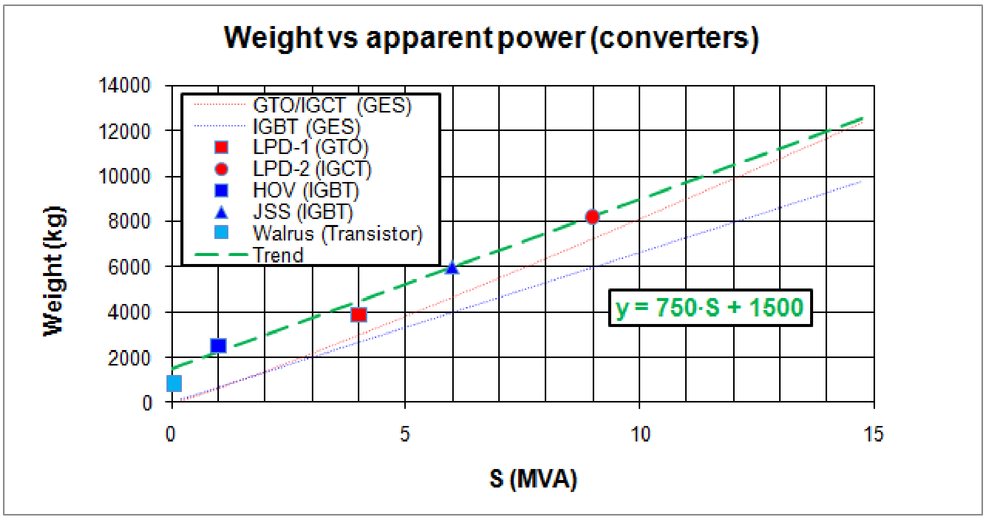


Figure 3.51: Weight (kg) vs apparent power (MVA) according to models in GES and of database converters

3.7.14 Operating speeds

Operating speed in the form of rotational speed does not apply to converters since there is no mechanical operation. The electrical equivalent of operating speed in this application is the frequency. The maximum output frequency that a converter can generate depends on the type of converter and on the maximum switching frequency of the semiconductors. Characteristics of the semiconductors are mentioned earlier.

The cyclo-converter is known to have a limited output frequency, between 0-35% of the input frequency. In a synchro-converter the output frequency is determined by the motor speed. The maximum output frequency is determined by the maximum switching frequency of the semiconductors. The same holds for PWM-converters which can go for high output frequencies, depending a little on the type of semiconductor that is used. A proper output can be produced up to approximately 1/3 to 1/2 of the maximum switching frequency of the semiconductors.

3.7.15 Efficiency

The efficiency of a converter, or the power factor, depends on the type. The PWM-converter has a high nominal efficiency which is rather constant throughout the speed range, normally above 95%. The cyclo- and synchro-converters have lower nominal power factors and the disadvantage that the power factor strongly varies with the speed. For cyclo-converters the power factor varies in the range 0.3-0.75, and for synchro-converters between 0.3-0.85.

There is no data available about the efficiency of the RNLN converters in the database. But in the AES study a relation between nominal efficiency and nominal real power of PWM-converters was derived (with P_{nom} in kW):

$$\eta_{nom} = \frac{1}{1.01 + \frac{0.527}{\sqrt{P_{nom}}}} \tag{3.98}$$

This formula is rewritten such that it can be used with S_{nom} in MVA as input. A power factor of 0.95 is assumed.

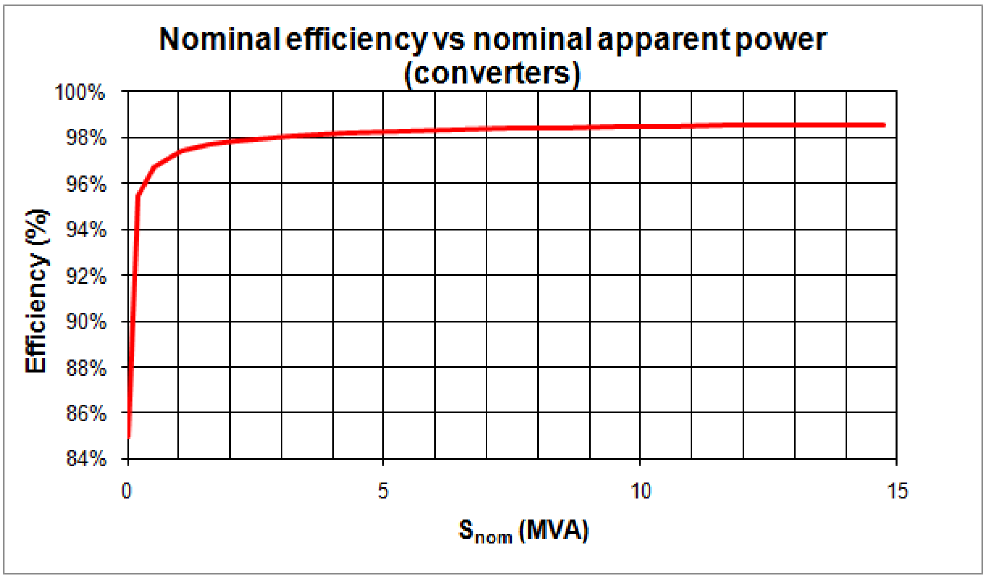


Figure 3.52: Nominal efficiency vs nominal apparent power (MVA) of converters according to model in GES

$$\boxed{\eta_{nom} = \frac{1}{1.01 + \frac{0.01622}{\sqrt{S_{nom}}}}} \tag{3.99}$$

The AES study also has models for part load efficiency of converters. Losses consist of a constant part ($\approx$ 10%), conductor losses proportional with current I ($\approx$ 45%) and switching losses proportional to I^2 ($\approx$ 45%).

$$P_{loss} = P_{loss,nom} \cdot \left(0.1 + 0.45\frac{I}{I_{nom}} + 0.45\frac{I^2}{I_{nom}^2}\right) \tag{3.100}$$

3.7.16 Signatures

Underwater noise

The converter itself produces very little noise, but the type of converter plays a major role in the underwater noise. An AC converter makes an output signal with a sine form. The way this is achieved differs per type of converter. The more perfect this sine form, the lower the higher harmonics in the power supply to the motor. The lower the higher harmonics, the lower the torque ripple on the output shaft of the electric motor. And the torque ripple causes vibrations thus noise. So it is important for the converter to produce as smooth as possible sine form, to have a silent drive.

PWM-converters produce a bad sine form, so are not very good in silent drive. Filtering measures can be taken to reduce the higher harmonics. Synchro- and cyclo-converters produce a much cleaner sine form thus less torque ripple. An example of the output of a PWM-converter and a cyclo-converter are presented in figure 3.53. A DC chopper doesn't create an alternating current and for that reason is able to drive a DC motor very silently.

Electro-magnetic

Converters are electrical components and also produce electro magnetic signatures. As long as the signature is predictable, some filtering measures can be taken. PWM-converters have

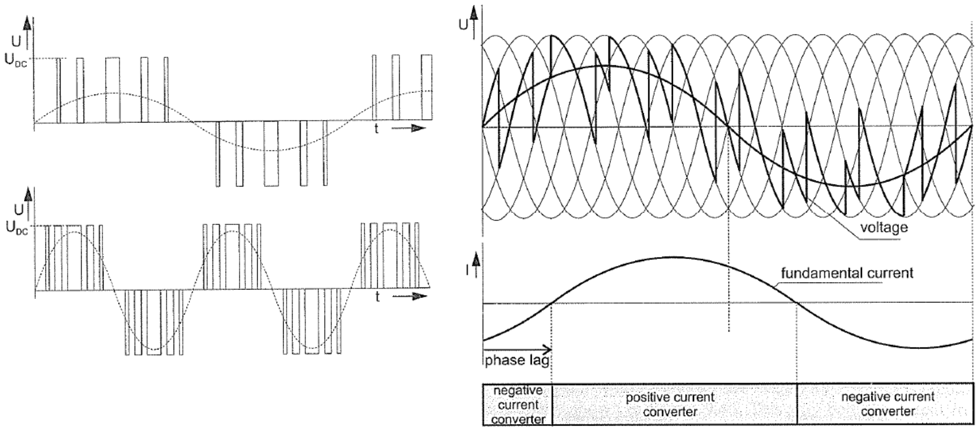


Figure 3.53: Left: output of a PWM-converter, Right: output of a cyclo-converter

(source: KleinWoud & Stapersma (2003))

predictable operating frequency, which may simplify filtering. PFM-converters don't have predictable operating frequencies, so electro magnetic interference (EMI) filtering is more difficult.

3.7.17 Shock resistance

The semiconductors in converters are very delicate pieces of power electronics, but not very sensitive to shockwaves. On the other hand, the cooling equipment in a converter is much more sensitive to shock, and if the cooling fails, the converter will fail. Converters are often placed on springs in ships with high shock requirements.

3.7.18 Maintainability

Maintenance on converters is low, compared to other components.

3.7.19 Reliability

The number of components in a converters says something about the reliability of the converter. The cyclo-converter has a large number of semiconductors, 12 per motorwinding. The PWM-converter only needs half of the semiconductors, 6 per motorwinding. The synchro-converter has the least number of semiconductors, only 4 per motorwinding. So one could say that the synchro-converter is the most reliable, followed by the PWM-converter, and the cyclo-converter has the lowest reliability.

The AES study mentions numbers for MTBF of PWM-converters: 12 years for IGBT and 10 years for GTO/IGCT. A MTTR of 5 hours is mentioned. It is hard to tell if these numbers are reasonable. Development on these power electronics goes fast, and the MTBF numbers might very well have increased.

3.7.20 Initial purchase costs

The initial purchase costs of converters are estimated with a Cost Estimating Relationship (CER). The Cost Analysis section at DMO has determined such a CER based on historical data from purchases and quotations. This CER is commercially confidential. The AES study also mentions some relationships. Distinction is made between IGBT converters and GTO/IGCT converters. These are a function of the converter power. Numbers are converted to €and corrected for inflation at a rate of 2% per year, because they date from 1998.

- IGBT: $\left(59 + \frac{1174}{5+P_{[MW]}}\right)$ €/kW

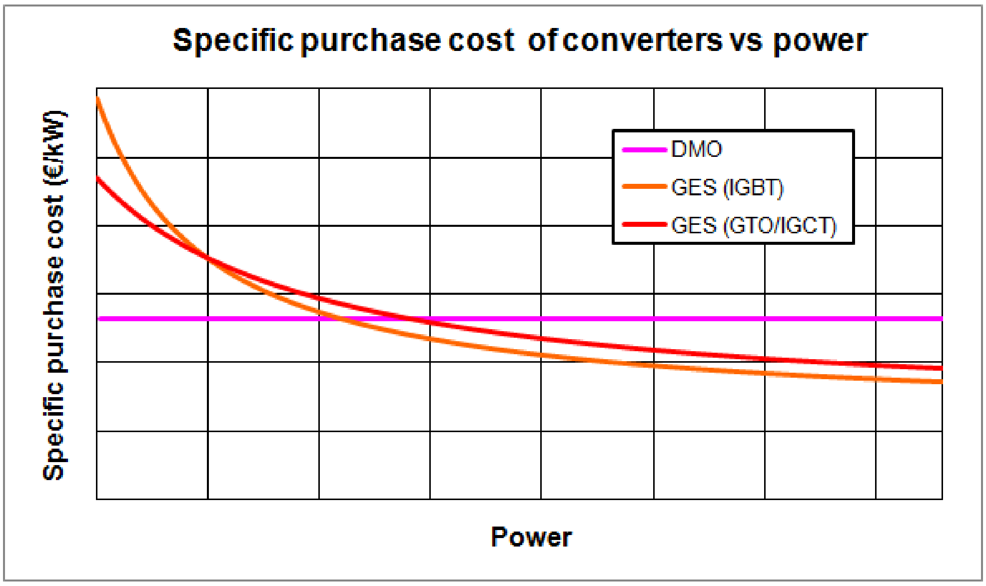


Figure 3.54: Specific purchase costs (€/kW) of converters vs power, according to different sources. Cell height is 50 €/kW

- GTO/IGCT: $\left(59 + \frac{1761}{10+P_{[MW]}}\right)$ €/kW

The CER's are presented in figure 3.54 together with the CER from DMO.

3.8 Cooling system

In the previous sections the main components to set up a power and propulsion plant are described. All components have heat dissipation which needs to be cooled away. There are two 'infinite' heat reservoirs available on a ship to which the dissipated heat can be released: environmental air and water. High power systems, as discussed in previous sections, are normally water cooled, because water has a higher heat capacity than air (≈ 4.2 vs ≈ 1.01 (kJ/kg·K)). Mostly the components are directly water cooled, but some systems indirectly, with air as the intermediate medium (especially in electronic equipment). The dissipated heat from the components can not be released directly to the seawater, because the temperature is far from constant and the salinity causes corrosion to the components. So, an intermediate fresh water cooling system is introduced. The fresh water cooling system is cooled with sea water via sea water coolers (normally plate heat exchangers) to a temperature of normally $38°C$. An overview is given in figure 3.55.

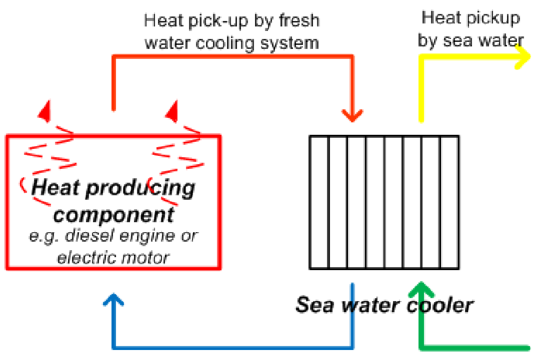


Figure 3.55: Schematic overview of cooling circuits onboard

The fresh water cooling system, and the sea water cooling system are mentioned, but a third cooling system can be distinguished: the chilled water system. The chilled water plant 'produces' water of normally $7°C$ that is used for cooling of SEWACO- and HVAC[11]-systems. In the cooler of a chilled water plant, the heat is withdrawn from the chilled water distribution system by evaporating a refrigerant (normally R134a). In a condenser the heat is transferred to the sea water cooling system. Normally the water in the chilled water distribution is not pure water but has some additives (like Glythermin®) to enhance heat capacity and protect from frost and corrosion. A chilled water plant has significant space consumption. As an example, some details about the chilled water plant of the LCF (consisting of 3 chilling sections) are listed:

Max. cooling capacity: 750 (kW)

Length x Width x Height: 4.2 x 2.4 x 2.0 (m)

Mass (excl. fluids): 9000 (kg)

Note: Because the chilled water plant has no direct connection to the choice of propulsion concept it is not further treated in this thesis. The fresh water and sea water cooling systems are inherently coupled to the propulsion machinery. Still, these systems are not further analyzed in this thesis; because these cooling systems are needed anyway, no matter what propulsion concept is chosen, and the difference in dimensions, weight, costs etc. per propulsion concept are assumed to be more or less constant. In comparing propulsion concepts it does not make sense to take into account a constant factor.

[11] Heating, Ventilation, Air Conditioning

3.9 Propeller

A propeller converts the rotating motion of the shaft into a translating motion of the ship, by creating a thrust force on the water. The rotation of an airfoil-shaped blade through the water creates a pressure difference between forward and rear surface, which makes the water accelerate and create a thrust force. The resultant velocity (v_R) and the angle of attack (α) of the water on the propeller blades are very important parameters in determining the thrust force (T) and the torque (Q)that is created.

The velocity that meets the propeller blade is the resultant of the advance velocity (v_A) and the circumferential speed of the propeller ($2\pi r \cdot n_p$). The angle of attack is the difference between the pitch angle (θ) of the propeller blade and the resultant flow angle (β) of the water on the blade. The thrust force and the torque(-force) are the axial and tangential resultants of the lift force (L) and the drag force (D) acting on the propeller blade. A schematic overview of all velocities, angles and forces is given in figure 3.56, copied from KleinWoud & Stapersma (2003).

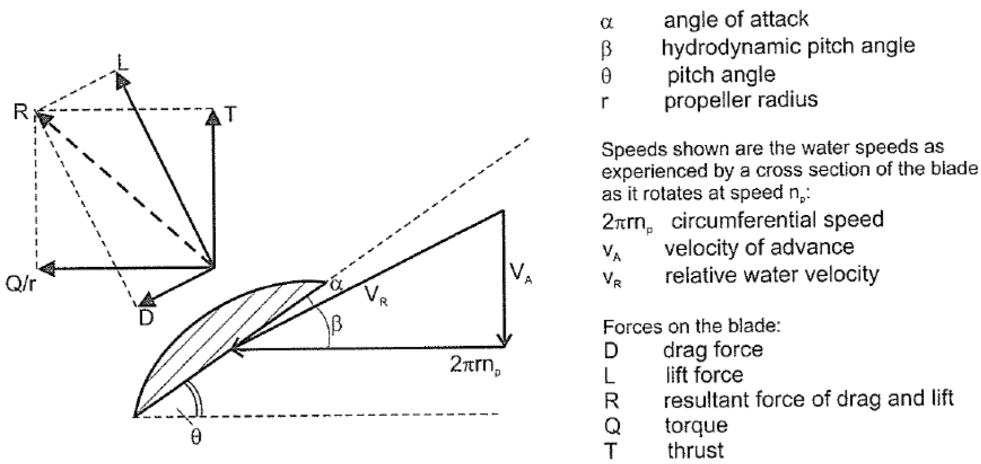


Figure 3.56: Propeller flow velocities and forces and angles on the blade

(source: KleinWoud & Stapersma (2003))

Performance

Performance of a propeller is normally expressed in three non-dimensional parameters (K_T,K_Q,J) and presented in a so-called open water diagram. An example is given in figure 3.57. K_T is the thrust coefficient and K_Q the torque coefficient. These are the thrust c.q. the torque made non-dimensional with the propeller rotational speed n_p, the propeller diameter D and the sea water density ρ, according to equation 3.101 and 3.102. J is the so-called advance ratio, which is the advance velocity v_A made non-dimensional with the circumferential speed op the propeller, see equation 3.103

$$K_T = \frac{T}{\rho \cdot {n_p}^2 \cdot D^4} \tag{3.101}$$

$$K_Q = \frac{Q}{\rho \cdot {n_p}^2 \cdot D^5} \tag{3.102}$$

$$J = \frac{v_A}{n_p \cdot D} \tag{3.103}$$

Also presented in the open water diagram is the open water propeller efficiency (η_O), given by equation 3.104. The open water efficiency is the efficiency of the propeller operating in undisturbed open water. Interaction with the hull is not included.

$$\eta_O = \frac{1}{2\pi} \cdot \frac{T \cdot v_A}{Q \cdot n_p} = \frac{1}{2\pi} \cdot \frac{K_T \cdot J}{K_Q} \tag{3.104}$$

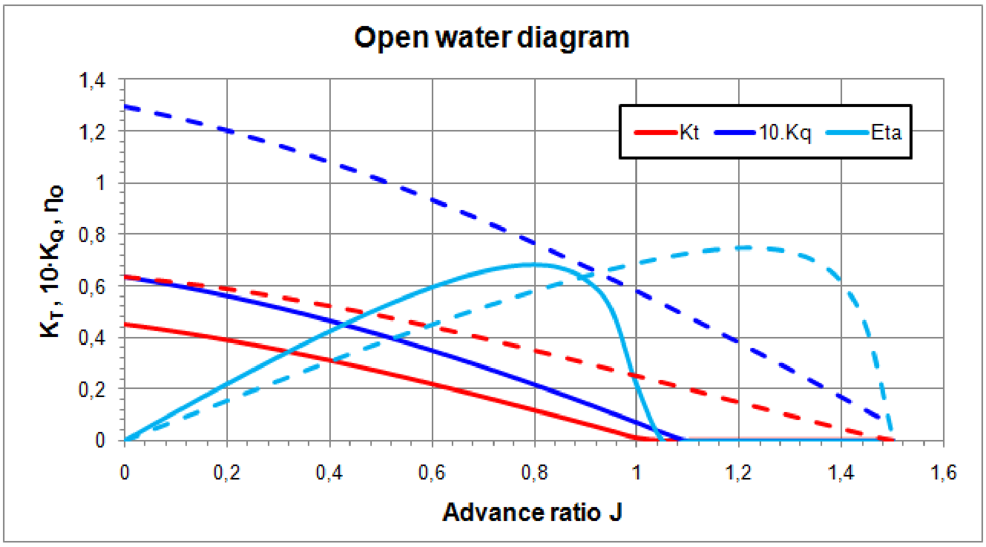


Figure 3.57: Propeller characteristics in open water diagram: Wageningen B5-75; full line P/D=0.96, dotted line P/D=1.4

The open water diagram in figure 3.57 is based on a propeller from the Wageningen B series. This is a series of propellers of which the characteristics are widely available (from MARIN) and said to be representative for all propellers. In Bernitsas *et al.* (1981), a general model is described with which the characteristics of all propellers from the Wageningen B series can be generated. Figure 3.57 is generated with the use of this model. The characteristics from the Wageningen B-series are still widely used nowadays. A few drawbacks of using these characteristics for today's projects are: propeller skew is not up-to-date, currently propellers with skew up to 50° are produced, the constant pitch distribution ratio and the restricted pitch ratio.

The *propeller diameter*, *speed* and *pitch* are the three main important parameters in propeller design that influence the performance, but also the number of blades , the blade thickness, blade-area and the Reynolds number. In figure 3.57 the open water performance is presented for two different values of the pitch to give an idea of the influence of pitch on the performance. Pitch is given as P/D. In which P is the distance a screw propeller with pitch angle θ would advance during one revolution, equal to $2\pi r \tan\theta$. D is the propeller diameter. The *diameter* is a fixed value that is well chosen in the design phase, taking into account a.o. hull form, maximum draught. The rotational *speed* of the propeller can be adjusted during operation, by in- or decreasing the torque of the engine/motor. But engines/motors have a certain operating enveloppe in which the load must be to prevent overload. The *pitch* was also a fixed value chosen in design phase, at least that was the case for a long time. Engineers found a way to adjust the pitch angle of a propeller. This gives rise to two different propeller types:

1. Fixed Pitch Propeller (FPP)

2. Controllable Pitch Propeller (CPP)

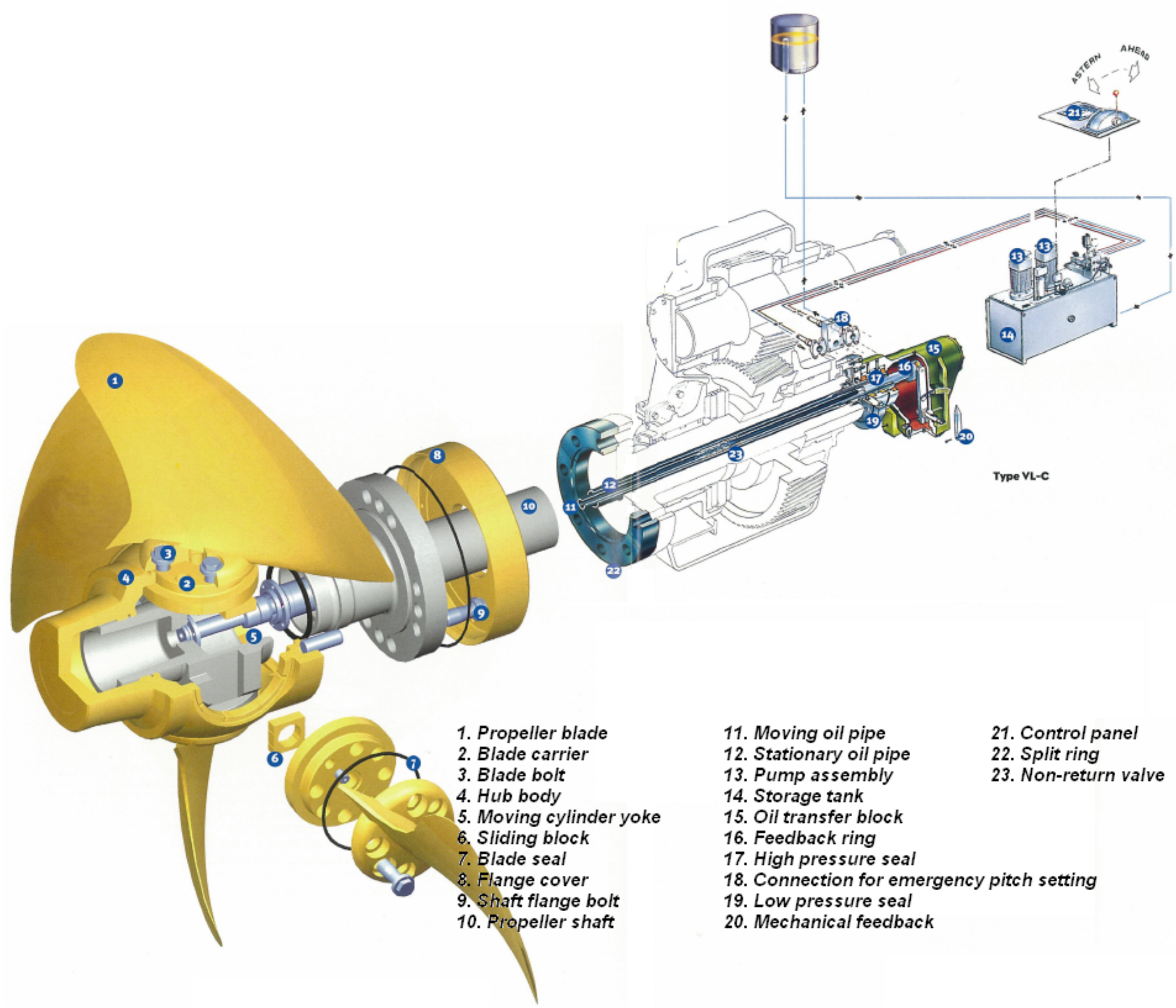


Figure 3.58: Schematic overview of controllable pitch propeller (CPP) installation

(source: JohnCrane-Lips (2001))

Controllable Pitch Propeller

A schematic overview of a CPP system is copied from JohnCrane-Lips (2001) and presented in figure 3.58.

With controllable pitch the performance of the propeller can be adjusted to the conditions. With a FPP, the only controlled variable is fuel flow to the engine. With a CPP there is an additional degree of freedom, in that thrust can be varied by changing propeller pitch. In this way, the engine speed can be adjust such that the engine can always operate in particular regions of its performance curve. An explanation with example from KleinWoud & Stapersma (2003): when the ship is in heavy seas or towing another ship, the velocity of advance decreases, the propeller speed stays the same, so the relative water velocity decreases slightly and the angle of attack increases. The increased angle of attack results in an increased lift force generated by the blade. This wil increase the thrust, and will increase the torque. This might cause an unfavourable situation for the engine as it has to develop a higher torque at the same rotational speed. This is a problem for diesel engines (especially when turbocharged). Electrical motors and gasturbines don't have this problem because they can deliver high torque at low speeds, so this problem does not apply for these machines. When diesel engines are used and a high grade of maneuverability is required, a CPP is inevitable.

With a CPP it is also possible to go astern without reversing the direction of rotation of the shaft. Just set the pitch to negative values and a reverse thrust for braking or going astern is created. The CPP improves maneuverability of a vessel driven by diesel engine or gasturbine. With the controllable pitch it is possible to fast change the propulsion direction. A reversing gear or reversible engine is not necessary when a CPP is used. Again this argument does not apply for ships with electrical propulsion, because an electrical motor can easily be reversed (four quadrant operation). An improvement to the maneuverability that holds for every propulsion motor, is the increased low-speed maneuverability with a CPP. If a FPP is installed, it is hardly possible to reduce the shaft speed until the ship speed is zero, because this causes problems with the lubrication of the shaft bearings at low speeds.

The system that controls the pitch is a hydraulic system consisting of a servo valve, hydraulic piston and a mechanical linking system, see figure 3.58. Hydraulic piston is normally incorporated in the propeller hub (nr. 5 in figure 3.58), which is the reason a CPP hub is bigger than a FPP hub. The servo valve is located inboard. The hydraulic oil is supplied to the hub through piping in the hollow propeller shaft.

A difficulty, but at the same time a great opportunity, is the control of a controllable pitch propeller. Control can be very simple to very complicated; the more complicated, the more input parameters thus measuring sensors. Simple way is only taking into account the shaft speed. Taking into account the load of the propeller (by measuring torque) or advance velocity or thermal loading of the propulsion engine gives already much more opportunities for good control. The desired pitch of the propeller is captured in a combinator curve, and depending on the input parameters (e.g. shaft speed, torque, exhaust gas temperature etc.) a certain value for the pitch is chosen from the combinator curve. The control of the pitch can be designed for optimum propulsive efficiency, constant shaft speed, preventing engine overload but also for optimizing cavitation free time, which is very interesting for warships with stringent requirements to signature profile. Vrijdag (2009) describes the study on this type of pitch control. The control strategy determines which input parameters are necessary.

Cavitation

Cavitation is a phenomenon that occurs when a propeller is vibrating and rapidly rotating in a liquid. Behind the rotating propeller blade a low pressure region occurs. Locally, the water starts to expand and even vaporize due to the low pressure (i.e. cold boiling), which causes bubbles. This is called cavitation inception, and the minimum speed at which this occurs is called cavitation inception speed. When the bubbles again enter higher pressure areas, the bubbles implode. The implosion causes strong shockwaves, which are powerfull enough to damage the propeller blades. Damage caused by cavitation is called cavitation erosion. Besides the damaging effect to the propeller blades, cavitation also has a bad effect on the underwater signature of the ship. The implosion of the bubbles causes a lot of underwater noise. Controllable pitch propellers are more prone to cavitation than fixed pitch propellers. Proper design of the propeller can increase the cavitation inception speed. Values of cavitation inception speeds are often found in the range 9-15 knots. A way to reduce cavitation is by placing the propeller in a decelerating duct. The pump-jet. The propeller is is fitted in a non-rotating nozzle. The nozzle reduces the inflow velocity of the water, whereby the pressure is increased. Increased pressure reduces cavitation. The blade area ratio, A_E/A_0, also has great influence on cavitation behaviour. Generally spoken, a larger blade area ratio is better for cavitation behaviour, but it cannot be chosen too large because a too large area ratio will cause thrust breakdown at full power. The formula of Keller gives a handsome indication of the required expanded blade area ratio:

$$A_E/A_0 = \frac{(1.3 + 0.3 \cdot Z) \cdot T}{(p - p_v) \cdot D^2} + k \tag{3.105}$$

With A_E is effective blade area, A_0 propeller disc area, Z number of blades, T is thrust, p is static pressure at the propeller shaft, p_v is vapour pressure of the water, D is propeller diameter and k represents a margin against cavitation which is taken zero for fast naval vessels, Kuiper (2007).

The material of which propellers are made varies a lot. The choice of the material depends on the requirements: weight, costs, cavitation erosion resistance, magnetic properties, ability to cast (for FPP), etc. For FPP often a copper alloy is used, other materials that are found are: nickel-bronze-aluminium alloy (NiBrAl), aluminium, stainless steel, titanium, composite material (a-magnetic).

> Well-known marine propeller manufacturers are: Wärtsilä former Lips BV, Berg propulsion.

Auxiliary systems

A fixed pitch propeller needs no auxiliary systems to operate, except for the shaft that rotates it. A controllable pitch propeller needs (besides the shaft) a hydraulic system, as described before, with a controlling automat.

Data analysis

It is not well described in van Dijk *et al.* (1998) how dimensions and weight of propellers is calculated in GES. Data from a small number of RNLN propeller is used to derive models of dimensions and weight of propellers. Three fixed pitch propellers are selected and four controllable pitch propellers.

Fixed pitch: Walrus class, LPD-1 (*Hr. Ms. Rotterdam*), JSS (*Hr. Ms. Karel Doorman*)

Controllable pitch: OPV, M-frigate, LCF, AOR (*Hr. Ms. Amsterdam*)

3.9.1 Available power

As explained in the previous section the delivered (or absorbed) power of the propeller depends on diameter, speed and pitch. Speed, and in case of CPP pitch, are variable, but diameter is chosen in design phase. In the design phase, the maximum delivered power of the propeller is determined and the propeller diameter is calculated with the help of open water diagrams. If the calculated diameter is too large for reasons of draught, it is an option to increase speed or pitch, but it will always be a trade off with propeller efficiency. Another option if calculated diameter is too large, is going for more propellers. In warships it is very common to have at least two shaftlines for redundancy reasons, but also to limit propeller diameter and still being able to deliver a lot of power.

Propeller power is depending on the diameter and the maximum propeller loading. A figure from a propeller manufacturer is presented in figure 3.60 with numbers of propeller loading for different ship types. The maximum loading on a propeller depends on the design and the material. The largest fixed pitch propellers that is available has a maximum output of 66 MW and weighs 94.5 tons. Current controllable pitch propeller designs can tolerate only a maximum output of 44 MW, according to the Wikipedia site on CPP's. Propeller power of CPP's is limited, because of the bolted-on rotatable blades and the supply of hydraulic oil to control the pitch.

3.9.2 Dimensions

The important dimension in propeller design is the diameter. As explained in the previous subsections, the propeller diameter necessary for delivering a certain amount of power, depends a.o. on the shaftspeed (n), the pitch ratio (P/D), number of blades (Z) and the blade area ratio (A_E/A_0). The propeller diameter can be estimated by fitting the dimensionless ships required thrust curve ($K_{T,ship}$) in the K_T-diagram of a propeller. Both K_T curves can be plotted as a function of diameter. The intersection of $K_{T,ship}$ and $K_{T,prop}$ indicates the required diameter. In this method a lot of assumption have to be made.

With the models described in Bernitsas *et al.* (1981) the open water characteristics of any Wageningen B series propeller can be generated as a function of J if P/D, Z, A_E/A_0 are chosen. J is a function of v_A, n and D according to equation 3.107c. With fixed values of v_A and n, J is directly related to D, so K_T of the propeller can be plotted as function of D. The ships K_T-curve can also be plotted as a function of D, with T_{ship}, ρ and n constant (at nominal point) and varying D, according to equation 3.106. The intersection point between $K_{T,ship}$ and K_T of the propeller gives the required propeller diameter for the given conditions. An example of this method is presented in figure 3.59 for the SFC vessel at 30 knots with 2 shafts at 200 rpm with Wageningen B5-65 for two different values of P/D, red: $P/D = 0.9$ and lightblue: $P/D = 1.4$. From this figure can be concluded that for $P/D = 0.9$ the propeller diameter should be 5m and for $P/D = 1.4$ should be 4.2m. With this method the minimum required propeller diameter in the nominal point can be determined.

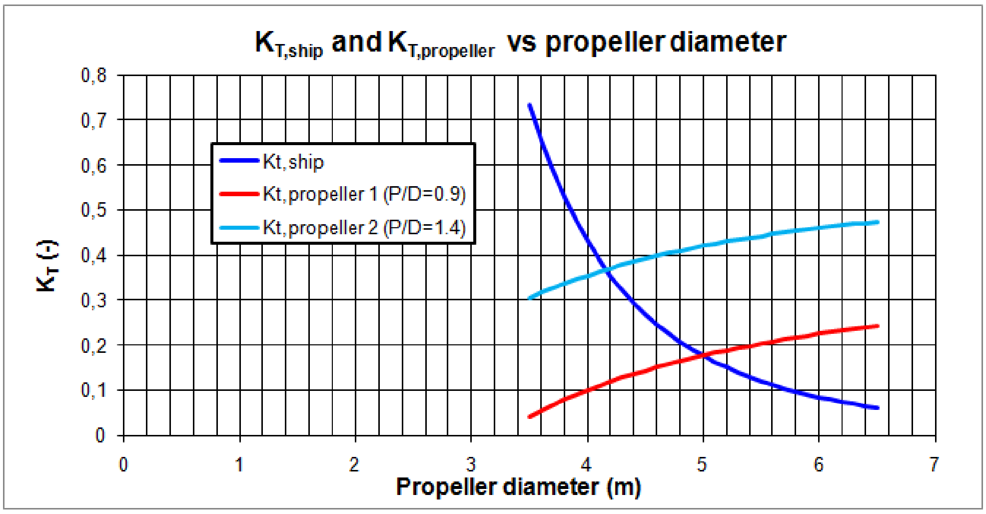


Figure 3.59: Propeller diameter determination in K_T vs D plot, with data from table 2.1 at 30 knots, with 2 shafts at 200 rpm with Wageningen B5-65, red: $P/D = 0.9$ and lightblue: $P/D = 1.4$

$$K_{T,ship} = \frac{T_{ship}}{\rho \cdot n^2 \cdot D^4} \tag{3.106}$$

With:

$$T_{ship} = \frac{R}{k_p \cdot (1 - t)} \tag{3.107a}$$

$$R = c \cdot {v_s}^2 = c \cdot \left(\frac{v_A}{1-w}\right)^2 \tag{3.107b}$$

$$J = \frac{v_A}{n \cdot D} \tag{3.107c}$$

it follows that:

$$K_{T,ship} = \left(\frac{1}{\rho \cdot D^2} \cdot \frac{c}{k_p \cdot (1-t) \cdot (1-w)^2}\right) \cdot J^2 \tag{3.108}$$

In which R is ship resistance, k_p is number of propellers, t is thrust deduction factor, c is a factor that describes the relation between ship resistance and squared ship speed (is often assumed constant in early stage design and determined at nominal point), v_A is advance velocity, w is the wake factor.

The above described method is rather intensive. Another possibility is to catch propeller power density in a ball-park figure. In figure 3.60 it is shown that a mean value for power density of propellers can be determined. From this chart 1.3 MW/m^2 is a representative power density value for frigates. With this value the diameter D can be calculated, with a known propeller power P_p. In this figure are also presented the power densities of some RNLN ships based on brake propulsion power P_B and propeller disc area A_0: $\frac{P_B}{A_0}$. The effective blade area ratio A_E/A_0 is not taken into account because it is not clear if this was taken into account in the figure from JohnCrane-Lips (2001). Because the values of the RNLN propellers are structurally lower than the indicated values from the propeller power chart, the values should probably be corrected for the effective blade ratio. A representative value for A_E/A_0 of frigates would be between 0.6 and 0.7.

From the RNLN data, a representative value for the (disc area) propeller loading of a SFC type of ship would be **1 MW/m^2** (which is the mean value of M-frigate and LCF). The propeller power density as determined by the earlier described method with the minimal diameter is 1.67 MW/m^2. But for better efficiency and better cavitation characteristics it is best to go for a lower propeller loading, thus larger diameter. The maximum propeller diameter is limited by the design of the aftship or draught restrictions (Den Helder is approximately 8m). A general rule of thumb is to go for as large propeller diameter as possible for highest propeller efficiency and the lowest blade loading, but with the number for propeller loading (x) it can also be estimated with equation 3.109:

$$\boxed{D = \sqrt{\frac{P_B}{x} \cdot \frac{4}{\pi}}} \tag{3.109}$$

Based on the RNLN data and figure 3.60, three (disc area) propeller loading conditions are determined:

- Low propeller loading $x = 0.5$ MW/m^2
- Medium propeller loading $x = 1$ MW/m^2
- High propeller loading $x = 1.5$ MW/m^2

Remarkable is that the FPP's are in the range of low propeller loading, and CPP's in medium to high propeller loading.

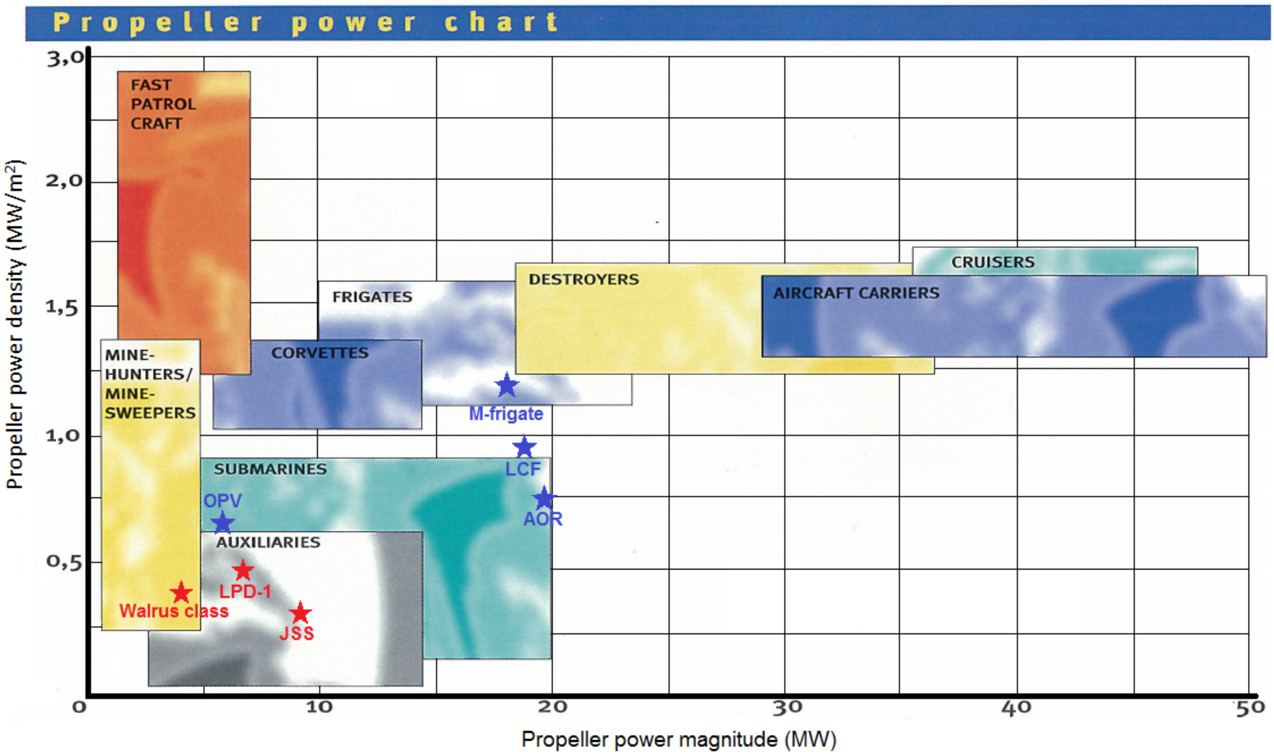


Figure 3.60: Propeller power density chart for several naval ship types including the data from some RNLN ships (red: FPP, blue: CPP)

(source: JohnCrane-Lips (2001), edited by author)

3.9.3 Weight

Weight of a propeller depends on many parameters. The diameter, the number of blades, the propeller type (CPP or FPP), manufacturer, blade thickness, material etc. The type of propeller is a large weight-driver, because for a CPP the hub contributes to more than half of the propeller weight. The material (NiBrAl) and the number of blades (5 blades) are pretty constant for propellers on the RNLN vessels. The number of blades is primarily determined by the need to avoid harmful resonant frequencies of the ship structure and the machinery. The diameter and the blade thickness depend on the power the propeller has to deliver. Blade thickness is increasing the last years for reasons of reducing cavitation.

So, based on the things above, there is a strong feeling that the propeller weight should be a function of power and diameter, with a distinction between CPP and FPP. The weights of the RNLN propellers are put in figure 3.61 as a function of propeller loading, but no clear trend is recognized.

Figure 3.62, shows the result of propeller weight versus propeller diameter (including trendlines). In this figure the power loading of the propeller is not taken into account, and weight is related to only the diameter. This relation is much more clear than the relation with power loading. It seems more useful to use a relation between weight and solely diameter.

FPP:

$$\boxed{\text{Weight} = 337 \cdot D^{2.3}} \tag{3.110}$$

CPP:

$$\boxed{\text{Weight} = 39 \cdot D^{3.9}} \tag{3.111}$$

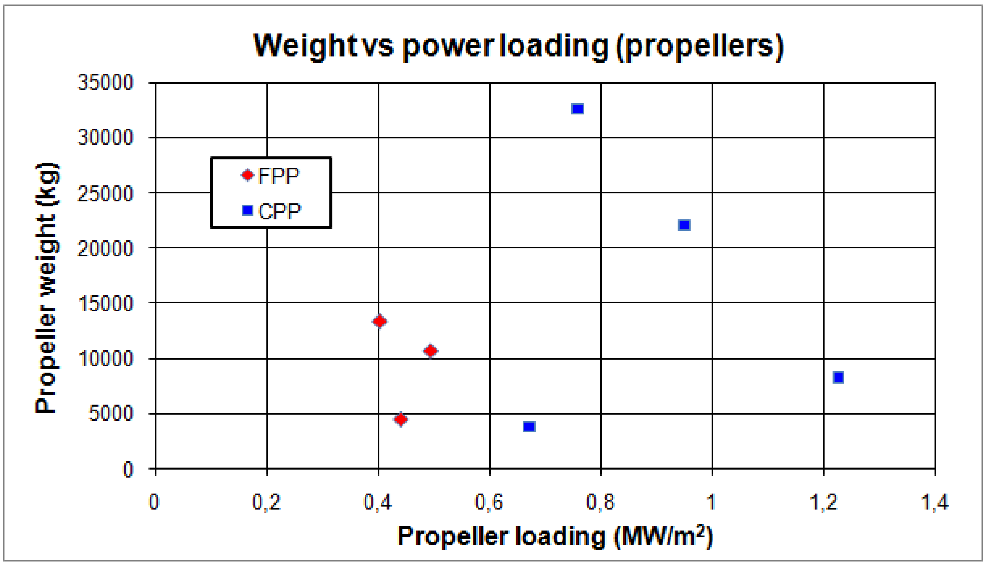


Figure 3.61: Propeller weight (kg) of some RNLN ships vs propeller loading (MW/m^2)

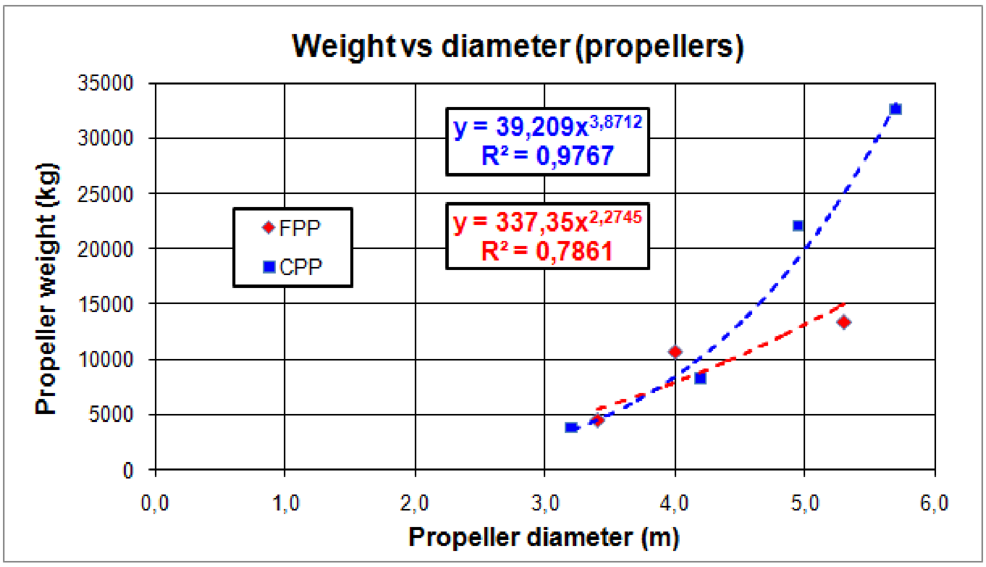


Figure 3.62: Propeller weight (kg) of some RNLN ships vs propeller diameter (m)

Valid for diameters between 3 m and 6 m, with D is propeller diameter in meters and weight in kilograms.

3.9.4 Operating speeds

The operating speed is closely related to the diameter of the propeller. If the diameter is chosen smaller, the maximum rotational speed of the propeller (1) *must* be higher to deliver the same amount of power, and (2) *can* be higher and still be below a limit value of circumferential speed. (1) To generate a certain thrust force, an amount of water must be accelerated by the propeller; if the disc area is smaller, the speed must be higher to generate the same thrust force. (2) For reasons of cavitation and shockwaves, circumferential speed of the propeller can not exceed a certain limit value. If the operating speed of the propeller is too high it can not deliver thrust anymore and efficiency decreases dramatically until zero. Speed can also not be too low, because than it can not deliver thrust as well.

In general, the designer will pick a propeller with as large as possible diameter (determined by the aftship and draught restrictions) for having the best propeller efficiency, and will then determine the propeller speed with the highest efficiency from the open water diagram. With this data an engine and/or gearbox can be picked.

3.9.5 Efficiency

The propeller efficiency depends, especially, on the speed of advance v_A, thrust force T, rate of revolution n, diameter D and, moreover, on the design of the propeller, i.e. the number of blades, disk area ratio, and pitch/diameter ratio. The propeller efficiency can vary between approx. 35% and 75%, with the high value being valid for propellers with a high speed of advance v_A. In MANB&W (2010) a picture was found with propeller efficiencies in an open

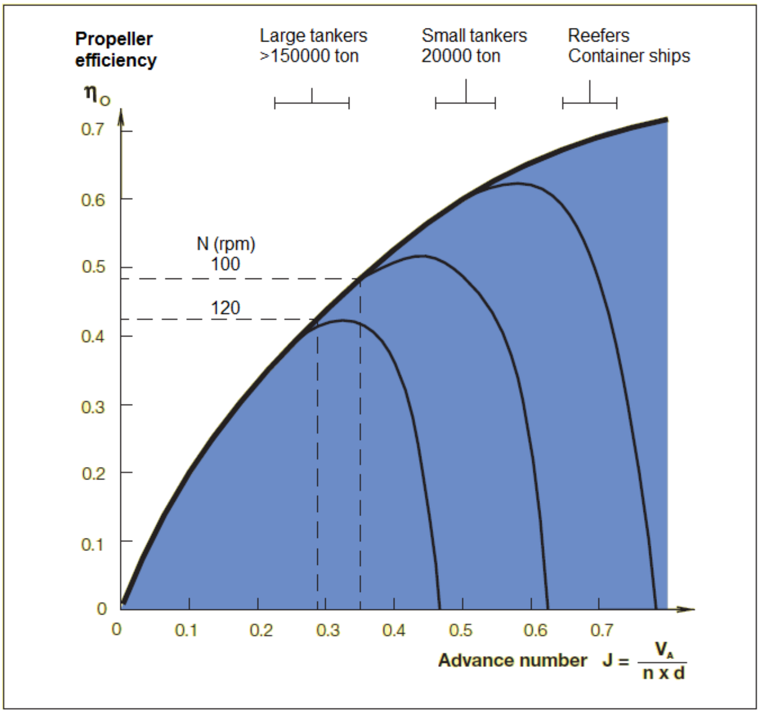


Figure 3.63: Obtainable propeller efficiency for some ship types

(source: MANB&W (2010))

water diagram of some shiptypes, see figure 3.63.

A method to calculate open water efficiency of a propeller is to calculate the ideal axial efficiency η_i with the non-dimensional thrust loading coefficient C_T, as mentioned in chapter 10 of KleinWoud & Stapersma (2003), and apply a certain correction factor. A commonly used correction factor for a state of the art propeller is 0.175.

$$\begin{aligned}\eta_O &= \eta_i - \text{correction} \\ &= \frac{2}{1+\sqrt{1+C_T}} - 0.175\end{aligned} \tag{3.112}$$

With:

$$C_T = \frac{T}{\frac{\pi}{4}\cdot D^2 \cdot \frac{1}{2}\cdot\rho\cdot {v_A}^2} = \frac{K_T\cdot\rho\cdot {n_p}^2\cdot D^4}{\frac{\pi}{4}\cdot D^2\cdot\frac{1}{2}\cdot\rho\cdot {v_A}^2} = \frac{8\cdot K_T}{\pi\cdot J^2} \tag{3.113}$$

In this equation T is thrust, D is propeller diameter, ρ is water density, v_A is advance speed, K_T is non-dimensional thrust coefficient, n_p is propeller speed and J the advance coefficient. The result of this approach is presented in figure 3.64, the lightblue line gives efficiency of a Wageningen B-series described with the polynomials from Bernitsas *et al.* (1981) and the green line gives the efficiency described with equation 3.112. It can be seen that this approach does not describe the efficiency very well, though it is often used this way.

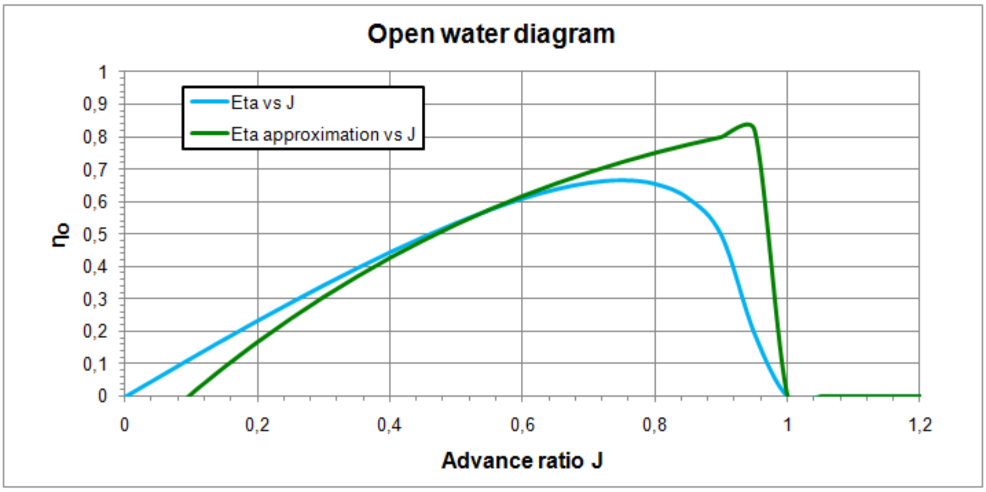


Figure 3.64: Propeller efficiency of Wageningen B-565 vs advance coefficient J according to Bernitsas *et al.* (1981) and approximation according to equation 3.112

A general rule of thumb is: the larger the propeller diameter, the lower the the thrust loading on the propeller, the higher the maximum efficiency will be. In general, maximum efficiency of a CPP is lower than of a FPP. A FPP is more efficient than a CPP under a specific rotational speed and load condition. At that particular rotational speed and load, a FPP can transmit power more efficiently than a CPP of equal pitch and diameter. This is mainly because of the larger hub diameter of the CPP. Also, the efficiency of a given blade whose pitch has changed is less than the efficiency of a propeller that has been designed for that pitch. At other rotational speed or loading the FPP will be either over-pitched or under-pitched, resulting in not being more efficient anymore. KleinWoud & Stapersma (2003) states that unless the load characteristics of the ship are varying strongly, it may not be expected that a CPP improves the propulsive efficiency.

Part load efficiency of a certain propeller can be read from its open water diagram as a function of the advance ratio J. The CPP gives the highest propulsive efficiency over a broad range of speeds and load conditions because it can adjust the pitch to optimize the efficiency at every condition. Nevertheless, a CPP needs a decent controller that chooses the right pitch at every load condition to really be more efficient. With bad control the desired effect is lost.

An advantage of the CPP is the 'vane'-stance, which is useful with combined motor vessels. The pitch is adjusted such that the propeller gives the least water resistance when not using the propeller (e.g. trailing shaft).

3.9.6 Signatures

Propeller noise has a great part in the underwater noise signature of a ship. Cavitation is the main reason for underwater noise. A proper designed propeller which has a high cavitation inception speed is very important in reducing underwater noise. The underwater noise caused by cavitation is very loud and has a high frequency (10^3 to 10^4 Hz). But also below cavitation inception speed the propeller produces noise. Typically in the low frequency range (10^1 Hz), dictated by the blade rate (speed and number of blades).

A CPP is more prone to cavitation than a FPP, which gives the CPP a disadvantage over FPP. The blades of a FPP can be optimized for as high as possible cavitation inception speed. Fixed pitch propellers have higher cavitation inception speeds. A propeller with a cavitation inception speed of 16 to 18 knots is most representative for a state-of-the-art propeller design. But caviation inception speeds up to 20 knots are achievable nowadays.

To increase the cavitation inception speed besides a proper propeller design, the outboard part of the propeller shafts need to be enclosed by a fairing[12] in order to avoid disturbances of the inflow field of the propeller. An advanced type of propeller with very good cavitation reducing properties is the so-called skewback propeller. The blades look like a saber: the blade tips of a skewback propeller are swept back against the direction of rotation. In addition, the blades are tilted rearward along the longitudinal axis, giving the propeller an overall cup-shaped appearance. This design preserves thrust efficiency while reducing cavitation, and thus makes for a quiet, stealthy design. Another cavitation reducing method is the pump-jet, as mentioned before. By the design of the enclosing nozzle, the inflow velocity of the water is decreased, whereby the pressure is increased. Increased pressure reduces cavitation.

Another design feature that reduces cavitation and noise, and can also be applied on CPP's, is the Pressurized Air System, see JohnCrane-Lips (2001). A pressurised air supply source blows continuous layers of air onto both sides of the propeller blade from tiny holes bored at the leading edge of the blades. Blocking valves prevent sea water from penetrating the system when it is shut down. Though, Hendriks *et al.* (2011), advises against this system, because at low speeds (where this system is not needed) it generates high noise levels when it is not switched off (which is often the case to avoid system degradation). The propeller also causes an EM signature. The rotation of the metal propeller causes an electric field in the seawater which causes ionic currents. Propellers can be coated to lower these underwater-currents. Another option is a composite propeller, which has no magnetic signature, but is very expensive.

3.9.7 Shock resistance

The propeller under the ship is not sensitive to shockwaves. Although, it is a rather thin construction with a large area the shockwave will not have much impact on the construction because the propeller is entirely surrounded by water. Shockwaves have large impact on the ships hull which forms the border between water and air.

[12]Dutch: stroomlijningskap

3.9.8 Maintainability

Maintenance on a propeller is very low. Especially the FPP almost needs no maintenance, there are no moving parts. Every few years it is cleaned from fouling. If the propeller is exposed to a lot of cavitation, the blades might be damaged by erosion and need repair. If the propeller needs repair the ship has to dry-dock.

For a CPP holds the same, but because of the controllable blades some more maintenance is to be done. On a CPP the blades are bolted on the propeller hub. The seals need replacement every few years. Further, some maintenance is needed on the hydraulic plant that adjusts the pitch. An advantage of a CPP is that it is possible to replace blades underwater, see JohnCrane-Lips (2001). So, the ship does not need to dry-dock when a blade is damaged.

3.9.9 Reliability

The reliability of a FPP is very high. There are no moving parts. If the FPP is operated properly within the operating range, the MTBF time goes far beyond the life of the ship. Reliability of a CPP is lower than of a FPP. The moving parts and the pressurized hydraulic lines and couplings are exposed to wear and tear. Still, JohnCrane-Lips (2001) states that MTBF of a CPP is 45 years, which is about the lifetime of the ship.

3.9.10 Initial purchase costs

Purchase costs of the propeller are estimated with the Cost Estimating Relation (CER) that is determined by the Cost Analysis section of the DMO. The CER is a function of the propeller weight. No distinction is made between FPP and CPP. The CER is commercially confidential information, and for that reason is not explicitly mentioned here. It can be found in the confidential appendix of this thesis.

There were no other sources found that give CER's for propellers, so no comparison can be made. MANB&W (2010) mentions that a CPP is 3-4 times as expensive as a FPP. But this is in contradiction with equations 3.110 and 3.111, because a CPP is not 3-4 times as heavy as a FPP.

3.10 Waterjet

This section gives a description of the waterjet and an explanation why this component is not further considered in the propulsion study for the Surface Combatant (SFC) project.

Besides the propeller, the waterjet is a propulsor that is often used. Especially on fast vessels. In a waterjet propulsion system, thrust is produced by accelerating a mass of water, as with propellers. For that reason the waterjet may be considered a special type of propeller in which a high speed rotor or impeller of relatively small diameter is located in a long tubing system. As a matter of fact it is an extreme form of the earlier mentioned pump-jet: a centrifugal pump in a nozzle. A waterjet consist of an inlet, which is a large hole in the hull that directs a huge amount of water to the the pump. The shape of this inlet is very important for the performance of the waterjet, for that reason is designed by the waterjet manufacturer instead of hull designer. The pump consists of a stator and an impeller, that is driven by an engine with a shaft. The impeller can be of the radial or axial type, the latter having smaller diameter and lower weight. Behind the pump is a jet/nozzle that accelerates the water and causes the thrust force. The outlet is just under or just above water level. At the end of the jet a bucket is placed that can be hydraulically steered. In this way the jet of water can be directed and even be reversed (reversing bucket). This bucket is called the jetavator. This gives the ship a very high rate of maneuverability, without rudders. It even cancels out the need for a bowthruster.

A waterjet has some advantages, but also some disadvantages which causes it to not be further taken into account for the SFC project. Some advantages are: high power density with respect to volume, low draught which gives improved shallow water capabilities, very good maneuverability, lower hull resistance because the lack of rudders and other appendages (e.g. shafts, struts), increased cavitation inception speed (due to higher pressure), protection of the rotating element (safer with divers).

Some disadvantage are: less efficient than propellers at low speeds (<25-30 knots), much more expensive than propellers, impact on hull design (inlet, outlet), huge underwater noise.

A waterjet could be considered for this project, but only for high speed operation. Than a propulsion concept like the South-African navy MEKO A class corvettes could be used, the so-called CODAG-WARP propulsion concept (Combined Diesel And Gasturbine - Waterjet And Refined Propeller) presented in figure 3.65. This concept has two CPP's driven by diesel engines for the lower speeds, and a booster waterjet driven by a gasturbine for high speeds up to approximately 30 knots. At low speeds the waterjet is too inefficient in comparison with a propeller (CPP or FPP). For the SFC project there are stringent underwater signature requirements at operational speeds, for Anti-Submarine Warfare (ASW) operations. When a WARP concept would be used on this project, still the underwater noise is significant at low speeds, because of the flow noise of the water past the huge inlet 'hole' in the hull. A solution to this could be to close the inlet with a large lid, but the inlet is so big that this is not considered an option.

The efficiency benefits only pay out for ship speeds above 25-30 knots. The desired speed of the SFC is 30 knots, so it would be on the edge. The expectation is that the ship will sail these high speeds only small part of the time (<5% of time), so the efficiency benefit will probably not cover the extra investment. Plus, it is very unfavourable for the waterjet to stand still, because of fouling. Another very important disadvantage, that makes the waterjet an unlikely option for this project is that the impact on the hull (inlet, outlet) makes the presence of the required launch and recovery ramp for unmanned surface vehicles impossible. An example of the impact on the aftship is given in figure 3.65.

> Some well-known waterjet manufacturers are: Lips (part of Wärtsilä), KaMeWa (part of Rolls-Royce) and HamiltonJet.

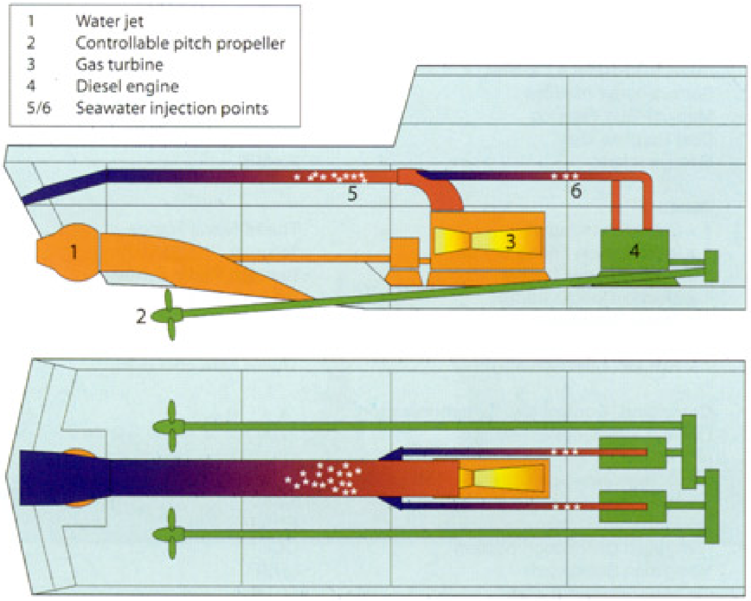


Figure 3.65: Schematic overview of CODAG-WARP propulsion concept on South African Navy MEKO A-200 corvette, with waterline exhaust system

(source: http://www.naval-technology.com/projects/meko/meko4.html)

Auxiliary systems

To operate a waterjet, some auxiliary machinery is necessary. An oil lubrication system and a hydraulic powerpack to steer and reverse the waterjet. Further a control system is needed which translates the inputs given on the bridge to steering commands for the water jet.

3.10.1 Available power

Performance of a waterjet is presented in net thrust force T in (kN), in which the ships speed is an important parameter. In an ideal waterjet, the developed thrust is equal to the change in velocity of the water over the pump times the massflow of water:

$$T = \dot{m} \cdot (v_{out} - v_{in}) \tag{3.114}$$

The ships speed is important in determining the velocity of the water entering the pump, that is why performance diagrams of a waterjet are given as a function of ships speed v_0. An example of such a diagram is given in figure 3.66. The effective thrust power $P_{T,e}$ delivered by a waterjet is calculated with:

$$P_{T,e} = T \cdot v_0 \tag{3.115}$$

Waterjets can be used in a wide power range. For example the Wärtsilä waterjets, the axial pump designs for vessel speeds up to 55 knots is available in power range of approximately 500 kW to 26 MW. The non-axial pump design, for extreme high speeds of 70 knots or more, are available up to approximately 40 MW.

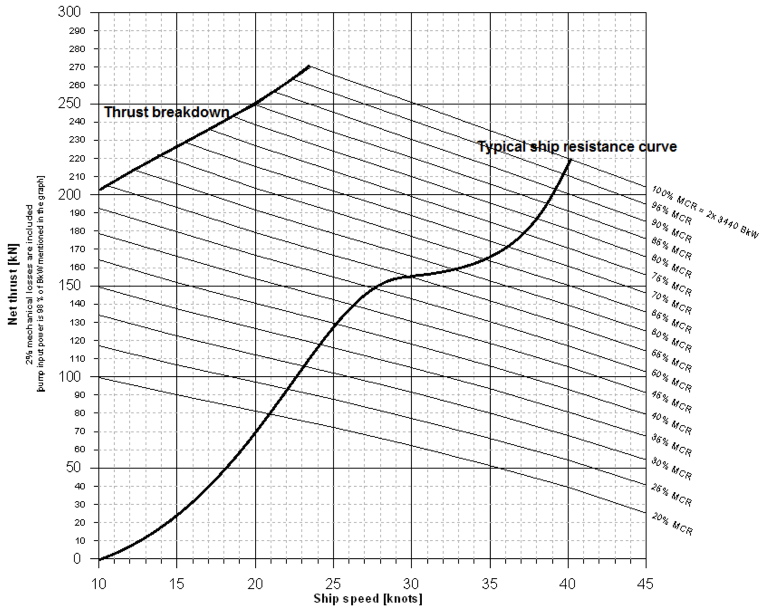


Figure 3.66: Example of operational characteristics of a waterjet including a typical ship resistance curve

(source: figure from Wärtsilä presentation, edited by author)

3.10.2 Dimensions

Dimensions of a waterjet are primarily determined by the impeller diameter, and the impeller diameter is proportional to the power that the waterjet can deliver. The main dimensions of a waterjet that can be distinguished are: the impeller diameter, the jet diameter (also called transom flange diameter), the inboard length of the waterjet including inlet channel, and the outboard length of the waterjet including jetavator. The different dimensions are presented in figure 3.67.

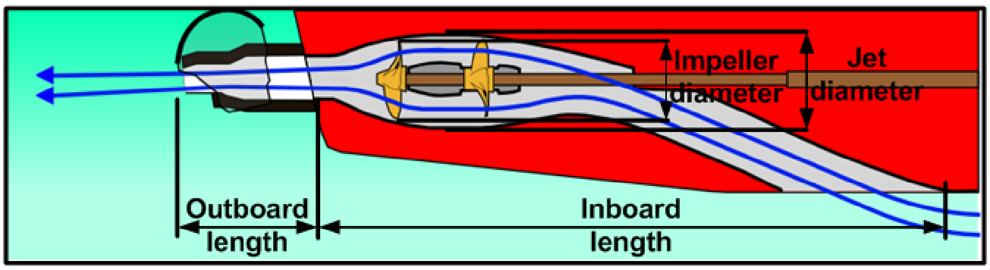


Figure 3.67: Outline of waterjet system with different dimensions

(source: http://nl.wikipedia.org/wiki/Waterjet, edited by author)

From a presentation by waterjet manufacturer Wärtsilä it follows that typically the inboard length of a waterjet is about 8-10 times the impeller diameter and the jet diameter is 28% larger than the impeller diameter for axial waterjets and 69% larger for radial waterjets. The outboard length of a radial waterjet is typically 2.7 times the impeller diameter and for an axial pump type 2.5. Common waterjet types have impeller diameters varying between approximately 0.4-2.1 m. As said before, the impeller diameter is proportional to the power of the waterjet:

$$P \propto c \cdot D^2 \tag{3.116}$$

In which c is a sizing factor of approximately 1 for radial pump types and approximately 0.65 for axial pump types. D is the impeller diameter. A typical impeller diameter for a vessel speed of 40 knots with 5000-6000 kW power is 1m. To give an idea of the sizing. Further analysis on the relations between ships speed, propulsive power and impeller diameter is not done because of the irrelevance of the waterjet for the SFC project.

3.10.3 Weight

Waterjets are heavy. The waterjet unit consists of a substantial mechanical component, plus the inlet assembly. These weights are given by the jet manufacturer in their ctalogs. In addition to the weight of the jet unit, the naval architect must also deal with the weight of the water entrained within the unit, which can be a substantial amount (25600 liters for Wärtsilä LJ200E). The water weight can also be obtained from manufacturers catalogs. The weight of two series of waterjets from Wärtsilä are presented in figure 3.68, and it is concluded that the weight is related to the third power of the impeller diameter. The waterjet system with axial pump type has a lower weight than the waterjet with non-axial pump type.

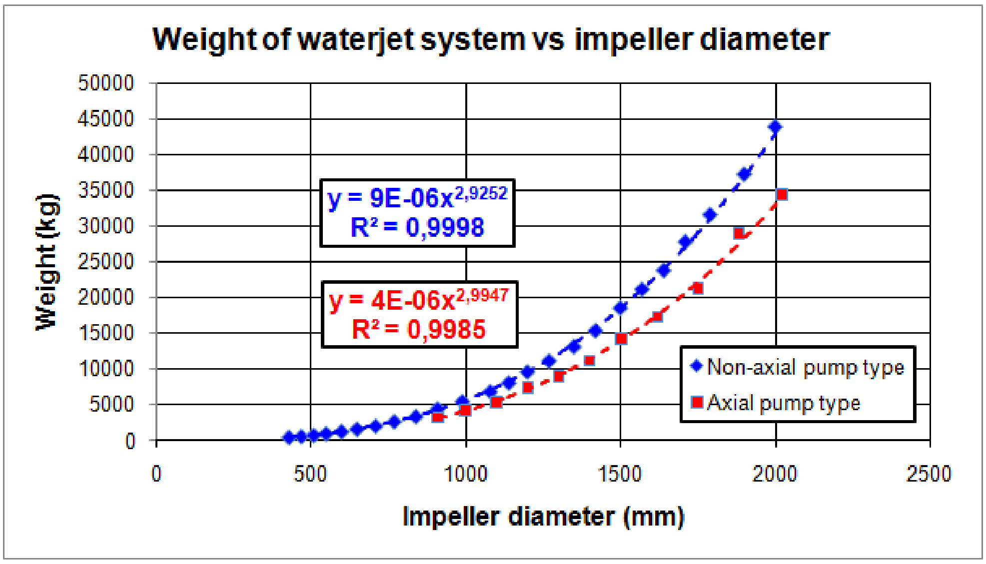


Figure 3.68: Weight (kg) of Wärtsilä axial and non-axial waterjets (incl. jetavator, excl. entrained water) vs impeller diameter (mm)

3.10.4 Operating speeds

The operating speed of a waterjet is higher than of a propeller, because the diameter of a waterjet impeller is significantly smaller than of a propeller that delivers the same thrust. Operating speeds of waterjets vary between 200 rpm for very large waterjets and 2000 rpm for smaller jets on yachts for example. Operating speed of the Lips LJ210E waterjet on the MEKO A-200 corvettes is 300 rpm.

3.10.5 Efficiency

The following analysis on waterjet efficiency is based on van Terwisga (Unknown date) and van Terwisga (1997).

The most important part of the jet system governing its overall efficiency, is the nozzle. This part of the waterjet converts the potential energy (pressure) in the flow, into kinetic energy that produces the thrust. For a given thrust and speed requirement, the nozzle area determines the thrust loading coefficient C_{Tn}, equation 3.117.

$$C_{Tn} = \frac{T}{\frac{1}{2}\rho \cdot {v_0}^2 \cdot A_n} \tag{3.117}$$

With T is the thrust as defined by equation 3.114, ρ is the water density, v_0 is ship speed and A_n is nozzle exit area.

Ideal efficiency η_I of the jet system only depends on the magnitude of C_{Tn}, see equation 3.118. This implies an increase in efficiency with increasing nozzle area. In analogy with propellers, the lower the thrust loading, the higher the efficiency of the system.

$$\eta_I = \frac{4}{3 + \sqrt{1 + 2 \cdot C_{Tn}}} \tag{3.118}$$

Ideal efficiency of conversion from hydaulic power to effective thrust η_I represesents an important part of the overall efficiency. Additional energy losses occur in the ducting system (η_{duct}) and in the pump system (η_{pump}). Pump efficiency o a well designed high efficiency pump has a value of approximately 0.90 and the ducting efficiency adopts values between approximately 0.90 and 0.95. All together this gives an open water efficiency of the waterjet η_O.

$$\eta_O = \eta_I \cdot \eta_{duct} \cdot \eta_{pump} \tag{3.119}$$

Open water characteristics of a waterjet are, like the propeller, plotted in an open water diagram. In such a diagram efficiencies and thrust loading coefficient are plotted versus Nozzle Velocity Ratio NVR. This is a coefficient characterizing the jet system working point.

$$NVR = \frac{v_n}{v_0} \tag{3.120}$$

With v_n is average nozzle velocity in the direction of the nozzle centreline and v_0 is ship speed.

At high speeds (>30 knots) the efficiency of a waterjet exceeds the values that are reached with propellers. With propellers, efficiencies of about 75% are reached, but for higher vessel speeds, above 30 knots, this value drops rapidly. This can also be seen in figure 3.63: above a certain value of the advance number J (which is the non-dimensional speed) the efficiency drops. Typical efficiency curves for a conventional propeller and a waterjet system are presented in figure 3.69. This figure gives an indication on the relative efficiencies of the two propulsors versus the ship speed.

Overall efficiencies of waterjets are in the range of 60-70%. Some numbers from Wärtsilä: at 60 knots 72%, 45 knots 69%, 55% around 20 knots.

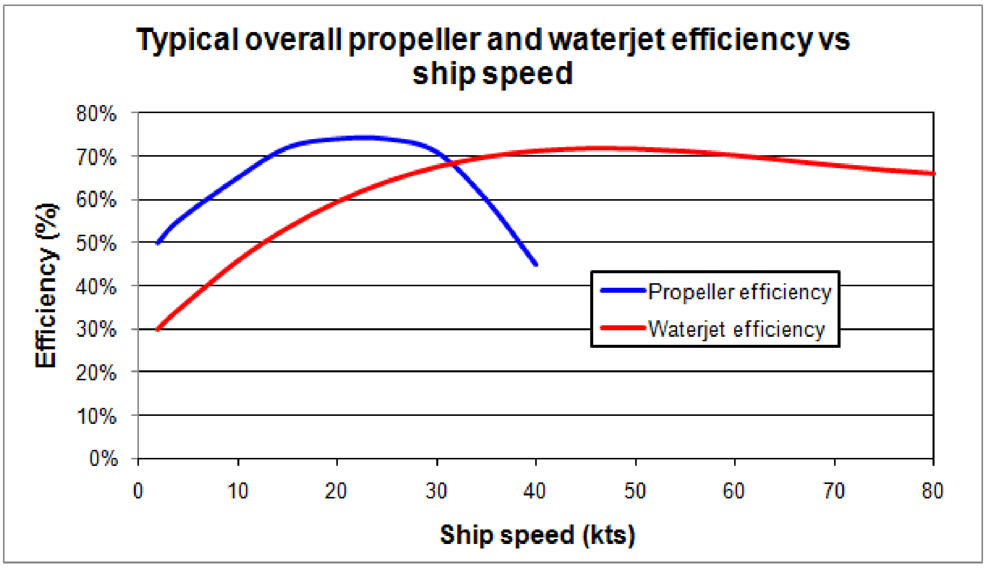


Figure 3.69: Typical efficiency curves of propeller and waterjetsystem vs ship speed

3.10.6 Signatures

The waterjet produces a lot of underwater noise, which makes it not suitable for ASW operations. It is not only the jet of water that produces so much noise, but also the hydraulics for the steering of the jetavator. When the waterjet is solely used as a boost propulsor, the ASW operations can be done with propellers. But also in such a configuration the presence of the waterjet has a bad effect on the underwater noise, because the flow noise of the ship is increased by the enormous water inlet duct in the ships hull. The inlet is too big to close it with some kind of valve or door.

3.10.7 Shock resistance

Like the propeller, the waterjet has not to fear from underwater shockwaves, because the waterjet is entirely filled and surrounded by water. If the waterjet is not entirely filled with water or is only partly submerged, shockwaves might be damaging to the waterjet.

3.10.8 Maintainability

A waterjet is designed for operation during the lifetime of the ship, but it needs some maintenance and overhaul. In comparison with a propeller, a waterjet needs more maintenance. Table 3.12 lists the main maintenance tasks with their interval.

Maintenance job	**Interval** (year)
Overhaul jetavator bearings	5
Overhaul steer & reverse cylinders	2.5
Replace hydraulic hoses	2.5
Overhaul stator bearing set	2.5
Overhaul of thrust bearing	10
Overhaul of shaft seal	2.5
Replace zinc anodes on jetavator	0.5
Replace hydraulic and lub oil + filters	1

Table 3.12: Typical maintenance tasks with intervals on a waterjet

(based on: Wärtsilä presentation)

3.10.9 Reliability

Numbers on reliability of waterjet are not available to the author, but given the relative complexity compared to a propeller, it is assumed that reliability of a waterjet is lower than of a propeller. The waterjet is more complex than the open water propeller due to the greater number of components including propulsion pump, thrust nozzle, thrust vectoring and reversing mechanisms, ducting, debris grill and fairings for the mounting of the inlets.

3.10.10 Initial purchase costs

For the waterjet, the purchase costs are estimated with a Cost Estimating Relationship (CER) from the Cost Analysis section of the DMO. The CER is a function of the power of the waterjet. The CER is mentioned in the confidential appendix. No other sources were found that give information about the purchase costs of waterjets, so no comparison can be made.

3.11 Podded propulsor

Podded propulsors, also called pods, are a configuration of ship propellers placed in a rotatable underwater body (pods). This rules out the need for a rudder and gives the ship an increase in maneuverability. The absence of the rudder and the fact that the aftship can have another shape, makes that the use of pods can result in lower ship resistance and higher hull efficiency. Pods can also have some benefits for underwater noise of the ship, because of better streamline under the ship, there is less cavitation. Cavitation inception speed can go up according to Trouwborst (1998). Experience learns that some measures (e.g. resilient mounting) need to be taken to make the electric motor operate quietly in the pod, because there is the risk of resonance on the pod hull.

The propellers in the pod can be driven mechanically, with a L-drive or a Z-drive than it is often called an azimuth thruster, or electrically, by an electric motor that is also in the pod. The mechanical solution suffers from large mechanical losses in the transmission. The electrical solution has the advantage that the propulsion motor is not within the ship, but outside in the pod. Of course the electrical energy has to be generated by a diesel- or gasturbine generatorset, but still the electrical pod results in an increased ship space and flexible lay-out. Within the RNLN, the LPD-2 *Johan de Witt*, has electrical pods. The use of pods on cruise ships is very common. Pods are available in different versions: with E-motor inside the pod, or with permanent magnet E-motor inside, with 1 propeller (pushing or pulling) or with 2 propellers (1 pushing, 1 pulling).

> Some well-known manufacturers of electrical pods are: Rolls Royce (Mermaid^{TM} series), Schottel.

With the current technology the pods are pretty large and heavy, and there is some doubt about the shock resistance of pods. This makes the application of pods on surface combatants questionable. As an example the dimensions and weights of two pods from the Mermaid^{TM} series (15 MW and 18 MW) are listed in table 3.13.

The SFC will need about 36 MW propulsion power, to be able to go 30 knots. This would mean two 18 MW pods of 10.5 m in length. This will not fit under the ship, because the typical beam of this type of vessel lies between 13-19 m. So it would not be possible to turn the two pods to

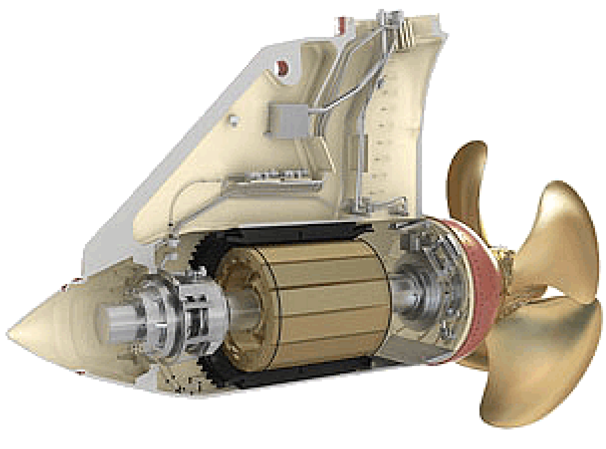


Figure 3.70: Schematic view of a podded propulsor

(source: http://www.rolls-royce.com/marine/products/propulsors/podded/index.jsp)

	Podsize 210	**Podsize 232**
Power	15 MW	18 MW
Propeller diameter	4.5 m	4.8 m
Shaft speed	185 rpm	175 rpm
Outboard weight (incl. motor)	129 tons	154 tons
Inboard weight (incl. steering gear, seating)	56 tons	70 tons
Overall length	10.2 m	10.5 m
Depth under ship	6 m	6.4 m

Table 3.13: Outline of dimensions and weights of two pods from Mermaid^{TM} series

the 90° position at the same time without hitting each other. The HTS technology, as described in section on electrical machines, offers opportunities for smaller and lighter pods.

About the doubt on shock resistance of pods: currently there are no shock-proof pods available on the market. TNO did studies about shock resistance of pods, van den Eikhoff (2002) being an example, which describes results from a finite element analyses. It was found that stresses were too high at the positions where the pods are connected to the ship, when a certain shock wave was applied. Also the relative displacement of the rotor with respect to the stator and exciter exceeded the clearance. Rolls-Royce also did analysis in the past, and they had the feeling that pods can be made shock proof up to certain levels if required, but it will require significant work (thus costs).

For the reasons that are explained above (dimensions, weight and shock resistance) the podded propulsor is not considered a reasonable option for the SFC project. Nevertheless, in the future, when developments are further and the mentioned issues are solved, pods might be considered a serious option, because they increase maneuverability substantially.

3.12 Summary of components

In this section a summary of the characteristics of the components, as discussed in the previous sections, are presented in the table on the next page. The quantitative estimation methods (formulas) that were derived for estimating dimensions, weight and nominal efficiency are listed. Also the scores on the more qualitative properties are presented by means of + and – signs, where – – – is the worst score possible and + + + the best. The score on purchase cost is also presented on the symbol scale. The cost models that are used in this study are not very detailed and do not distinguish between different types. The cost models are only used for indicative numbers in an early stage. The mutual score is determined by experiences from the past or are logical, for example the price of a gearbox is estimated based on the weight, a twin gear weights more so it has a higher price, thus a lower score on the symbolic scale.

		Available power	Dimensions: Length	Dimensions: Width	Dimensions: Height	Weight	Speed	Nominal efficiency	Signatures: Noise	Signatures: IR	Signatures: EM	Shockproof	Maintainability	Reliability	Purchase costs
		(MW)	(m)	(m)	(m)	(ton)	(rpm)	(-)	(-)	(-)	(-)	(-)	(-)	(-)	(-)
Prime movers															
Diesel engine	Slow speed (line)	1-84	$2.95 \cdot P_B^{0.49}$	$1.77 \cdot P_B^{0.28}$	$5.64 \cdot P_B^{0.26}$	$1281.2 \cdot N^{-0.76} \cdot P_B$	< 300	$0.69 \cdot N^{-0.07}$	−−	−	+	++	−−	++	−−−
	Medium speed (line)	0.5-35	$2.95 \cdot P_B^{0.49}$	$1.77 \cdot P_B^{0.28}$	$1.70 \cdot P_B^{0.46}$	$1281.2 \cdot N^{-0.76} \cdot P_B$	$300 - 1000$	$0.69 \cdot N^{-0.07}$	−−	−	+	++	−−−	+	−−
	Medium speed (vee)	0.5-35	$1.94 \cdot P_B^{0.66}$	$1.26 \cdot P_B^{0.42}$	$1.70 \cdot P_B^{0.46}$	$1281.2 \cdot N^{-0.76} \cdot P_B$	$300 - 1000$	$0.69 \cdot N^{-0.07}$	−−	−	+	++	−−−	+	−−
	High speed (vee)	0-9	$1.94 \cdot P_B^{0.66}$	$1.26 \cdot P_B^{0.42}$	$1.70 \cdot P_B^{0.46}$	$1281.2 \cdot N^{-0.76} \cdot P_B$	> 1000	$0.69 \cdot N^{-0.07}$	−	−	+	+	−−−	0	−
Gasturbine	Simple cycle	3.5-45	$3.25 \cdot P_B^{0.28}$	$1.5 \cdot P_B^{0.18}$	$2.33 \cdot P_B^{0.10}$	$5.97 \cdot P_B^{0.41}$	3600-7000	$0.27 \cdot P_B^{0.11}$	++	−−	+++	++	−	0	−−−
	ICR cycle	25	$3.25 \cdot P_B^{0.28}$	$1.5 \cdot P_B^{0.18}$	$3.50 \cdot P_B^{0.10}$	$11.9 \cdot P_B^{0.41}$	3600	$0.27 \cdot P_B^{0.11} + 0.05$	++	−	+++	++	−−−	−−	−−−−
Electrical machines															
	DC	≤ 5	$0.54704 \cdot T^{\frac{1}{3}}$	$0.45586 \cdot T^{\frac{1}{3}}$	$0.71607 \cdot T^{\frac{1}{3}}$	$1.5 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	+++	+++	0	+	−−−	0	−
Synchronous	AC	≤ 50	$0.61264 \cdot T^{\frac{1}{3}}$	$0.38290 \cdot T^{\frac{1}{3}}$	$0.75182 \cdot T^{\frac{1}{3}}$	$1.2 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	+	+++	−−	+	−	0	−
	AC (HTS)	≤ 36.5	$0.27311 \cdot T^{\frac{1}{3}}$	$0.27311 \cdot T^{\frac{1}{3}}$	$0.32175 \cdot T^{\frac{1}{3}}$	$1.2 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	+	+++	−	+	0	+	−−
	PM	≤ 5	$0.29420 \cdot T^{\frac{1}{3}}$	$0.29420 \cdot T^{\frac{1}{3}}$	$0.34659 \cdot T^{\frac{1}{3}}$	$2.65 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	++	+++	−−	+	0	+	−−−
Asynchronous	AC	≤ 25	$0.61264 \cdot T^{\frac{1}{3}}$	$0.38290 \cdot T^{\frac{1}{3}}$	$0.75182 \cdot T^{\frac{1}{3}}$	$1.2 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	0	+++	−−	−−	+	++	+
	AC (advanced)	≤ 20	$0.32183 \cdot T^{\frac{1}{3}}$	$0.26819 \cdot T^{\frac{1}{3}}$	$0.35809 \cdot T^{\frac{1}{3}}$	$2.65 \cdot$ Volume	100-5000	$\frac{1}{1.026+\frac{0.003162}{\sqrt{P}}}$	+	+++	−−	−−	+	++	−−
Conversion machines															
Gearbox	Single gear	n/a	$0.54\left(\frac{P}{N}\right)^{0.37}$	$0.84\left(\frac{P}{N}\right)^{0.27}$	$0.98\left(\frac{P}{N}\right)^{0.27}$	$0.3\left(\frac{P}{N}\right)$	n/a	0.98-0.99	−−−	+++	+	++	+++	+++	−
	Twin gear	n/a	$0.54\left(\frac{P}{N}\right)^{0.37}$	$\sum 0.84\left(\frac{P}{N}\right)^{0.27}$	$0.98\left(\frac{P}{N}\right)^{0.27}$	$0.6\left(\frac{P}{N}\right)$	n/a	0.95-0.97	−−−	+++	+	++	+++	++	−−
Converter	Cyclo	± 100	>PWM	>PWM	>PWM	>PWM	$< 0.35 \cdot f_{in}$	0.75	+++	+++	−	++	+++	−−	?
	Synchro	< 100	<PWM	<PWM	<PWM	<PWM	$< 0.5 \cdot f_{switch}$	0.85	+	+++	−	++	+++	++	?
	PWM (GTO/IGCT)	≤ 25	$0.7 + 0.06 \cdot S$	$\frac{7.5+0.7 \cdot S}{2.3 \cdot (0.7+0.06 \cdot S)}$	2.3	$1.5 + 0.75 \cdot S$	$< 0.5 \cdot f_{switch}$	$\frac{1}{1.01+\frac{0.01667}{\sqrt{S_{nom}}}}$	−	+++	−	++	+++	+	?
	PWM (IGBT)	≤ 10	$0.7 + 0.09 \cdot S$	$\frac{7.5+2 \cdot S}{2.3 \cdot (0.7+0.09 \cdot S)}$	2.3	$1.5 + 0.75 \cdot S$	$< 0.5 \cdot f_{switch}$	$\frac{1}{1.01+\frac{0.01667}{\sqrt{S_{nom}}}}$	−	+++	−	++	+++	+	?
Switchboards															
	High voltage	> 1kV	1.7	$0.65 \cdot n_f$	2.6	$1 \cdot n_f$	n/a	0.995	+	+++	−−	−−−	+++	++	−−
	Low voltage	< 1kV	1	$0.65\left(n_{f,in} + \frac{1}{4} n_{f,out}\right)$	2.2	$0.5 n_{f,in} + 0.45 \cdot \frac{1}{4} n_{f,out}$	n/a	0.995	+	+++	−−	−−−	+++	++	−
Propulsors															
Propeller	FPP	≤ 66	n/a	D= $\sqrt{\frac{P_p}{1} \cdot \frac{4}{\pi}}$	n/a	$0.337 \cdot D^{2.3}$	100-500	$\frac{2}{1+\sqrt{1+C_T}} - 0.175$	−	+++	-	+++	+++	+++	−
	CPP	≤ 44	n/a	D= $\sqrt{\frac{P_p}{1} \cdot \frac{4}{\pi}}$	n/a	$0.039 \cdot D^{3.9}$	100-500	$\frac{2}{1+\sqrt{1+C_T}} - 0.175$	−−−	+++	−	++	+	+	−−−

Part II

Concepts and analysis

Chapter 4

Propulsion Concepts

The components that are described and characterized extensively in chapter 3 are the 'building blocks' of ship propulsion concepts. In this chapter some examples of propulsion concepts are described. Components are put together to be able to deliver a certain amount of power to the water to make the ship accelerate and move up to the required speed. In the composition of propulsion concepts, the speed requirement is normally not the only consideration. Subjects like maintainability and fuel consumption are also very important characteristics of a propulsion concept. Combining different machinery might be a smart solution to combine the good characteristics of each component. On the other hand, more components increase complexity, space, weight and maintenance. The chosen propulsion configurations are judged on a number of criteria, which enables a multiple criteria analysis between the different concepts in the next chapter.

4.1 Methodology and considerations

First the considerations in the composition of the propulsion concepts is explained and the methodology to compare and assess them.

4.1.1 Design considerations

The number of combinations is almost endless, so not all options can be considered. A choice has to be made about what the most relevant concepts are. The starting points for this selection are the requirements on the ship which are all listed in chapter 2. One of these requirements is that proven technologies should be used. With this requirement in mind, propulsion concepts of existing ships of this type from other navies are collected and analyzed. Another important requirement is the reduced signature profile for ASW operations. This requirement had a major role in the selection of the components. The third major player in the composition process is fuel efficiency. Components are chosen such that engine loading is acceptable at all speeds.

Propulsion concepts are put together with components without tuning it to the exact offer at the market. Especially for gasturbines, where the offer is rather small, this might lead to engines that are not available. The philosophy is to first determine what you desire as ship designer and marine engineer, and after that look if it is available. The offer on the market might change in the near future or an investment can be done to develop new products. Only if the picture of the desires is clear, one can determine if it is worth an investment for new products or a change in capability if products on the market are used.

An important choice in the design of a propulsion configuration, is the number of shaftlines. In the requirements it is described that there should be more than one shaftline, for reasons of redundancy. In merchant vessels, normally, one shaftline is used with one propeller, because this is the most efficient method. Nevertheless, for a surface combatant this is not considered an option for reasons of redundancy and maneuverability. Besides, it requires very large diameter propellers to deliver the relatively high power of this ship to one single propeller. Diameter is limited for these ship types because it also has to operate in shallow waters. Normally on this shiptype, 2 shaftlines are used. This number is adopted for this design study. It might also be a possibility to go for more, but this introduces extra components, costs and complexity.

4.1.2 Concept selection

In a brainstorm session with members of the taskgroup 'Propulsion concepts SFC' the requirements are considered and a short list of most potential propulsion concepts is produced. This list is produced with the requirements and with the experience of the taskgroup. A distinction is made between pure mechanical propulsion concepts, hybrid concepts (mechanical plus electrical) and pure electrical concepts. The list starts with the mechanical propulsion concept that is found on the existing RNLN frigates, namely the CODOG concept. This is the concept that the RNLN is very familiar with, and serves as a reference. Other full mechanical concepts that are examined are a CODOG concept with only 1 gasturbine instead of 2, and a CODAD concept with 4 diesel engines. Hybrid concepts that are examined are two CODLAG concepts with 1 or 2 gasturbines, a CODLADAD concept which is a pure diesel concept with a PTI/PTO possibility, and a CODLADOG concept which in essence is the CODOG concept with addition of 2 electric motors for slow speeds. Full electric concepts are also called Integrated Full Electrical Propulsion (IFEP) where the propellers are solely driven by electric motors. Variations are in the way the electric power is generated, one example is examined with 4 diesel-generators and 1 gasturbine-generator.

- Concept 1: CODOG (2 gasturbines, 2 diesel engines)
- Concept 2: CODOG (1 gasturbine, 2 diesel engines)
- Concept 3: CODAD (4 diesel engines)
- Concept 4: CODLAG (2 gasturbines, 2 electric motors)
- Concept 5: CODLAG (1 gasturbine, 2 electric motors)
- Concept 6: CODLADAD (4 diesel engines, 2 electric motors/shaft generators)
- Concept 7: CODLADOG (2 gasturbines, 2 diesel engines, 2 electric motors)
- Concept 8: IFEP (2 electric motors)

Initially, the outcome of the brainstorm session held some concepts with waterjets and podded propulsors. During the components study it was concluded that these components were no further considered for different reasons, so these propulsion concepts were deleted from the list.

4.1.3 Assessment

The propulsion concepts are judged on four main criteria, so-called '*parent criteria*': **Operational characteristics**; **Integration in ship**; **Availability** and **Costs**. These criteria symbolize the cost versus benefit. The operational characteristics represent the benefit: *What can I do with it?*. The other parent criteria represent the costs: *What does it cost me to do and keep doing it?*. Integration in ship represents the costs in terms of space and weight on the ship, availability in terms of how long it can be used and what you have to do to keep using it, and costs are the financial consequences of purchasing and using it. These parent criteria are subdivided into direct criteria, or so-called '*child criteria*', to better give value to the parent criteria. The child criteria show great similarity with the characteristics that were used in chapter 3 to describe the components. Table 4.1 shows the structure of parent and child criteria. In this chapter the value of the child criteria is determined per propulsion concept. The outcomes are used in a multiple criteria analysis in the next chapter.

The models, that were derived in chapter 3 for calculating dimensions, weight, costs and nominal efficiency are used to quantify the scores on some of the child criteria. These models are all programmed in an Excel worksheet to ease and automate the calculation, see appendix F. The

	Parent criteria	Child criteria
Benefit	**Operational characteristics**	Maneuverability Signature profile (*susceptability*) Redundancy (*survivability*)
Cost	**Integration in ship**	Nr. of components Space consumption Weight Fuel capacity
	Availability	Reliability Maintainability Shock-proofness (*vulnerability*)
	Costs	Initial purchase costs Fuel costs Maintenance costs

Table 4.1: Criteria on which propulsion concepts are assessed

basic properties and numbers of the components need to be filled in and rough estimations of dimensions, weight, efficiency and initial purchase costs roll out. Not all criteria can be assessed quantitatively, some have qualitative scores. The nature of the available information determines if a criteria can be assessed quantitatively. The qualitative scores are based on subjective feelings from past experience and knowledge, and are assessed on a symbolic [– – –...+++]-scale in comparison to a reference concept. The CODOG concept (section 4.2) is adopted on the other frigates of the RNLN, so this concept acts as the reference concept.

- Operational characteristics
  - The **maneuverability** of a propulsion concept is assessed qualitatively in comparison to the reference concept, based on experiences and component characteristics. Maneuvering capability is in this sense defined as acceleration and slow speed maneuvering.
  - The **signature profile** of the ship is not only depending on the propulsion concept, but the contribution of the propulsion concept is judged qualitatively in comparison to the reference concept, because quantification is not possible in an early stage.
  - **Redundancy** can be defined as spatial or functional redundancy. The positioning of the components in the ship is a matter of proper ship design. Functional redundancy is determined by the choice of machinery. In this assessment the level of redundancy is expressed as the minimum number of components to be fail in a worst case scenario before the ship is not able to sail 10 knots anymore with full electrical power demand of 1350 kWe. These values were chosen, because 10 knots is the lowest operational speed, and with 1350 kWe electrical power the ship is able to use all its SEWACO systems to defend itself. So, at these conditions the ship is still operationally employable. Failure of the shafts + propellers is not taken into account, because in all concepts a failure of both propellers leads to failure of the entire propulsion system.
- Integration in ship
  - The **number of components** is just a matter of counting the components that are assigned as main components in this study.
  - **Space consumption** of the propulsion concepts is evaluated quantitatively with the help of the Excel worksheet. With the estimated dimensions the volume is calculated. The sum of all volumes is the space consumption. The volume of inlet, outlet and

cooling ducts is not taken into account because that depends on the position of the machinery in the ship.

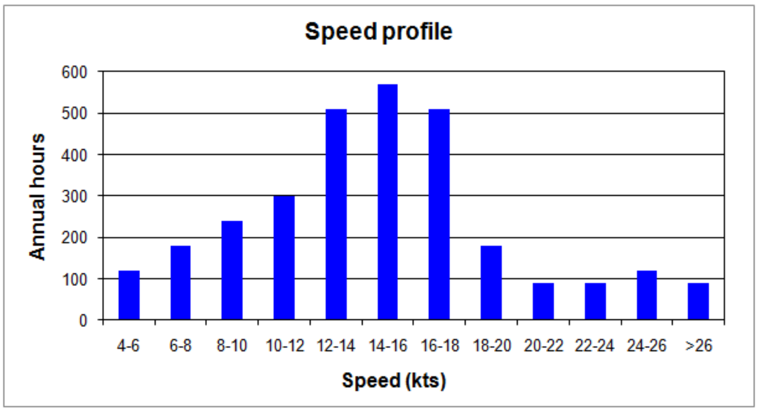
- The **weight** of the machinery is also calculated with the Excel worksheet, according to the models from chapter 3.
- The **fuel capacity** is the calculated minimum amount of fuel that the ship has to carry to do 5000 nautical miles at 18 knots. This is the requirement as mentioned in the Operational Concept. In this is an auxiliary powerload of 1100 kW assumed. Based on models for nominal and partload efficiencies of the components a calculation of burned fuel is made. This results in a mass of fuel that is required for the 5000 nm range. With an average value for fuel density of 850 kg/m^3 this calculated into a required tankvolume in m^3.

- Availability
  - **Reliability** of the machinery is very difficult to judge on. A qualitative judgement in comparison to the reference concept is made based on reliability of components and number of components and experiences.
  - **Maintainability** is also difficult to quantify. For some components a model for maintenance costs was determined but this was not possible for all components. Based on the maintenance descriptions and experiences the level and the complexity of the regular maintenance tasks will be assessed in comparison to the reference concept.
  - The **shock-proofness** of the concept is not evaluated extensively, because almost every concept can be made shock-proof with the right measures. Still, it is tried to assess the vulnerability of a concept qualitatively in comparison to the reference concept.

- Costs
  - The **initial purchase costs** are calculated with Cost Estimating Relationships (CER) from the Cost Analysis section of the DMO. These CER's are commercially confidential, and are given in a confidential appendix. It should be noted that these are very crude estimates.
  - The **fuel costs** are calculated with the fuel consumption and a fuel price of 620 €/ton[1]. This is calculated for a period of 1 year. Annual fuel consumption is calculated with a certain reference operating profile as input. There is not yet a required speed profile available from the staff requirement. An imaginary speed profile (mentioned below), based on the staff requirement of the M-frigate, is now used as an input for fuel consumption calculations, with 3000 sailing hours per year (is 125 days). Additionally, a certain auxiliary power profile is used, to also take into account the fuel consumption due to auxiliary power. The resistance curve is used to calculate the required thrust power. With the efficiencies on page 9, the delivered power P_D is calculated (incl. seastate and windfactors). With the models for gearbox efficiency and diesel, gasturbine, electromotor and generator efficiency this required thrust power is calculated back to a certain fuel burn. All this is programmed in an Excel worksheet so it can easily be calculated and compared between different concepts (see appendix F). In the calculation, the most efficient operating mode at every ship speed is used.

Speed (kts)	4-6	6-8	8-10	10-12	12-14	14-16	16-18	18-20	20-22	22-24	24-26	>26
Aux. power (kW)	800	800	800	1350	1350	1350	1100	1100	1100	1100	1100	1100
Time	4%	6%	8%	10%	17%	19%	17%	6%	3%	3%	4%	3%

[1]MGO, Rotterdam, June 2011, 881 $/ton, 1.42 $/€



– Life cycle costs also consist of **maintenance costs** of all components. Unfortunately, not for every component an estimation model could be derived for the maintenance costs. For diesel engines and gasturbines (the prime movers) this is tried and models are derived. Expected is that these have the largest share in the total maintenance costs. With these models, based on installed power of the machinery and the runninghours, the annual maintenance costs (of only the prime movers) are estimated. These costs are calculated in the same Excel worksheet as in which the fuel costs are calculated.

Note: *Manning costs have the largest share in the life cycle costs of the ship, but these are not accounted for, because the required manning is largely determined by the level of process automation. In principal can all concepts have a higher or lower level of process automation, so with an extra investment on process automation the required manning of each concept can be reduced. But a general rule of thumb is: the less prime movers, the less manning. Because in practice the maintenance on prime movers is the most intensive.*

4.2 Concept 1/CODOG/Full mechanical

The first propulsion concept in this study is the reference concept that is found on the current Royal Netherlands Navy frigates (M-frigate and Air-defence and Command Frigate). It is the so-called CODOG (COmbined Diesel Or Gasturbine) concept. It is a full mechanical propulsion concept with diesel generators to generate electrical power. A schematic view is presented in figure 4.1.

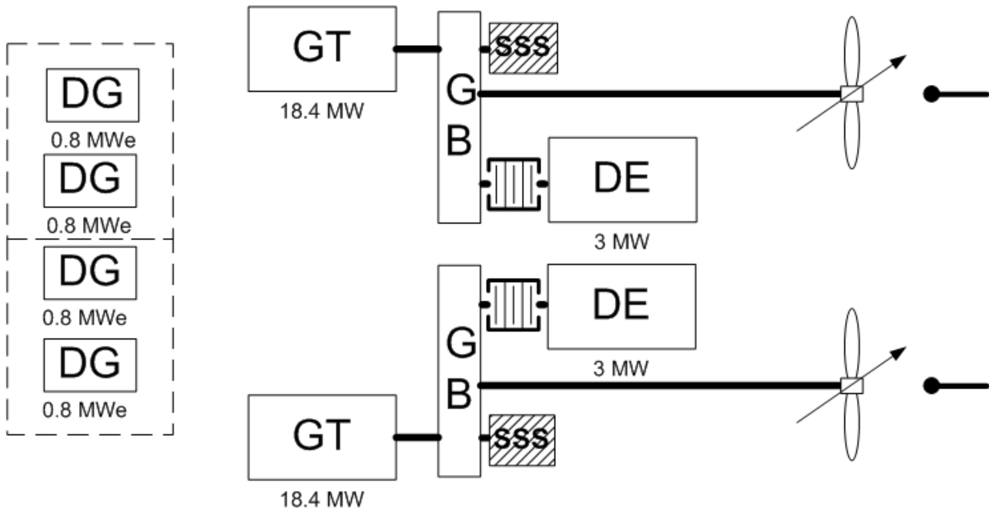


Figure 4.1: Propulsion concept 1, CODOG

With this configuration there are 5 different operating modes:

1. High speed mode (2 gasturbines on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.
2. Cruise speed mode (2 diesel engines on 2 shafts). For travelling long distances. Max. speed ≈18 knots.
3. High speed on 1 shaft mode (1 gasturbine on 1 shaft, other shaft trailing). Max. speed ≈25.5 knots.
4. Cruise speed on 1 shaft mode (1 diesel engine on 1 shaft, other shaft trailing). For doing operations at low speeds. Max. speed ≈14 knots.
5. High speed on 1 shaft, cruise speed on other shaft (1 gasturbine on 1 shaft, 1 diesel engine on other shaft). Power per shaft is limited to max. diesel power. If 1 diesel engine is unserviceable for whatever reason, but still power on 2 shafts is required. Max. speed ≈18 knots.

4.2.1 Components

Main components in this propulsion concept are 2 controllable pitch propellers both driven by a diesel engine **or** a gasturbine through a combination gearbox. SSS-clutches are used for the coupling of the gasturbines, which enables the possibility to smoothly overtake propulsion with gasturbines. The diesel engines are coupled through multiplate friction clutches in combination with flexible coupling or through fluid couplings. Electric power supply by 4 diesel generator sets.

Main components Concept 1 (CODOG)

Component	Number	Power (MW)
Controllable pitch propeller	2	18.4
Diesel engine	2	3
Gasturbine	2	18.4
Gearbox	2	18.4
Diesel-generator	4	0.8
Switchboard	2	1.6
	14	

Controllable pitch propeller

To propel the ship, controllable pitch propellers are used. In this propulsion configuration CPP's are inevitable to ensure the desired maneuverability. Sailing very slowly and go astern without reversing shaft speed is not possible in this configuration without a CPP or a multispeed gearbox. The latter is not taken into account because this is considered too complex and unreliable, so it leaves a CPP. The CPP has a lower cavitation inception speed, but it has the advantage that propeller pitch can be optimized for cavitation behaviour at off-design conditions.

The propeller is used as the starting point to match the propulsion configuration. The propeller dimensions are determined at first. A comparable ship with almost the same amount of power on the propellers and the same propulsion configuration is the LCF. The LCF has propeller diameter 5m. From the conceptual design, the maximum diameter could approximately be 4.5 m. Each propeller receives a maximum of 18.4 MW propulsion power from the shaft. In practice this amount will never be reached because of the losses in the gearbox and shaftline. But using this figure as keynote creates a safe margin. The propeller loading (based on disc area A_0 and brake power P_B) is now 1.16 MW/m^2, which is comparable to M-frigate propeller (1.23 MW/m^2). A typical propeller that is used on other frigates, has 5 blades and a blade area ratio of approximately 0.65. With this information the Wageningen B-565 is used as a base. The open water diagram of this propeller is modelled in Excel with the polynomials from Bernitsas *et al.* (1981) and matched with the dimensionless thrust diagram of the hull. This results in a best open water efficiency at 30 knots of 0.71, with P/D ratio 1.3. With this P/D ratio the propeller has to rotate at 212 rpm at 30 knots. The method to come to these results is explained in Appendix D.

Power	18.4 MW
Speed / type	212 rpm / CPP
Diameter	4.5 m
Estimated weight	13.8 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	515 k€
Nominal efficiency	0.71

Diesel engine

Two propulsion diesel engines are installed, together delivering 6 MW. The diesel engines can deliver the power to the propellers up to transit speed, including a 7% margin. Diesel engines are used because these are more fuel efficient than gasturbines and the ship will sail most of the time at transit speed or lower. In order to retain an acceptable engine efficiency, low speeds will be sailed on only one shaft, with one trailing shaft.

For the diesel engines, the choice can be made between low, medium or high speed diesels. The lower the speed the bigger and heavier the engine gets, but on the other hand the lower the number of cylinders (less maintenance), the more efficient and reliable, generally spoken. Figure C.1 gives a clear picture of the trade-off between weight and efficiency. Normally on naval ships the middle course is adopted, and medium speed engines are chosen with speeds around 1000 rpm. Another consideration that has to be made is the choice between line or vee engine. Vee engines are more compact. Since space is scarce on this type of ship, Vee engines are chosen.

Power	3 MW
Speed / type	1000 rpm / V-engine
L x W x H	4.01 x 2.00 x 2.82 m
Inlet / Outlet duct	0.243 / 0.228 m^2
Weight	20.2 ton
Estimated costs	1360 k€
Nominal efficiency	0.43

Gasturbine

For high speeds (above transit speed) gasturbines are used, because they have a much higher power density than diesel engines. Besides that, gasturbines deliver fast power and have a wider operating envelope, which results in improved maneuverability. The lower efficiency of gasturbines is accepted because the gasturbine will in practice only be used for small part of time. In the example speed profile it will be used 19% of time. To be able to sail 30 knots, the gasturbines should deliver 36.8 MW, which means 18.4 MW each. Choice can be made between simple cycle or improved cycle gasturbine. In practice the simple cycle gasturbines show better reliability and lower purchase costs, but because the gasturbines are only used for small part of time in this configuration, the improved efficiency will not weigh against these major disadvantages of the ICR cycle gasturbine. So, simple cycle is chosen.

Power	18.4 MW
Speed / type	5600 rpm / Simple cycle
L x W x H	7.35 x 2.53 x 3.12 m
Inlet / Outlet duct	3.551 / 3.036 m^2
Weight	19.7 ton
Estimated costs	6400 k€
Nominal efficiency	0.37

Gearbox

Gearbox is necessary to reduce the shaft speed of diesel engine and gasturbine to values suitable for driving the propeller. Besides, the gearbox makes it possible to drive one shaft by two different machines, but not at the same time, because it is an or-configuration. The gearbox includes clutches to couple and decouple the diesel engine and gasturbine to the outgoing shaft. The diesel shaft has a friction plate clutch or a fluid coupling for better noise reduction. The gasturbine is coupled via a so-called SSS-clutch (self-shifting-synchronous). With this type of clutch it is possible to smoothly overtake shaft drive from the diesel engine with the gasturbine.

Maximum input speed from the gasturbine is 5600 rpm. The 4.5 m Wageningen B-565 should be operated at $P/D \approx 1.2$ to have best efficiency, the matching propeller speed is 212 rpm. This means the gearbox should have a gear ratio i_{GB} of 26.42 for the gasturbine drive. The maximum input speed of the diesel engine is 1000 rpm. The propeller speed at 18 knots with both propellers running is 130 rpm, which means a gear ratio of 7.69 for diesel drive. Power/speed ratio $\frac{P}{N}$ in (kW/rpm) for gasturbine and diesel drive are 86.8 respectively 23.1.

The gearing wheels have double helical teeth to lower the noise production. To further decrease underwater noise, the gearbox should be placed on stiff resilient mountings.

Power/speed ratio (GT-drive)	86.8 kW/rpm
Power/speed ratio (DE-drive)	23.1 kW/rpm
Type	Twin, multiple stage, double helical
L x W x H	2.82 x 4.76 x 3.27 m
Weight	54.7 ton
Estimated costs	2220 k€
Nominal efficiency	0.98

Diesel-generator

For generation of electrical power there are diesel generator sets. Peak electrical load in operational condition is approximately 1350 kWe. On the maximum power consumption a margin of 250 kWe (18.5%) is calculated to ensure sufficient power for possible future modifications and for temporary peak loads. It is common practice to generate electrical power with a minimum of 2 generator sets at sea, to prevent total power failure in case one generator set fails. In the harbour, power demand is maximal 800 kWe. So it seems most logical to install 800 kWe diesel generator sets to meet all power demands with one or two DG-sets.

The losses in the generator and distribution network have to be taken into account, so the diesel engine should deliver more than 800 kW. With an efficiency of 96% (97% generator efficiency and 99% network efficiency) this means the diesel should deliver approximately 850 kW mechanical power. To be sure the generator will not be overloaded in case of temporary diesel engine overload, the maximum power of the generator should be higher than the maximum delivered power of the engine. A margin 0f 7.5% of the nominal diesel power is added, which means the generator should have nominal power 915 kW.

Depending on the desired level of redundancy, 3 or 4 DG-sets could be installed. In both cases there is some redundancy in power generation. But similar to other ships of this type, 4 DG-sets are chosen. One of the DG-sets should comply with the emergency generator requirements, then there is no need for an extra emergency generator.

The net frequency onboard is 60 Hz, according to NATO standards. According to equation 3.61 (page 67), the speed of the diesel engine should be 3600, 1800, 1200, 900, 720, 600 rpm, or even lower as long as p in equation 3.61 is an integer number. To safe space the engine speed should be as high as possible, because both diesel engine and generator become larger with slower speeds. 3600 rpm is too high for diesel engines of this power level, 1800 rpm V-engine is then the most compact option. Though, on page 37, it was stated that there is some relation between engine speed and maintenance costs. So, from maintenance point of view, slower speeds would be prefered. Trade-off will be between maintenance costs and space consumption. This trade-off can not be analyzed properly, because the relation between speed and maintenance costs is not satisfactory. Regarding the rather limited power of the DG-set, 900 rpm is suggested. For higher power DG-sets a higher power density is more important to safe space.

The generated electrical power is of low voltage (440V). It is not necessary to generate high voltage because there are no big users (like electric propulsion motors or bowthrusters).

L/D value of the generators is chosen such that the height of the generator is equal to the height of the diesel engine: $L/D = 1.6$.

Power (diesel)	0.85 MW
Power (generator)	0.92 MWe
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(1.74 + 1.41) x 1.18 (0.81) x 1.60 m
Inlet / Outlet duct	0.069 / 0.065 m^2
Weight	6.2 + 2.2 ton
Estimated costs	1173 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

The electrical power generated by the diesel generator sets is directly fed to the main switchboard. For redundancy reasons there are (at least) two main switchboards which can be coupled. These are low voltage switchboards (440V). The switchboards have 4 incoming fields (2 from diesel generator set, 1 from bus-coupler and 1 from shore connection). The number of outgoing fields is estimated as explained on page 87, and is set to 8.

For better shock-resistance the switchboards are mounted on springs.

Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

D=Depth.

Auxiliary equipment

Important auxiliary equipment that comes with this propulsion configuration are: high pressure air starting systems for diesel engines and gasturbines, fuel feed pumps for diesel engines and gasturbines, at least one fuel seperator is needed, a fresh water cooling system is required to cool the diesel engines and generators, the CPP comes with a hydraulic control unit, the diesel engines and in particular the gasturbines have large in- and outlet ducts. It has to be taken into account that exhaust gas after treatment equipment might be installed in the future to reduce emissions, so-called *provisions for* should be provided.

4.2.2 Operational characteristics

Maneuverability	reference
Signature profile	reference
Redundancy	2

The maneuverability of this propulsion concept is set as reference. In diesel mode maneuverability is rather limited due to narrow operating envelope of the engine. The diesel engine is approximately a constant torque machine, which is dramatically narrowed by the surge limit of the turbocharger, which means it can not deliver full torque at low speeds. Sequential turbocharging solves great part of this problem. The diesel engine also has a minimum speed

(25-35% of nominal speed) and a minimum torque (25-40% of nominal torque) below which it will not run smoothly and causes damage to the engine. The CPP increases the maneuvering capabilities of the ship. In gasturbine mode maneuvrability is better, because the gasturbine (with free power turbine) is approximately a constant power machine, which can deliver full power at all speeds.

Signature profile of this concept is set as reference. The presence of a gearbox , which is also in use at slow speeds, is not advantageous for the signature profile.

Redundancy of this concept is such, that in a worst case scenario with failure of 2 gearboxes the propulsion stops. So, number of components to fail before the ship isn't able to sail 10 knots anymore is 2. The failure of 2 propellers is not taken into account because this is the same for all concepts.

4.2.3 Integration in ship

Nr. of main components	14
Total space consumption	286 m^3
Total weight	256 ton
Fuel capacity	302.4 ton / $3.56 \cdot 10^5$ m^3

4.2.4 Availability

Reliability	reference
Maintainability	reference
Shock resistance	reference

The reliability, level and complexity of maintenance and shock resistance of this concept are set as reference.

4.2.5 Costs

Initial purchase	26.1 M€
Annual fuel cost / consumption	3.42 M€/ 3887 ton
Annual maintenance cost	0.168 M€

4.3 Concept 2/CODOG/Full mechanical

The second propulsion concept is also a CODOG, according to Concept 1, but with only one gas-turbine and a cross-connection gearbox. This concept is found on the Australian/New Zealand ANZAC class. This concept could also be fitted as COD***A***G as found on the Norwegian Fridtjof Nansen class and the German Sachsen class. The CODAG layout adds significant complexity to the gearing system, and a proper load sharing control is essential. For this reason the OR solution is preferred.

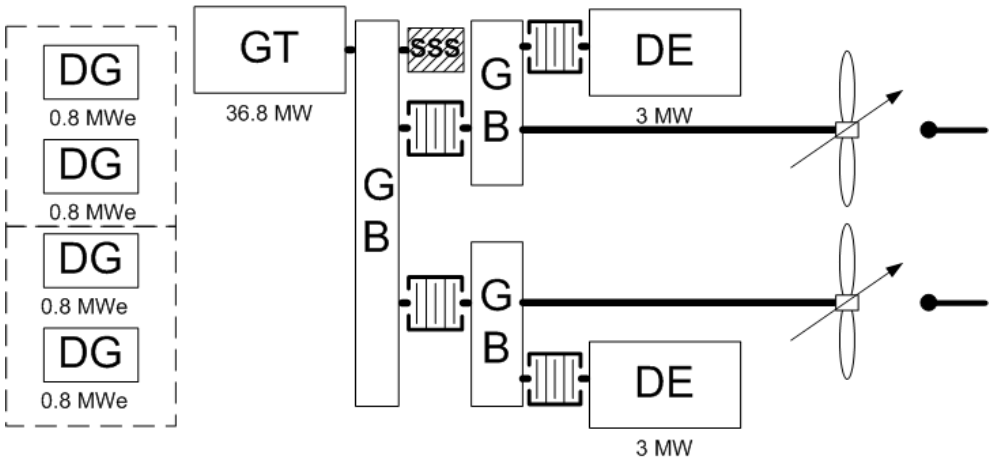


Figure 4.2: Propulsion concept 2, CODOG

With this configuration there are 6 different operating modes:

1. High speed mode (1 gasturbines on 2 shafts). For sprinting at top speed. Max. speed $\approx$30 knots.

2. Cruise speed mode (2 diesel engines on 2 shafts). For travelling long distances. Max. speed $\approx$18 knots.

3. High speed on 1 shaft mode (1 gasturbine on 1 shaft, other shaft trailing). Max. speed $\approx$25.5 knots. If one shaft or gearbox is unserviceable for whatever reason. Gasturbine power is limited to the capacity of the clutches, gearboxes, shaft and propeller, if these are designed for full power, max. speed could be $\approx$30 knots.

4. Cruise speed 1 engine on 2 shafts mode (1 diesel engine on 2 shafts). For doing operations at low speeds. Max. speed $\approx$15 knots. But it turns out from fuel consumption calculations that it is more efficient to sail slow with 1 engine on 1 shaft.

5. Cruise speed on 1 shaft mode (1 diesel engine on 1 shaft, other shaft trailing). For doing operations at low speeds. If one shaft or gearbox is unserviceable for whatever reason. Max. speed $\approx$14 knots.

6. High speed on 1 shaft, cruise speed on other shaft (1 gasturbine on 1 shaft, 1 diesel engine on other shaft). Power per shaft is limited to max. diesel power. If 1 diesel engine is unserviceable for whatever reason, but still power on 2 shafts is required. Max. speed $\approx$18 knots.

4.3.1 Components

Main components in this propulsion concept are 2 controllable pitch propellers, which are driven by one or two diesel engines **or** a gasturbine through a cross-connecting gearbox. In this concept, there is no need for a trailing shaft at low speeds because one diesel can drive two shafts, but the cross-connection gear can also be decoupled to reduce underwater noise. A SSS-clutch is used for the coupling of the gasturbine, the diesel engines are coupled through multiplate friction clutches in combination with flexible coupling or through fluid couplings. Electric power supply by 4 diesel generator sets, identical to concept 1.

Main components Concept 2 (CODOG)

Component	Number	Power (MW)
Controllable pitch propeller	2	18.4
Diesel engine	2	3
Gasturbine	1	36.8
Cross-connect gearbox	1	36.8
Reduction gearbox	2	3
Diesel-generator	4	0.8
Switchboard	2	1.6
	14	

Controllable pitch propeller

Description according to concept 1.

Power	18.4 MW
Speed / type	212 rpm / CPP
Diameter	4.5 m
Estimated weight	13.8 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	515 k€
Nominal efficiency	0.71

Diesel engine

Description according to concept 1.

Power	3 MW
Speed / type	1000 rpm / V-engine
L x W x H	4.01 x 2.00 x 2.82 m
Inlet / Outlet duct	0.243 / 0.228 m^2
Weight	20.2 ton
Estimated costs	1360 k€
Nominal efficiency	0.43

Gasturbine

Description according to concept 1, with the difference that in this concept only 1 gasturbine is used to deliver 36.8 MW.

Power	36.8 MW
Speed / type	3600 rpm / Simple cycle
L x W x H	8.92 x 2.87 x 3.34 m
Inlet / Outlet duct	7.10 / 6.07 m^2
Weight	26.2 ton
Estimated costs	12806 k€
Nominal efficiency	0.40

Gearbox

This concept has a special gearing arrangement, actually consisting of three gearboxes. In diesel drive up to transit speed, the smaller gears reduce the shaft speed of the diesel engine with a small wheel and a large wheel in the right teeth ratio. At speeds where only one diesel engine could deliver the power, one of the engines is decoupled, and the cross-connection gear is coupled. The cross-connection gear delivers half of the torque to the other shaft, and does not reduce the speed. Now two shafts are driven at the same speed by one engine. Though, at sonar operations it might be better to sail with one trailing shaft for reasons of underwater noise. At high speeds, the gasturbine is coupled through the SSS-clutch to the cross-connection gear, and the diesel engines are decoupled from the small gears. In the cross-connection gear, the gasturbine shaft speed is reduced to the desired speed, so the small gears don't have a function in this operating mode. This means the small gears only have to be designed for diesel power, but must be able to handle the high gasturbine shaft speed. The multiplate clutches that couple the cross-connection gear to the other gearboxes have to be very large because they are on the low speed/high torque side.

Maximum input speed from the diesel engines is 1000 rpm. The 4.5 m Wageningen B-565 should be operated at $P/D \approx 1.1$ to have best efficiency, the matching propeller speed is 130 rpm. This means the gearbox should have a gear ratio i_{GB} of $1000 : 130 = 7.69$. Power/speed ratio $\frac{P}{N}$ in (kW/rpm) is 23.1. The cross-connection gear has a gear ratio of $3600 : 212 = 17$, and a power/speed ratio 173.6. To determine the dimensions and weight of the cross-connect gear it is represented as a twin gear with two times a power/speed ratio of 173.6.

A risk of having two gears in line, is shaft misalignment. The ships structure should probably be stiffened at the the place of the gearbox to prevent misalignment.

The gearing wheels have double helical teeth to lower the noise production. To further decrease underwater noise, the gearbox should be placed on stiff resilient mountings.

Power/speed ratio (GT-drive)	173.6 kW/rpm
Power/speed ratio (DE-drive)	23.1 kW/rpm
Type	Cross-connect, multiple stage, double helical
L x W x H (Small gears)	1.72 x 1.96 x 2.29 m
L x W x H (Cross-connect gear)	3.64 x 6.76 x 3.94 m
Weight (Small gears)	6.7 ton
Weight (Cross-connect gear)	109 ton
Total estimated costs	4670 k€
Nominal efficiency	0.94

Diesel-generator

Description according to concept 1.

Power (diesel)	0.85 MW
Power (generator)	0.92 MWe
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(1.74 + 1.41) x 1.18 (0.81) x 1.60 m
Inlet / Outlet duct	0.069 / 0.065 m^2
Weight	6.2 + 2.2 ton
Estimated costs	1173 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

Description according to concept 1.

Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

D=Depth.

Auxiliary equipment

Description according to concept 1 with the addition of 2 extra high torque multiplate friction clutches (or fluid couplings).

4.3.2 Operational characteristics

Maneuverability	0
Signature profile	0
Redundancy	2

The maneuverability of this propulsion concept is comparable to the reference concept 1. Probably the power delivery of the gasturbine will be slightly slower because it is a bigger engine. But at low speeds the maneuvrability will probably be slightly better because it is possible to drive two shafts instead of one, but from fuel consumption calculations it follows that one trailing shaft is more efficient. Differences in maneuverability with concept 1 will be marginal, that's why value *0* is attached.

Signature profile is comparable to the reference concept 1. When silent operation is required, it is best to decouple the cross-connect gear and sail with one trailing shaft.

The level of redundancy of this concept is equal to that of concept 1. In this concept also the minimum number of components to fail before losing ability to sail 10 knots is 2. In a worst case scenario with failure of 2 gearboxes or 2 clutches the propulsion stops. If the cross-connect gearbox would not be isolated from the smaller gears with high-torque clutches, the redundancy level would be 1, because then with failure of one gearbox the whole propulsion would be dead.

4.3.3 Integration in ship

Nr. of main components	14
Total space consumption	281 m^3
Total weight	256 ton
Fuel capacity	306.5 ton / $3.61 \cdot 10^5$ m^3

The required fuel capacity for doing 5000 nm at 18 knots is slightly higher for this concept, compared to concept 1, because the gearbox efficiency is lower. Because the gearboxes are designed for higher power, they operate at lower part load with lower efficiency at 18 knots.

4.3.4 Availability

Reliability	−
Maintainability	+
Shock resistance	0

The reliability of this concept is estimated to be somewhat lower, because of the more complex cross-connect gearbox arrangement. Besides there are some extra clutches, which give an extra risk of failure. Level of maintenance will be less, because there is one gasturbine less to maintain. The extra gearbox does not require significant extra maintenance. The complexity of the maintenance is equal to that of concept 1. Shock resistance is also equal to the reference concept.

4.3.5 Costs

Initial purchase	26.4 M€
Annual fuel cost / consumption	3.55 M€/ 4025 ton
Annual maintenance cost	0.225 M€

This concept has higher investment costs because of the complex gearbox. The investment cost calculation even does not take into account the cost of clutches. This would even make this concept more expensive, because it has more clutches, and two high torque clutches.

Fuel consumption is higher than with concept 1, because the cross-connection gear adds some extra transmission losses, and for speeds from approximately 18-26 knots, the gasturbine runs on rather low load which is very inefficient. In concept 1 you could choose for 1 gasturbine on 1 shaft, with the other shaft trailing. This is more fuel efficient. Higher annual fuel consumption results in higher annual fuel costs.

4.4 Concept 3/CODAD/Full mechanical

The third full mechanical propulsion concept is conform the Danish Iver-Huitfeldt class, but with "father-son" diesel engines to have more optimal engine loading at al conditions. It is a CODAD concept, COmbined Diesel And Diesel, with 4 diesel engines in two different sizes.

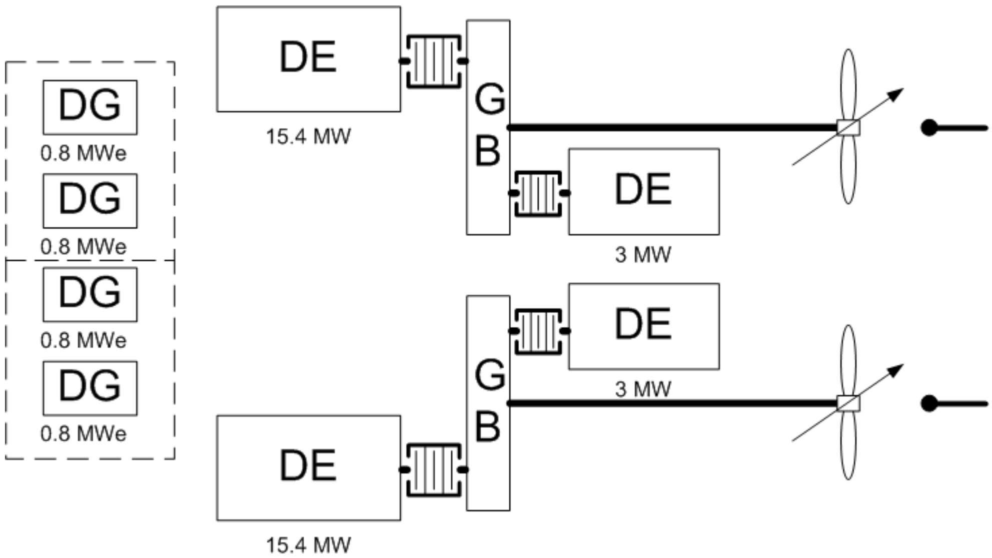


Figure 4.3: Propulsion concept 3, CODAD

With this configuration there are 6 different operating modes:

1. High speed mode (2 father-engines and 2 son-engines on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.
2. Cruise speed mode (2 son-engines on 2 shafts). For travelling long distances. Max. speed ≈18 knots.
3. High speed on 1 shaft mode (1 father-engine and 1-son-engine on 1 shaft, other shaft trailing). Max. speed ≈25.5 knots. If one shaft or gearbox is unserviceable for whatever reason.
4. Cruise speed on 1 shaft mode (1 son-engine on 1 shaft, other shaft trailing). For doing operations at low speeds or if one shaft or gearbox is unserviceable for whatever reason. Max. speed ≈14 knots.
5. High speed on 1 shaft, cruise speed on other shaft (1 father-engine on 1 shaft, 1 son-engine on other shaft). Power per shaft is limited to max. son-engine power. If 1 engine is unserviceable for whatever reason, but still power on 2 shafts is required. Max. speed ≈18 knots.

4.4.1 Components

Main components in this propulsion concept are 2 controllable pitch propellers both driven by a small diesel engine of 3 MW (son-engine) **and/or** a big diesel engine of 15.4 MW (father-engine) through a combination gearbox. The **and**-possibility of the gearbox adds some extra complexity and requires proper load-sharing-control between father- and son-engines. The diesel engines are all coupled through multiplate friction clutches in combination with flexible coupling or through a fluid coupling. Electric power supply by 4 diesel generator sets of equal power.

Main components Concept 3 (CODAD)

Component	Number	Power (MW)
Controllable pitch propeller	2	18.4
Diesel engine (son)	2	3
Diesel engine (father)	2	15.4
Gearbox	2	18.4
Diesel-generator	4	0.8
Switchboard	2	1.6
	14	

Controllable pitch propeller

Description according to concept 1.

Power	18.4 MW
Speed / type	212 rpm / CPP
Diameter	4.5 m
Estimated weight	13.8 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	515 k€
Nominal efficiency	0.71

Diesel engine

In total 4 propulsion diesel engines are installed, together delivering the full amount of 36.8 MW. The son-engines together deliver 6 MW, which is enough to do transit speed. For operational speeds, with power demand below 3 MW, only one diesel engine is operated, in order to retain an acceptable engine efficiency. At high speeds the father-engines are also operated. The diesel engine has a more favourable part load efficiency than a gasturbine, so transit speeds can also be run on one father-engine to save engine running hours.

The speed of the son-engines is 1000 rpm. This choice is driven by the trade-off between dimensions, weight, efficiency and reliability. 1000 rpm engines represent the middle course. The father-engines are of much higher power with bigger cylinders and are not available at this relatively high speed. Medium speed diesel engines of this power are available with maximum speeds of approximately 500 rpm. Another consideration that has to be made is the choice between line or vee engines. Checking the diesel engine database, all 4-stroke diesel engines in this power range are vee engines. The power level is such that a large number of cylinders is required. To keep it as compact as possible, vee engines are chosen.

Son-engine	
Power	3 MW
Speed / type	1000 rpm / V-engine
L x W x H	4.01 x 2.00 x 2.82 m
Inlet / Outlet duct	0.243 / 0.228 m^2
Weight	20.2 ton
Estimated costs	1360 k€
Nominal efficiency	0.43

Father-engine	
Power	15.4 MW
Speed / type	500 rpm / V-engine
L x W x H	11.79 x 3.97 x 5.98 m
Inlet / Outlet duct	1.247 / 1.170 m^2
Weight	152.7 ton
Estimated costs	4582 k€
Nominal efficiency	0.44

Gearbox

The gearbox is necessary to reduce the shaft speed of the diesel engines to values suitable for driving the propeller (130 rpm at 18 knots, 212 rpm at 30 knots). The gearbox has an *and*-configuration, so both engines can drive the propeller at the same time. This means that the gear ratio of the son-engines has to be suitable to drive the propeller at 212 rpm, but also at 130 rpm. Running at 130 rpm is of course possible with a fixed gear ratio for 212 rpm, but then the engine has to run slower. At slower engine speeds the full torque can not be delivered. With the pitch control of the propeller the load has to be adjusted such that the engine can deliver enough torque and run efficiently at cruise mode. Otherwise the son-engine should have two different gear ratios, one for cruise mode and one for high speed mode. Assuming the first option, the gear ratio i_{GB} should be $1000 : 212 = 4.72$. Maximum input speed from the father-engines is 600 rpm. The gear ratio i_{GB} for the father-engines should be $600 : 212 = 2.83$.

To estimate dimensions and weight, the ratio between input power and output speed $\frac{P}{N}$ in (kW/rpm) is used. For the father-engines this value is 72.6, and for the son-engines 14.2.

To couple and decouple the diesel engines there are clutches needed. Every engine has a friction plate clutch or a fluid coupling for better noise reduction.

The gearing wheels have double helical teeth to lower the noise production. To further decrease underwater noise, the gearbox should be placed on stiff resilient mountings.

Power/speed ratio (father-engines)	72.6 kW/rpm
Power/speed ratio (son-engines)	14.2 kW/rpm
Type	Twin, multiple stage, double helical
L x W x H	2.64 x 4.39 x 3.12 m
Weight	45.8 ton
Estimated costs	1935 k€
Nominal efficiency	0.98

Diesel-generator

Description according to concept 1.

Power (diesel)	0.85 MW
Power (generator)	0.92 MWe
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(1.74 + 1.41) x 1.18 (0.81) x 1.60 m
Inlet / Outlet duct	0.069 / 0.065 m^2
Weight	6.2 + 2.2 ton
Estimated costs	1173 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

Description according to concept 1.

Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

D=Depth.

Auxiliary equipment

Description according to concept 1, but without auxiliary equipment for gasturbines. The fuel tolerance of diesel engines is higher than of gasturbines, so a separator will not be needed.

4.4.2 Operational characteristics

Maneuverability	−−
Signature profile	−
Redundancy	2

The maneuverability of this propulsion concept is significantly lower when compared to the reference concept. At low speeds the maneuverability is the same, but at higher speeds, the father-engines are used instead of gasturbines in the reference concept. Diesel engines have slower reaction times and can not deliver full torque at all speeds.

Signature profile is worse than the reference concept in particular when looked at underwater noise, because diesel engines produces louder and lower frequency structure-borne noise than gasturbines. However, the difference is only noticeable at higher speeds, and in this range the underwater noise requirements are not applicable.

The minimum redundancy level is determined by the 2 gearboxes, as with the reference concept.

4.4.3 Integration in ship

Nr. of main components	14
Total space consumption	715 m^3
Total weight	549 ton
Fuel capacity	301.7 ton / $3.55 \cdot 10^5$ m^3

4.4.4 Availability

Reliability	++
Maintainability	−−
Shock resistance	−−

Reliability of this concept is higher than of the reference concept, because a diesel engine has a higher reliability than a gasturbine. On the other hand, diesel engines need more maintenance. And because most of the maintenance on diesel engines is done by the crew it adds complexity to the maintenance for the crew. This concept has 8 diesel engines, so maintainability of this concept is lower than reference. Shock resistance of this concept is lower, because of the big diesel engines. The large surface of the engines makes them vulnerable to shockwaves.

4.4.5 Costs

Initial purchase	21.9 M€
Annual fuel cost / consumption	3.03 M€/ 3434 ton
Annual maintenacne cost	0.119 M€

This concept has low investment costs when compared to the reference concept. Gasturbines are very expensive machines in comparison with medium and high speed diesel engines. The absence of gasturbines saves a lot of investment costs in this propulsion concept. Besides that the annual fuel costs of this concept are lower, because the diesel engines have higher efficiency than gasturbines thus use less fuel.

Annual fuel consumption, thus fuel cost is lower than the reference concept. High speeds are sailed with the father-engines. The diesel engines are more fuel efficient than the gasturbines in the reference concept.

4.5 Concept 4/CODLAG/Hybrid

The fourth concept combines mechanical and electrical propulsion, which makes it a hybrid concept. It is a CODLAG concept, which means COmbined Diesel-eLectric And Gasturbine. This means that the propellers can be driven solely electrical, or in combination with gasturbines. Two variants are shown, 4.a (figure 4.4)and 4.b (figure 4.5), which are in essence the same, but are shown to point out the possibilities. The electric motor can be placed directly on the shaft, concept 4.a, as found on the United Kingdom Type 23 frigates. But the electric motor can also be placed in front of the gearbox without being coupled to the gearbox (via hollow shaft arrangement), concept 4.b. This special gearbox arrangement is also found on the German F125 class frigate, but only with 1 gasturbine. Advantage of this arrangement is that the shaft can still be used in case the electric motor is unserviceable. This is not possible in configuration 4.a.

A hybrid concept like this gives high flexibility. Depending on the required power more or less diesel-generators are in service. Diesel-generators are in two different sizes to have optimal engine loading at all points and to have lower fuel consumption.

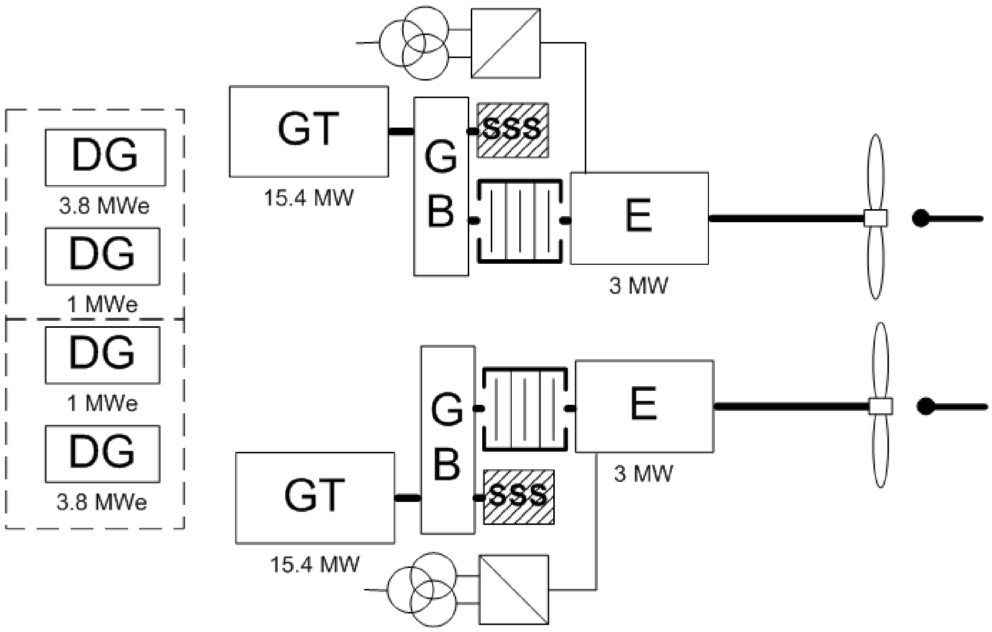


Figure 4.4: Propulsion concept 4a, CODLAG, electric motor on the shaft

With this configuration there are 4 different operating modes:

1. High speed mode (2 gasturbines and 2 E-motors on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.

2. Cruise speed mode (2 E-motors on 2 shafts). For travelling long distances and performing operations. Max. speed ≈18 knots.

3. High speed on 1 shaft mode (1 gasturbine and 1 E-motor on 1 shaft, other shaft trailing). Max. speed ≈25.5 knots. If one shaft or gearbox is unserviceable for whatever reason.

4. Cruise speed on 1 shaft mode (1 E-motor on 1 shaft, other shaft trailing). If one shaft or gearbox is unserviceable for whatever reason. Max. speed ≈14 knots.

Configuration 4.b. has an extra operating mode:

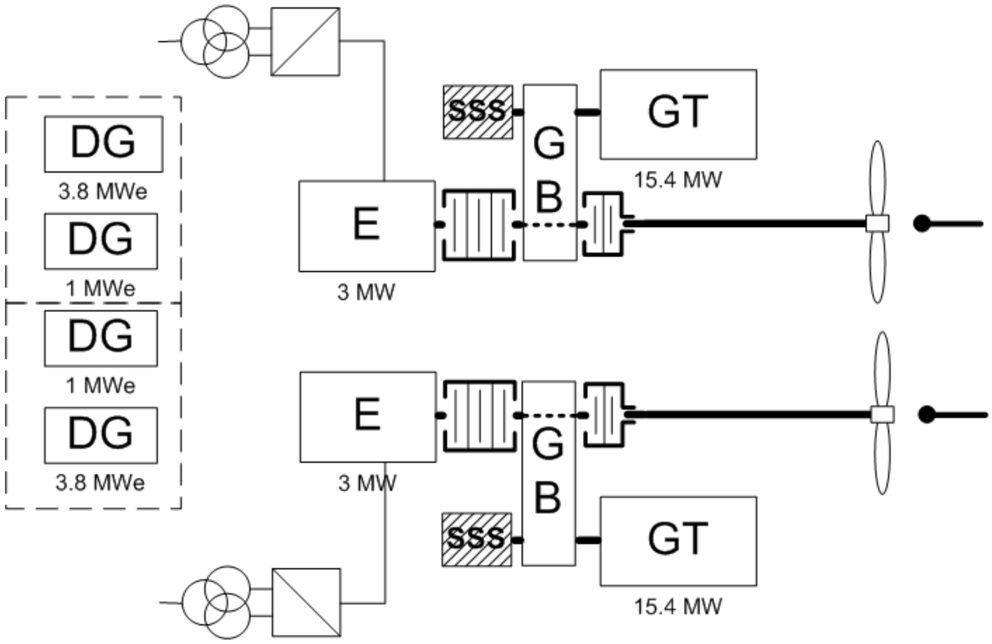


Figure 4.5: Propulsion concept 4b, CODLAG, electric motor in front of the gearbox

5. Cruise speed on 1 shafts, high speed on other shaft (1 E-motor on 1 shaft, 1 gasturbine on other shaft). Gasturbine power is limited to E-motor power. If one E-motor is unserviceable for whatever reason and power on 2 shafts is required. Max. speed ≈18 knots.

4.5.1 Components

Main components in this propulsion concept are 2 fixed pitch propellers both driven by an electric motor of 3 MW **and/or** a gasturbine of 15.4 MW. Gasturbine speed is reduced in a reduction gearbox, which are coupled to each other through a SSS-clutch. Electric motors are slow speed and directly coupled to the shaft with high-torque multiplate friction clutches. Electric power supply by 4 diesel generator sets in 2 different power outputs, for optimal engine loading at most common operational conditions.

Main components Concept 4 (CODLAG)

Component	Number	Power (MW)
Fixed pitch propeller	2	18.4
E-motor	2	3
Gasturbine	2	15.4
Gearbox	2	15.4
Converter	2	3
Diesel-generator	2 / 2	3.9 / 1.0
Switchboard	2 / 2	7.6 / 1.6
	18	

Fixed pitch propeller

Fixed pitch propellers are used to propel the ship. In combination with electric motors, which can deliver full torque at all speeds in four quadrants, the desired level of maneuverability can be achieved with FPP's. In the previous concepts, CPP's were inevitable because of the diesel engines. The FPP can be designed for higher cavitation inception speeds, but because the pitch can't be controlled the efficiency will be somewhat lower at off-design speeds. Depending on the speed profile of the ship, the propeller efficiency is optimized for the most common speed. In this speed profile 15 knots is the most common speed. From analysis on Wageningen B data, $P/D = 1.1$ seems to give the best efficiency results at all speeds, with a peak efficiency of 0.72 at 15 knots. With this P/D-value, the propeller rotates with 225 rpm at 30 knots and 130 rpm at 18 knots. The method to determine these values is explained in Appendix D.

Power	18.4 MW
Speed / type	225 rpm / FPP
Diameter	4.5 m
Estimated weight	10.7 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	411 k€
Nominal efficiency	0.72

Electric motor

Two electric motors are installed, together delivering a maximum of 6 MW, which is enough to power the propellers up to transit speed (incl. 7% margin). Propeller speed is 130 rpm. But because the E-motor is directly coupled to the shaft and also has to deliver power at higher speeds, the motor should be able to rotate at higher speeds. Through field weakening the torque is decreased and the speed increased, and still delivering the maximum amount of power, but now at higher speeds.

The electric power to the E-motors is provided by diesel-generator sets. The big advantage of this configuration is the low noise, because the diesel engines can be placed higher up in the ship on double resilient mountings and even in an acoustic enclosure, and the E-motors are relatively silent. It depends a bit on the type of E-motor and the combination with the converter how much noise is produced by the E-motor. From signatures point of view the DC-motor seems most favourable or otherways the AC synchronous motor. The DC-motor has more intensive maintenance, but smaller and cheaper converter. With the right measures, the AC synchronous motor can probably be as silent as the DC-motor. An extensive cost analysis, taking also into account the cost of the matching converter and maintenance, should show which solution is in the end the best. The cost models that were used in this thesis are not accurate enough to make this analysis. The choice is now made for AC synchronous machines, based on the fact that these have less weight, which could imply lower purchase costs, and have less maintenance, which means lower life cycle costs. If in the near future high temperature superconduction (HTS) technology is well established, this could be a more interesting option. Permanent magnet machines are also a very interesting option, because of their higher power density, but they are about twice the price of a conventional machine. For now, conventional technology is assumed, considering the reliability. Considering the rather high power of the E-motors, they are fed with high voltage electrical power to limit the currents through the distribution network and through the motor windings. A separate high voltage network has the advantage that the disturbances on this network, caused by the motors, are not directly noticeable on the low voltage network, where for instance the SEWACO users are.

Power	3 MW
Design speed / type	130 rpm / AC synchronous
L x W x H	3.70 x 2.31 x 4.54 m
Weight	46.7 ton
Estimated costs	3126 k€
Nominal efficiency	0.973

A L/D-ratio of 1.6 is assumed. For comparison: a HTS motor would be approximately: 2.26 x 1.41 x 1.66 m and 6.4 tons, and a PM motor: 2.43 x 1.52 x 1.79 m and 17.5 tons, according to the models.

Gasturbine

Description according to concept 1, with the difference that power of the gasturbines is now 15.4 MW instead of 18.4 MW, because the remaining 3 MW is delivered by the E-motors.

Power	15.4 MW
Speed / type	5600 rpm / Simple cycle
L x W x H	6.99 x 2.45 x 3.06 m
Inlet / Outlet duct	2.97 / 2.54 m^2
Weight	18.3 ton
Estimated costs	5360 k€
Nominal efficiency	0.36

Gearbox

In this propulsion configuration there are only gearboxes needed to reduce the shaftspeed of the gasturbines. It has no combining function, because the E-motors are directly on the shaft.

Configuration 4.a. has a simple reduction gearbox, with a SSS-clutch to couple the gasturbine to the gearbox and a high-torque multiplate friction clutch to couple the E-motor plus propeller to the gearbox. With this configuration it is possible to sail slow speeds on the electric motors with the gearbox uncoupled, for reasons of underwater noise.

Configuration 4.b. has a more complex gearbox arrangement. The E-motor is coupled to the propeller through a high-torque friction clutch. This shaft goes through the gearbox without being coupled to the gearing wheels, with a hollow shaft arrangement. When the gasturbine is operated, the hollow shaft, which is fixed to the gearing wheel, is coupled to the propeller shaft with another friction clutch, and the gasturbine is coupled to the gearbox through an SSS-clutch. The advantage of this arrangement is that there is more flexibility in the placing of the E-motor, and the E-motor can be isolated from the propulsion train when it is broken or in maintenance.

Maximum input speed from the gasturbine is 5600 rpm. The 4.5 m Wageningen B-565 should be operated at maximum 225 rpm. This means the gearbox should have a gear ratio i_{GB} of 24.89. Ratio between input power and output speed, $\frac{P}{N}$ in (kW/rpm), is 68.44.

Power/speed ratio	68.4 kW/rpm
Type	Single, single stage, double helical
L x W x H	2.58 x 2.63 x 3.07 m
Weight	19.9 ton
Estimated costs	1014 k€
Nominal efficiency	0.99

Converter

The power and speed of the electric motors needs to be controlled by voltage and frequency. For this purpose, a converter is needed. The electrical machines are only used as motor, so the regenerative energy from the motor has to be dissipated in the converter. The electric motors are of the conventional AC synchronous type. Such motors can be controlled by different types of converters: cyclo-converter, synchro-converter or PWM-converter. Synchro-converter is not preferred for its highly variable power factor, high torque ripple and higher harmonics on the net. Cyclo-converter is especially for low speed applications, which is the case, and produces a smooth sine, which lowers the underwater noise of the motor. Downside of the cyclo-converter is the lower reliability in relation to PWM converters, because of the high number of switching elements. PWM-converters also have a higher power density than cyclo-converters, constant high power factor (> 0.95) and are more commonly used. Besides, there were no relations derived for dimensions, weight and costs of cyclo-converters, because this converter type is not found on RNLN ships. To make a good comparison, a more detailed cost model is needed. With the data known from this thesis the choice is made for PWM-converters.

The electric motors deliver a maximum of 3 MW mechanical power. Assuming a nominal motor efficiency of 97%, this means 3.1 MWe. This electrical power is delivered by the converter, which itself also has some losses. Assuming a power factor of 0.95, 2 converters with apparent power of 3.26 MVA are installed to drive the E-motors. The choice can be made for the type of semiconductors that is used in the PWM-converter. From the dimensions analysis, it was found that converters with GTO or IGCT as semiconductors are smaller than with IGBT's.

Apparent power	3.3 MVA
Type	PWM, GTO/IGCT
D x W x H	0.90 x 4.75 x 2.30 m
Weight	4.0 ton
Estimated costs	436 k€
Nominal power factor	0.98

D=Depth.

Diesel-generator

Because this concept has a partly electric propulsion, there is larger electrical power demand, but only when the E-motors are operated. Up to transit speed, the E-motors are used for propulsion, with a maximum power demand of 6.2 MWe (assuming 97% motor efficiency). Maximum auxiliary power is 1.35 MWe, including a safe margin of 0.25 MWe for future upgrades or extra high cooling demands in tropical areas, gives 1.6 MWe. Preferably this amount of power is generated by two diesel-generator sets, such that there are two redundant sets. To deliver 7.8 MWe with two DG's, they need to deliver 3.9 MWe each, which is pretty large. If four 3.9 MWe DG's are installed there is full redundancy of power supply. The problem then is that at high speeds (on only gasturbines) or in harbour, when the power demand is much lower, these engines run on very low partload (20% in harbour). This is bad for the engines and efficiency. So, to also have a proper engine loading in these conditions, smaller DG's should be installed next to the bigger ones. 1 MW seems reasonable, because then there is a 88% engine loading in harbour, which is optimal for fuel consumption, at operational speed 1 with 3.3 MWe propulsion power demand plus a maximum of 1.6 MWe auxiliary power, the power can be delivered by one big and one small DG, at gasturbine mode the auxiliary power is delivered by two small DG's with 80% engine loading. So, with 3.9 MWe father DG's and 1 MWe son DG's all conditions can be served with power in an 'engine friendly' way. But there is a lower level of redundancy, because if one big DG fails or is in maintenance, the ship is not able to do transit speed on E-motors together with full auxiliary load anymore. This should be kept in mind.

The losses in the generator and distribution network should be taken into acount by selecting the diesel engine. The losses are assumed to be 4% ($0.97 \cdot 0.99$). Which means the diesel engines have to deliver 4.1 MW and 1.1 MW mechanical power. To prevent generator overload in case of temporary diesel engine overload, the generator is a little overdimensioned: +7.5% of nominal diesel engine power. This means a generator with 4.4 MW nominal power and 1.2 MW. Diesel engines of the 4.1 MW power level are normally medium speed line engines with speeds around 600 to 750 rpm. Because of 60 Hz generation, the engine speed is chosen 720 rpm. The 1.1 MW diesel engines are 900 rpm V-engines.

Because this propulsion concept has high voltage E-motors, the generated electrical power should be of high voltage, to limit the current in the system. There is no rule for the voltage to use, but through the years a certain commonly accepted standard voltage of 6.6 kV is adapted.

The L/D ratio of the generators is chosen such that the height of the generator (incl. cooling) is equal to the height of the diesel engine: $L/D = 1.2$ for father DG's and $L/D = 1.5$ for son DG's.

Father DG's	
Power (diesel)	4.1 MW
Power (generator)	4.4 MW
Speed / type (diesel)	720 rpm / Line engine
Speed / type (generator)	720 rpm / Conventional AC synchronous
L x W x H	(5.89 + 2.12) x 2.63 (1.63) x 3.25
Inlet / Outlet duct	0.332 / 0.312 m^2
Weight	35.4 + 13.4 ton
Estimated costs	2970 k€
Nominal efficiency (diesel)	0.44
Nominal efficiency (generator)	0.97

Son DG's	
Power (diesel)	1.1 MW
Power (generator)	1.2 MW
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(2.07 + 1.48) x 1.31 (0.91) x 1.78
Inlet / Outlet duct	0.089 / 0.084 m^2
Weight	8.0 + 2.9 ton
Estimated costs	1388 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

The diesel-generators in this concept generate high voltage electrical power, which means that there are high voltage switchboards needed. For redundancy reasons there are two, both capable of distributing the maximum amount of 7.8 MWe (2 motors + maximum auxiliary load). Both high voltage boards have 5 input fields (4 DG inputs, 1 bus coupler) and 4 outgoing fields (2 E-motor, 2 low voltage switchboard). In this way the power distribution is fully redundant. If one switchboard fails the other can take over full capacity. Between the high and low voltage switchboard is a transformer, which transforms from 6.6 kV to 440 V. This transformer is not further considered in this analysis. There are 2 low voltage switchboards both capable of distributing the maximum amount of 1.6 MWe to the auxiliary systems. The low voltage boards have 4 incoming fields (2 transformer inputs, 1 bus coupler, 1 shore connection). The number of outgoing fields is estimated to be 8.

For better shock-resistance the switchboards are mounted on springs.

D=Depth.

High voltage switchboard	
Power	7.8 MWe
Nr. of incoming fields	5
Nr. of outgoing fields	4
D x W x H	1.70 x 5.85 x 2.60 m
Weight	5.0 ton
Estimated costs	1087 k€
Nominal efficiency	0.995

Low voltage switchboard	
Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

Auxiliary equipment

The auxiliary equipment that is needed for this propulsion concept are: high pressure air starting systems for diesel engines and gasturbines, fuel feed pumps for diesel engines and gasturbines, at least one fuel seperator, fresh water cooling system for diesel-generators, electric motors and converters, voltage transformers, in- and outlet ducts for diesel engines and gasturbines, and exhaust gas after treatment equipment to reduce emissions.

4.5.2 Operational characteristics

Maneuverability	+++
Signature profile	++
Redundancy	3

The maneuverability of this propulsion concept is significantly better than the reference concept. The E-motors have the advantage that they can deliver full torque at all speeds in four quadrants. This gives very high maneuverability, because the E-motors can be used to rapidly reverse the shaft rotation and brake the ship, or to rapidly accelerate from all speeds. The gasturbines also give a high maneuverability due to their wide operating envelope and fast power delivery.

The signature profile of this propulsion concept is better than the reference. The operational speeds are sailed on electric motors, which give lower underwater noise than propulsion diesels with a gearbox. In particular the gearbox is known to be an important producer of underwater noise. The diesel-generators that are used to generate electrical power can be placed higher up in the ship, on double resilient mountings and even in an acoustic enclosure to minimize the noise. A fixed pitch propeller is used which can have higher cavitation inception speeds. At high speeds the gasturbines are driving the propeller through the gearbox, this produces considerably more noise, but at these high speeds the underwater noise is of less concern.

Redundancy of this concept is such, that in a worst case scenario with failure of 2 father diesel-generator sets and 1 son diesel-generator, the ship is not able to sail 10 knots anymore and deliver 1350 kWe. The failure of 2 gearboxes is not the limiting factor in this concept, because the ship can always continue on solely E-motors. So, the minimum number of components to fail before the ship isn't able to sail 10 knots anymore with 1350 kWe auxiliary power is 3.

4.5.3 Integration in ship

Nr. of main components	18
Total space consumption	450 m^3
Total weight	334 ton
Fuel capacity	312.4 ton / $3.68 \cdot 10^5$ m^3

The required fuel capacity to reach 5000 nm at 18 knots, is with this propulsion concept slightly higher than with the reference concept. This is caused by the higher losses of the electrical propulsion due to the larger number of energy conversion steps. At 18 knots, the cruise speed engines of the reference concept have an optimal loading, so at this speed the electrical propulsion can't compete with direct mechanical propulsion.

4.5.4 Availability

Reliability	+
Maintainability	++
Shock resistance	−

This concept has less prime movers when compared to the reference concept. The electric motors and equipment have a rather high reliability when compared to prime movers. This concepts reliability is estimated somewhat higher than the reference concept, although it has more components and the rather complex gearboxes and clutches. The fixed pitch propeller has a higher reliability than a controllable pitch propeller. The level of maintenance is lower, because the electrical components need very few maintenance, where the diesel engines relatively need a lot of maintenance. This concept has less diesel engines than reference concept. Though, in practice the maintenance on electrical equipment is experienced as more difficult.

Shock resistance of this concept is estimated worse. There are more components which can suffer from shockwaves, and in particular sensitive electronical equipment. Switches in the switchboards are very sensitive to shockwaves. The electric motors have low tolerances between rotor and stator, and a large displacement can cause severe damage to the motor. To decrease the sensitivity to shockwaves, the equipment is placed on springs, preferably high up in the ship.

4.5.5 Costs

Initial purchase	32.0 M€
Annual fuel cost / consumption	3.44 M€/ 3903 ton
Annual maintenance cost	0.140 M€

The investment costs of this concept are higher than the reference concept. It is generally known that electrical propulsion equipment has higher investment costs, but because fuel consumption is lower and it gives a high grade of flexibility the higher investment might be earned back within reasonable time, especially when the ship sails more slow speeds on E-motors.

Annual fuel consumption of this concept is lower than the reference concept. The E-motors have a high efficiency at all loads, and by using more or less diesel-generators to power the E-motors, the engine loading can be optimized to go for lower fuel consumption. Disadvantage of the electrical propulsion is the larger number of energy conversion steps which gives higher losses. But as can be seen, the annual fuel consumption is still lower than the reference concept.

4.6 Concept 5/CODLAG/Hybrid

This hybrid concept is very much the same as the previous concept, only with 1 gasturbine instead of 2. There are also 2 variants shown. Concept 5.a (figure 4.6) is found on the Italian FREMM class and concept 5.b (figure 4.7) is found on the German F125 class. In figure 4.7 the gasturbine is drawn right of the gearbox, but it could as well be placed left of the gearbox.

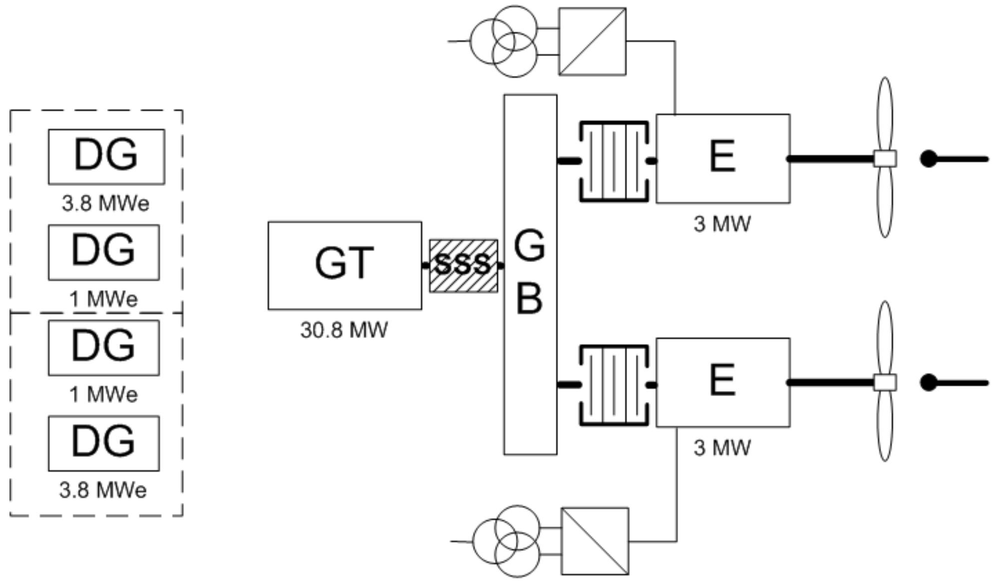


Figure 4.6: Propulsion concept 5a, CODLAG, electric motor on the shaft

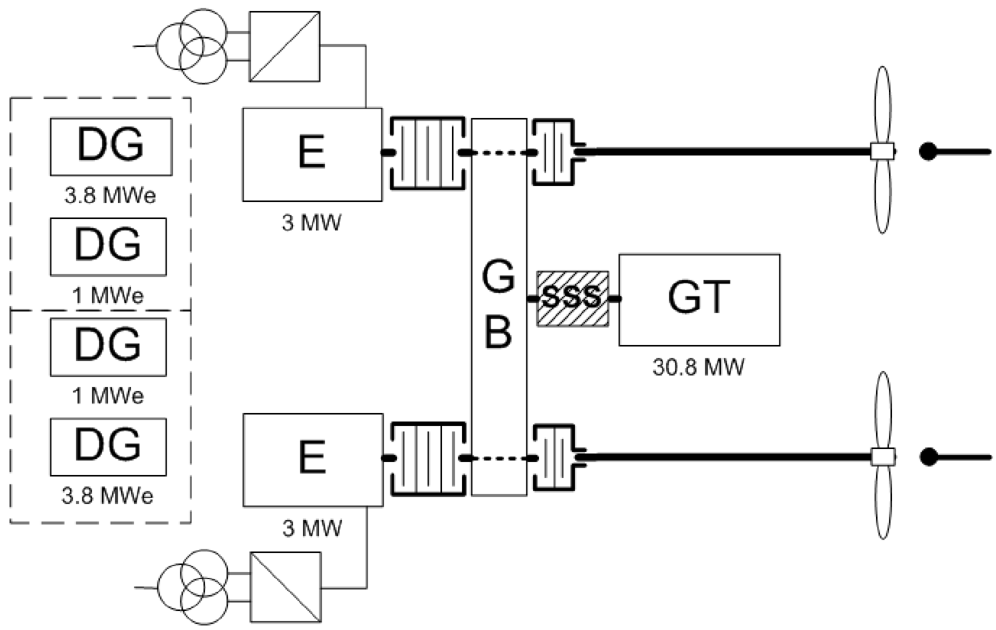


Figure 4.7: Propulsion concept 5b, CODLAG, electric motor before the gearbox

With this configuration there are 4 different operating modes:

1. High speed mode (1 gasturbine and 2 E-motors on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.

2. Cruise speed mode (2 E-motors on 2 shafts). For travelling long distances and performing operations. Max. speed ≈18 knots.
3. High speed on 1 shaft mode (1 gasturbine and 1 E-motor on 1 shaft, other shaft trailing). Max. speed ≈25.5 knots. If one shaft is unserviceable for whatever reason. Power of the gasturbine is limited by the capacity of the clutches, gearbox, shaft and propeller, if these are designed for full power, max. speed could be ≈29 knots.
4. Cruise speed on 1 shaft mode (1 E-motor on 1 shaft, other shaft trailing). If one shaft is unserviceable for whatever reason. Max. speed ≈14 knots.

Configuration 5.b. has two extra operating modes:

5. Cruise speed on 1 shafts, high speed on other shaft (1 E-motor on 1 shaft, 1 gasturbine on other shaft). Gasturbine power is limited to E-motor power. If one E-motor is unserviceable for whatever reason and power on 2 shafts is required. Max. speed ≈18 knots.
6. Cruise speed 1 E-motor on 2 shafts. Could also be performed with configuration 5.a, but doesn't make sense, because partload efficiency of E-motors is very good. In configuration 5.b this mode is used if one E-motor is unserviceable for whatever reason and power on 2 shafts is required. Max. speed ≈14 knots.

4.6.1 Components

Main components in this configuration are 2 fixed pitch propellers both driven by an electric motor of 3 MW **and/or** by one shared gasturbine of 30.8 MW. Gasturbine power is divided over two shaftlines in a cross-connection gearbox in which also the speed is reduced. Electric motors are slow speed and directly coupled to the shaft. Electric power supply by 4 diesel generator sets in 2 different power outputs, for optimal engine loading at most common operational conditions, like concept 4.

Main components Concept 5 (CODLAG)

Component	Number	Power (MW)
Fixed pitch propeller	2	18.4
E-motor	2	3
Gasturbine	1	30.8
Cross-connect gearbox	1	30.8
Converter	2	3
Diesel-generator	2 / 2	3.9 / 1.0
Switchboard	2 / 2	7.6 / 1.6
	16	

Fixed pitch propeller

Description according to concept 4.

Power	18.4 MW
Speed / type	225 rpm / FPP
Diameter	4.5 m
Estimated weight	10.7 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	411 k€
Nominal efficiency	0.72

Electric motor

Description according to concept 4.

Power	3 MW
Design speed / type	130 rpm / AC synchronous
L x W x H	3.70 x 2.31 x 4.54 m
Weight	46.7 ton
Estimated costs	3126 k€
Nominal efficiency	0.973

A L/D-ratio of 1.6 is assumed. For comparison: a HTS motor would be approximately: 2.26 x 1.41 x 1.66 m and 6.4 tons, and a PM motor: 2.43 x 1.52 x 1.79 m and 17.5 tons, according to the models.

Gasturbine

Description according to concept 1, with the difference that there is only one gasturbine with a rated power of 30.8 MW, the remaining 6 MW to reach 30 knots is delivered by the two E-motors.

Power	30.8 MW
Speed / type	3600 rpm / Simple cycle
L x W x H	8.49 x 2.78 x 3.28 m
Inlet / Outlet duct	5.94 / 5.08 m^2
Weight	24.3 ton
Estimated costs	10718 k€
Nominal efficiency	0.39

Gearbox

This concept has a cross-connection gearbox with a speed reduction for the gasturbine. The power of the gasturbine is divided over two shafts by the cross-connection gear. Further description of concept 5.a and 5.b according to concept 4.a and 4.b.

Maximum input speed from the gasturbine is 3600 rpm. The 4.5 m Wageningen B-565 should be operated at maximum 225 rpm. This means the gearbox should have a gear ratio i_{GB} of 16. Ratio between input power and output speed, $\frac{P}{N}$ in (kW/rpm), is 136.9. To determine the dimension, the gearbox is modeled as a twin gear with two time a power/speed ratio of 136.9.

Power/speed ratio	136.9 kW/rpm
Type	Cross-connect, multiple stage, double helical
L x W x H	3.33 x 6.34 x 3.70 m
Weight	86.2 ton
Estimated costs	3158 k€
Nominal efficiency	0.97

Converter

Description according to concept 4.

Apparent power	3.3 MVA
Type	PWM, GTO/IGCT
D x W x H	0.90 x 4.75 x 2.30 m
Weight	4.0 ton
Estimated costs	436 k€
Nominal power factor	0.98

Diesel-generator

Description according to concept 4.

Father DG's	
Power (diesel)	4.1 MW
Power (generator)	4.4 MW
Speed / type (diesel)	720 rpm / Line engine
Speed / type (generator)	720 rpm / Conventional AC synchronous
L x W x H	(5.89 + 2.12) x 2.63 (1.63) x 3.25
Inlet / Outlet duct	0.332 / 0.312 m^2
Weight	35.4 + 13.4 ton
Estimated costs	2970 k€
Nominal efficiency (diesel)	0.44
Nominal efficiency (generator)	0.97

Son DG's	
Power (diesel)	1.1 MW
Power (generator)	1.2 MW
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(2.07 + 1.48) x 1.31 (0.91) x 1.78
Inlet / Outlet duct	0.089 / 0.084 m^2
Weight	8.0 + 2.9 ton
Estimated costs	1388 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

Description according to concept 4.

High voltage switchboard	
Power	7.8 MWe
Nr. of incoming fields	6
Nr. of outgoing fields	4
D x W x H	1.70 x 5.85 x 2.60 m
Weight	5.0 ton
Estimated costs	1087 k€
Nominal efficiency	0.995

Low voltage switchboard	
Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

Auxiliary equipment

Description according to concept 4.

4.6.2 Operational characteristics

Maneuverability	+++
Signature profile	++
Redundancy	3

The maneuverability of this propulsion concept is comparable to that of concept 4, which is significantly better than the reference concept. Description according to concept 4.

The signature profile of this propulsion concept is comparable to that of concept 4, which is significantly better than the reference concept. Description according to concept 4.

4.6.3 Integration in ship

Nr. of main components	16
Total space consumption	459 m^3
Total weight	368 ton
Fuel capacity	312.4 ton / $3.68 \cdot 10^5$ m^3

4.6.4 Availability

Reliability	+
Maintainability	+++
Shock resistance	−

Description according to concept 4. Maintainability is rated higher, because there is one gasturbine less to maintain. In combination with the low-maintenance electrical equipment, this gives the concept a superior score on maintainability.

4.6.5 Costs

Initial purchase	33.2 M€
Annual fuel cost / consumption	3.48 M€/ 3950 ton
Annual maintenance cost	0.176 M€

Higher investment costs than concept 4, primarily because of the cross-connection gearbox. Higher fuel consumption because of the rather bad loading of the gasturbine at speeds between 18-26 knots.

4.7 Concept 6/CODLADAD/Hybrid

The sixth concept is also a hybrid concept, the CODLADAD concept, COmbined Diesel-eLectric And Diesel And Diesel. It is comparable to concept 3, but it has a Power Take In/Power Take Out (PTI/PTO) possibility on the shaft. At low speeds the propulsion power is delivered by electric motors, which can be used as shaft generators when the power is delivered by the diesel engines. A comparable concept is found on the Korean Coast Guard KCG3000 vessel. In figure 4.8, the electric motors are positioned in front of the gearbox, but they might as well be positioned after the gearbox, see also Concept 4.a (figure 4.4) and Concept 4.b (figure 4.5).

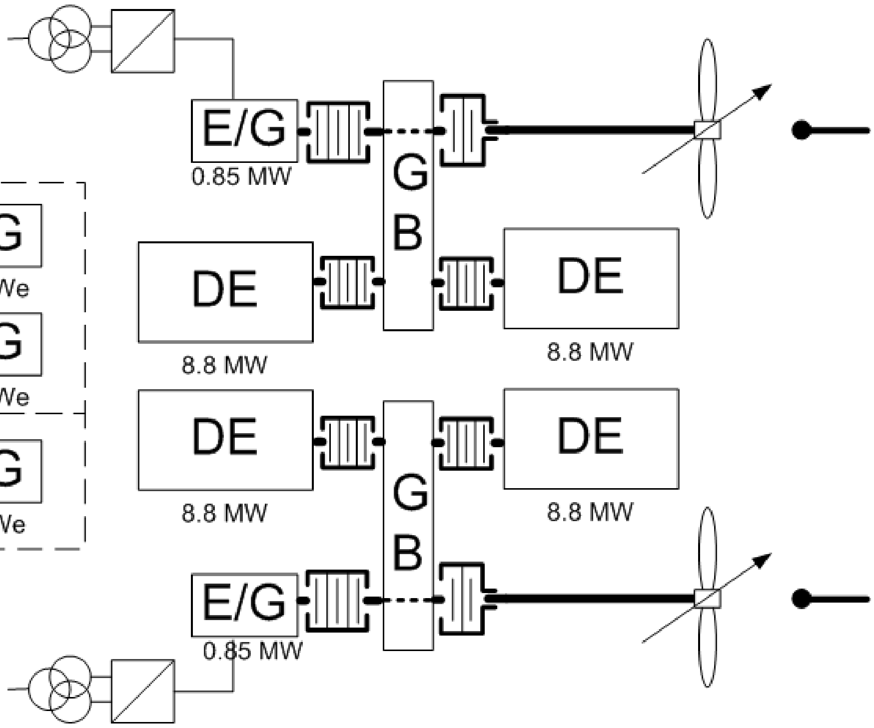


Figure 4.8: Propulsion concept 6, CODLADAD

With this configuration there are 8 different operating modes:

1. High speed mode (4 diesel engines and 2 electric motors on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.

2. High speed mode (4 diesel engines and 2 generators on 2 shafts). Max. speed ≈29.5 knots.

3. Semi-high speed on 1 shaft mode (2 diesel engines and 1 generator on 1 shaft, other shaft trailing). Max. speed ≈25 knots. If one shaft or gearbox is unserviceable for whatever reason and high speeds are required.

4. Semi-high speed on 2 shaft mode (2 diesel engines and 2 generators on 2 shafts). Max. speed ≈25 knots. If semi-high speeds are required this is the most efficient mode.

5. Cruise speed mode (1 diesel engine and 1 generator on 1 shaft, other shaft trailing). For travelling long distances. Max. speed ≈20 knots.

6. Cruise speed mode (1 diesel engine and 1 electric motor on 1 shaft, other shaft trailing). Max. speed ≈21 knots.

7. Slow speed on 2 shaft mode (2 electric motors on 2 shafts). Max. speed ≈12 knots. If silent slow speed operations are performed.

8. Slow speed on 1 shaft mode (1 electric motor on 1 shaft, other shaft trailing). Max. speed ≈9 knots. If silent very slow speed operations are performed, or if one shaft or gearbox is unserviceable for whatever reason.

It is clear from the large number of operating modes that this concept offers a great operating flexibility.

4.7.1 Components

Main components in this propulsion concept are 2 controllable pitch propellers both driven through a combination gearbox by maximal 2 diesel engines of 8.8 MW **and/or** directly driven by a slow speed electric motor, which can also be operated as generator. The **and**-possibility of the gearbox adds some extra complexity and requires proper load-sharing-control between the 2 diesel engines on 1 shaft. The diesel engines are all coupled through multiplate friction clutches in combination with flexible coupling or through a fluid coupling to the gearbox. With a hollow shaft arrangement and another multiplate friction clutch the gearbox is coupled to the shaft. The electric motor/generator is coupled with multiplate friction clutches directly to the shaft. Electric power is supplied by the shaft generators in diesel drive mode or by a maximum of 3 diesel generator sets in a father-son configuration in electric drive mode.

Main components Concept 6 (CODLADAD)

Component	Number	Power (MW)
Controllable pitch propeller	2	18.4
E-motor/Shaft generator	2	0.85
Diesel engine	4	8.8
Gearbox	2	18.4
Converter	2	0.9
Diesel-generator	2 / 1	1.7 / 1
Switchboard	2	3.4
	17	

Controllable pitch propeller

Description according to concept 1.

Power	18.4 MW
Speed / type	212 rpm / CPP
Diameter	4.5 m
Estimated weight	13.8 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	515 k€
Nominal efficiency	0.71

Electric motor/Shaft generator

This propulsion concept has 2 electric motors that are also used as shaft generators. They both deliver a maximum mechanical power of 0.85 MW, together 1.7 MW which means that they can deliver power up to operational speed 2 (12 knots). In generator mode a maximum of 1.7 MW mechanical power is absorbed and approximately 1.6 MWe is generated, assuming 97% efficiency for the generator and also for the frequency converter. The motor/generator is coupled directly to the propeller shaft, which means it has to be a low speed machine. It is designed to deliver maximum torque at 90 rpm, this is the propeller speed at 12 knots, but it also has to be able to deliver/absorb power at higher rotational speeds (maximum 212 rpm). Through field weakening the torque is decreased and the speed increased, and still delivering the maximum amount of power, but now at higher speeds.

The motors/generators are of the AC synchronous type. As explained earlier in concept 4, for now conventional technology is assumed, but in the near future when HTS technology is well established that can be applied. Considering the rather limited power of the motors/generators, they are fed with/generating low voltage electrical power, so no separate high voltage network has to be installed. This saves an extra electrical network with extra switchboards and the crew does not need extra education. A disadvantage is that the motors, which produce some disturbance on the net, are on the same network as for instance SEWACO equipment, which is sensitive to harmonic distortion.

Power	0.85 MW
Design speed / type	90 rpm / AC synchronous
L x W x H	2.75 x 1.72 x 3.37 m
Weight	19.1 ton
Estimated costs	1025 k€
Nominal efficiency	0.971

A L/D-ratio of 1.6 is assumed. For comparison: a HTS machine would be approximately: 1.68 x 1.05 x 1.23 m and 2.6 tons, and a PM machine: 1.80 x 1.13 x 1.33 m and 7.2 tons, according to the models.

Diesel engine

In total 4 propulsion diesel engines are installed with equal power of 8.8 MW, together delivering 35.2 MW. At top speeds all diesel engines are operated together with the electric motors, but at operational speeds and transit speed only 1 diesel engine needs to be operated. This offers great flexibility and redundancy, because 4 engines are available.

At speeds above 12 knots, the power is delivered by the diesel engines instead of the electric motors. Power demand is minimal 1.7 MW, this means an engine loading of 20% of nominal power, which is low. Too low loading may cause damaging and fouling of the engine and is bad for specific fuel consumption. To increase the engine loading, the shaft generator can also absorb 0.85 MW, which gives a 29% engine loading.

The speed of the diesel engines will be medium speed. High speed diesel engines are available in this power range, but they have a large number of cylinders, which is not preferred from reliability and maintainability point of view. A representative speed would be 700 rpm. Then there is the choice between line and vee engines. The line engines have somewhat larger dimensions, but the vee engines normally have higher number of cylinders which results in more maintenance and lower reliability. The difference in dimensions is small according to the models, so line engines are selected.

Power	8.8 MW
Speed / type	700 rpm / Line engine
L x W x H	8.56 x 3.25 x 4.62 m
Inlet / Outlet duct	0.713 / 0.669 m^2
Weight	77.6 ton
Estimated costs	3023 k€
Nominal efficiency	0.44

Gearbox

The gearbox has two functions. It combines the power input of the two diesel engines and it reduces the speed. At 30 knots, the selected propeller rotates 212 rpm. Nominal engine speed is 700 rpm, which means that the gear ratio is 700 : 212 = 3.30.

The gearbox has an *and*-function, to deliver the power of two engines together to the outgoing shaft. This requires a proper load-sharing control because otherwise the one engine will drive the other engine. The engines are coupled to the gearbox through multiplate friction clutches in combination with a flexible coupling for some vibration reduction. Another option could be to couple the engines through fluid couplings to the gearbox. This type combines the clutch and vibration reduction function. The electric motor can also drive the shaft but not through a gearing wheel. The electric motor is directly coupled to the shaft. In figure 4.8 the electric motor/generator is positioned in front of the gearbox, this requires the more complex hollow shaft arrangement through the gearbox with an extra clutch. This arrangement was already explained in Concept 4, and has advantage the the motor/generator can be isolated from the shaftline, for example when maintenance is done. If the motor/generator is positioned after the gearbox, on the shaft, this more complex gearbox arrangement is not necessary.

The gearbox is a twin input, double helical, single stage gear with a power/speed ratio of 41.5 kW/rpm on both shafts. Double helical gearing wheels are applied to lower the underwater noise. Single stage reduction is possible because the gear ratio is rather low.

Power/speed ratio	41.5 kW/rpm
Type	Twin, single stage, double helical
L x W x H	2.14 x 4.59 x 2.68 m
Weight	26.2 ton
Estimated costs	1255 k€
Nominal efficiency	0.98

Converter

The power and speed of the electric motors needs to be controlled by voltage and frequency. For this purpose, a converter is needed. But the electric motors are also operated as generators, so the converter also has the function to control the voltage and frequency of the generated power that is delivered to the net. Especially the frequency is highly variable, because the shaft speed is highly variable due to waves etc. The converter has to operate in four quadrants. All AC converter types can operate in four quadrants, on condition that no diode-bridges are used but only semi-conductors. In Concept 4 is already explained for what reasons the PWM-converter is chosen, for this concept this is no different. The price of a converter that works both ways is probably higher, but no detailed information on this type is available.

Both electric motors/generators have their own converter. The converter needs to deliver a maximum of approximately 0.9 MWe to the motors. The converter itself also introduces some losses, assuming a minimum power factor of 0.95 the nominal apparent power of the converters is 0.95 MVA. The choice can be made for the type of semiconductors that is used in the PWM-converter. From the dimensions analysis, it was found that converters with GTO or IGCT as semiconductors are smaller than with IGBT's.

Apparent power	0.95 MVA
Type	Regenerative PWM, GTO/IGCT
D x W x H	0.76 x 4.69 x 2.30 m
Weight	2.3 ton
Estimated costs	132 k€
Nominal power factor	0.97

D=Depth.

Diesel-generator

Because this concept has a partly electric propulsion, there is larger electrical power demand, but only when the electric motors are operated. Up to 12 knots, the E-motors are used for propulsion, with a maximum electric power demand of 1.8 MWe. The maximum auxiliary power demand is 1.35 MWe, but a margin is accounted, so maximum 1.6 MWe. Together this adds up to a maximum of 3.4 MWe which has to be generated by the diesel-generator sets. For lower speeds the power demand of the motors is lower, and for higher speeds the power demand is also lower, because then the electric motors are not operated anymore. The choice can be made to generate the maximum amount of power by a number (2 or 3) of small diesel-generators or by 1 big diesel-generator. The problem if bigger sets are installed is that in the harbour these sets are too low loaded. Besides that you would like to minimize the number and the amount of overcapacity, because there are also the 2 shaft generators that have to be kept in mind. A maximum of 3 diesel-generators is set, which brings the total to 5 generators. Normally is 4.

Three equal diesel-generators would be preferred from maintenance and commonality point of view. This would mean 1.15 MWe diesel-generators. The criticism to this is that there is no redundancy in the power generation, to reach 3.4 MWe if 1 diesel-generator fails. On the other hand, you could always choose to sail this speed on diesel engines in that case. An alternative could be to generate the maximum amount of power by 2 diesel-generators of 1.7 MWe. For redundancy reasons the third should also be 1.7 MWe. The criticism to this choice is that the diesel-generator is very low loaded in harbour (47%), which is bad for fuel consumption. Optimal engine loading would be 80% for the harbour diesel-generator. A solution is to have 2 bigger (father) and 1 smaller (son) diesel-generator of 1.7 MWe and 1 MWe. These could probably be the same diesel engines only with less cylinders, which is still good for maintenance and commonality. From fuel consumption calculation it was also found that this solution is more fuel efficient than with three 1.15 MWe diesel-generators. With the father-son layout the electric power at 10 knots is generated by 1 father and 1 son diesel-generator. At speeds above 12 knots, the electric power is generated by the shaft generators. If this is not enough the son diesel-generator is standby to deliver the rest.

The losses in the generator and distribution network should be taken into acount by selecting the diesel engine. The losses are assumed to be 4% $(0.97 \cdot 0.99)$. Which means the diesel engines have to deliver 1.77 MW and 1.05 MW mechanical power. To prevent generator overload in case of temporary diesel engine overload, the generator is a little overdimensioned: +7.5% of nominal diesel engine power. This means a generator with 1.9 MW nominal power and 1.15 MW.

The L/D ratio of the generators is chosen such that the height of the generator (incl. cooling) is equal to the height of the diesel engine: $L/D = 1.3$ for father DG's and $L/D = 1.5$ for son DG's.

Father DG's	
Power (diesel)	1.8 MW
Power (generator)	1.9 MW
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(2.86 + 1.58) x 1.61 (1.12) x 2.23
Inlet / Outlet duct	0.146 / 0.137 m^2
Weight	13.1 + 4.7 ton
Estimated costs	1839 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Son DG	
Power (diesel)	1.05 MW
Power (generator)	1.15 MWe
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(2.00 + 1.45) x 1.29 (0.90) x 1.74
Inlet / Outlet duct	0.085 / 0.080 m^2
Weight	7.7 + 2.7 ton
Estimated costs	1340 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

The electrical power generated by the diesel generator sets and the shaft generators is directly fed to the main switchboard. For redundancy reasons there are (at least) two main switchboards which can be coupled. These are low voltage switchboards (440V). The switchboards have 7 incoming fields (3 from diesel generators, 2 from shaft generator set, 1 from bus-coupler and 1 from shore connection). The incoming field of the shaft generator also is an outgoing field to the electric motors. The number of outgoing fields for the rest of the auxiliary load is estimated as explained on page 87, and is set to 8. With the switchboard layout as described there is 100% redundancy.

For better shock-resistance the switchboards are mounted on springs.

Power	3.4 MWe
Nr. of incoming fields	7
Nr. of outgoing fields	8
D x W x H	1.00 x 5.85 x 2.20 m
Weight	3.9 ton
Estimated costs	486 k€
Nominal efficiency	0.995

D=Depth.

Auxiliary equipment

The auxiliary equipment that is needed for this propulsion concept are: high pressure air starting systems for diesel engines, fuel feed pumps for diesel engines, fresh water cooling system for diesel engines, electric motors and converters, in- and outlet ducts for diesel engines, and exhaust gas after treatment equipment to reduce emissions.

4.7.2 Operational characteristics

Maneuverability	+
Signature profile	+
Redundancy	3

The maneuverability of this propulsion concept is rated slightly higher than the reference concept. At slow speed, maneuverability is better with the electric motors. In diesel mode maneuverability is comparable to the reference concept. At high speeds the acceleration of the reference concept will be better with the gasturbines. The electric motors are used to brake the ship or to quickly reverse, because the electric motors can deliver torque at all speeds in four quadrants.

The score on signature profile for this concept is higher than the reference concept, because at slow speeds (when the underwater noise is important) electric motors are used, uncoupled from the gearbox. At higher speeds the underwater noise is comparable, but it is less important at those speeds. The infrared signature will also be better because exhaust gas temperature of a diesel engine is normally lower than of a gasturbine.

The score on redundancy is 3, because a maximum of 3 components in a worst case scenario can fail, before the ship isn't able to do 10 knots anymore. In a worst case scenario the two gearboxes can fail and one of the electric motors, before the ship loses its ability to sail 10 knots.

4.7.3 Integration in ship

Nr. of main components	17
Total space consumption	677 m^3
Total weight	488 ton
Fuel capacity	309.6 ton / $3.64 \cdot 10^5$ m^3

The fuel capacity that is necessary to reach 5000 nm at 18 knots is larger than with the reference concept. This is caused by rather unfavourable engine loading at this speed, while in the reference concept the diesel engines are loaded such at 18 knots that they run most efficient.

4.7.4 Availability

Reliability	+
Maintainability	−−
Shock resistance	−−

Reliability of this concept is rated higher than the reference concept. The diesel engines have a high reliability when compared to gasturbines. The electrical motors also have high reliability. On the other hand there is a larger number of components in this concept and there is the rather complicated gearbox arrangement which reduce the reliability. All over the score is rated one step higher than the reference concept.

The large number of diesel engines (7) requires a lot of maintenance, because most of the maintenance on diesel engines is done by the crew. So, the level of maintenance on this concept is high, which gives a low score. There are less diesel engines than in concept 3, so the score is higher than concept 3.

Shock resistance of this concept is lower, because of the big diesel engines. The large surface of the engines makes them vulnerable to shockwaves. The propulsion engines are all low in this ship where they are extra vulnerable to shock.

4.7.5 Costs

Initial purchase	23.93 M€
Annual fuel cost / consumption	3.03 M€/ 3435.8 ton
Annual maintenance cost	0.097 M€

Initial purchase costs of this concept are estimated lower than the reference concept, because medium speed diesel engines have lower purchase costs than gasturbines. On the other hand, the electrical equipment is expensive, but because it are rather small components the impact on total costs is small. The initial purchase costs are about 1 M€higher than full diesel concept 3.

The annual fuel consumption, thus fuel costs of this propulsion concept and the given operation profile is significantly lower than the reference concept. The diesel engines have a lower fuel consumption than gasturbines, which can be clearly recognized in the annual fuel consumption.

4.8 Concept 7/CODLADOG/Hybrid

The hybrid CODLADOG concept, COmbined Diesel-eLectric And Diesel Or Gasturbine combines three machine types. Low speeds on electric motors (on the shaft or in front of the gearbox), transit speed on diesel engines and high speeds on gasturbines and electric motors.

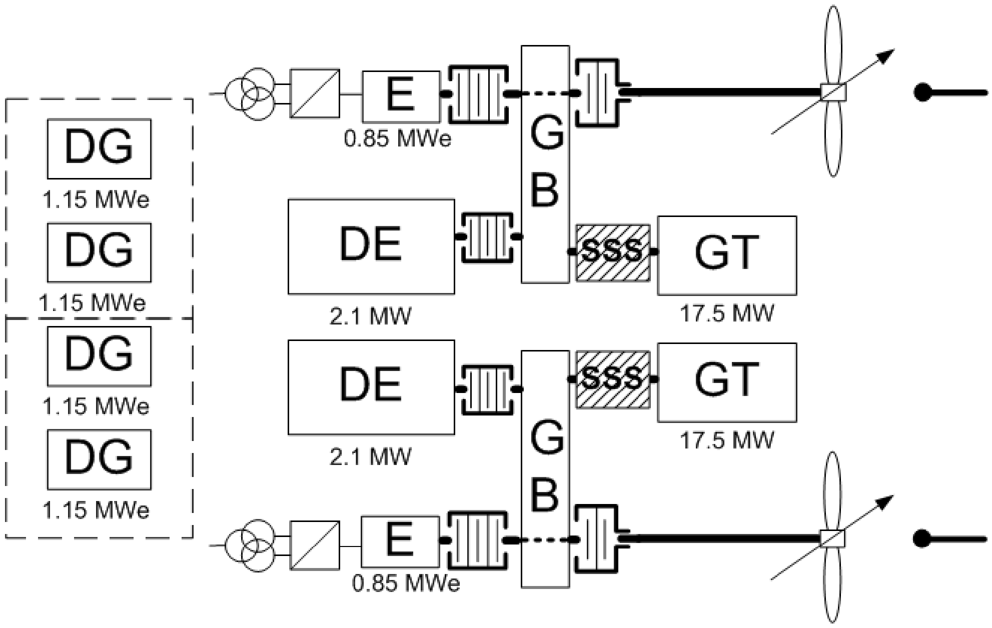


Figure 4.9: Propulsion concept 7, CODLADOG

With this configuration there are 10 different operating modes:

1. High speed mode (2 gasturbines and 2 electric motors on 2 shafts). For sprinting at top speed. Max. speed ≈30 knots.

2. High speed mode (2 gasturbines on 2 shafts). If the electric motors are unserviceable for whatever reason and high speeds are required. Max. speed ≈29.5 knots.

3. Semi-high speed on 1 shaft mode (1 gasturbine and 1 electric motor on 1 shaft, other shaft trailing). Max. speed ≈25.5 knots. If one shaft or gearbox is unserviceable for whatever reason and high speeds are required.

4. Semi-high speed on 1 shaft mode (1 gasturbine on 1 shaft). Max. speed ≈25 knots.

5. Cruise speed mode (2 diesel engines and 2 electric motors on 2 shafts). For travelling long distances. Max. speed ≈18.5 knots.

6. Semi-cruise speed mode (2 diesel engines on 2 shafts). For travelling long distances at a max. speed ≈16.5 knots, or if one or two electrical motors are unserviceable for whatever reason (this only holds if electrical motors are in front of gearbox, see figure 4.9).

7. Semi-cruise speed on 1 shaft mode (1 diesel engine and 1 electric motor on 1 shaft, other shaft trailing). If one shaft is unserviceable for whatever reason and medium speeds are required up to max. speed ≈14.5 knots.

8. Semi-cruise speed on 1 shaft mode (1 diesel engine on 1 shaft, other shaft trailing). If one shaft is unserviceable for whatever reason. Max. speed ≈13 knots.

9. Slow speed mode on 2 shafts mode (2 electric motors on 2 shafts). If silent slow speed operations are performed. Max. speed ≈12 knots.
10. Slow speed mode on 1 shaft mode (1 electric motor on 1 shaft, other shaft trailing). If silent very slow speed operations performed, or if one shaft or gearbox is unserviceable for whatever reason. Max. speed ≈9 knots.

It is clear from the large number of operating modes that this concept offers a great operating flexibility.

4.8.1 Components

Main components in this propulsion concept are 2 controllable pitch propellers both driven directly by a slow speed electric motor **and/or** a diesel engine **or** a gasturbine through a combination gearbox. The diesel engines are coupled through multiplate friction clutches in combination with flexible coupling or through a fluid coupling to the gearbox, the gasturbines are coupled through SSS-clutches to the gearbox. With a hollow shaft arrangement and another multiplate friction clutch the gearbox is coupled to the shaft. The electric motor is coupled with multiplate friction clutches directly to the shaft. Electric power for the electric motors and auxiliary equipment is supplied by 4 diesel generator sets of equal power.

Main components Concept 7 (CODLADOG)

Component	Number	Power (MW)
Controllable pitch propeller	2	18.4
E-motor	2	0.9
Diesel engine	2	2.1
Gasturbine	2	17.5
Gearbox	2	17.5
Converter	2	0.9
Diesel-generator	4	1.15
Switchboard	2	3.4
	18	

Controllable pitch propeller

Description according to concept 1.

Power	18.4 MW
Speed / type	212 rpm / CPP
Diameter	4.5 m
Estimated weight	13.8 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	515 k€
Nominal efficiency	0.71

Electric motor

This propulsion concept has 2 slow speed electric motors that are used to deliver power to the shaft, especially at slow speeds, but they also deliver part of the power at higher speeds. They both deliver a maximum mechanical power of 0.9 MW. Assuming 97% efficiency of the combination of motor and frequency converter, makes that a maximum 0.93 MWe has to be delivered to the motor. The motor is designed to deliver maximum torque at 90 rpm, this is the propeller speed at 12 knots, but it also has to be able to deliver power at higher rotational speeds (maximum 212 rpm). Through field weakening the torque is decreased and the speed increased, and still delivering the maximum amount of power, but now at higher speeds. The motors are not operated as generators in this concept, because the power of the other propulsion engines is well matched to the operating conditions, so there is not much overpower. The extra investment on the converter and possible trouble with control strategies is assumed to be not worth the benefit of the electric power generated on the shaft.

The motors are AC synchronous machines. As explained earlier in concept 4, for now conventional technology is assumed, but HTS and PM technology could be considered.

Considering the rather limited power of the motors, they are fed with low voltage electrical power, so no separate high voltage network has to be installed. This saves an extra electrical network with extra switchboards and the crew does not need extra education. A disadvantage is that the motors, which produce some disturbance on the net, are on the same network as for instance SEWACO equipment, which is sensitive to harmonic distortion, so extra filters could be necessary.

Power	0.9 MW
Design speed / type	90 rpm / AC synchronous
L x W x H	2.80 x 1.75 x 3.44 m
Weight	20.2 ton
Estimated costs	1078 k€
Nominal efficiency	0.971

A L/D-ratio of 1.6 is assumed. For comparison: a HTS machine would be approximately: 1.71 x 1.07 x 1.26 m and 2.8 tons, and a PM machine: 1.84 x 1.15 x 1.35 m and 7.6 tons, according to the models.

Diesel engine

Two propulsion diesel engines are installed, together delivering 4.2 MW. Medium speed V-engines are chosen with a nominal speed of 1000 rpm.

Power	2.1 MW
Speed / type	1000 rpm / V-engine
L x W x H	3.17 x 1.72 x 2.39 m
Inlet / Outlet duct	0.17 / 0.16 m^2
Weight	14.1 ton
Estimated costs	1042 k€
Nominal efficiency	0.43

Gasturbine

Description according to concept 1, with the difference that nominal power of the gasturbines is now 17.5 MW instead of 18.4 MW, because the remaining 1.8 MW is delivered by the E-motors.

Power	17.5 MW
Speed / type	5600 rpm / simple cycle
L x W x H	7.24 x 2.51 x 3.10 m
Inlet / Outlet duct	3.38 / 2.89 m^2
Weight	19.3 ton
Estimated costs	6090 k€
Nominal efficiency	0.37

Gearbox

The gearbox reduces the rotational speeds of the outgoing shafts of the diesel engine and the gasturbine. At 30 knots, the selected propeller rotates 212 rpm. At this speed the power is delivered by the gasturbine. The speed of the gasturbine is 5600 rpm, so a reduction ratio of 5600 : 212 = 26.42 is required. This is big reduction which will be achieved in multiple stages. The maximum ship speed at which the diesel engines deliver power to the shaft is approximately 18 knots, where the selected propeller rotates with 130 rpm. Nominal engine speed is 1000 rpm, which means that the gear ratio is 1000 : 130 = 7.69. The diesel engines are coupled to the gearbox through multiplate friction clutches in combination with a flexible coupling for some vibration reduction. Another option could be to couple the engines through fluid couplings to the gearbox. This type combines the clutch and vibration reduction function. The gasturbines are coupled through SSS-clutches, which enables the possibility to smoothly overtake propulsion by the gasturbines from the diesel engines. The electric motors can also drive the shaft but not through a gearing wheel. The electric motor is directly coupled to the shaft. In figure 4.9 the electric motor is positioned in front of the gearbox, this requires the more complex hollow shaft arrangement through the gearbox with an extra clutch. This arrangement was already explained in Concept 4, and has advantage the the motor can be isolated from the shaftline, for example when maintenance is done. If the motor is positioned after the gearbox, on the shaft, this more complex gearbox arrangement is not necessary.

The gearbox is a twin input, double helical, multiple stage gear with a power/speed ratio of 82.6 kW/rpm on the gasturbine shaft and 16.2 kW/rpm on the diesel shaft. Double helical gearing wheels are applied to lower the underwater noise.

Power/speed ratio (GT-drive)	82.6 kW/rpm
Power/speed ratio (DE-drive)	16.2 kW/rpm
Type	Twin, multiple stage, double helical
L x W x H	1.51 x 4.55 x 2.08 m
Weight	52.0 ton
Estimated costs	2136 k€
Nominal efficiency	0.98

Converter

Description according to Concept 4, with the difference that the delivered electric power by the converter is 0.93 MWe, assuming a minimum power factor of 0.95 this means that delivered apparent power should be 1 MVA.

Apparent power	1 MVA
Type	PWM, GTO/IGCT
D x W x H	0.77 x 4.69 x 2.30 m
Weight	2.3 ton
Estimated costs	145 k€
Nominal power factor	0.97

D=Depth.

Diesel-generator

Electrical power is generated by diesel-generator sets. Because this is a hybrid propulsion concept, the propulsion power is partly delivered electrically by the diesel-generator sets, which makes that the electrical load is higher. Peak electrical load is 1350 kWe. As mentioned before a certain margin is taken into account, so the peak load to calculate with is 1.6 MWe. Maximum electrical demand from the E-motors is 1.9 MWe. Together this means a maximum demand of 3.5 MWe which has to be delivered by diesel-generator sets. A total of 4 sets is assumed, so to ensure a certain level of redundancy, this amount should be delivered by maximal 3 sets. This means that each set should deliver 1.15 MWe. In the harbour this set is loaded for 70%, which is acceptable. Electrical power demand in transit mode (without electric motors) can be delivered by only 1 set.

The losses in the generator and distribution network have to be taken into account, so the diesel engine should deliver more than 1150 kW. With an assumed efficiency of 96% $(0.97 \cdot 0.99)$ this means the diesel should deliver 1.2 MW mechanical power. To be sure the generator will not be overloaded in case of temporary diesel engine overload, the maximum power of the generator should be higher than the maximum delivered power of the engine. A margin of 7.5% of the nominal diesel power is added, which means the generator should have nominal power 1.29 MW. The diesel engines are medium speed engines that run at 900 rpm. The generated electrical power is of low voltage (440V). It is not necessary to generate high voltage because the electric propulsion motors have limited power, but the network must be capable of handling temporary high current peaks (in-rush currents).

L/D value of the generator is chosen such that the height of the generator is equal to the height of the diesel engine: $L/D = 1.4$.

Power (diesel)	1.2 MW
Power (generator)	1.3 MWe
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(2.20 + 1.45) x 1.37 (0.96) x 1.86 m
Inlet / Outlet duct	0.098 / 0.092 m^2
Weight	8.8 + 3.2 ton
Estimated costs	1455 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Switchboard

The electrical power generated by the diesel generator sets is directly fed to the main switchboards. For redundancy reasons there are (at least) two main switchboards which can be coupled. These are low voltage switchboards (440V). The switchboards have 6 incoming fields (4 from diesel generators, 1 from bus-coupler and 1 from shore connection). Each switchboard has 2 outgoing fields going to both electrical motors. The rest of the auxiliary load divided over 8 outgoing fields. This number is estimated as explained on page 87. For better shock-resistance the switchboards are mounted on springs.

Power	2.5 MWe
Nr. of incoming fields	6
Nr. of outgoing fields	10
D x W x H	1.00 x 5.53 x 2.20 m
Weight	4.1 ton
Estimated costs	229 k€
Nominal efficiency	0.995

D=Depth.

Auxiliary equipment

The auxiliary equipment that is needed for this propulsion concept are: high pressure air starting systems for diesel engines and gasturbines, fuel feed pumps for diesel engines and gasturbines, at least one fuel seperator, fresh water cooling system for diesel-generators, electric motors and converters, in- and outlet ducts for diesel engines and gasturbines, and exhaust gas after treatment equipment to reduce emissions.

4.8.2 Operational characteristics

Maneuverability	++
Signature profile	+
Redundancy	3

The maneuverability of this propulsion concept better than the reference concept, because at slow speed maneuverability is better with the electric motors. They are used to brake the ship or to quickly reverse, because the electric motors can deliver torque at all speeds in four quadrants. In diesel and gasturbine mode maneuverability is slightly better than the reference concept, because the electric motors can be used to rapidly speed up or brake.

The score on signature profile for this concept is higher than the reference concept, because at slow speeds (when the underwater noise is important) electric motors are used, uncoupled from the gearbox. At higher speeds, when the diesel engines propel the ship, the underwater noise is comparable, but it is less important at those speeds.

The score on redundancy is 3, because a maximum of 3 components in a worst case scenario can fail, before the ship isn't able to do 10 knots anymore. In a worst case scenario the two gearboxes can fail and one of the electric motors, before the ship loses its ability to sail 10 knots. Or if 3 diesel generators fail, the amount of 1350 kWe can no longer be generated.

4.8.3 Integration in ship

Nr. of main components	18
Total space consumption	275 m^3
Total weight	300 ton
Fuel capacity	302.3 ton / $3.56 \cdot 10^5$ m^3

4.8.4 Availability

Reliability	–
Maintainability	–
Shock resistance	–

Reliability of this concept is rated lower than the reference concept. This concept has more components, which increases the chance of failure. Besides that, the complex gearbox arrangement with hollow shaft is assumed to have a lower reliability than the gearbox in the reference concept.

Maintainability of this concept is estimated slightly lower than the reference, because there are more components. The difference will not be large, because the electrical components do not need much maintenance.

Shock resistance of this concept is rated lower, because of the vulnerable electric motor. The electric motors have low tolerances between rotor and stator, and a large displacement can cause severe damage to the motor. To decrease the sensitivity to shockwaves, the equipment is placed on springs. The converters and switchboards are preferably placed high up in the ship.

4.8.5 Costs

Initial purchase	28.3 M€
Annual fuel cost / consumption	3.38 M€/ 3835.5 ton
Annual maintenance cost	0.158 M€

4.9 Concept 8/IFEP/Full electrical

The IFEP concept, Integrated Full Electrical Propulsion, is an example of full electric propulsion. Propellers are solely driven by electric motors. This propulsion concept is found on Netherlands Joint Support Ship (JSS).

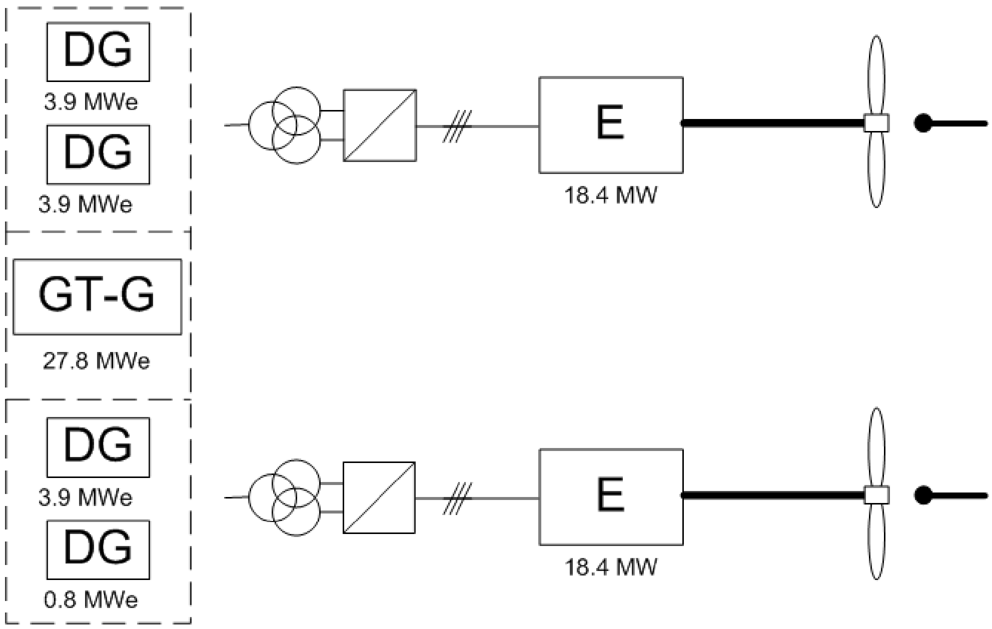


Figure 4.10: Propulsion concept 8, IFEP

With this configuration there are 2 different operating modes:

1. Slow/Cruise/High speed mode (2 electric motors on 2 shafts). Max. speed ≈30 knots.
2. Slow/Cruise/Semi-high speed mode (1 electric motor on 1 shaft, other shaft trailing). If 1 electric motor or shaft is unserviceable for whatever reason. Max. speed ≈25.5 knots.

4.9.1 Components

Main components in this propulsion concept are 2 fixed pitch propellers both directly driven by a slow speed electric motor. No gearbox or clutches are needed, which saves some mechanical losses. All propulsion power is provided as electric power to the electric motors. Electric power is generated by 4 diesel generator sets in a father-son configuration and a gasturbine-generator set for boost power.

Main components Concept 8 (IFEP)

Component	Number	Power (MW)
Fixed pitch propeller	2	18.4
E-motor	2	18.4
Converter	2	18.4
Diesel-generator	3 / 1	3.9 / 0.8
Gasturbine-generator	1	27.8
Switchboard	2 / 2	36.8 / 1.6
	15	

Fixed pitch propeller

Description according to Concept 4.

Power	18.4 MW
Speed / type	225 rpm / FPP
Diameter	4.5 m
Estimated weight	10.7 ton
Propeller disc loading	1.16 MW/m^2
Estimated costs (incl. shaft)	411 k€
Nominal efficiency	0.72

Electric motor

Two large slow speed electric motors are installed, together delivering the full amount of 36.8 MW, which is enough to power the propellers up to top speed with a maximum speed of 225 rpm. This makes that these motors have to deliver very high torque, which makes them very bulky. With the right drive technology the produced underwater noise of these motors can be much lower than with diesel engines or gasturbines. It depends a bit on the type of E-motor and the combination with the converter how much noise is produced by the E-motor. From signatures point of view the DC-motor seems most favourable or otherways the AC synchronous motor. The DC-motor has more intensive maintenance, but smaller and cheaper converter. With the right measures, the AC synchronous motor can probably be as silent as the DC-motor. An extensive cost analysis, taking also into account the cost of the matching converter and maintenance, should show which solution is in the end the best. The cost models that were used in this thesis are not accurate enough to make this analysis. The choice is now made for AC synchronous machines, based on the fact that these have less weight, which could imply lower purchase costs, and have less maintenance, which means lower life cycle costs. If in the near future high temperature superconduction (HTS) technology is well established, this is a very interesting option. Permanent magnet machines are also a very interesting option, because of their higher power density, but they are about twice the price of a conventional machine. Especially for these high power motors such technologies with higher power densities should be taken into considerations. The lack of detailed cost models makes this consideration impossible. For now, conventional technology is assumed.

The E-motors are fed with high voltage electrical power to limit the currents through the distribution network and through the motor windings. A commonly used voltage is 6.6 kV. A separate high voltage network has the advantage that the disturbances on this network, caused by the motors (in-rush currents), are not directly noticeable on the low voltage (440V) network, where for instance the SEWACO users are.

Power	18.4 MW
Design speed / type	212 rpm / AC synchronous
L x W x H	5.75 x 3.60 x 7.06 m
Weight	175.4 ton
Estimated costs	15545 k€
Nominal efficiency	0.973

A L/D-ratio of 1.6 is assumed. The dimensions as shown above are in case the cooling is placed as one cubic box on top of the motor. In practice the cooling can also be placed as two units on the 11 and 1 o'clock position of the motor. This decreases the height somewhat. For comparison: a HTS motor would be approximately: 3.51 x 2.19 x 2.58 m and 23.9 tons, and a PM motor: 3.78 x 2.36 x 2.78 m and 65.9 tons, according to the models. This ones more clarifies why these technologies are very interesting, especially for such large electrical machines.

Converter

Description according to Concept 4, with a different power level. Each motor demands a maximum of 18.4 MW. The choice can be made to install 1 big converter that feeds both motors, this saves money but the big disadvantage is the absence of any redundancy. This is undesirable on a warship, which is active on the battlefield. Each motor should have its own converter. PWM converters are available up to these high powers, but at powers above 10 MW only IGCT's are used as semiconductors. The converters need to deliver a maximum of 19 MWe (if 97% or higher motor efficiency is assumed). The converter itself also introduces some losses, assuming a minimum power factor of 0.95, the nominal apparent power of the converters is 20 MVA.

Apparent power	20 MVA
Type	PWM, IGCT
D x W x H	1.90 x 4.92 x 2.30 m
Weight	16.5 ton
Estimated costs	2640 k€
Nominal power factor	0.99

D=Depth.

Diesel-generator

The full electric propulsion has a very high electrical power demand, because propulsion power is also delivered electrically at all speeds. The maximum power demand, at top speed assuming a motor efficiency of 97%, including full auxiliary power (1.6 MWe) is approximately 39.5 MWe. If all this electrical power was generated by diesel-generator sets, this would become very large sets, or it would be a lot of them. For that reason also a gasturbine-generator is installed. The aim is to generate electrical power up to transit speed with diesel-generator sets and above transit speed a gasturbine-generator provides power.

The decreasing motor efficiency at partload has to be taken into account. This can be estimated according to the partload efficiency model in section 3.5.5. The maximum power demand at transit speed is 5.6 MW, which corresponds with 6.2 MWe electrical power demand for propulsion with 90% motor efficiency, plus 1.6 MWe maximum auxiliary load is 7.8 MWe. This amount of power should be generated by a number n of diesel-generators. For reasons of redundancy you would like to have at least *n+1* sets. For logistical and maintainability reasons the total number of engines should be limited. A limit of 4 diesel-generators is used. With these boundary conditions, the 7.8 MWe should be delivered by 3 sets of 2.6 MWe. With such a diesel-generator, the engine would run at very low load in harbour conditions (30%), so a smaller set is needed. A father-son solution could be chosen, with two larger and two smaller engines or 3 larger engines and 1 smaller harbour/emergency engine. For full redundancy at transit speed in case 1 set fails, without having to turn to the gasturbine, the latter option is selected. This means 3 diesel-generator sets of 3.9 MWe and 1 small harbour/emergency set of 0.8 MWe.

Taking into account the losses in the generator and distribution network, the diesel engines should deliver 4.1 MW and 0.85 MW (assumed 96% efficiency). To prevent the generator from overload a margin of 7.5% of the nominal diesel power is calculated, which means the generators have nominal power 4.4 MW and 0.92 MW. The large diesel engines are medium speed line engines that run at 720 rpm, and the small diesel is a 900 rpm V-engine. The generated electrical power is of high voltage (6.6 kV).

L/D value of the generator is chosen such that the height of the generator is equal to the height of the diesel engine: $L/D = 1.2$ for the large sets and $L/D = 1.7$ for the small set.

Power (diesel)	4.1 MW
Power (generator)	4.4 MW
Speed / type (diesel)	720 rpm / Line engine
Speed / type (generator)	720 rpm / Conventional AC synchronous
L x W x H	(5.89 + 2.12) x 2.63 (1.63) x 3.25
Inlet / Outlet duct	0.332 / 0.312 m^2
Weight	35.4 + 13.4 ton
Estimated costs	2970 k€
Nominal efficiency (diesel)	0.44
Nominal efficiency (generator)	0.97

Harbour/emergency DG	
Power (diesel)	0.85 MW
Power (generator)	0.92 MW
Speed / type (diesel)	900 rpm / V-engine
Speed / type (generator)	900 rpm / Conventional AC synchronous
L x W x H	(1.74 + 1.47) x 1.18 (0.80) x 1.58
Inlet / Outlet duct	0.069 / 0.065 m^2
Weight	6.2 + 2.2 ton
Estimated costs	1188 k€
Nominal efficiency (diesel)	0.43
Nominal efficiency (generator)	0.97

Gasturbine-generator

Up to transit speed the diesel-generator sets can deliver enough electrical power for the propulsion motors and auxiliary equipment on the ship. For high speeds a gasturbine-generator is installed to generate electrical power. With 2 of the 3 larger diesel-generators, 7.8 MWe can be generated, the rest should be generated by the gasturbine-generator, if 1 diesel-generator is saved as redundant set. With the earlier calculated 39.5 MWe at full load, this means the gasturbine-generator should generate 31.7 MWe. An alternative is to install a lower power gasturbine-generator (27.8 MWe) and use 3 diesel-generator sets. There is still 1 redundant diesel-generator set, but with limited power. Disadvantage of course is that there is no redundancy in large diesel-generator sets for top speeds. Considering the investment costs and the limited space onboard, this alternative seems better.

Taking into account the losses in the generator (97%) and distribution network (99%), the gasturbine should deliver 29 MW. To prevent the generator from overload a margin of 7.5% of the nominal gasturbine power is calculated, which means the generator has nominal power 31.2 MW. The powerturbine of a gasturbine of this powerlevel is estimated to rotate at 3600 rpm. This makes direct coupling of a generator with 2 polepairs possible. The low polepair number makes that the produced noise increases the signature profile. This effect could be decreased by placing the gasturbine higher up in the ship. The generated electrical power is of high voltage (6.6 kV).

L/D value of the generator is chosen such that the height of the generator is equal to the height of the gasturbine: $L/D = 1.6$

Power (gasturbine)	29 MW
Power (generator)	31.2 MW
Speed / type (gasturbine)	3600 rpm / Simple cycle
Speed / type (generator)	3600 rpm / Conventional AC synchronous
L x W x H	(8.34 + 2.89) x 2.75 (1.67) x 3.26
Inlet / Outlet duct	5.597 / 4.785 m^2
Weight	23.7 + 19.0 ton
Estimated costs	14196 k€
Nominal efficiency (gasturbine)	0.39
Nominal efficiency (generator)	0.97

Switchboard

The diesel- and gasturbine-generators in this concept generate high voltage electrical power, so high voltage switchboards are needed to distribute the power to the motors and the low voltage network. For redundancy reasons there are 2 high voltage switchboards, both capable of distributing the maximum amount of 39.5 MWe to both converters and the low voltage net. In this way full redundancy is ensured in case 1 switchboard fails. An alternative is to feed only 1 converter per switchboard, this limits the power on the switchboard, thus dimensions and costs, but in case of failure of 1 switchboard only 1 motor can be operated. This alternative is not preferred. So, both high voltage boards have 6 input fields (4 DG inputs, 1 GT-G input, 1 bus coupler) and 4 outgoing fields (2 E-motor, 2 low voltage switchboard). Between the high and low voltage switchboard is a transformer, which transforms from 6.6 kV to 440 V. This transformer is not further considered in this analysis. There are (at least) 2 low voltage switchboards both capable of distributing the maximum amount of 1.6 MWe to the auxiliary systems. The low voltage boards have 4 incoming fields (2 transformer inputs, 1 bus coupler, 1 shore connection). The number of outgoing fields is estimated to be 8 based on the amount of power that is distributed, according to the method mentioned on page 87

For better shock-resistance the switchboards are mounted on springs.

D=Depth.

High voltage switchboard	
Power	39.5 MWe
Nr. of incoming fields	6
Nr. of outgoing fields	4
D x W x H	1.70 x 6.50 x 2.60 m
Weight	6.0 ton
Estimated costs	5649 k€
Nominal efficiency	0.995

Low voltage switchboard	
Power	1.6 MWe
Nr. of incoming fields	4
Nr. of outgoing fields	8
D x W x H	1.00 x 3.90 x 2.20 m
Weight	2.9 ton
Estimated costs	229 k€
Nominal efficiency	0.995

Auxiliary equipment

The auxiliary equipment that is needed for this propulsion concept are: high pressure air starting systems for diesel engines and gasturbines, fuel feed pumps for diesel engines and gasturbines, at least one fuel seperator, fresh water cooling system for diesel-generators, electric motors and converters, voltage transformers, in- and outlet ducts for diesel engines and gasturbines, and exhaust gas after treatment equipment to reduce emissions.

4.9.2 Operational characteristics

Maneuverability	+++
Signature profile	+++
Redundancy	2

The maneuverability of this concept is superior to the reference concept, because of the operating envelope of the electric motors. Electric motors can deliver maximum torque at all speeds and can operate in four quadrants, so they can be used to brake the ship and easily reverse shaft speed and have maximum acceleration at all speeds. The limitations are determined by the power delivery of the generator sets, but these operate at constant nominal speed where they can deliver maximum torque at all times.

Signature profile of this concept is very good when compared to other concepts, because all prime movers can be placed on foundation that are double resilient mounted for best damping of structure-borne noise. Only the electric motors are directly coupled to the shaft, and can probably not be resilient mounted because of their weight. With a proper control of the motors by the converter the torque ripple on the shaft is minimized and so does the noise. Another big advantage of this concept with respect to signature profile is the absence of a gearbox, which always introduces some underwater noise. Though, the big electric motors will increase the electro-magnetic signature of the ship. Active reduction measures can reduce these signatures.

In terms of redundancy, this concept is not superior, because there are only 2 propulsion motors. If both fail for whatever reason the ship is a 'sitting duck'. Other weak points were only 2 pieces are available are the converters and the switchboards. So, in a worst case scenario the ship isn't able to sail anymore after failure of 2 components.

4.9.3 Integration in ship

Nr. of main components	15
Total space consumption	448 m^3
Total weight	401 ton
Fuel capacity	327.4 ton / $3.85 \cdot 10^5$ m^3

4.9.4 Availability

Reliability	+++
Maintainability	+++
Shock resistance	+

This concept has the lowest number of prime movers (5), together with concept 5. This concept doesn't have a complex gearing arrangement or clutches which decrease the reliability. The electric motors have a relatively high reliability when compared to the prime movers. This concept has a FPP, which is much more reliable than a CPP. All together makes that this concepts is estimated to have best reliability of all concepts. Maintainability of this concept is also very good, because of the low number of prime movers, which need a lot of maintenance when compared to electrical equipment.

Shock proofness of this concept is rather good, because all the prime movers can be positioned flexible throughout the ship (as long as ship stability allows it). The engines can be placed higher up in the ship to reduce the vulnerability, and because they are not directly couple to the shaft, they can be mounted on soft springs. The weakest point are the electric motors which have low tolerances between rotor and stator and are rather vulnerable to shock. The AC synchronous machine has a better shock resistance than a asynchronous machine because of a bigger airgap. To improve shock resistance the motors can be placed on springs.

4.9.5 Costs

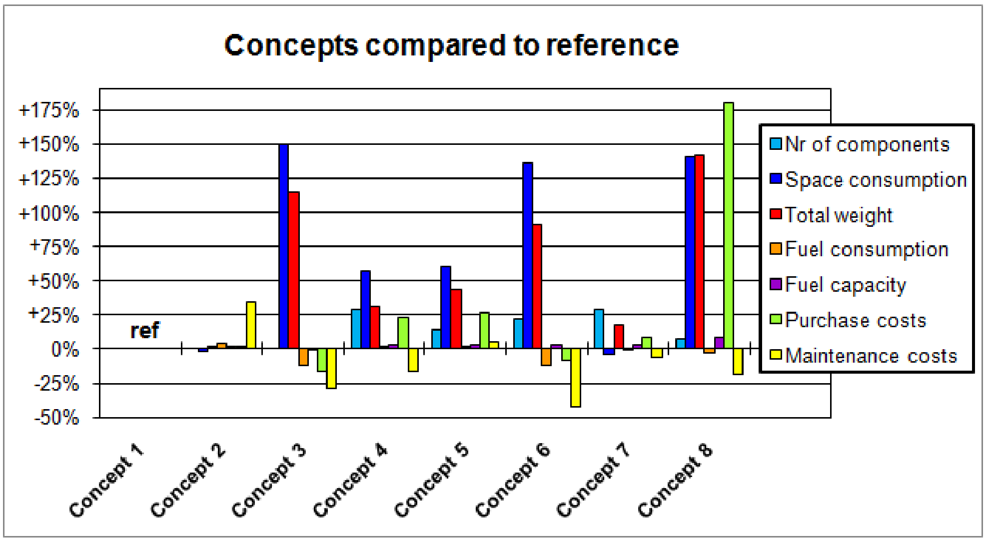
Initial purchase	73.2 M€
Annual fuel cost / consumption	3.34 M€/ 3793 ton
Annual maintenance cost	0.137 M€

The cost calculation confirms every prejudice that exists about full electric propulsion being expensive. This concept has by far the highest initial purchase costs of all concepts in this study. The high purchase costs are not translated into significant lower fuel costs, because the difference is only small in comparison to the reference concept, and there are even concepts that have lower fuel costs. At least, for the speed profile that is used in this study. The maintenance costs are not calculated because the lack of proper calculation models.

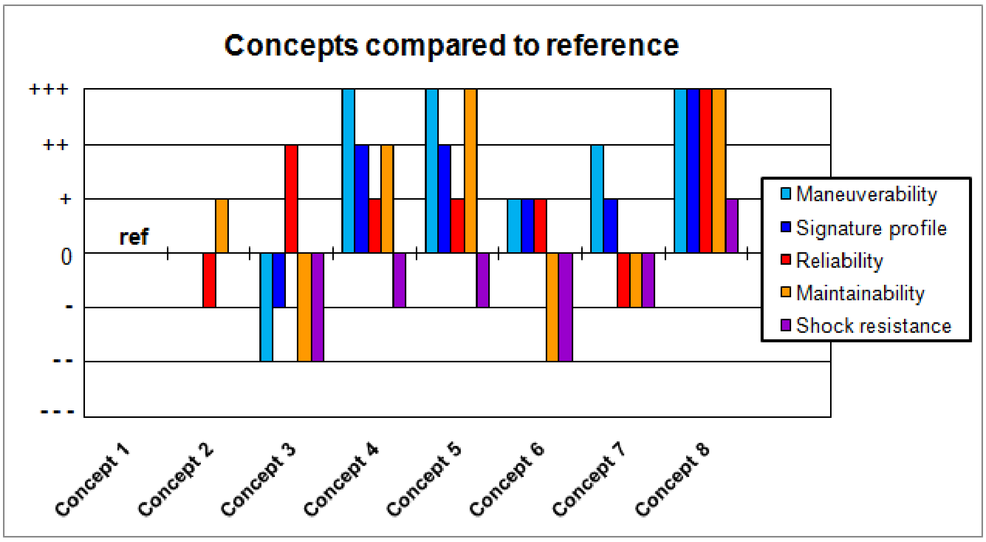
Fuel consumption is lower than the reference concept, but scores worse than some other concepts. This is caused by the losses in the various conversion steps in the generator, the converter and the electric motor. In direct mechanical propulsion there are less conversion steps, thus less losses. Fuel capacity required for the desired range at 18 knots is higher than any other concept. This is also caused by the losses in the various conversion steps, and the the rather poor partload efficiency of the electric motors (88%) at this ship speed. Where all other concepts are optimized for fuel consumption at transit speed, this concept has a relatively bad fuel consumption. In the fuel consumption calculations all speeds are sailed on 2 shafts, to improve the loading of the machinery to have lower fuel consumption, the lower speeds might be sailed on 1 shaft with the other shaft trailing. If all speeds up to 26 knots are sailed on 1 shaft, annual fuel consumption goes down to 3729 tons, but required fuel capacity even goes up to 329 tons.

4.10 Summary of concepts

In this section the results from the concept analysis are summarized. In the previous sections the scores on the different criteria were calculated and explained, a summary is given in the table on the next page. This data is used as input for the multiple criteria analysis in the next chapter. In figure 4.11 below, the relative scores compared to the reference concept are presented graphically. From this figure it is clear that Concept 8 has by far the highest purchase costs, almost 3 times the purchase costs of the reference concept, but it also has the highest scores on several characteristics. Concept 3 has the lowest purchase cost, and also the lowest fuel consumption, but on the other hand it has the highest space consumption and weight. From this figure can't be concluded which concept is the 'best', this all depends on the weighing factors for each criteria. This process is explained in chapter 5.



(a) Quantitative criteria



(b) Qualitative criteria on (− − −...+++)-scale

Figure 4.11: Relative scores on quantitative and qualitative criteria of all concepts compared to the reference concept

	Concept 1	**Concept 2**	**Concept 3**	**Concept 4**	**Concept 5**	**Concept 6**	**Concept 7**	**Concept 8**
Maneuverability	ref	0	−−	+++	+++	+	++	+++
Signature profile	ref	0	−	++	++	+	+	+++
Redundancy	2	2	2	3	3	3	3	2
Nr of components	14	14	14	18	16	17	18	15
Space consumption	286.3	280.6	715.0	450.3	459.3	677.2	274.8	690.2
Total weight	255.9	256.1	549.4	334.1	368.4	487.7	299.6	620.4
Fuel capacity	302.4	306.5	301.7	312.4	312.4	309.6	309.1	327.4
Reliability	ref	−	++	+	+	+	−	+++
Maintainability	ref	+	−−	++	+++	−−	−	+++
Shock resistance	ref	0	−−	−	−	−−	−	+
Purchase costs	26.14	26.37	21.93	32.04	33.17	23.93	28.29	73.24
Fuel consumption	3886.9	4024.5	3434.3	3903.1	3950.0	3435.8	3875.0	3792.8
Annual fuel costs	3.42	3.55	3.03	3.44	3.48	3.03	3.41	3.34
Maintenance costs	0.168	0.225	0.119	0.140	0.176	0.097	0.158	0.137

Chapter 5

Multi Criteria Analysis

This chapter describes the process that should lead to the 'best' solution. With a Multiple Criteria Analysis (MCA) the characteristics of all propulsion concepts are weighted and judged upon a number of criteria. The characteristics of the propulsion concepts are described in the previous chapter. The problem in such decisionmaking is not how to calculate what is the 'best' solution, but how to define 'best'. This definition is different for everyone who looks at the decisionmaking problem. The employee from cost analysis section will probably say that the cheapest is the best, but the employee from signatures section will choose the concept with the least signature production. The relative importance of all characteristics in the decisionmaking problem is defined in terms of weighing factors. The choice of the weighing factors is a personal matter, and will be different for everyone. For that reason it is important to have insight in the sensitivity of these weighing factors on the final outcome of the multiple criteria analaysis. TNO developed a tool, called TOPSYStem, that helps in the decisionmaking, and that can visualize the sensitivity of the outcome on the change of weighing factors.

5.1 About TOPSYS

TOPSYStem is a software system developed by TNO running under MS-DOS, which helps making TOPping decisions by identifying The Option Preferred. Several alternatives are assessed and compared on several criteria. These elements are arranged in a hierarchical format with the ultimate goal, the 'best' choice in this case, as the top-node. The alternatives are judged on the direct criteria with a certain score. This score can be quantitative or qualitative. The direct criteria may be clustered into several higher hierarchy level criteria. The relative importance of the direct criteria on the higher level criterion, is defined in terms of weighing factors. Finally, TOPSYStem offers a way to synthesize all scores and weights, thus arriving at an aggregate judgement about the alternatives at the top level of the hierarchy. This aggregate judgement is presented in a set of final scores, on a so-called scorecard, indicating how well each alternative performs with respect to the final goal. In addition, it offers some tools to analyse the results and their sensitivity to changes in scores or weights. Wijnmalen (1999) is an user guide to the program.

There are several methods of evaluation to establish a score of an alternative or a weight of a criteria. The methods differ with the nature of the required information and the way of expressing it. This largely depends on whether the available information is based on objective data or on subjective feelings from past experience and knowledge. Distinction is made between quantitative and qualitative evaluation methods and also between direct comparison and pairwise comparison. Quantitative methods use a cardinal scale, and the numbers have a physical meaning. The physical numbers are transformed by the program into scores on a scale from 0 to 10. Qualitative methods use an ordinal scale, which is a matter of ranking (by numbers or symbols). There are no aggregation methods in TOPSYStem which can deal with ordinal weights, so when aggregating all qualitative weights are always transformed to quantitative values by the program. The difference between direct comparison and pairwise comparison, is that with direct comparison all alternatives in the poule are directly compared to each other on the selected criterion, and with pairwise criterion you compare two elements at a time. In Wijnmalen (1999) it is mentioned that pairwise comparison is probably a more accurate method, but it is more time consuming and pairwise comparisons may cause inconsistencies: A better than B, B better than C, C better than A, while C should be worse than A when perfectly consistent. A list of all evaluation methods within TOPSYStem and their nature (quantitative vs qualitative, direct vs pairwise comparison) and scale transformation formulas are presented in appendix E.

5.2 Hierarchy

The hierarchy of the MCA is built up such that it matches with the assesment criteria in chapter 4. The final criteria is the 'best' solution which is determined by the four parent criteria:

- Operational characteristics
- Integration in ship
- Availability
- Costs

These parent criteria are fed by the quantitative and qualitative scores on the child criteria:

- Maneuverability
- Signature profile
- Redundancy
- Number of components
- Space consumption
- Weight
- Fuel capacity
- Reliability
- Maintainability
- Shock-proofness
- Initial purchase costs
- Fuel costs
- Maintenance costs

The MCA hierarchy is visualized in figure 5.1. The MCA is implemented in TOPSYStem according to this hierarchy.

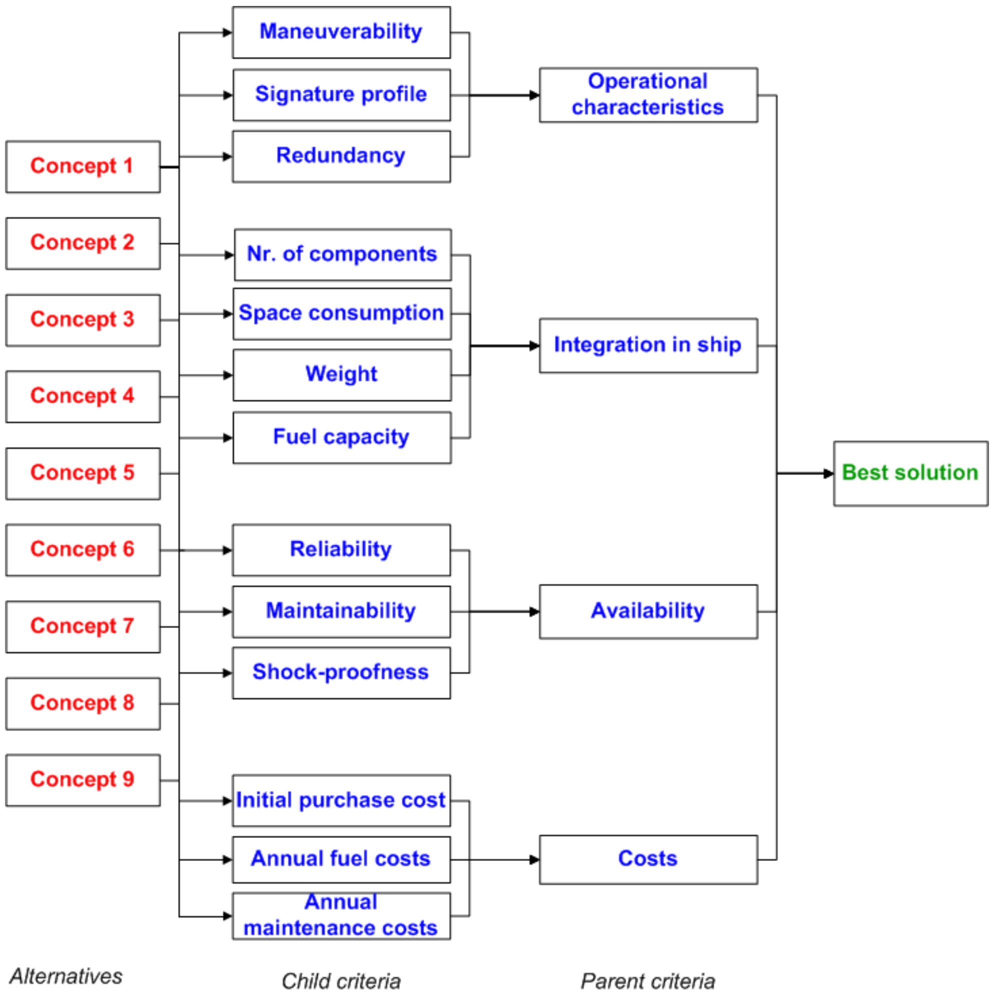


Figure 5.1: Hierarchy of multiple criteria analysis on propulsion concepts, as implemented in TOPSYStem

5.3 Evaluation methods

A list of the evaluation methods within TOPSYStem is presented in appendix E. For each criteria, one of these evaluation methods is chosen which best suits the available data. See table 5.1. For the quantitative criteria the decreasing [$min...max$]-scale is used. Decreasing means that a high value on the scale gets a low score, because this is unpreferred. The range is determined by the minimum and maximum value from all propulsion concepts on that specific criterion. The qualitative data is evaluated on the [– – –...+++]-symbolic scale. The evaluation method re-calculates all data on a scale from 0 to 10, and feeds this to the higher level criteria. On the higher level criteria, the weighted sum of the lower level criteria is calculated and a score on the scale from 0 to 10 is determined. On the top level criterion, the weighted sum of the higher level criteria is calculated and the score per alternative is presented on a 0 to 10 scale. The alternative with the highest score should be the 'best' solution for that situation. Outcome may change if other scores and weighing factors are assigned.

	Criterion	Evaluation method	Range
Child	Maneuverability	[– – –...+++]-symbolic scale	[– – –...+++]
	Signature profile	[– – –...+++]-symbolic scale	[– – –...+++]
	Redundancy	[*min*...*max*]-scale, decreasing	[1...4]
	Nr. of components	[*min*...*max*]-scale, decreasing	[14...18]
	Space consumption	[*min*...*max*]-scale, decreasing	[274...715](m^3)
	Weight	[*min*...*max*]-scale, decreasing	[255...550](ton)
	Fuel capacity	[*min*...*max*]-scale, decreasing	[301...328](ton)
	Reliability	[– – –...+++]-symbolic scale	[– – –...+++]
	Maintainability	[– – –...+++]-symbolic scale	[– – –...+++]
	Shock-proofness	[– – –...+++]-symbolic scale	[– – –...+++]
	Purchase costs	[*min*...*max*]-scale, decreasing	[21...74](M€)
	Fuel costs	[*min*...*max*]-scale, decreasing	[3000...3600](k€)
	Maintenance costs	[*min*...*max*]-scale, decreasing	[90...230](k€)
Parent	Operational characteristics	[0...*max*]-scale	[0...10]
	Integration in ship	[0...*max*]-scale	[0...10]
	Availability	[0...*max*]-scale	[0...10]
	Costs	[0...*max*]-scale	[0...10]
	Best solution	[0...*max*]-scale	[0...10]

Table 5.1: Used evaluation methods in multiple criteria analysis

5.4 Weighing factors

The choice of the weighing factors plays a major role in the outcome of the multiple criteria analysis. The height of these factors are determined by the importance of each criteria. This directly emphasizes the difficulty and the subjectivity in this process, because every party in the design process has other importances and sometimes they are contradicting. The importance of some criteria might also change along the design process, for example the costs because of shrinking budgets.

The choice of weighing factors as they are made by the author are explained hereafter. They are defined on a scale from 0 to 10. Weight factor 0 means that the criterion is of no importance, weight factor 1-3 minor importance, 4-6 moderate importance, 7-9 major importance and 10 means most imporant. The values that are mentioned are suggestions based on the authors interpretation of the requirements and experiences, and the situation at that time. At first the weighing factors of the parent criteria on the final solution are explained, followed by the weight factors of the child criteria on the parent criteria.

5.4.1 Parent criteria

Operational characteristics

The operational characteristics represents the benefit in the cost-benefit analysis. It shows what the propulsion is capable of. The operational characteristics describe the ability of the propulsion configuration to bring the ship into the operational theatre that the ship is meant to operate in. This criterion is most important.

The suggested weight factor is 10

Integration in ship

Integration in ship summarizes the impacts on the ship design. All space and weight that is occupied by machinery or fuel is not available for payload. The ship is the carrier of sensor and weapon systems that are used in the operating theatre. Space is scarce on these type of ships. This criterion is of major importance.

THE SUGGESTED WEIGHT FACTOR IS 7

Availability

This parent criterion represents the availability of the machinery. The system can be unavailable due to maintenance, failures or shock damage. With respect to other criteria, this criterion is considered to be of moderate importance.

THE SUGGESTED WEIGHT FACTOR IS 4

Costs

Costs contain of purchase costs and usage costs and are very important in the decisionmaking. Though, functionality is even more important. The navy is a public service, non-profit organisation, but the costs have to fit in the target budget. In this early stage the cost estimates are very fluid, which is reason to not weigh costs too high. The costs are of major importance.

THE SUGGESTED WEIGHT FACTOR IS 8

5.4.2 Child criteria

Maneuverability

The maneuverability of a warship is very important. In case a torpedo or missile is launched towards the ship it needs to perform certain dodging maneuvers, which include accelerating, steering and decelerating. When urgent action of the ship is required, it needs to be operational within short notice. Also at slow speeds, for example during ASW operations, the ship needs good maneuverability to track contacts. Further, the ship might be assigned to escort a high valuable unit (HVU) like for example an aircraft carrier. These ships have high cruising speeds, so the escorting ships need to keep up with the HVU. In short, the maneuverability is of major importance.

THE SUGGESTED WEIGHT FACTOR IS 8

Signature profile

One of the intended tasks of the SFC is ASW operations. For that reason the ship will be equiped with high-tech sonar equipment. It is very important that the underwater noise of the ship is as low as possible, on the one hand to be able to use the sonar and on the other hand to not be detected by the enemy. The ship can also be detected by infrared-, visual-, electro-magnetic- or radar profile. For the tasks this ship is intended to be used for, it is of major importance that the ship is not detected by an enemy before it has detected the enemy itself.

THE SUGGESTED WEIGHT FACTOR IS 9

Redundancy

The ships redundancy determines the survivability. If the ship is in operation and some machinery breaks due to failure or a missile hit, the level of redundancy can determine the remaining operational employability. The probability of a missile hit is very low, but realistic on a battlefield. The probability of failure of main components is much higher. It is of major importance that the ship has some level of redundancy to improve survivability.

THE SUGGESTED WEIGHT FACTOR IS 7

Number of components

The number of components might say something about the complexity of the propulsion concept, thus the reliability, and about the space consumption and weight. All these things are assessed separately, so that makes this criteria more or less superfluous. Though, the number of components also says something about the number of manufacturers or subcontractors, thus about potential logistical, communication and installation problems, but also about the number of maintenance contracts. Besides that it might also say something about the required manning of the ship, because the more components, the more maintenance. The criterion is of a minor importance.

The suggested weight factor is 2

Space consumption

The configurations need a lot of space for their equipment. The space that is occupied by machinery is not available for payload and will largely influence the ship design. Payload for a surface combatant consists of sensor- and weaponsystems and the crew. Especially on this kind of ship, space is scarce. In order not to increase the size of the ship too much, which has an increase in power demand as result, the space consumption should be as small as possible, though with proper ship design space demand can be solved. The criteria is of moderate importance.

The suggested weight factor is 5

Weight

The buoyancy of a ship is determined by the size and shape of the hull and the weight. The machinery occupies a great part of the total ship weight. An increase in weight is considered to be a design risk, as well as the position of the center of gravity. An increased weight also has positive effects because it improves ship stability as long as the heavy machinery is positioned low in the ship. The weight of the machinery is of major importance.

The suggested weight factor is 7

Fuel capacity

The required fuel capacity is determined by the requirement of 5000 nm at 18 knots. It is dependent on the fuel efficiency of the machinery. The fuel capacity determines the size of the fuel tanks and the weight of the carried fuel. The space that is occupied by fuel tanks can't be used to carry payload. The fuel weight is a substantial part of the weight of the ship, about 5% in this case. Because fuel is a liquid it has the negative effect of free fluid surface, which can do harm to the ships stability. In short, there are many reasons why fuel capacity should be as low as possible, and this criterion is of major importance.

The suggested weight factor is 7

Reliability

The reliability of the components determine the probability of failure. A failure of components may have dramatic consequences. Within the RNLN a very strict maintenance regime prevails. Good maintenance and regular overhaul increases the reliability of the macihnery, and lower the probability of failure. Besides that, it is very difficult to assess reliability, so large uncertainty is in the value of reliability. If reliability would have a too high weight factor, these uncertainties would weigh too much. In short, the weight of reliability should have a moderate value.

The suggested weight factor is 4

Maintainability

Maintainability gives an indication of the effort for the crew to keep the machinery running. It might say something about the size of the crew. On naval ships it is common to have an outnumbered crew. Though, future policy dictates reduced crews. Maintainability also says something about the availability of the machinery, during maintenance the machinery is unavailable. But there is always some level of redundancy to overcome the temporary unavailability due to maintenance. All together, the maintainability criterion is of moderate importance.

The suggested weight factor is 6

Shock resistance

The shock resistance determines the vulnerability of the propulsion concept to shockwaves, due to for example mine explosions. The probability of a mine explosion is low, but realistic. Most of the machinery can be mounted on springs to decrease shockwave impact. The greatest trouble for shock occurs at hull valves, rigid piping and electrical switches. A lot can be done to improve shock resistance, in almost every propulsion concept, besides that it is difficult to assess shock resistance. This criterion is of minor importance.

The suggested weight factor is 3

Initial purchase costs

The initial purchase costs are very important in the selection of a propulsion concept. The acquisition of such a large project is bound by a target budget. The total costs of the project have to fit within the budget. It is a trade-off between functionality and costs. In this early stage it is very difficult, almost impossible, to do reliable estimations on purchase costs, this uncertainty should be taken into account in applying a weighing factor. Besides, the initial purchase costs are only a part of the through life costs of the ship. Maintenance and fuel costs also play a major role in the total costs. In a later stadium of the design process the purchase costs become more important, and more accurate estimates can be made, nevertheless it is important not to select a propulsion configuration that will never fit the budget in this early stage. Within the parent criterion of costs, the initial purchase costs play a major role because of the budgetary system the ministry is bound to.

The suggested weight factor is 7

Fuel costs

Fuel efficiency is becoming more and more important nowadays. Burning fossil fuel has the emission of pollutants as a result. Regulations are becoming more strict nowadays, and great pressure is on shipbuilders and marine engineers to lower fuel consumption and pollutant emissions. Fuel consumption is dependent on the way the ship is operated, in other words the speed profile, but also on the efficiency of the machinery. The costs of fuel are a substantial part of the through life costs of a ship. Regarding the threatening depletion of fossil fuel resources, the price of fuel might increase even more in the near future. A combination of fuel efficient machinery onboard and awareness of the crew in the usage of the machinery must lead to lower fuel costs in the future. Fuel costs are of major importance.

The suggested weight factor is 9

Maintenance costs

Life cycle costs are gaining interest in the selection of ships machinery. Part of the life cycle costs are the maintenance costs. This would suggest a major weight factor, but the problem is that estimation of these costs are very difficult and inadequate. In this thesis, only estimation models are derived for the prime movers (diesel engine and gasturbine) and the accuracy of these models is doubtful. The maintenance costs of these components are only a part of the total maintenance costs, and for that reason the weight of this criterion is low. In this multi criteria analysis, the maintenance costs are of minor importance.

THE SUGGESTED WEIGHT FACTOR IS 2

Summary

	Criterion	Local weight	Global weight
Child	Maneuverability	8	0.115
	Signature profile	9	0.129
	Redundancy	7	0.101
	Nr. of components	2	0.023
	Space consumption	5	0.058
	Weight	7	0.081
	Fuel capacity	7	0.081
	Reliability	4	0.042
	Maintainability	6	0.064
	Shock-proofness	3	0.032
	Purchase costs	7	0.107
	Fuel costs	9	0.138
	Maintenance costs	2	0.031
Parent	Operational characteristics	10	0.345
	Integration in ship	7	0.241
	Availability	4	0.138
	Costs	8	0.276
	Best solution	10	1

Table 5.2: Local and global weight factors of all criteria

5.5 Aggregation

All detailed information regarding concept scores and criterion weights should be combined now in order to arrive at aggregate scores. These aggregate values can be presented as either a rank ordering of the alternatives, or a quantitative appreciation (degree of preference) per alternative. Five different aggregation methods are offered by TOPSYStem:

1. Weighted sum
2. Weighted product
3. Quantitative concordance
4. Qualitative concordance
5. Mixed concordance

The first two methods, weighted sum and weighted product, are the most common MCA methods and are used to calculate quantitative appreciation. In this method the qualitative scores have to be calculated into quantitative scores, as explained in table E.1 on page 229. The degree of preference of each alternative is presented as a score on a user-defined scale (standard [0...10]). The difference between the methods is that with the weighted product method the low-valued scores are stressed.

With the weighted sum method the aggregate score is calculated by summing the weighted scores of each concept. If x_j is the weight factor for criterion j, and $Y_{i,j}$ is the score of alternative i on criterion j, than the aggregate score Z is calculated with:

$$Z_i = \sum_{j}^{n} x_i \cdot Y_{i,j} \quad \text{for } i = 1, 2, 3 \ldots, m$$

m indicates the number of alternatives and n the number of criteria.

The weighted product method is the multiplicative variant of the weighted sum method. Each score is raised to the power equal to the weight factor, all results are then multiplied:

$$Z_i = \prod_{j=1}^{n} {Y_{i,j}}^{x_j} \quad \text{for } i = 1, 2, 3 \ldots, m$$

Method 3, 4 and 5 are concordance methods and work differently. These methods calculate an aggregate score based on rank ordering. Concordance is based on a pairwise comparison of scores, and the final scores are relative to each other among the alternatives. This matches with the way the qualitative criteria are assessed; these are relative scores in comparison to the reference concept and can not be seen as absolute scores.

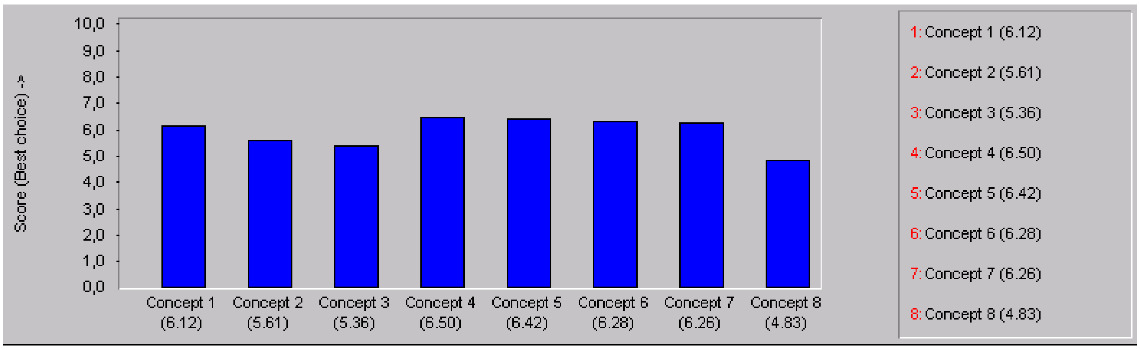
The concordance methods use a [-10...10] scale to express the final score. A score of 0 on the [-10...10] scale means that the weighted positive differences between that alternative and all other alternatives, just outweigh the weighted negative differences. A score of 10 means that the alternative has the best score on every criterion compared to the other alternatives, and -10 that the alternative has the worst score on every criterion. Each alternative is compared to each other alternative on all criteria. The difference values between two alternatives are weighted by the global weight factor per criterion. The sum of these weighted difference values are normalized on the [-10...10] and determine the aggregate score. This score is a measure for the overall, relative performance of the alternative.

$$Z_i = \sum_{j=1}^{n} x_j \cdot (Y_{i,j} - Y_{k,j}) \quad \text{for } i \text{ and } k = 1, 2, 3 \ldots, m \text{ and } i \neq k$$

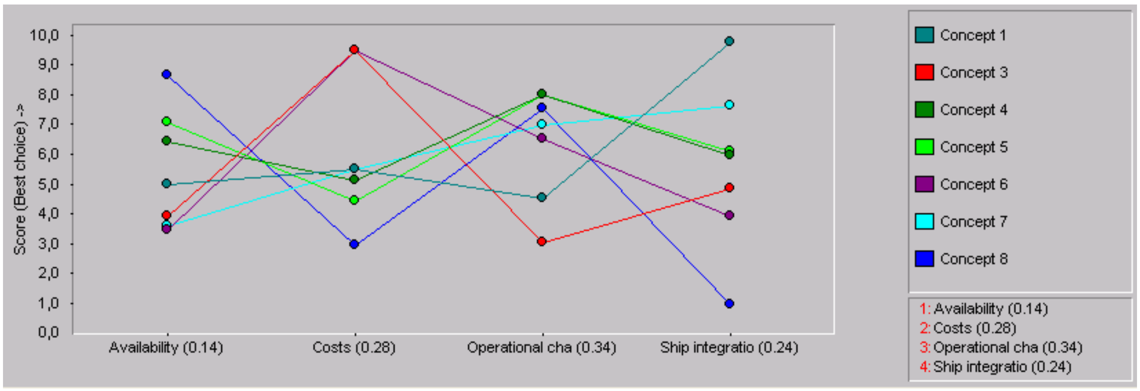
Method 3 and 4, respectively quantitative and qualitative concordance, are concordance methods that are used if either all scores are quantitative or all scores are qualitative. These two methods are not applicable for this study, because the multi criteria analysis in this thesis is a combination of quantitative and qualitative scores. The mixed concordance method combines method 3 and 4 by applying the quantitative concordance method on the quantitative scores, and the qualitative concordance method on the qualitative scores. According to the TOPSYStem manual in Wijnmalen (1999), this is the method to be used when there are both quantititave and qualitative scores.

5.6 Results

The hierarchy, concept scores and weight factors are all programmed into TOPSYStem and aggregate scores are calculated. In the previous section it was explained that the mixed concordance aggregation method should be used to calculate aggregate scores. In practice, often the weighted sum method is used, even when there is a combination of quantitative and qualitative scores. Weighted sum is the most common and straight-forward method for performing MCA's, but mixed concordance is stated to be the most suitable method for this MCA with



(a) Final scores graph



(b) Alternative profile graph

Figure 5.2: Multiple criteria analysis results with weighted sum method

both quantitative and qualitative criteria. To see if there is any difference in the outcome, both aggregation methods are performed.

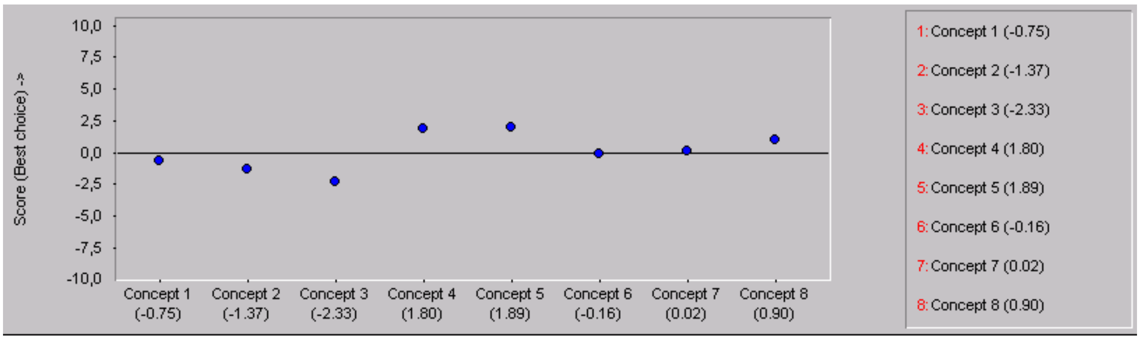
Weighted sum method

Figure 5.2a shows the results as presented by TOPSYStem, after calculating the weighted sum. With this aggregation method (and the choice of weight factors and scores as it is explained before), the rank order is:

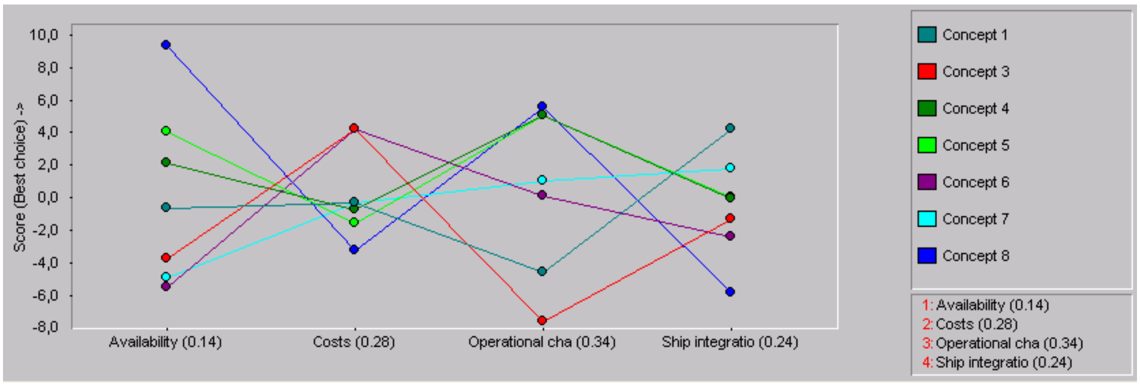
1. Concept 4 *(6.50)*
2. Concept 5 *(6.42)*
3. Concept 6 *(6.28)*
4. Concept 7 *(6.26)*
5. Concept 1 *(6.12)*
6. Concept 2 *(5.61)*
7. Concept 3 *(5.36)*
8. Concept 8 *(4.83)*

Figure 5.2b presents the scores of the alternatives[1] per parent criterion. It shows how well a concept scores on the parent criteria and gives insight in the contribution of the parent criteria on the final score of each concept. It can be seen that the 'best solution', Concept 4, only scores best on *Operational characteristics*, but this is the most important criterion. Based on this, it might be expected that the outcome changes if the weight of *Operational characteristics* is decreased.

[1]Concept 2 excluded, because TOPSYStem is limited to presenting only 7 alternatives. Concept 2 is almost equal to Concept 1



(a) Final scores graph



(b) Alternative profile graph

Figure 5.3: Multiple criteria analysis results with mixed concordance method

Mixed concordance method

The aggregation scores are also determined with the mixed concordance method, to see if there is any difference on the outcome with weighted sum method. The mixed concordance method weighs the ranking order per criterion. For a multiple criteria analysis with both quantitative and qualitative this is the method to use, according to the TOPSYStem manual. The results from the mixed concordance method are presented in figure 5.3a. Figure 5.3b presents the scores on the parent criterion, to see the contribution to the final score. It can be seen that Concept 8 scores best on two parent criteria. But overall Concept 5 scores better and ends up as 'best solution' with this aggregation method, while it doesn't score best on any of the parent criteria.

The rank ordering according to the mixed concordance method is:

1. Concept 5 *(1.89)*
2. Concept 4 *(1.80)*
3. Concept 8 *(0.90)*
4. Concept 7 *(0.02)*
5. Concept 6 *(-0.16)*
6. Concept 1 *(-0.75)*
7. Concept 2 *(-1.37)*
8. Concept 3 *(-2.33)*

It can be seen that with both aggregation methods the final concept scores are close to each other, which means there is not a very obvious winner. The fact that the scores are so close might also say something about the stability of the solution. In the next section a sensitivity analysis will be performed to test if the outcome is sensitive to changes in the weight factors or the scores.

Both methods, give completely different outcomes, which endorses the importance of selecting the right aggregation method. The most remarkable difference between the two methods is the position of Concept 8. In the weighted sum method this concept scores lowest, but with the concordance method this concept scores much better, and ends up third. In both methods Concepts 4 and 5 end up on positions 1 and 2, and the differences in final scores are very small. Based on these results, one of these two concepts should be the propulsion concept on the SFC. The stability of the solution determines the validity of this outcome. Stability is determined in a sensitivity analysis.

5.7 Sensitivity analysis

In this section the robustness of the outcome will be tested by varying the concept scores within a certain accuracy range, and the weight factors of parent and child criteria. There is a high level of uncertainty and subjectivity in the selection of the weight factors and qualitative scores, but also in the estimation of the quantitative scores there are uncertainties. It has now to be investigated how sensitive the final outcome is to the change of these. First the stability of the solution is tested to a change of parent criteria weight factors, secondly to child criteria weight factors, and last to a change of the concept scores. The results of the mixed concordance method are used in the sensitivity analysis, because this method is best suitable for this MCA with both quantitative and qualitative criteria.

5.7.1 Parent criteria

At first the sensitivity of the final score on a change of weight factors of parent criteria is investigated. In figure 5.3b it was already seen that Concept 8 scores best on two parent criteria. Especially on *Availability* the score is much higher than all other concepts. It might be expected that if the weight factor of this criterion is increased, the outcome will change to Concept 8 as 'best solution'.

The sensitivity of the final outcome to the change of weight factor is tested per parent criteria (one by one). This can easily be done with TOPSYStem. The results of the sensitivity analysis are summarized in table 5.3. It mentions the current value of the weight factor (bold printed), which gives Concept 5 as the best choice, and the values to which the weight factor has to be decreased (left) or increased (right) to cause a change in outcome. The upper number (in range 1-10) indicates the local weight, and the lower number (in the range 0-1) the global weight. In some cases an increase of the local weight factor to 10 (maximum) does not result in another outcome, but if the global weight increases (by means of decreasing other weight factors) it does result in another outcome. The value to which the global weight has to be increased is mentioned in the utmost right column. The accompanying trade-off figures will clarify the table and are found in Appendix E.2, figure E.1.

From table 5.3 and figure E.1 it can be concluded that the most sensitive parent criterion is *Availability*. A decrease of the weight factor from 4 to 2.9 results in another 'best' solution: Concept 4. This means the accuracy in assigning the weight factors of parent criteria has to lay within a range of ± 1.1 to make this solution stable. The stability of all other parent criteria is far outside this critical range. A change of the weight factor of *Costs* will never result in another outcome. Only if the global weight is increased to 0.35 by decreasing the weight of other parent criteria the outcome can change to Concept 4.

The critical range of parent criteria weight factors is ± 1.1 and is determined by the stability of parent criterion *Availability*. The accuracy in assigning the initial weight factors is assumed to be ± 1. Within this range a variation of weight factors is considered realistic. The sensitivity of all parent criteria, lies outside this critical sensitivity range, so this means:

*The solution of this MCA is considered **stable** with respect to a change in parent criteria weight factors*

Operational	Concept 1		Concept 5		Concept 5		
	1.9	←	**10**	→	10		Concept 8
	0.09	←	0.34	→	0.34	→	0.79
Ship integration	Concept 8		Concept 5		Concept 5		
	2.0	←	**7**	→	10		Concept 1
	0.08	←	0.24	→	0.31	→	0.54
Availability	Concept 4		Concept 5		Concept 8		
	2.9	←	**4**	→	9.5		
	0.10	←	0.14	→	0.28		
Costs	Concept 5		Concept 5		Concept 5		
	0	←	**8**	→	10		Concept 4
	0	←	0.28	→	0.32	→	0.35

Table 5.3: Stability of the MCA to the change of parent criteria weight factors, see also accompanying figure E.1 in Appendix E.2

5.7.2 Child criteria

Secondly, the influence of the weight factors of the child criteria is investigated. This sensitivity analysis is performed in exactly the same way as with the parent criteria: the initial child criteria weight factors are varied (one by one) within the range [0-10], and the influence on the 'best solution' is observed. Table 5.4 summarizes the results, the accompanying figures are presented in Appendix E.2, figure E.2–E.5.

The most sensitive child criterion is *Number of components*. The critical sensitivity range of this criterion is ±1.4, because a decrease of weight from 2 to 0.6 results in another 'best solution': Concept 4. Earlier the assumption is made that the accuracy in assigning the weight factors is ±1. Larger deviations are not considered realistic. None of the child criteria lies within the critical sensitivity range of ±1. A change in the weight factors of criteria *Maneuverability*, *Signatures profile*, *Redundancy*, *Space consumption*, *Weight*, *Fuel capacity* and *Fuel costs* will even never result in another outcome.

The solution of this MCA is considered ***stable*** *with respect to a change in child criteria weight factors*

	Concept 5		Concept 5		Concept 5		
Maneuverability	0	←	**8**	→	10		
	0	←	0.33	→	1		
	Concept 5		Concept 5		Concept 5		
Signature profile	0	←	**9**	→	10		Concept 8
	0	←	0.38	→	0.40	→	0.85
	Concept 5		Concept 5		Concept 5		
Redundancy	0	←	**7**	→	10		
	0	←	0.29	→	1		
	Concept 4		Concept 5		Concept 5		
Nr. of components	0.6	←	**2**	→	10		Concept 8
	0.03	←	0.10	→	0.34	→	0.52
	Concept 5		Concept 5		Concept 5		
Space consumption	0	←	**5**	→	10		Concept 4
	0	←	0.24	→	0.38	→	0.96
	Concept 5		Concept 5		Concept 5		
Weight	0	←	**7**	→	10		Concept 4
	0	←	0.33	→	0.42	→	0.53
	Concept 5		Concept 5		Concept 5		
Fuel capacity	0	←	**7**	→	10		
	0	←	0.33	→	1		
	Concept 5		Concept 5		Concept 4		
Reliability	0	←	**4**	→	9.6		
	0	←	0.31	→	0.52		
	Concept 4		Concept 5		Concept 5		
Maintainability	3.3	←	**6**	→	10		
	0.32	←	0.46	→	1		
	Concept 5		Concept 5		Concept 4		
Shock resistance	0	←	**3**	→	8.7		
	0	←	0.23	→	0.47		
	Concept 8		Concept 5		Concept 5		
Purchase costs	0.8	←	**7**	→	10		
	0.07	←	0.39	→	1		
	Concept 5		Concept 5		Concept 5		
Fuel costs	0	←	**9**	→	10		Concept 8
	0	←	0.5	→	0.53	→	0.92
	Concept 5		Concept 5		Concept 4		
Maintenance costs	0	←	**2**	→	5		
	0	←	0.11	→	0.24		

Table 5.4: Stability of the MCA to the change of child criteria weight factors, see also accompanying figures E.2–E.5 in Appendix E.2

5.7.3 Concept scores

Finally, the influence of the uncertainties in concept scores are investigated. Not only the choice of weight factors introduces large uncertainties in the solution of the MCA, because it is a very subjective procedure, but also the concept scores introduce uncertainties. The qualitative scores are subjective and not well-founded by numbers and the quantitative scores are based on estimation models which are sometimes very inaccurate. This spread on the input data has to be taken into account in the sensitivity analysis, because it might influence the outcome of the multi criteria analysis.

Qualitative scores

The qualitative scores are varied in a range of ±2 steps on the [– – –...+++]-scale and the influence on the 'best solution' is observed. The results of the effect on the position 1, 2 and 3 are summarized in table 5.5, but only for the cases where position 1 is changed. The left column presents the concept which score is changed, the middle column indicates the change in score that results in a changing outcome and the right column gives the 'new' rank ordering.

Maneuverability		
Concept 5	+++ → ++	1. Concept 4 2. Concept 5 3. Concept 8
Signatures		
Concept 4	++ → +++	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	++ → +	1. Concept 4 2. Concept 5 3. Concept 8
Reliability		
Concept 4	+ → ++	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+ → 0	1. Concept 4 2. Concept 5 3. Concept 8
Maintainability		
Concept 4	++ → +++	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+++ → ++	1. Concept 4 2. Concept 5 3. Concept 8
Shock resistance		
Concept 4	– → 0	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	– → – –	1. Concept 4 2. Concept 5 3. Concept 8

Table 5.5: Stability of the MCA outcome to the change of qualitative concept scores in a range of ±2 steps on the [– – –...+++]-scale

From table 5.5 it is concluded that if one of the qualitative scores of Concept 5 is decreased with at least 1 step, this directly results in a new 'best solution': Concept 4, because the final scores of Concept 4 and 5 are so close to each other. For all qualitative scores, the increase of the score of Concept 4 with at least 1 step also results in this concept to be 'best solution', except for *Maneuverability* because that score can't be increased. Each other change in qualitative score of a concept in a range of ±2 steps doesn't affect the final outcome of the MCA.

The solution of this MCA is considered ***unstable*** *with respect to a change in qualitative scores.*

Quantitative scores

The uncertainty of the quantitative scores is determined by the accuracy of the estimation models. The scores on redundancy and number of components are exact. On all other quantitative scores, the following inaccuracy ranges are applied:

Space consumption: ±30%

Weight: ±30%

Fuel capacity: ±10%

Purchase costs: ±50%

Fuel costs: ±10%

Maintenance costs: +150%

These inaccuracy ranges are determined on the accuracy of the estimation models and feeling. On the space consumption a spread of maximum 30% is applied, because some of the dimension models are pretty inaccurate, especially on electrical machines. The weight is also varied over a range of ±30% because some of the models are rather inaccurate, and weight is related to the volume. Fuel capacity and fuel costs are assumed to be rather accurate, so a spread of 10% is investigated. The purchase costs are varied over a wide range of plus or minus 40%, because estimation of costs in this early stage is very inaccurate, because there are so many variables influencing the purchase costs. The maintenance costs are for sure estimated too low, because it only includes the maintenance costs of prime movers. For the hybrid and full electric concepts an amount of +150% is added to see if this has any influence on the outcome. This amount serves as an estimation on extra maintenance costs for electrical equipment.

It turned out that the increase of maintenance costs for hybrid and full electric concepts has no influence on the outcome of the MCA. The results of the other variations in concept scores are summarized in table 5.6. The left column presents the concept which score is changed, the middle column indicates the change in score (within the specified range) that results in a changing outcome and the right column gives the 'new' rank ordering.

From the results in table 5.6 it is concluded that if one of the quantitative scores of Concept 5 is decreased within the predefined range, this directly results in a new 'best solution': Concept 4. This also holds for an increase of one of the scores of Concept 4 within the predefined range. A change of any quantitative score of any other concept (within the predefined range) doesn't result in another outcome.

The solution of this MCA is considered ***unstable*** *with respect to a change in quantitative scores.*

Space consumption		
Concept 4	-12.28%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+12.35%	1. Concept 4 2. Concept 5 3. Concept 8
Weight		
Concept 4	-10.22%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+9.13%	1. Concept 4 2. Concept 5 3. Concept 8
Fuel capacity		
Concept 4	-1.10%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+0.82%	1. Concept 4 2. Concept 5 3. Concept 8
Purchase costs		
Concept 4	-11.35%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+10.95%	1. Concept 4 2. Concept 5 3. Concept 8
Fuel costs		
Concept 4	-0.89%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 5	+0.92%	1. Concept 4 2. Concept 5 3. Concept 8
Concept 7	-9.98%	1. Concept 5 2. Concept 4 3. Concept 7

Table 5.6: Stability of the MCA outcome to the change of quantitative concept scores

5.7.4 Summary on sensitivity analysis

Some critical sensitivity ranges were assumed for the variation of parent criteria weight factors, child criteria weight factors, qualitative concept scores and quantitative scores. Variations of weight factors and concept scores are assumed to be realistic within these critical sensitivity ranges. Weight factors and concept scores are varied, and changes in the final outcome of the MCA were observed (and summarized in tables). If a variation in weight factor or concept score within the critical sensitivity range results in another outcome, that criterion is called unstable.

It is concluded from the sensitivity analysis that the solution of the MCA is stable to a reasonable change of every weight factor. But, either an increase of a concept score of Concept 4 or a decrease of a concept score of Concept 5, results in another 'best solution': Concept 4. Which makes the solution of the MCA, strictly speaking, unstable. However, the MCA gives a very stable solution, but the solution comprises of two possible propulsion concepts: Concept 4 or Concept 5.

Chapter 6

Conclusions and recommendations

6.1 Conclusions

In the introduction of this study, Chapter 1, the main question, that forms the basis of this study, was deposited:

Main question: *What propulsion and power generation configuration is best for this ship?*

This is a general question that raises at every design study of a ship. The question was worked out for the example of the current design study of the M-frigate replacement within the RNLN. A systematical approach is worked out to come from a list of requirements to a deliberate advise to the designer. In order to arrive at this advise a number of subquestions were answered.

Subquestion 1: *What are the requirements for the ship?*

The process starts with formulating the requirements and boundary conditions that affect the propulsion and power generation configuration. Most important are the required amount of propulsion power and auxiliary power. The propulsion power is determined by the maximum speed of the ship and the towing resistance. The towing resistance is determined by the hull shape and size, and can be estimated with the Holtrop & Mennen method. Within the DMO a corrected Holtrop & Mennen method is used that is based on naval vessels. The required amount of auxiliary power is determined by the maximum power consumption of all electrical systems. The sensor-, weapons- and communication systems (SEWACO) have a very large share in the total auxiliary power consumption. The auxiliary power can be estimated based on comparable ships with comparable SEWACO systems. It is very useful, for optimization purposes, to define a number of standard operating conditions. The layout of the configuration can be optimized to those standard conditions, in order to have best performance at common conditions. For the SFC this data is summarized in table 2.1.

Besides the main requirements on propulsion and auxiliary power there are more requirements to the ship that affect the propulsion concept. For the SFC the relevant requirements are listed in Chapter 2.3. Typical military requirements are a reduced signature profile, improved shock resistance, high level of redundancy. The requirement on signatures has great effect on the design of the propulsion configuration, because some components, like a gearbox, could best be avoided to reduce the signature profile of the ship. The reduced underwater noise signature has a high priority in the speed range from 0 to 15 knots, so propulsion and power generation needs to be silent at least up to 15 knots.

Subquestion 2: *What are the component characteristics?*

To select the right components that meet the requirements, the engineer has to know the available components and their characteristics. The most common components are selected: diesel engine, gasturbine, fuel cell, electric motor + generator, gearbox, switchboard, converter, cooling system, propeller, waterjet and podded propulsor. Of all components a brief description of the working principle is given. The performance of the components is described by a number of characteristics that either say something about their effectiveness (how well they perform) or about their efficiency (what the performance costs): available power, dimensions, weight, operating speed, (energy) efficiency, signature profile, maintainability, reliability and purchase costs.

To describe the characteristics of the components, a lot of data is collected, and estimation models are derived to estimate component performance based on basic information (e.g. power and speed). The AES study – a comparable study from 1998/1999 under supervision of the Netherlands Institute for Maritime research (NIM) – also derived such models. Initially, these models were used, but most of the models turned out to be very inaccurate when they were compared with the collected data. So, new models needed to be derived. A lot of data is collected and trends are recognized from the data.

For the diesel engine a lot of data is available, and rather accurate models are derived for dimensions, weight and efficiency. For the gasturbine less data is available, but the models are still rather accurate. The models for the electrical machine are derived from a physical approach, but the large variety in types makes it difficult to determine accurate models. Still, the result is satisfying. For gearboxes a model was found in literature that relates dimensions and weight to offset distance. This method didn't give satisfactory results. A new approach is developed based on power/speed ratio of the gearbox. This model returns remarkably accurate results. For switchboards, converters and propellers there is very limited data available and models are expected to be less accurate.

The signature profile of components is difficult to quantify. Only a qualitative description could be given. For maintainability, the maintenance tasks are described, and it is tried to derive a model for maintenance costs. Almost no data is available, except for the diesel engine and gasturbine. Models on maintenance costs are expected to be very inaccurate. Reliability is also very difficult to quantify because data is scarce and unreliable. Numbers from the AES study are adopted. For the purchase costs it is difficult to derive models because data is very scarce. Manufacturers don't easily give away information. Besides that, the purchase costs are depending on a lot of side factors which makes it hard to derive accurate models. From literature some information was found, and with the help of DMO's Cost Analysis section, Cost Estimating Relations (CER) were determined.

All results of the component analysis are summarized on page 123. An Excel worksheet is developed which easily calculates the quantitative characteristics of the components.

Subquestion 3: *What are the characteristics of the propulsion and power generation concept?*

Eight suitable propulsion and power generation concepts were put together with the components. Three full mechanical concepts: 2 Combined Diesel or Gasturbine (CODOG) concepts, with 2 diesels and 1 or 2 gasturbines, and a Combined Diesel And Diesel (CODAD) concept with 4 diesel engines. Electrical power generation by diesel-generator sets. Four hybrid concepts: 2 Combined Diesel eLectric And Gasturbine (CODLAG) concept, with 2 electric motors and 1 or 2 gasturbines, 1 Combined Diesel eLectric And Diesel And Gasturbine (CODLADAG) concept with 2 electric motors, 2 diesels and 2 gasturbines, and a Combined Diesel eLectric And Diesel Or Gasturbine (CODLADOG) concept with 2 electric motors, 2 diesels and 2 gasturbines. Electric power generation by diesel-generator sets. One full electric concept: Integrated Full Electric Propulsion (IFEP) concept with 2 electric motors and electric power generation by diesel-generator sets and a gasturbine-generator set. The power levels of the machines are optimized for common operating conditions.

All concepts are assessed on a number of assessment criteria: maneuverability, signature profile, level of redundancy, number of components, space consumption, total weight, required fuel capacity, reliability, maintainability, shock resistance, purchase costs, annual fuel costs and maintenance costs of prime movers. The manning costs are not taken into account, because this largely determines on the level of automation. But actually the number of prime movers says something about the required manning: the more prime movers, the more manning, because maintenance on prime movers is the most intensive.

The scores of the concepts on the criteria are determined by the characteristics of the components. The CODOG concept, that is found on the other RNLN frigates, serves as a reference. The hybrid and full electric concepts score better on maneuverability and signature profile. The CODAD concept scores best on fuel consumption and purchase costs. The full electric concept is by far the most expensive.

All results are summarized on page 184. An Excel worksheet is developed which easily calculates the quantitative characteristics of the propulsion concepts.

Subquestion 4: *What propulsion and power generation configuration is best for the ship?*

In order to determine the best propulsion and power generation configuration for the SFC, a Multiple Criteria Analysis (MCA) is done with the help of the software tool TOPSYStem. The assessment criteria of the concepts are the input criteria for the MCA. In a hierarchical structure, the assessment criteria, now child criteria, are subdivided into 4 parent criteria. Each criteria, child and parent, has its own weight factor that indicates the relative importance of that criteria. The choice of weight factors is very subjective. The suggested choice of weight factors by the author is explained. Maneuverability, signature profile and fuel costs have the highest weight factors. Maneuverability and signature profile determine the capabilities of the ship, and the fuel costs mainly determine the usage costs. Purchase costs is also very important, but gets a lower weight factor because the estimates are not very accurate in this early stage and it makes more sense to look at through life costs than only at purchase costs. The number of components is the least important criterion. All weight factors that were suggested by the author are summarized on page 194.

Once all concept scores and weight factors are put into TOPSYStem, the program calculates the aggregate score. The program has 5 different aggregation methods, of which 3 are suitable: weighted sum, weighted product and mixed concordance method. Weighted sum is the most common MCA aggregation method, but the mixed concordance method is more suitable for MCA's with both quantitative and qualitative criteria, which this MCA is. The aggregate score of this MCA is calculated with both methods, and the results are totally different. Though, with both methods Concept 4 and Concept 5 hold the first and second position. But with the weighted sum method the winner is Concept 4, and with the mixed concordance method it is Concept 5. The most remarkable difference between the methods is the position of Concept 8. With the weighted sum method, this concept scores lowest, and with the mixed concordance method it ends up third position. The aggregate scores of both methods are found on page 196.

Because there is a high level of subjectivity in assigning the weight factors, a sensitivity analysis is carried out. Sensitivity analysis is only carried out on the mixed concordance method. All weight factors are varied until they cause a changing result. The most sensitive is the parent criterion *Availability*. A decrease of this weight factor by 1.1 (from 4 to 2.9) results in a changing result (from Concept 5 to Concept 4). Such a decrease of this criterion is very unrealistic. The second most sensitive criterion is the child criterion *Number of components*. A decrease of this weight factor by 1.4 (from 2 to 0.6) results in a changing result (from Concept 5 to Concept 4).

This decrease is also very unrealistic. All other criteria need much higher in- or decreases to result in another outcome. All results of the sensitivity analysis on weight factors are summarized on pages 199 and 200.

A sensitivity analysis is also carried out on the concept scores, because there is some level of subjectivity in assigning the qualitative concept scores, and there is inaccuracy in the estimation of quantitative scores. The qualitative scores are varied in a range of ±2 steps on the [−−−...+++]-scale. Every decrease of qualitative score of Concept 5 results in Concept 4 to be the new winner, so does every increase of qualitative score of Concept 4. All results of the sensitivity analysis on qualitative concept scores are summarized on page 201. For the variation of the quantitative scores some accuracy ranges were defined based on the accuracy of the estimation models. Every decrease of quantitative score within the accuracy range of Concept 5 results in Concept 4 to be the new winner, so does every increase of quantitative score within the accuracy range of Concept 4. All results of the sensitivity analysis on quantitative concept scores are summarized on page 203.

All together, the conclusion of this concept study is that on the basis of the used methodology Concept 5 is advised as 'best' propulsion and power generation configuration. This concept is not the indisputed winner of this study, because it is closely followed by Concept 4. Together they form a stable solution to the decisionmaking problem.

6.2 Recommendations

This study describes a methodology to design and evaluate propulsion and power generation configurations for surface combatant. But the methodology as it is described here is not without errors and incompletenesses.

The maintenance costs of the components are now only estimated for diesel engines and gasturbines. Ideally, the maintenance costs for all components are taken into account. The models that are derived for the maintenance costs are very inaccurate. The need for more accurate models exists. A lot of research is done on this topic, but it is experienced that this is a very complex topic to get hold on. Another important aspect is the required manning. The manning costs are probably the largest part of the through life costs, these are not taken into account now. The maintainability of a concept might say something about the required manning, and the costs should also be taken into account. Although, it is questionable if difference can be noticed between concepts, because the concept of manning a naval ship is based on redundancy.

The dimensions of diesel- and gasturbine-generator sets are now calculated by adding the dimensions of the diesel or gasturbine and the generator. In reality the set will be larger. Even more important than the dimensions, is the weight. The weight is now calculated by adding the weight of diesel or gasturbine and the generator. The weight of the bedplates is not taken into account, and this weight is significant. Recommendation is to treat diesel- and gasturbine-generators as separate components.

The signature profile of the components and concepts are now described qualitatively. It would be useful if numbers can be put to the signature profile. Information on signatures is often confidential, so it is difficult to get hold on this information.

In this study, the numbers on reliability are adopted from the AES study. These are very old numbers and probably not applicable anymore. No thorough research was done to more recent numbers.

Very large uncertainties are introduced in the estimation of purchase costs. Although, it is difficult to do accurate estimations in such an early design stage, the need exist for more accurate estimation models, but above all for more detailed models. With the current models no distinction can be made between different types of machines (e.g. conventional vs permanent magnet motor). On all components there is need for more detailed estimation models.

In characterizing the electrical converters, the DC chopper is not taken into account, because only 1 datapoint is available. To really investigate an option with DC motors, more information is needed on the chopper.

Maneuverability should be assessed based on dynamical simulations. Now is only looked qualitatively at power, acceleration, deceleration and steerability. This could well be quantified by the results of dynamic simulations.

Finally, in this study are only investigated conventional concepts. This is caused by the requirement in the OC that states that conventional techniques should be used. Nevertheless, it would be very interesting to investigate more non-conventional concepts with for instance a large number of micro turbine-generators throughout the ship in a full electric concept, or even a small nuclear plant.

References

Aalbers, A. Unknown date. *Evaluation of ship design alternatives.* Tech. rept. Delft University of Technology.

Bernitsas, M.M., Ray, D., & Kinley, P. 1981. *K_T, K_Q and efficiency curves for the Wageningen B-series propellers.* Tech. rept. University of Michigan: Department of Naval Architecture and Marine Engineering.

Bosklopper, J.J. 2009. *Systeemstudie van elektrische/hybride aandrijving.* Thesis: Royal Netherlands Naval College.

de Jong, E.C.W., & Hoeijmakers, M.J. 2002 (March). *Prepatory study of acoustics of electrical machines intended for marine propulsion (DGT-RPT-020040).* Tech. rept. TNO.

Dill, R.B., & Evenblij, B.H. 2003 (October). *Haalbaarheid modellering EM velden (DGT-RPT-020141)[confidential].* Tech. rept. TNO.

Flusund, I.B. 2011 (May). Propulsion and Electrical power systems efficiency - Vessels operating on fixed routes and multi-purpose cargo vessels for short sea shipping. *In: The 33^rd Motorship Propulsion and Emissions Conference.*

Frouws, J.W. 2008. *AES decision model.* Tech. rept. Delft University of Technology.

Henderson, K. 2010. *Jane's Marine Propulsion: issue 26.* Coulsdon: IHS.

Hendriks, B.R., Kätow, L., & Hasenpflug, H. 2011 (February). *Future Surface Combatant signature concept study (CSSM-2011-01)[confidential].* Tech. rept. Centre for ship signature management.

Hodge, C.G., & Mattick, D.J. 2001. The electric warship VI. *Trans IMarE*, **113**(2), 49–63.

Holtrop, J., & Mennen, G.G.J. 1982. An approximate power prediction. *Pages 166–170 of: International Shipbuilding Progress 29.*

JohnCrane-Lips. 2001 (September). *Advanced Naval Propulsion Systems.* Brochure of John Crane-Lips Defence Propulsion Systems.

KleinWoud, H., & Stapersma, D. 2003. *Marine Engineering; Design of propulsion and electric power generation systems.* Vol. 2nd print. London: IMarEST.

Kok, R. 1997 (December). *A comparison of advanced cycle gas turbines with medium speed diesel engines in cruise ship applications.* M.Phil. thesis, Delft University of Technology.

Kuiper, G. 2007. Resistance and propulsion I. *In: Hydromechanics III lecture notes.* TU Delft.

Lee, R. *Core materials.* www.vias.org/eltransformers/lee_electronic_transformers_03_03.html.

MANB&W. 2010. *Basic principles of ship propulsion.* www.manbw.com/files/news/filesof3859/P254-04-04.pdf.

Miller, T.J.E. 1989. *Brushless permanent-magnet and reluctance motor drives.* Oxford: Oxford Science publications.

Moss, N.J., & Thomson, J.M. 1994. The impact of ICR marine gas turbines on future warship design. *In: INEC 94; Cost Effective Martitime Defence.*

Newell, J.M., Mattick, D.J., & Hodge, C.G. 1998. The electric warship IV. *Trans IMarE*, **111**(1), 25–39.

Nijsen, L. 2010 (October). *E-mail contact about indicative costs of slow-speed diesel engines.* MAN B&W.

Oudenaller, R.A. 2011. *Onderhoud en gebruik van gasturbines bij het CZSK.* Thesis: Royal Netherlands Naval College.

Plumb, C.M. 1987. *Warship Propulsion System Selection.* London: Marine Management (Holdings) Ltd.

Rolls-Royce. 1997. *SM1C Propulsion Unit: Technical Description.*

Rolls-Royce. 1999. *WR21: Fact Sheet.*

Stapersma, D. 1994. The importance of (e)mission profiles for naval ships. *In: INEC 94; Cost Effective Martitime Defence.*

Stapersma, D. 2001. *Prime movers in an AES powerplant: number of units, size and cost.* Tech. rept. Royal Netherlands Naval College.

Stapersma, D. 2009a. *Diesel engine database in Excel.*

Stapersma, D. 2009b. Volume 1: Performance analysis and turbocharging. *In: Diesel Engines lecture notes*, vol. 7th print. TU Delft & NLDA.

Stapersma, D. 2010. Volume 4: Emissions and heat transfer. *In: Diesel Engines lecture notes*, vol. 6th print. TU Delft & NLDA.

Takken, E.H. 2010. *SurFace Combatant: Concept VIIIa Revision 2.* Tech. rept. Defense Materiel Organization.

Trouwborst, W. 1998. *Non-conventional ship propulsion; literature study on contra-rotating propellers and podded propulsion (98-CMC-R1337).* Tech. rept. TNO.

Unknown. 2002. Optimal electric ship propulsion solution. *Marine Reporter.*

van den Eikhoff, J. 2002. *Shock resistance of a podded propulsion system for naval vessels (2002-CMC-R060)[confidential].* Tech. rept. TNO.

van Dijk, J.C., Meerman, E.R.W., & Vaessen, P.T.M. 1998 (June). *Komponenten en parameters; Taken 030 en 040.* Tech. rept. Projectbureau AES.

van Terwisga, T.J.C. 1997. A parametric propulsion prediction method for waterjet driven craft. *In: 4th International Conference on Fast Sea Transportation.*

van Terwisga, T.J.C. Unknown date. *MARIN training course on waterjet propulsion (selected chapters).* Tech. rept. MARIN.

van Vugt, J., Jansen, S., & van Dijk, E. 1998 (June). *Voorstudie All Electric Ship; Taak 045: Modelleren van de energiesysteemcomponenten (1998-CMC-R0371).* Tech. rept. TNO.

Vrijdag, A. 2009. *Control of propellor cavitation in operational conditions.* Ph.D. thesis, Delft University of Technology.

Vrijdag, A. 2011 (February). *M-Class replacement power and propulsion study.* Propulsion study by Rolls-Royce.

Wärtsilä. 2003. *Marine project guide W26A.*

Webster, J.S., Fireman, H., Allen, D.A., Mackenna, A.J., & Hootman, J.C. 2007a. Alternative propulsion methods for surface combatants and amphibious warfare ships. *Transaction of the Society of Naval Architects and Marine Engineers*, **115**, 224–244.

Webster, J.S., Fireman, H., Allen, D.A., Mackenna, A.J., & Hootman, J.C. 2007b. US Navy studies on alternative fuel sources and power and propulsion methods for surface combatants and amphibious warfare ships. *Naval Engineers Journal*, 35–47.

Wijnmalen, D.J.D. 1999. *User's guide to TOPSYStem 4 for Windows, program for multi-criteria decision support (FEL-97-H136).* Tech. rept. TNO.

Part III

Appendices

Appendix A

Abbreviations

AAW	Anti-Air Warfare
AC	Alternating Current
ADGT	Aero-Derivative Gasturbine
AES	All Electric Ship
AIM	Advanced Induction Machine
ASuW	Anti-Surface Warfare
ASW	Anti-Submarine Warfare
AUV	Autonomous Underwater Vehicle
BN	Belgian Navy
BSc	Bachelor of Science
CER	Cost Estimating Relationship
CODAD	Combined Diesel And Diesel
CODAG	Combined Diesel And Gasturbine
CODAG-WARP	Combined Diesel And Gasturbine - Waterjet And Refined Propeller
CODLADAD	Combined Diesel eLectric And Diesel And Diesel
CODLADOG	Combined Diesel eLectric And Diesel Or Gasturbine
CODLAG	Combined Diesel eLectric And Gasturbine
CODLOD	Combined Diesel eLectric Or Diesel
CODOG	Combined Diesel Or Gasturbine
COTS	Commercial Of The Shelf
CPP	Controllable Pitch Propeller
CSI	Current Source Inverter
CVF	Cooling Volume Factor
DC	Direct Current
DE	Diesel Engine
DG	Diesel-Generator
DMO	Defence Materiel Organisation
DOBBP	Directorate of Operational Policy, Requirements and Plans
DP	Dynamic Positioning
DTC	Direct Torque Control
DUT	Delft University of Technology
ECA	Emission Controlled Area
EMI	Electro Magnetic Interference
FPP	Fixed Pitch Propeller
GB	Gearbox
GES	Gentegreerde Energy Systemen/Integrated Energy Systems
GT	Gasturbine

GT-G	Gasturbine-Generator
GTO	Gate Turn-Off thyristor
HFO	Heavy Fuel Oil
HOV	Hydrografic Research Vessel
HTS	High Temperature Superconduction
HVAC	Heating, Ventilation and Air Conditioning
HVU	High Valuable Unit
ICR	Inter-Cooled Regenerative
IFEP	Integrated Full Electric Propulsion
IGBT	Insulated Gate Bipolar Transistor
IGCT	Integrated Gate-Commutated Thyristor
IMO	International Maritime Organisation
IPC	Initial Purchase Costs
IR	Infrared
JSS	Joint Support Ship
LCF	Airdefence and Command Frigate
LFAS	Low Frequency Active Sonar
LHV	Lower Heating Value
LNG	Liquefied Natural Gas
LPD	Landing Platform Dock
MCA	Multiple Criteria Analysis
MIMO	Multiple Input Multiple Output
MIO	Maritime Interdiction Operations
MISO	Multiple Input Single Output
MoD	Ministry of Defence
MSc	Master of Science
MSO	Maritime Security Operations
MTBF	Mean Time Between Failure
MTTR	Mean Time To Repair
MW	Mega Watt
NIM	Netherlands Institute for Maritime research
NLDA	Netherlands Defence Academy
NOR	Nitrogen Oxide Reducer
OC	Operational Concept
OPV	Oceangoing Patrol Vessel
PEMFC	Proton Exchange Membrane Fuel Cell
PFM	Pulse Frequency Modulation
PM	Permanent Magnet
PWM	Pulse Width Modulation
RCS	Radar Cross-Section
RHIB	Rigid Hull Inflatable Boat
RN	Royal Navy
RNLN	Royal Netherlands Navy
RPG	Rocket-Propelled Grenade
SCR	Selective Catalytic Reduction
SEWACO	Sensor-, Weapons- and Communication systems
SFC	Surface Combatant
SISO	Single Input Single Output
SSS	Self Shifting Synchronous clutch
SUMC	Specific Unit Maintenace Costs
TAS	Towed Array Sonar

TDC	Top Dead Center
TNO	Toegepast Natuurwetenschappelijk Onderzoek/Applied Science Research
TRV	Torque per Rotor Volume
UAV	Unmanned Aerial Vehicle
UKMOD	United Kingdom Ministry of Defence
USSV	Unmanned Sea Surface Vehicle
USV	Unmanned Surface Vehicle
UUV	Unmanned Underwater Vehicle
VSI	Voltage Source Inverter

Appendix B

Nomenclature

ϵ	Pressure ratio	(-)
η	Efficiency	(-)
κ	Isentropic coefficient	(-)
λ	Air excess ratio	(-)
λ_S	Stroke/bore diameter	(-)
ω	Angular velocity	(rad/s)
ρ	Density	(kg/m^3)
σ	Air/fuel ratio	(-)
σ	Airgap shear stress	(kN/m^2)
τ	Temperature ratio	(-)
A	Area	(m^2)
A	Electric loading	(A/m)
A_E/A_0	Effective blade area ratio	(-)
B	Flux density	(T)
c_m	Mean piston speed	(m/s)
c_p	Specific heat	(kJ/kg/K)
C_T	Thrust loading coefficient	(-)
D	Depth	(m)
D_B	Bore diameter	(m)
EMF	Electromagnetive force	(N)
f	Frequency	(Hz)
Fn	Froude number	(-)
g	Gravitational acceleration	(m/s^2)
H	Height	(m)
H	Enthalphy	(kJ/kg)
i	Number of cylinders	(-)
I	Current	(A)
i	Gearbox ratio	(-)
J	Advance ratio	(-)
k	Number of revolutions per cycle	(-)
k_p	Number of propellers	(-)
K_Q	Torqe coefficient	(-)
K_T	Thrust coefficient	(-)
L	Length	(m)
l	Length	(m)
L/D	Length/diameter ratio	(-)
L_S	Stroke length	(m)

LHV	Lower heating value	(kJ/kg)
$\dot{m}$	Mass flow	(kg/s)
M	Torque	(Nm)
n	Rotational speed	(Hz)
N	Rotational speed	(rpm)
n_f	Number of fields	(-)
p	Number of pole pairs	(-)
P/D	Pitch/diameter ratio	(-)
P_B	Brake power	(MW)
p_{me}	Mean effective pressure	(Pa)
R	Ship resistance	(kN)
r	Radius	(m)
S	Apparent power	(VA)
s	Split ratio	(-)
sfc	Specific fuel consumption	(g/kWh)
SM	Sea margin	(-)
T	Thrust	(kN)
t	Thrust deduction factor	(-)
T	Draught	(m)
T	Torque	(Nm)
U	Voltage	(V)
v	Velocity	(m/s)
V	Volume	(m^3)
V_S	Stroke volume	(m^3)
w	Wake fraction	(-)
W	Width	(m)
Z	Number of blades	(-)

Appendix C

Components

In this appendix are found some additional figures that did not directly fit within the text of chapter 3.

C.1 Diesel engines

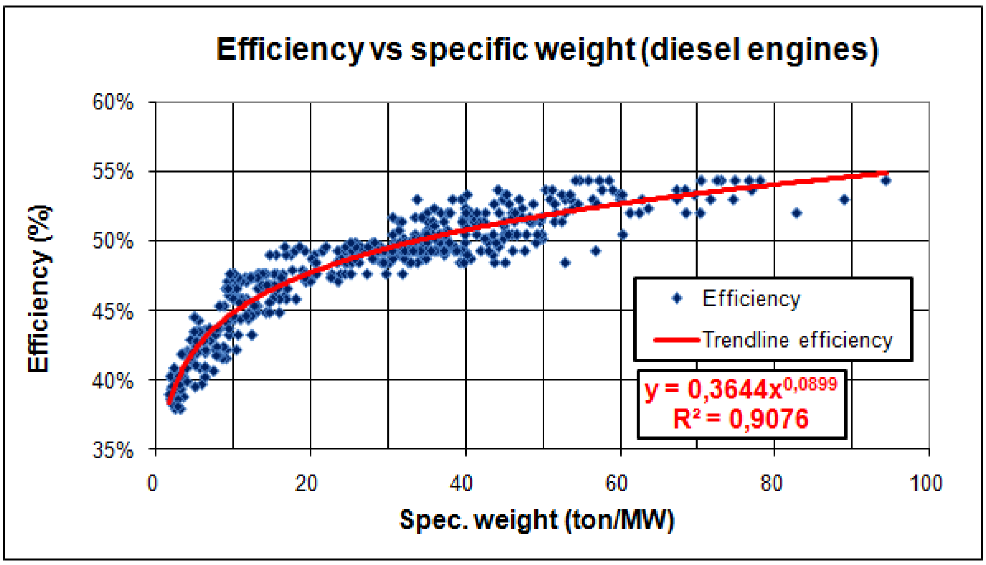


Figure C.1: Nominal engine efficiency (%) of database diesel engines vs specific weight (ton/MW)

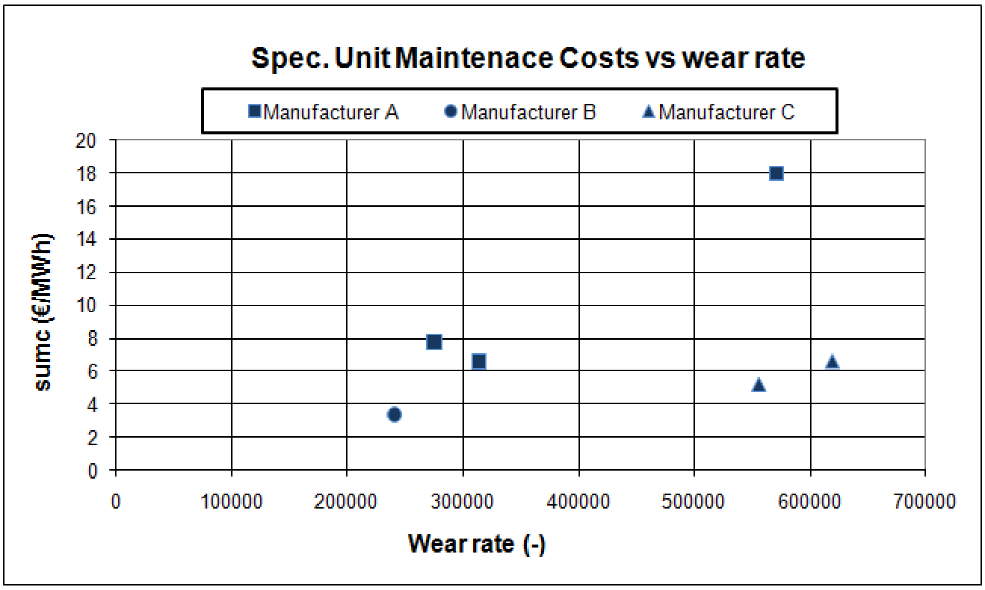


Figure C.2: Manufacturers data for Specific Unit Maintenance Costs (€/MWh) vs wear rate

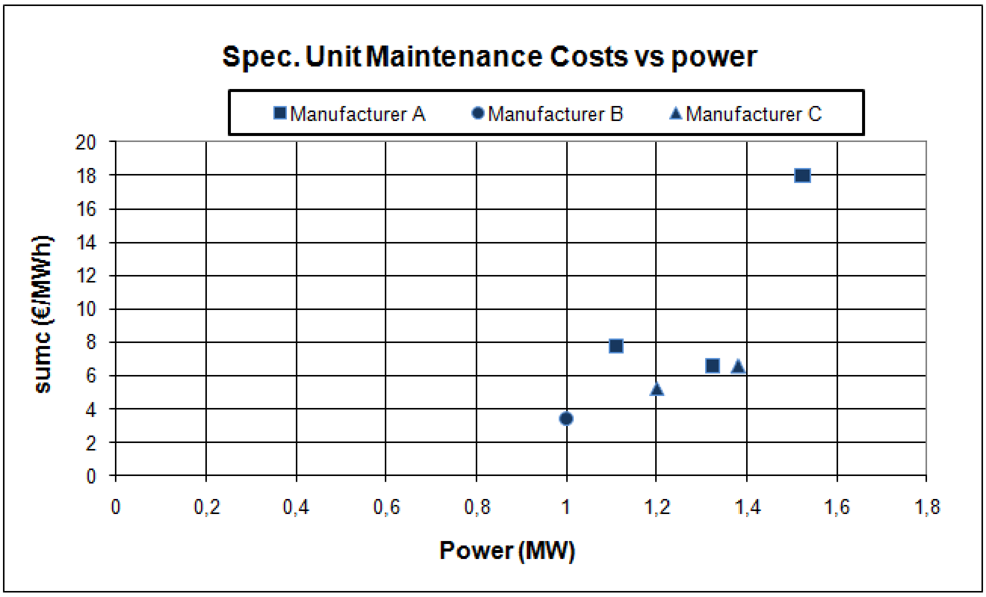


Figure C.3: Manufacturers data for Specific Unit Maintenance Costs (€/MWh) vs power (MW)

C.2 Gasturbines

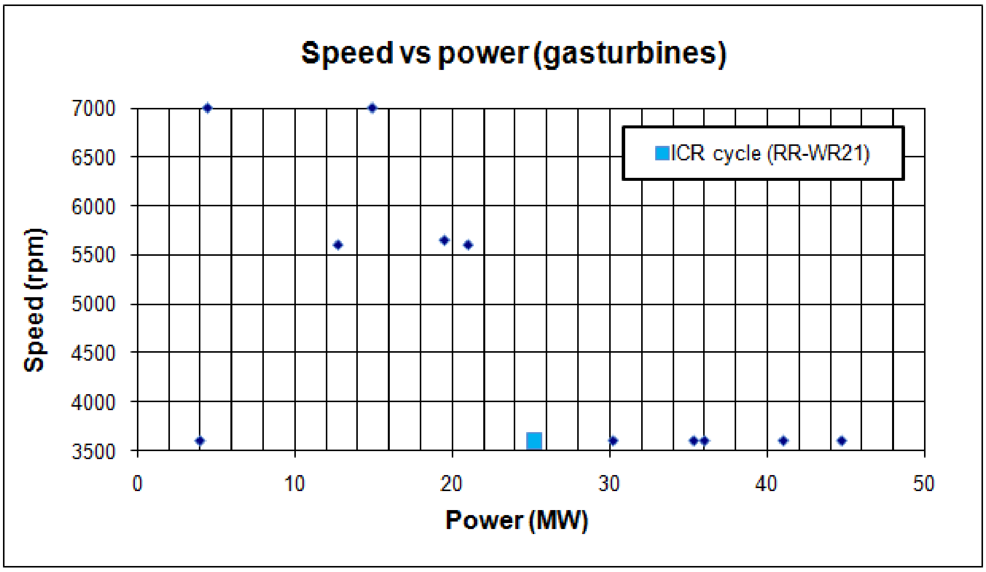


Figure C.4: Nominal speed (rpm) of database gasturbines vs nominal power (MW)

Appendix D

Propeller matching

This section illustrates the method to determine propeller efficiency and propeller rotational speed as applied in chapter 4.

With the polynomials from Bernitsas *et al.* (1981), the open water diagrams of the Wageningen B-series can be determined. This propeller series is used as reference to calculate shaft rpm and propeller efficiencies. The Wageningen B565 is used as reference for this study, because comparable ship types have propellers with 5 blades, and a 0.65 is a typical blade area ratio for frigate propellers.

With the resistance curve for the SFC from the Holtrop & Mennen method the thrust force at all speeds can be calculated. The thrust is presented dimensionless as $K_{T,ship}$ in the open water diagram of the propeller. The intersection of $K_{T,propeller}$ and $K_{T,ship}$ gives the operating point J. With a given diameter (4.5 m), the propeller rotational speed n is calculated, with equation 3.107c. Figure D.1 gives an example of this method.

The optimal P/D-ratio of the propeller at a certain speed is determined by applying the above described method at that specific ship speed, and determine the open water efficiency of the propeller at the intersection of $K_{T,propeller}$ and $K_{T,ship}$. This can be repeated for a number

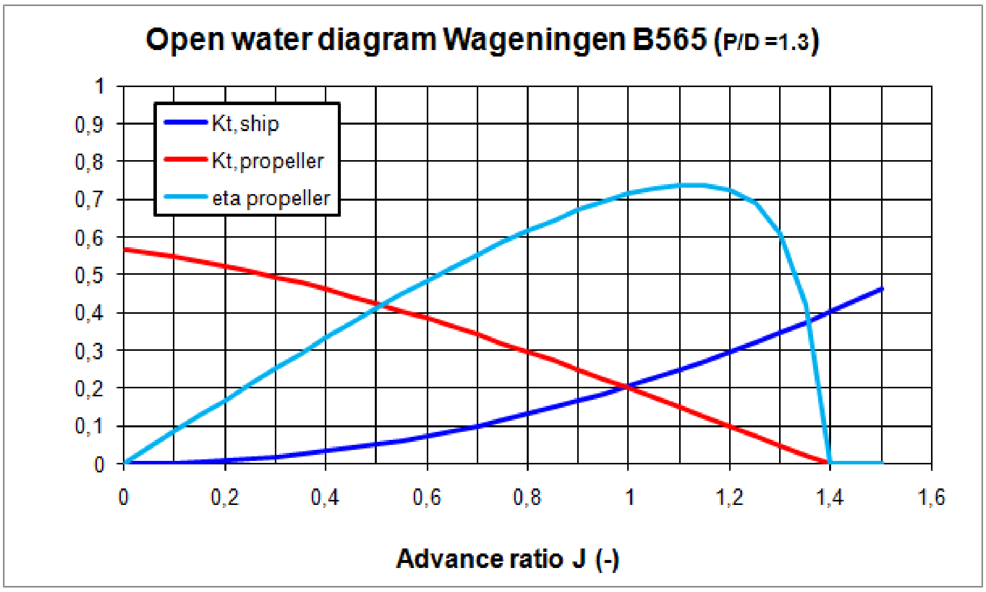


Figure D.1: Open water diagram of Wageningen B565 with P/D=1.3, ship's curve is for a speed of 30 knots

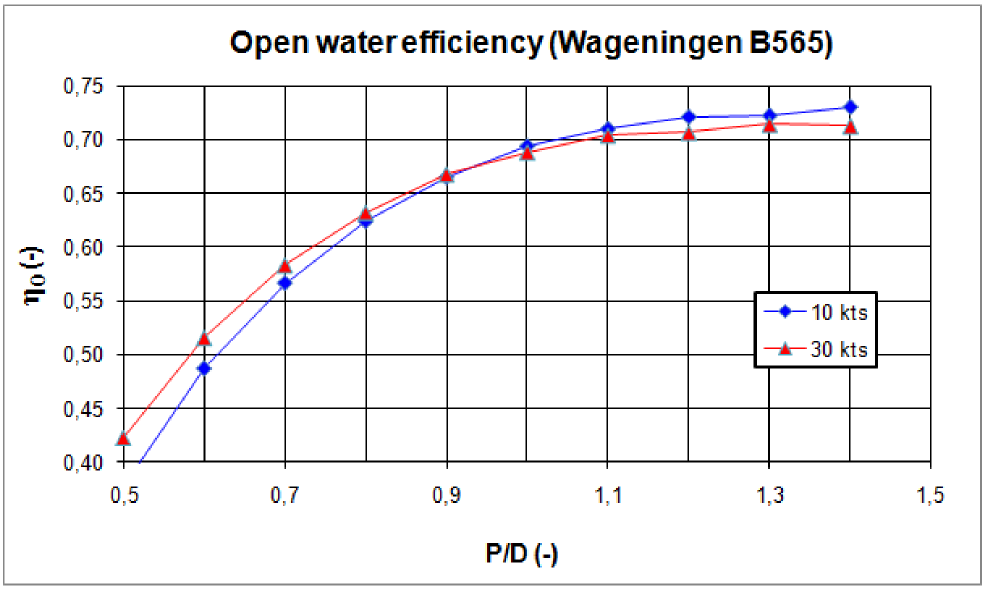


Figure D.2: Open water efficiency vs P/D-ratio of Wageningen B565 at 10 and 30 knots

of P/D-ratios and plot in a figure. Figure D.2 presents propeller open water efficiency versus P/D-ratio at two different ship speeds.

Appendix E

TOPSYS

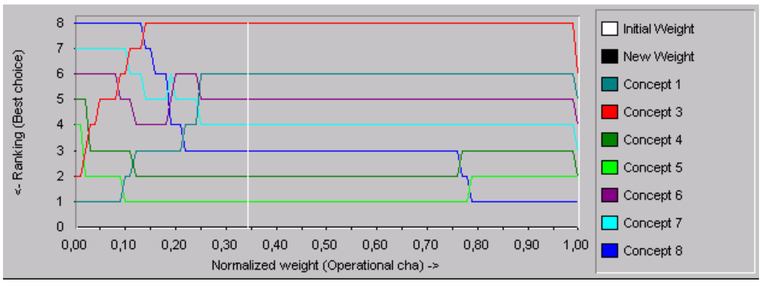
E.1 Evaluation methods

In table E.1 all evaluation methods within TOPSYStem are pointed out with their nature and scale transformation to the [*0...10*]-scale.

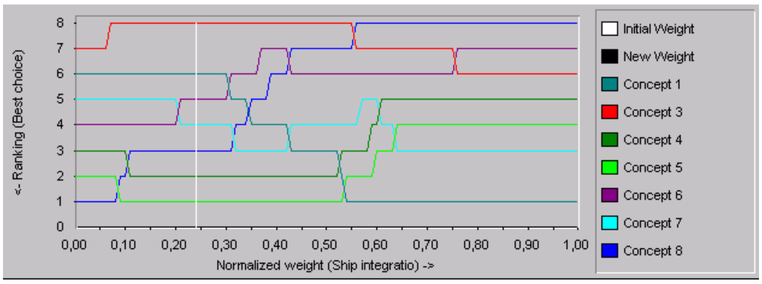
Evaluation method	Nature	Scale transformation
[*min...max*]-scale	Quantitative with value transformation; direct comparison	$$10\left(\frac{X-min}{max-min}\right)^{\alpha}$$ (increasing value) $$10\left(1-\frac{X-min}{max-min}\right)^{\alpha}$$ (decreasing value)
[*0...max*]-scale	Quantitative; direct comparison	$$\frac{10 \cdot X}{max}$$
[$---$...$+++$]-symbolic scale	Qualitative with pre-defined quantitative values; direct comparison	$--- = \frac{10}{14}$; $-- = \frac{30}{14}$; $- = \frac{50}{14}$; $0 = \frac{70}{14}$; $+ = \frac{90}{14}$; $++ = \frac{110}{14}$; $+++ = \frac{130}{14}$
[*1, 2,...n*]-ordinal scale	Qualitative with pre-defined quantitative values according to rank-ordering scheme; direct relative comparison	$$X \cdot \frac{10}{max}$$
Semantic (free verbal) scale	Qualitative according to user-defined scale (ordinal) or quantitative; direct comparison	User-defined scores on [*0...10*]-scale
[*1...max*]-scale	Quantitative; pairwise comparison, assign a value to how many times A is better than B	Compute geometric mean of pairwise comparisons, example in Appendix **??**
[*0...1*]-graphical scale	Quantitative; pairwise comparison, assign a graphical value to how many times A is better than B	Compute geometric mean of pairwise comparisons after quantifying, example in Appendix **??**
[*extremely less...extremely more*]-verbal scale (Saaty)	Quantitative interpretation of verbal statements, with pre-defined quantitative values; pairwise comparison, assign a verbal value to how many times A is better than B	Compute geometric mean of pairwise comparisons after quantifying, example in Appendix **??**
[*<,=,>1*]-ordinal/binary scale	Qualitative with pre-defined quantitative values according to rank-ordering scheme; pairwise comparison, assign a value to relation between A and B	Compute geometric mean of pairwise comparisons after quantifying, example in Appendix **??**

Table E.1: Overview of evaluation methods in TOPSYStem

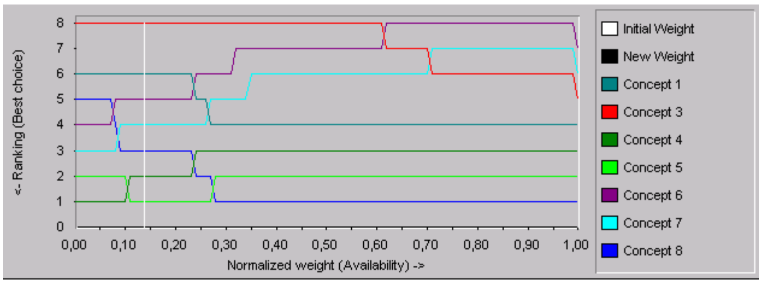
E.2 Sensitivity analysis figures



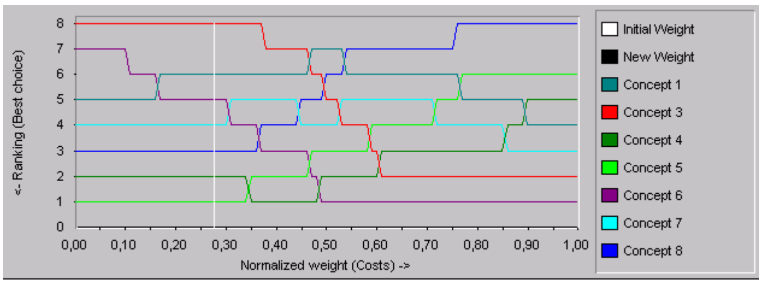
(a) Operational characteristics



(b) Ship integration

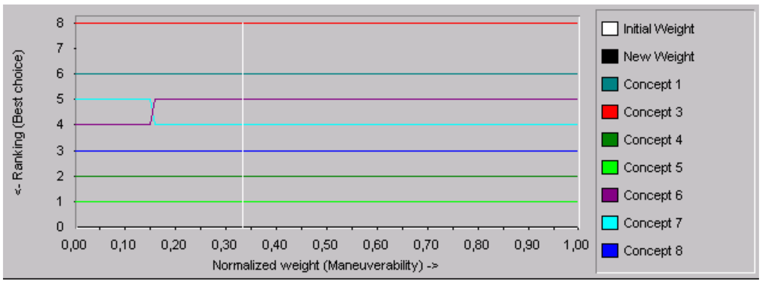


(c) Availability

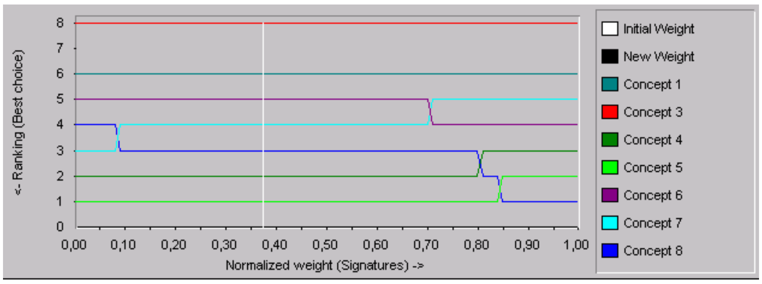


(d) Costs

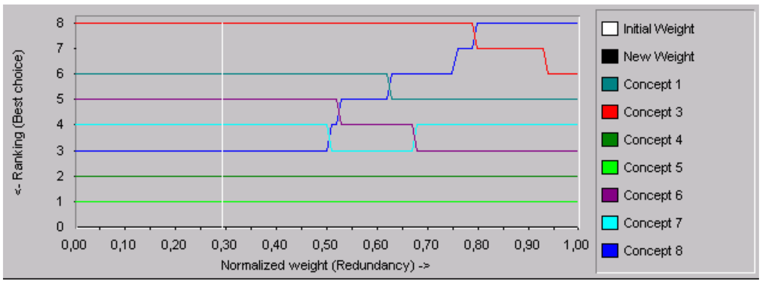
Figure E.1: Trade-off sensitivity analysis to the change of parent criteria weight factors (mixed concordance method)



(a) Maneuverability

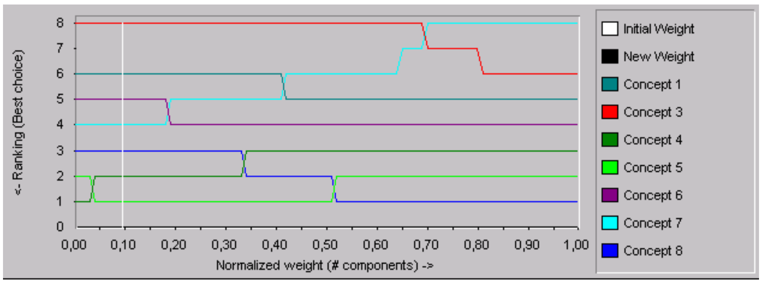


(b) Signature profile

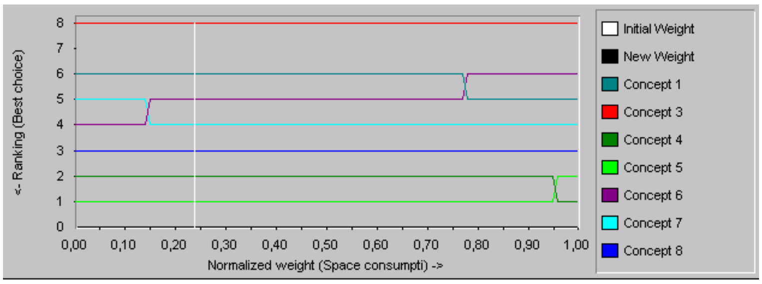


(c) Redundancy

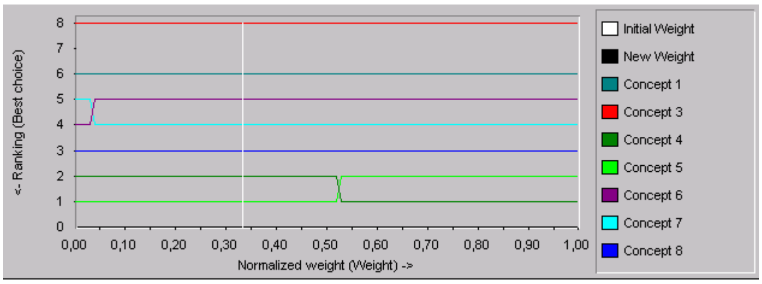
Figure E.2: Trade-off sensitivity analysis to the change of child criteria weight factors under the parent criterion 'Operational characteristics'(mixed concordance method)



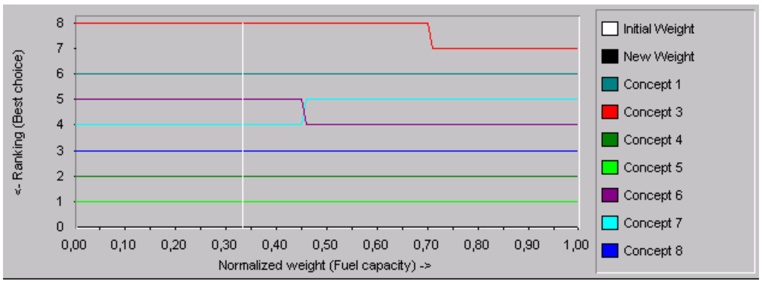
(a) Number of components



(b) Space consumption

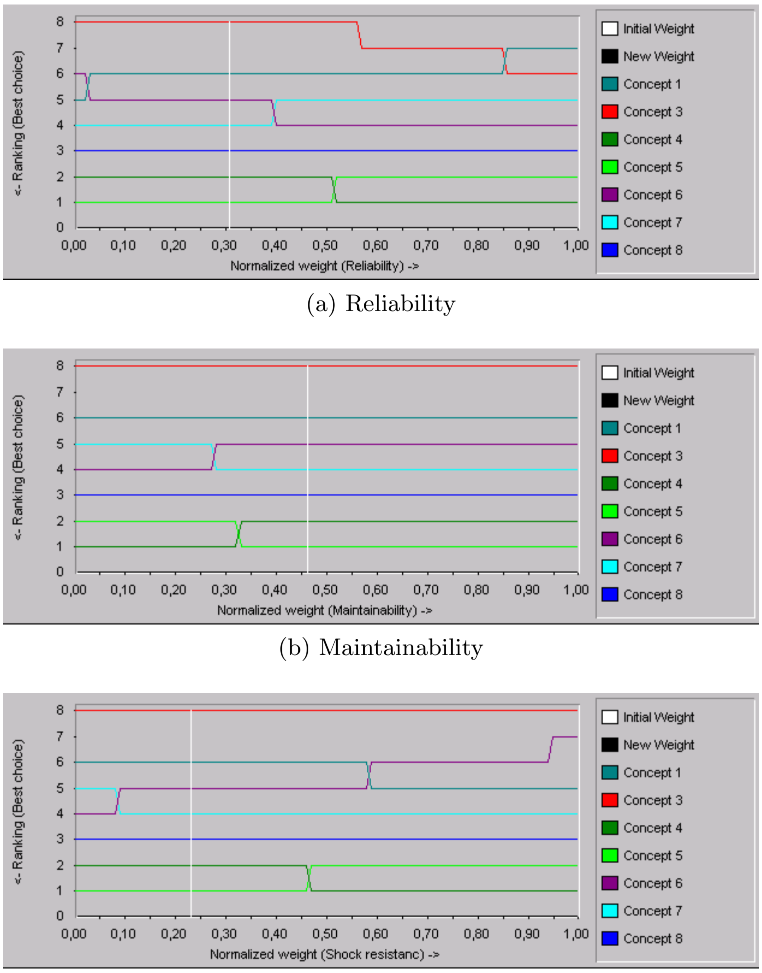


(c) Weight



(d) Fuel capacity

Figure E.3: Trade-off sensitivity analysis to the change of child criteria weight factors under the parent criterion 'Integration in ship'(mixed concordance method)

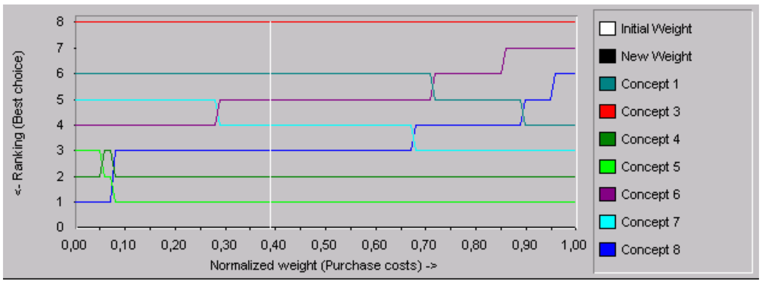


(a) Reliability

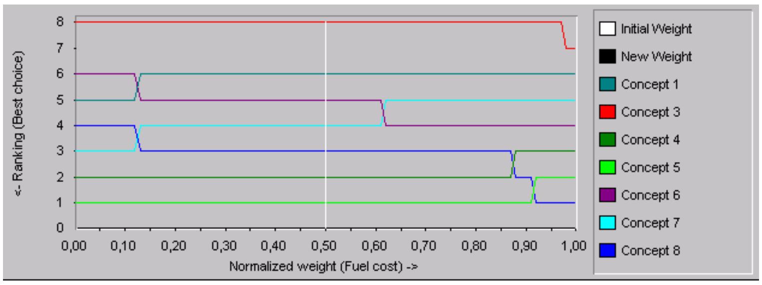
(b) Maintainability

(c) Shock resistance

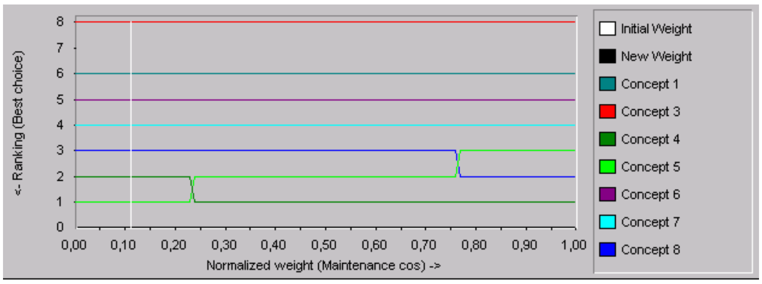
Figure E.4: Trade-off sensitivity analysis to the change of child criteria weight factors under the parent criterion 'Availability'(mixed concordance method)



(a) Initial purchase costs



(b) Fuel costs



(c) Maintenance costs

Figure E.5: Trade-off sensitivity analysis to the change of child criteria weight factors under the parent criterion 'Costs'(mixed concordance method)

Appendix F

Concept analysis: Excel sheets

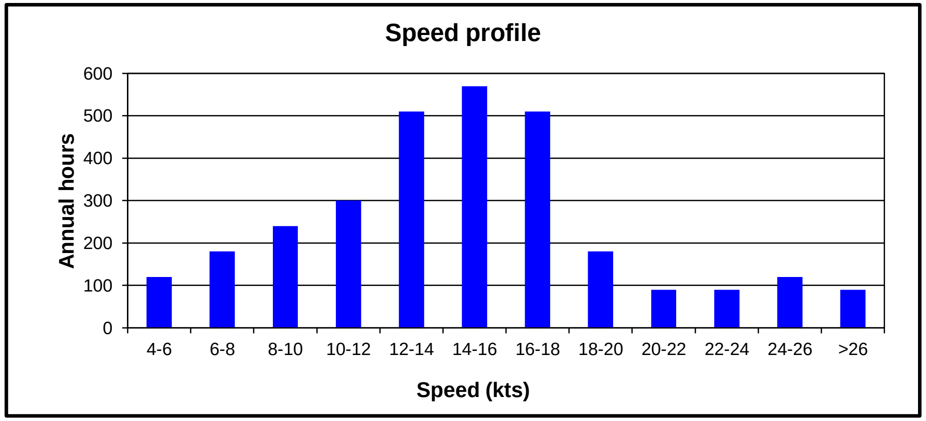
This final appendix holds the Excel worksheets that were developed and used in the concept analysis. Each concept comprises of 2 Excel sheets: the first one is for calculation of dimensions, weight, efficiency and purchase costs, and the second one is for calculation of the annual fuel costs and maintenance costs based on the input speed profile. The speed profile is defined on the very first page of the Excel workmap. The last sheet is the scorecard, which summarizes all results from all 8 concepts. The concept scores are presented as absolute scores per criterion, also as a score on a scale from 1 to 10 and the relative score in comparison with the reference concept: Concept 1.

For different propulsion concepts, dimensions, volume, weight, efficiencies and fuel consumption can be estimated with this Excel tool.

For every concept, the machinery needs to be selected and specified. It is important that at all times, the YELLOW marked cells are filled in. For some cells there is a drop-down box from which an option can be selected. When all YELLOW cells are filled in, the approximate dimension, weight etc. are calculated and filled in.

In the first column (A) the number of different types of a certain component selected. Example: if 6 diesel engines are installed, 3 high speed 2 MW engines and 3 medium speed 6 MW engines, you fill in 2 in column A, and 3 in column F for both engine types.

In a separate sheet, the fuel consumption calculations are made. In this sheet you also need to enter the main characteristics of the machinery. The efficiency is calculated, and at all speeds the engine load is calculated. The engine loading together with the time and the partload fuel consumption gives the total fuel consumption. On this sheet the speedprofile needs to be filled in.



These cells need to be filled in!
These cells contain global values from the input sheet
These cells were manually changed
These cells indicate a possible engine overload

TOTAL ANNUAL HOURS

3000

SPEED PROFILE			AUXILIARY POWER
(knots)	*(% of time)*	*(hours)*	*(MWe)*
4-6	0,04	120	0,800
6-8	0,06	180	0,800
8-10	0,08	240	0,800
10-12	0,1	300	1,350
12-14	0,17	510	1,350
14-16	0,19	570	1,350
16-18	0,17	510	1,100
18-20	0,06	180	1,100
20-22	0,03	90	1,100
22-24	0,03	90	1,100
24-26	0,04	120	1,100
>26	0,03	90	1,100

Name: **Concept 1/CODOG**

# of different types	Component		Extra info			#	INPUT		DIMENSIONS						WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
							Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	80%	50%	30%	Noise	EM	IR	MTBF	MTTR	Avail
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	*For generator?*	*High/medium/slow speed*	*Vee/Line*	*2-stroke/4-stroke*																					
	1	*No (propulsion)*	Medium speed (300-1000 rpm)	Vee	4-stroke	2	3,0	1000	4,01	2,00	2,82	22,56	0,243	0,228	20,17	1358,55	0,4255	0,4178	0,3820	0,3286				10000	5	0,9995
2	*2*	*Yes (generator)*	Medium speed (300-1000 rpm)	Vee	4-stroke	4	0,85	900	1,74	1,18	1,58	3,24	0,069	0,065	6,19		0,4286	0,4210	0,3851	0,3318				10000	5	0,9995
					Total:	6	9,4					58,1	0,8	0,7	65,1	2717,1										
	Gasturbine	*For generator?*	*Simple cycle/ICR cycle*																							
	1	*No (propulsion)*	Simple cycle			2	18,4	5600	7,35	2,53	3,12	58,03	3,551	3,036	19,70	6403,20	0,3720							4500	8	0,998225
1																										
					Total:	2	36,8					116,1	7,1	6,1	39,4	12806,4										
	Electric motor		*Asynchronous/Synchronous*	*AC/DC/PM/HTS*	***L/D ratio***																					
													-	-												
													-	-												
													-	-												
					Total:								-	-												
	Generator		*Diesel/Gasturbine*	*AC/DC/PM/HTS*	***L/D ratio***																					
	1		Diesel driven	Conventional AC	1,6	4	0,9	900	1,41	0,81	1,60	1,82	-	-	2,19	1172,80	0,9715	0,9634	0,9562	0,9423				201480	120	0,999405
1													-	-												
													-	-												
					Total:	4	3,6					7,3	-	-	8,7	4691,2										
	Gearbox		*Horizontal/Vertical offset*	*Single/Twin input*			*P_{in}*	*N_{out}*																		
	1		Horizontal	Twin input gear	Input #1	2	18,4	212	2,82	2,80	3,27	43,87	-	-	54,68		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		3,0	130		1,96			-	-												
1					*Subtotal:*	2			2,82	4,76	3,27	43,87			54,68	2220,23										
													-	-												
													-	-												
					Subtotal:																					
					Total:	2						87,74	-	-	109,36	4440,46										
	Switchboard		*Low voltage/Medium voltage*	*Nr. of incoming fields*	*Nr. of outgoing fields*				**Length=depth*																	
	1		*Low voltage (<1 kV)*	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
1													-	-												
													-	-												
					Total:	2	3,2					17,2	-	-	5,8	457,6										
	Converter		*Chopper/PWM*	*GTO/IGCT/IGBT*			***!!!***		**Length=depth*																	
													-	-												
													-	-												
													-	-												
					Total:								-	-												
	Propeller		*Fixed pitch/Controllable pitch*	*Low/Medium/High loading*			***!!!***			**Width=diameter*																
	1		Controllable pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	13,76	513,68										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	27,52	1027,36										
	Waterjet																									
						14					Total:	286,31			255,94	26,14										
												m^3			ton	M€										

Concept 1

#								
2	Diesel	3	MW	1000	rpm	η_{nom}	0,425	
2	Gasturbine	18,4	MW	5600	rpm	η_{nom}	0,372	
			MW		rpm	η_{nom}		

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	No	η_{nom}	1

4	Dieselgenerator	0,9	Mwe	900	rpm	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
			Mwe		rpm	$\eta_{m,nom}$		$\eta_{e,nom}$	

Speed	Time	Aux. Power	P_D	P_D trailing shaft	Trailing shaft	Geared	Cross-connect gear	Delivered mechanical	Delivered electrical
(kts)	(hrs)	(MWe)	(MW)	(MW)	Y/N	Y/N	Y/N	(MW)	(MWe)
4-6	120	0,800	0,277	0,288	Y	Y	N	0,342	0,800
6-8	180	0,800	0,449	0,470	Y	Y	N	0,558	0,800
8-10	240	0,800	0,755	0,794	Y	Y	N	0,900	0,800
10-12	300	1,350	1,250	1,323	Y	Y	N	1,439	1,350
12-14	510	1,350	1,959	2,082	Y	Y	N	2,217	1,350
14-16	570	1,350	2,936	3,129	N	Y	N	3,088	1,350
16-18	510	1,100	4,238	4,525	N	Y	N	4,424	1,100
18-20	180	1,100	6,114	6,538	Y	Y	N	6,789	1,100
20-22	90	1,100	8,532	9,134	Y	Y	N	9,453	1,100
22-24	90	1,100	11,459	12,277	Y	Y	N	12,680	1,100
24-26	120	1,100	15,410	16,520	Y	Y	N	17,036	1,100
>26	90	1,100	33,854	36,339	N	Y	N	34,823	1,100

Speed	Diesel	Loading per engine	Fuel consumption	Maintenance cost	Gasturbine	Loading per engine	Fuel consumption	Maintenance cost		Loading per engine	Fuel consumption	Maintenance cost
(kts)	(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
4-6	1	0,114	12,0	1691	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	1	0,186	24,3	2536	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	1	0,300	46,0	3382	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	1	0,480	85,9	4227	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	1	0,739	221,0	7186	0	0,000	n/a	0	0	0,000	n/a	n/a
14-16	2	0,515	348,0	16062	0	0,000	n/a	0	0	0,000	n/a	n/a
16-18	2	0,737	441,0	14372	0	0,000	n/a	0	0	0,000	n/a	n/a
18-20	0	0,000	n/a	0	1	0,369	336,9	21528	0	0,000	n/a	n/a
20-22	0	0,000	n/a	0	1	0,514	214,7	10764	0	0,000	n/a	n/a
22-24	0	0,000	n/a	0	1	0,689	271,7	10764	0	0,000	n/a	n/a
24-26	0	0,000	n/a	0	1	0,926	467,1	14352	0	0,000	n/a	n/a
>26	0	0,000	n/a	0	2	0,946	714,4	21528	0	0,000	n/a	n/a

Dieselgenerator	Loading per engine	Fuel consumption	Maintenance cost		Loading per engine	Fuel consumption	Maintenance cost
(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
1	0,889	19,4	872	0	0,000	n/a	n/a
1	0,889	29,1	1308	0	0,000	n/a	n/a
1	0,889	38,9	1744	0	0,000	n/a	n/a
2	0,750	81,7	4360	0	0,000	n/a	n/a
2	0,750	138,8	7412	0	0,000	n/a	n/a
2	0,750	155,1	8284	0	0,000	n/a	n/a
2	0,611	113,7	7412	0	0,000	n/a	n/a
2	0,611	40,1	2616	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	26,7	1744	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a

ANNUAL FUEL BURN:	**3886,9**	ton
Fuel capacity 5000 nm at 18 kts:	**302,4**	ton
	3,56E+05	m^3
ANNUAL FUEL COSTS:	**3,42**	M€
ANNUAL MAINTENANCE COSTS:	**0,17**	M€

Name: **Concept 2/CODOG**

# of different types	Component		Extra info			#	Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	~~80%~~	~~50%~~	~~30%~~	Noise	EM	IR	MTBF (Reliability)	MTTR	Availability
							INPUT		DIMENSIONS						WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	No (propulsion)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	3,0	1000	4,01	2,00	2,82	22,56	0,243	0,228	20,17	1358,55	0,4255	0,4178	0,3820	0,3286				10000	5	0,99950025
2	2	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	4	0,85	900	1,74	1,18	1,58	3,24	0,069	0,065	6,19		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
					Total:	6	9,4					58,1	0,8	0,7	65,1	2717,1										
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
	1	No (propulsion)	Simple cycle			1	36,8	3600	8,92	2,87	3,34	85,55	7,102	6,072	26,18	12806,40	0,4014							4500	8	0,99822538
1																										
					Total:	1	36,8					85,5	7,1	6,1	26,2	12806,4										
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
													-	-												
1													-	-												
													-	-												
					Total:								-	-												
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,6	4	0,9	900	1,41	0,81	1,60	1,82	-	-	2,19	1172,80	0,9715	0,9634	0,9562	0,9423				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	4	3,6					7,3	-	-	8,7	4691,2										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
	1		Horizontal	Single input gear	Input #1	2	3,0	130	1,72	1,96	2,29	7,73	-	-	6,69		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
													-	-												
2					Subtotal:	2			1,72	1,96	2,29	7,73			6,69	437,66										
	2		Horizontal	Twin input gear	Input #1	1	36,8	212	3,64	3,38	3,94	97,03	-	-	109,36		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		36,8	212		3,38			-	-												
					Subtotal:	1			3,64	6,76	3,94	97,03			109,36	3794,23										
					Total:	3						112,49	-	-	122,74	4669,56										
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		Low voltage (<1 kV)	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
1													-	-												
													-	-												
					Total:	2	3,2					17,2	-	-	5,8	457,6										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
													-	-												
1													-	-												
													-	-												
					Total:								-	-												
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!			*Width=diameter																
	1		Controllable pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	13,76	513,68										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	27,52	1027,36										
	Waterjet																									
						14					Total:	280,55			256,10	26,37										

Concept 2

#									
2	Diesel	3	MW	1000	rpm	η_{nom}	0,425		
1	Gasturbine	36,8	MW	5600	rpm	η_{nom}	0,401		
			MW		rpm	η_{nom}			
4	Dieselgenerator	0,9	Mwe	900	rpm	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
			Mwe		rpm	$\eta_{m,nom}$		$\eta_{e,nom}$	

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	Yes	η_{nom}	0,97

Speed (kts)	Time (hrs)	Aux. Power (MWe)	P_D (MW)	P_D trailing shaft (MW)	Trailing shaft Y/N	Geared Y/N	Cross-connect gear Y/N	Delivered mechanical (MW)	Delivered electrical (MWe)
4-6	120	0,800	0,277	0,288	Y	Y	N	0,342	0,800
6-8	180	0,800	0,449	0,470	Y	Y	N	0,558	0,800
8-10	240	0,800	0,755	0,794	Y	Y	N	0,943	0,800
10-12	300	1,350	1,250	1,323	Y	Y	N	1,531	1,350
12-14	510	1,350	1,959	2,082	Y	Y	N	2,303	1,350
14-16	570	1,350	2,936	3,129	N	Y	N	3,166	1,350
16-18	510	1,100	4,238	4,525	N	Y	N	4,499	1,100
18-20	180	1,100	6,114	6,538	N	Y	Y	6,817	1,100
20-22	90	1,100	8,532	9,134	N	Y	Y	9,355	1,100
22-24	90	1,100	11,459	12,277	N	Y	Y	12,430	1,100
24-26	120	1,100	15,410	16,520	N	Y	Y	16,583	1,100
>26	90	1,100	33,854	36,339	N	Y	Y	35,977	1,100

Speed (kts)	Diesel (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Gasturbine (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	1	0,114	12,0	1691	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	1	0,186	24,3	2536	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	1	0,314	47,8	3382	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	1	0,510	90,9	4227	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	1	0,768	229,7	7186	0	0,000	n/a	0	0	0,000	n/a	n/a
14-16	2	0,528	356,2	16062	0	0,000	n/a	0	0	0,000	n/a	n/a
16-18	2	0,750	448,7	14372	0	0,000	n/a	0	0	0,000	n/a	n/a
18-20	0	0,000	n/a	0	1	0,185	410,4	43056	0	0,000	n/a	n/a
20-22	0	0,000	n/a	0	1	0,254	245,1	21528	0	0,000	n/a	n/a
22-24	0	0,000	n/a	0	1	0,338	293,8	21528	0	0,000	n/a	n/a
24-26	0	0,000	n/a	0	1	0,451	480,5	28704	0	0,000	n/a	n/a
>26	0	0,000	n/a	0	1	0,978	681,6	21528	0	0,000	n/a	n/a

Dieselgenerator (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
1	0,889	19,4	872	0	0,000	n/a	n/a
1	0,889	29,1	1308	0	0,000	n/a	n/a
1	0,889	38,9	1744	0	0,000	n/a	n/a
2	0,750	81,7	4360	0	0,000	n/a	n/a
2	0,750	138,8	7412	0	0,000	n/a	n/a
2	0,750	155,1	8284	0	0,000	n/a	n/a
2	0,611	113,7	7412	0	0,000	n/a	n/a
2	0,611	40,1	2616	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	26,7	1744	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a

ANNUAL FUEL BURN:	**4024,5**	ton
Fuel capacity 5000 nm at 18 kts:	**306,5**	ton
	3,61E+05	m^3
ANNUAL FUEL COSTS:	**3,55**	M€
ANNUAL MAINTENANCE COSTS:	**0,23**	M€

Name: Concept 3/CODAD

							INPUT		DIMENSIONS						WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
# of different types	Component		Extra info			#	Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	~~80%~~	~~50%~~	~~30%~~	Noise	EM	IR	MTBF (Reliability)	MTTR	Availability
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	No (propulsion)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	3,0	1000	4,01	2,00	2,82	22,56	0,243	0,228	20,17	1358,55	0,4255	0,4178	0,3820	0,3286				10000	5	0,99950025
3	2	No (propulsion)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	15,40	500	11,79	3,97	5,98	280,16	1,247	1,170	175,36	4581,90	0,4466	0,4390	0,4031	0,3498				10000	5	0,99950025
	3	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	4	0,9	900	1,74	1,18	1,58	3,24	0,069	0,065	6,19		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
					Total:	8	40,2					618,4	3,3	3,1	415,8	11880,9										
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
					Total:																					
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
													-	-												
													-	-												
													-	-												
					Total:								-	-												
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,6	4	0,9	900	1,41	0,81	1,60	1,82	-	-	2,19	1172,80	0,9715	0,9634	0,9562	0,9423				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	4	3,6					7,3	-	-	8,7	4691,2										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
	1		Horizontal	Twin input gear	Input #1	2	15,4	212	2,64	2,67	3,12	36,08	-	-	45,76		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		3,0	212		1,72			-	-												
1					Subtotal:	2			2,64	4,39	3,12	36,08			45,76	1934,81										
													-	-												
													-	-												
					Subtotal:																					
					Total:	2						72,15	-	-	91,53	3869,63										
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		Low voltage (<1 kV)	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
1													-	-												
													-	-												
					Total:	2	3,2					17,2	-	-	5,8	457,6										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
													-	-												
													-	-												
													-	-												
					Total:								-	-												
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!			*Width=diameter																
	1		Controllable pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	13,76	513,68										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	27,52	1027,36										
	Waterjet																									
						14					Total:	714,98			549,41	21,93										
												m^3			ton	M€										

Concept 3

#								
2	Diesel	3	MW	1000	rpm	η_{nom}	0,425	
2	Diesel	15,4	MW	500	rpm	η_{nom}	0,447	
			MW		rpm	η_{nom}		

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	No	η_{nom}	1

4	Dieselgenerator	0,9	Mwe	900	rpm	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
			Mwe		rpm	$\eta_{m,nom}$		$\eta_{e,nom}$	

Speed	Time	Aux. Power	P_D	P_D trailing shaft	Trailing shaft	Geared	Cross-connect gear	Delivered mechanical	Delivered electrical	Diesel	Loading per engine	Fuel consumption	Maintenance cost
(kts)	(hrs)	(MWe)	(MW)	(MW)	Y/N	Y/N	Y/N	(MW)	(MWe)	(-)	(%)	(ton)	(€)
4-6	120	0,800	0,277	0,288	Y	Y	N	0,342	0,800	1	0,114	12,0	1691
6-8	180	0,800	0,449	0,470	Y	Y	N	0,556	0,800	1	0,185	24,2	2536
8-10	240	0,800	0,755	0,794	Y	Y	N	0,885	0,800	1	0,295	45,4	3382
10-12	300	1,350	1,250	1,323	Y	Y	N	1,426	1,350	1	0,475	85,1	4227
12-14	510	1,350	1,959	2,082	Y	Y	N	2,204	1,350	1	0,735	219,7	7186
14-16	570	1,350	2,936	3,129	N	Y	N	3,076	1,350	2	0,513	346,8	16062
16-18	510	1,100	4,238	4,525	N	Y	N	4,412	1,100	2	0,735	439,8	14372
18-20	180	1,100	6,114	6,538	Y	Y	N	6,776	1,100	0	0,000	n/a	0
20-22	90	1,100	8,532	9,134	Y	Y	N	9,441	1,100	0	0,000	n/a	0
22-24	90	1,100	11,459	12,277	Y	Y	N	12,667	1,100	0	0,000	n/a	0
24-26	120	1,100	15,410	16,520	N	Y	N	15,879	1,100	0	0,000	n/a	0
>26	90	1,100	33,854	36,339	N	Y	N	34,812	1,100	2	1,000	107,0	2536

Speed	Diesel	Loading per engine	Fuel consumption	Maintenance cost		Loading per engine	Fuel consumption	Maintenance cost
(kts)	(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
4-6	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	0	0,000	n/a	0	0	0,000	n/a	n/a
14-16	0	0,000	n/a	0	0	0,000	n/a	n/a
16-18	0	0,000	n/a	0	0	0,000	n/a	n/a
18-20	1	0,440	233,1	6236	0	0,000	n/a	n/a
20-22	1	0,613	158,6	3118	0	0,000	n/a	n/a
22-24	1	0,823	212,8	3118	0	0,000	n/a	n/a
24-26	2	0,516	358,9	8315	0	0,000	n/a	n/a
>26	2	0,935	487,2	6236	0	0,000	n/a	n/a

Dieselgenerator	Loading per engine	Fuel consumption	Maintenance cost		Loading per engine	Fuel consumption	Maintenance cost
(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
1	0,889	19,4	872	0	0,000	n/a	n/a
1	0,889	29,1	1308	0	0,000	n/a	n/a
1	0,889	38,9	1744	0	0,000	n/a	n/a
2	0,750	81,7	4360	0	0,000	n/a	n/a
2	0,750	138,8	7412	0	0,000	n/a	n/a
2	0,750	155,1	8284	0	0,000	n/a	n/a
2	0,611	113,7	7412	0	0,000	n/a	n/a
2	0,611	40,1	2616	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a
2	0,611	26,7	1744	0	0,000	n/a	n/a
2	0,611	20,1	1308	0	0,000	n/a	n/a

ANNUAL FUEL BURN:	**3434,3**	ton
Fuel capacity 5000 nm at 18 kts:	**301,7**	ton
	3,55E+05	m^3
ANNUAL FUEL COSTS:	**3,03**	M€
ANNUAL MAINTENANCE COSTS:	**0,12**	M€

Name: **Concept 4/CODLAG**

# of different types	Component		Extra info			# (-)	INPUT Power (MW)	Speed (rpm)	DIMENSIONS Length (m)	Width (m)	Height (m)	Volume (m^3)	Inlet (m^2)	Outlet (m^2)	WEIGHT Weight (ton)	COST IPC (k€)	EFFICIENCY @ PARTLOAD Nominal eff (-)	80% (-)	50% (-)	30% (-)	SIGNATURES Noise (+/-)	EM (+/-)	IR (+/-)	RELIABILITY MTBF (Reliability) (hour)	MTTR (hour)	Availability (-)
	Diesel engine	*For generator?*	*High/medium/slow speed*	*Vee/Line*	*2-stroke/4-stroke*																					
	1	*Yes (generator)*	Medium speed (300-1000 rpm)	Vee	4-stroke	2	1,1	900	2,07	1,31	1,78	4,81	0,089	0,084	8,01		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
2	*2*	*Yes (generator)*	Medium speed (300-1000 rpm)	Line	4-stroke	2	4,10	720	5,89	2,63	3,25	50,35	0,332	0,312	35,39		0,4353	0,4277	0,3919	0,3385				10000	5	0,99950025
					Total:	4	10,4					110,3	0,8	0,8	86,8											
	Gasturbine	*For generator?*	*Simple cycle/ICR cycle*																							
	1	*No (propulsion)*	Simple cycle			2	15,4	5600	6,99	2,45	3,06	52,52	2,972	2,541	18,32	5359,20	0,3647							4500	8	0,99822538
1																										
					Total:	2	30,8					105,0	5,9	5,1	36,6	10718,4										
	Electric motor		*Asynchronous/Synchronous*	*AC/DC/PM/HTS*	***L/D ratio***																					
	1		Synchronous	Conventional AC	1,6	2	3,0	130	3,70	2,31	4,54	38,87	-	-	46,64	3126,24	0,9729	0,9648	0,9576	0,9436				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	2	6,0					77,7	-	-	93,3	6252,5										
	Generator		*Diesel/Gasturbine*	*AC/DC/PM/HTS*	***L/D ratio***																					
	1		Diesel driven	Conventional AC	1,2	2	4,4	720	2,12	1,63	3,21	11,13	-	-	13,36	2969,07	0,9732	0,9651	0,9579	0,9439				201480	120	0,99940476
2	*2*		Diesel driven	Conventional AC	1,5	2	1,2	900	1,48	0,91	1,79	2,43	-	-	2,91	1387,88	0,9719	0,9638	0,9566	0,9427				201480	120	0,99940476
													-	-												
					Total:	4	11,2					27,1	-	-	32,5	8713,9										
	Gearbox		*Horizontal/Vertical offset*	*Single/Twin input*			*P_{in}*	*N_{out}*																		
	1		Horizontal	Single input gear	Input #1	2	15,4	225	2,58	2,63	3,07	20,80	-	-	19,85		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
													-	-												
1					*Subtotal:*	2			2,58	2,63	3,07	20,80			19,85	1014,30										
													-	-												
													-	-												
					Subtotal:																					
					Total:	2						41,60	-	-	39,70	2028,60										
	Switchboard		*Low voltage/Medium voltage*	*Nr. of incoming fields*	*Nr. of outgoing fields*				**Length=depth*																	
	1		*High voltage (>1 kV)*	5	4	2	7,6		1,70	5,85	2,60	25,86	-	-	5,00	1086,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
2	*2*		*Low voltage (<1 kV)*	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
													-	-												
					Total:	4	18,4					68,9	-	-	15,8	2631,2										
	Converter		*Chopper/PWM*	*GTO/IGCT/IGBT*			***!!!***		**Length=depth*																	
	1		AC PWM-converter	GTO/IGCT		2	3,3		0,90	4,75	2,30	9,81	-	-	3,98	435,60	0,9812	0,9420	0,9457	0,9430						#DEEL/0!
1													-	-												
													-	-												
					Total:	2	6,6					19,6	-	-	8,0	871,2										
	Propeller		*Fixed pitch/Controllable pitch*	*Low/Medium/High loading*			***!!!***			**Width=diameter*																
	1		Fixed pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	10,72	410,89										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	21,43	821,78										
	Waterjet																									
						18					Total:	***450,31***			***334,14***	***32,04***										
												m^3			ton	M€										

Concept 4

#							
2	Electric motor	3	MW	130	rpm	η_{nom}	0,973
2	Gasturbine	15,4	MW	5600	rpm	η_{nom}	0,365
			MW		rpm	η_{nom}	

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	No	η_{nom}	1

2	Dieselgenerator	1	Mwe	900	rpm	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
2	Dieselgenerator	3,9	Mwe	720	rpm	$\eta_{m,nom}$	0,435	$\eta_{e,nom}$	0,973

Speed	Time	Aux. Power	P_D	P_D trailing shaft	Trailing shaft	Geared	Cross-connect gear	Delivered mechanical	Delivered electrical	Electric motor	Loading per engine	Fuel consumption	Maintenance cost	Gasturbine	Loading per engine	Fuel consumption	Maintenance cost		Loading per engine	Fuel consumption	Maintenance cost
(kts)	(hrs)	(MWe)	(MW)	(MW)	Y/N	Y/N	Y/N	(MW)	(MWe)	(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
4-6	120	0,800	0,277	0,288	N	N	N	0,000	1,188	2	0,047	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	180	0,800	0,449	0,470	N	N	N	0,000	1,349	2	0,076	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	240	0,800	0,755	0,794	N	N	N	0,000	1,655	2	0,127	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	300	1,350	1,250	1,323	N	N	N	0,000	2,711	2	0,210	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	510	1,350	1,959	2,082	N	N	N	0,000	3,440	2	0,330	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
14-16	570	1,350	2,936	3,129	N	N	N	0,000	4,448	2	0,494	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
16-18	510	1,100	4,238	4,525	N	N	N	0,000	5,543	2	0,713	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
18-20	180	1,100	6,114	6,538	Y	Y	N	6,776	1,100	0	0,000	n/a	n/a	1	0,440	326,1	18018	0	0,000	n/a	n/a
20-22	90	1,100	8,532	9,134	Y	Y	N	9,441	1,100	0	0,000	n/a	n/a	1	0,613	210,7	9009	0	0,000	n/a	n/a
22-24	90	1,100	11,459	12,277	Y	Y	N	12,667	1,100	0	0,000	n/a	n/a	1	0,823	269,5	9009	0	0,000	n/a	n/a
24-26	120	1,100	15,410	16,520	N	Y	N	15,879	1,100	0	0,000	n/a	n/a	2	0,516	490,0	24024	0	0,000	n/a	n/a
>26	90	1,100	33,854	36,339	N	Y	N	28,642	7,304	2	1,000	n/a	n/a	2	0,935	603,5	18018	0	0,000	n/a	n/a

Dieselgenerator	Loading per engine	Fuel consumption	Maintenance cost	Dieselgenerator	Loading per engine	Fuel consumption	Maintenance cost
(-)	(%)	(ton)	(€)	(-)	(%)	(ton)	(€)
2	0,594	28,9	1848	0	0,000	n/a	0
2	0,675	49,0	2772	0	0,000	n/a	0
2	0,827	80,2	3696	0	0,000	n/a	0
0	0,000	n/a	0	1	0,695	163,8	4883
0	0,000	n/a	0	1	0,882	354,3	8301
1	1,000	116,1	4389	1	0,884	396,9	9278
2	1,000	207,7	7854	1	0,909	365,4	8301
2	0,550	40,4	2772	0	0,000	n/a	0
2	0,550	20,2	1386	0	0,000	n/a	0
2	0,550	20,2	1386	0	0,000	n/a	0
2	0,550	26,9	1848	0	0,000	n/a	0
0	0,000	n/a	0	2	0,936	133,1	2930

ANNUAL FUEL BURN: **3903,1** ton

Fuel capacity 5000 nm at 18 kts: **312,4** ton

3,68E+05 m^3

ANNUAL FUEL COSTS: **3,44** M€

ANNUAL MAINTENANCE COSTS: **0,14** M€

Name: **Concept 5/CODLAG**

							INPUT		DIMENSIONS						WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
# of different types	Component		Extra info			#	Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	~~80%~~	~~50%~~	~~30%~~	Noise	EM	IR	MTBF (Reliability)	MTTR	Availability
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	1,1	900	2,07	1,31	1,78	4,81	0,089	0,084	8,01		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
2	2	Yes (generator)	Medium speed (300-1000 rpm)	Line	4-stroke	2	4,10	720	5,89	2,63	3,25	50,35	0,332	0,312	35,39		0,4353	0,4277	0,3919	0,3385				10000	5	0,99950025
					Total:	4	10,4					110,3	0,8	0,8	86,8											
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
	1	No (propulsion)	Simple cycle			1	30,8	3600	8,49	2,78	3,28	77,43	5,944	5,082	24,34	10718,40	0,3936							4500	8	0,99822538
1																										
					Total:	1	30,8					77,4	5,9	5,1	24,3	10718,4										
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
	1		Synchronous	Conventional AC	1,6	2	3,0	130	3,70	2,31	4,54	38,87	-	-	46,64	3126,24	0,9729	0,9648	0,9576	0,9436				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	2	6,0					77,7	-	-	93,3	6252,5										
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,2	2	4,4	720	2,12	1,63	3,21	11,13	-	-	13,36	2969,07	0,9732	0,9651	0,9579	0,9439				201480	120	0,99940476
2	2		Diesel driven	Conventional AC	1,5	2	1,2	900	1,48	0,91	1,79	2,43	-	-	2,91	1387,88	0,9719	0,9638	0,9566	0,9427				201480	120	0,99940476
													-	-												
					Total:	4	11,2					27,1	-	-	32,5	8713,9										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
	1		Horizontal	Twin input gear	Input #1	1	30,8	225	3,33	3,17	3,70	78,17	-	-	86,24		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		30,8	225		3,17			-	-												
1					Subtotal:	1			3,33	6,34	3,70	78,17			86,24	3157,79										
													-	-												
													-	-												
					Subtotal:																					
					Total:	1						78,17	-	-	86,24	3157,79										
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		High voltage (>1 kV)	5	4	2	7,6		1,70	5,85	2,60	25,86	-	-	5,00	1086,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
2	2		Low voltage (<1 kV)	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
													-	-												
					Total:	4	18,4					68,9	-	-	15,8	2631,2										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
	1		AC PWM-converter	GTO/IGCT		2	3,3		0,90	4,75	2,30	9,81	-	-	3,98	435,60	0,9812	0,9420	0,9457	0,9430						#DEEL/0!
1													-	-												
													-	-												
					Total:	2	6,6					19,6	-	-	8,0	871,2										
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!		*Width=diameter																	
	1		Fixed pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	10,72	410,89										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	21,43	821,78										
	Waterjet																									
						16					Total:	**459,27**			**368,38**	**33,17**										
												m^3			ton	M€										

Concept 5

#							
2	Electric motor	3	*MW*	130	*rpm*	η_{nom}	0,973
1	Gasturbine	30,8	*MW*	3600	*rpm*	η_{nom}	0,394
			MW		*rpm*	η_{nom}	

Gearbox?	No	η_{nom}	1
Cross-connect gearbox?	Yes	η_{nom}	0,97

2	Dieselgenerator	1	*Mwe*	900	*rpm*	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
2	Dieselgenerator	3,9	*Mwe*	720	*rpm*	$\eta_{m,nom}$	0,435	$\eta_{e,nom}$	0,973

Speed *(kts)*	Time *(hrs)*	Aux. Power *(MWe)*	P_D *(MW)*	P_D trailing shaft *(MW)*	Trailing shaft *Y/N*	Geared *Y/N*	Cross-connect gear *Y/N*	Delivered mechanical *(MW)*	Delivered electrical *(MWe)*
4-6	120	*0,800*	*0,277*	*0,288*	N	N	N	0,000	1,188
6-8	180	*0,800*	*0,449*	*0,470*	N	N	N	0,000	1,349
8-10	240	*0,800*	*0,755*	*0,794*	N	N	N	0,000	1,655
10-12	300	*1,350*	*1,250*	*1,323*	N	N	N	0,000	2,711
12-14	510	*1,350*	*1,959*	*2,082*	N	N	N	0,000	3,440
14-16	570	*1,350*	*2,936*	*3,129*	N	N	N	0,000	4,448
16-18	510	*1,100*	*4,238*	*4,525*	N	N	N	0,000	5,543
18-20	180	*1,100*	*6,114*	*6,538*	Y	N	Y	6,968	1,100
20-22	90	*1,100*	*8,532*	*9,134*	Y	N	Y	9,653	1,100
22-24	90	*1,100*	*11,459*	*12,277*	Y	N	Y	12,905	1,100
24-26	120	*1,100*	*15,410*	*16,520*	N	N	Y	16,133	1,100
>26	90	*1,100*	*33,854*	*36,339*	N	N	Y	28,978	7,304

Speed *(kts)*	Electric motor *(-)*	Loading per engine *(%)*	Fuel consumption *(ton)*	Maintenance cost *(€)*	Gasturbine *(-)*	Loading per engine *(%)*	Fuel consumption *(ton)*	Maintenance cost *(€)*	*(-)*	Loading per engine *(%)*	Fuel consumption *(ton)*	Maintenance cost *(€)*
4-6	2	0,047	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	2	0,076	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	2	0,127	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	2	0,210	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	2	0,330	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
14-16	2	0,494	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
16-18	2	0,713	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
18-20	0	0,000	n/a	n/a	1	0,226	390,7	36036	0	0,000	n/a	n/a
20-22	0	0,000	n/a	n/a	1	0,313	238,6	18018	0	0,000	n/a	n/a
22-24	0	0,000	n/a	n/a	1	0,419	291,6	18018	0	0,000	n/a	n/a
24-26	0	0,000	n/a	n/a	1	0,524	459,7	24024	0	0,000	n/a	n/a
>26	2	1,000	n/a	n/a	1	0,949	566,3	18018	0	0,000	n/a	n/a

Speed *(kts)*	Dieselgenerator *(-)*	Loading per engine *(%)*	Fuel consumption *(ton)*	Maintenance cost *(€)*	Dieselgenerator *(-)*	Loading per engine *(%)*	Fuel consumption *(ton)*	Maintenance cost *(€)*
4-6	2	0,594	28,9	1848	0	0,000	n/a	0
6-8	2	0,675	49,0	2772	0	0,000	n/a	0
8-10	2	0,827	80,2	3696	0	0,000	n/a	0
10-12	0	0,000	n/a	0	1	0,695	163,8	4883
12-14	0	0,000	n/a	0	1	0,882	354,3	8301
14-16	1	1,000	116,1	4389	1	0,884	396,9	9278
16-18	2	1,000	207,7	7854	1	0,909	365,4	8301
18-20	2	0,550	40,4	2772	0	0,000	n/a	0
20-22	2	0,550	20,2	1386	0	0,000	n/a	0
22-24	2	0,550	20,2	1386	0	0,000	n/a	0
24-26	2	0,550	26,9	1848	0	0,000	n/a	0
>26	0	0,000	n/a	0	2	0,936	133,1	2930

ANNUAL FUEL BURN:	**3950,0**	ton
Fuel capacity 5000 nm at 18 kts:	**312,4**	ton
	3,68E+05	m^3
ANNUAL FUEL COSTS:	**3,48**	M€
ANNUAL MAINTENANCE COSTS:	**0,18**	M€

Name: Concept 6/CODLADAD

							INPUT				DIMENSIONS				WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
# of different types	Component		Extra info			#	Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	80%	50%	30%	Noise	EM	IR	MTBF (Reliability)	MTTR	Availability
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	No (propulsion)	Medium speed (300-1000 rpm)	Line	4-stroke	4	8,8	700	8,56	3,25	4,62	128,81	0,713	0,669	77,59	3022,87	0,4362	0,4286	0,3927	0,3394				10000	5	0,99950025
3	2	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	1,80	900	2,86	1,61	2,23	10,27	0,146	0,137	13,11		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
	3	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	1	1,1	900	2,00	1,29	1,74	4,48	0,085	0,080	7,65		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
					Total:	7	39,9					540,3	3,2	3,0	344,2	12091,5										
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
0																										
					Total:																					
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
	1		Synchronous	Conventional AC	1,6	2	0,9	90	2,75	1,72	3,37	15,91	-	-	19,09	1024,92	0,9714	0,9633	0,9561	0,9422				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	2	1,7					31,8	-	-	38,2	2049,8										
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,3	2	1,9	900	1,58	1,12	2,21	3,93	-	-	4,71	1838,47	0,9725	0,9644	0,9572	0,9432				201480	120	0,99940476
2	2		Diesel driven	Conventional AC	1,5	1	1,1	900	1,45	0,90	1,76	2,29	-	-	2,74	1339,90	0,9718	0,9638	0,9565	0,9426				201480	120	0,99940476
													-	-												
					Total:	3	5,0					10,1	-	-	12,2	5016,8										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
	1		Horizontal	Twin input gear	Input #1	2	8,8	212	2,14	2,30	2,68	26,39	-	-	26,15		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		8,8	212		2,30			-	-												
1					Subtotal:	2			2,14	4,59	2,68	26,39			26,15	1255,30										
													-	-												
													-	-												
					Subtotal:																					
					Total:	2						52,78	-	-	52,30	2510,59										
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		Low voltage (<1 kV)	7	8	2	3,4		1,00	5,85	2,20	12,87	-	-	4,40	486,20	0,9950	0,9950	0,9950	0,9950				∞	0	1
1													-	-												
													-	-												
					Total:	2	6,8					25,7	-	-	8,8	972,4										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
	1		AC PWM-converter	GTO/IGCT		2	1,0		0,76	4,69	2,30	8,20	-	-	2,25	132,00	0,9740	0,9757	0,9773	0,9761						#DEEL/0!
1													-	-												
													-	-												
					Total:	2	2,0					16,4	-	-	4,5	264,0										
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!			*Width=diameter																
	1		Controllable pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	13,76	513,68										#DEEL/0!
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	27,52	1027,36										
	Waterjet																									
						17					Total:	677,15			487,71	23,93										#DEEL/0!
												m^3			ton	M€										

Concept 6

#								
2	Electric motor	0,85	*MW*	90	*rpm*	η_{nom}	0,971	
4	Diesel	8,8	*MW*	700	*rpm*	η_{nom}	0,436	
			MW		*rpm*	η_{nom}		

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	No	η_{nom}	1

2	Dieselgenerator	1,7	*Mwe*	900	*rpm*	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
1	Dieselgenerator	1	*Mwe*	900	*rpm*	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972

Speed (kts)	Time (hrs)	Aux. Power (MWe)	P_D (MW)	P_D trailing shaft (MW)	Trailing shaft Y/N	Geared Y/N	Cross-connect gear Y/N	Delivered mechanical (MW)	Delivered electrical (MWe)
4-6	120	*0,800*	*0,277*	*0,288*	N	N	N	0,000	1,107
6-8	180	*0,800*	*0,449*	*0,470*	N	N	N	0,000	1,284
8-10	240	*0,800*	*0,755*	*0,794*	N	N	N	0,000	1,599
10-12	300	*1,350*	*1,250*	*1,323*	N	N	N	0,000	2,662
12-14	510	*1,350*	*1,959*	*2,082*	Y	Y	N	3,063	0,483
14-16	570	*1,350*	*2,936*	*3,129*	Y	Y	N	4,132	0,483
16-18	510	*1,100*	*4,238*	*4,525*	Y	Y	N	5,561	0,241
18-20	180	*1,100*	*6,114*	*6,538*	Y	Y	N	7,626	0,241
20-22	90	*1,100*	*8,532*	*9,134*	N	Y	N	9,926	0,000
22-24	90	*1,100*	*11,459*	*12,277*	N	Y	N	12,930	0,000
24-26	120	*1,100*	*15,410*	*16,520*	N	Y	N	16,985	0,000
>26	90	*1,100*	*33,854*	*36,339*	N	Y	N	33,039	2,861

Speed (kts)	Electric motor (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Diesel (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	2	0,165	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
6-8	2	0,267	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
8-10	2	0,448	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
10-12	2	0,743	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	n/a
12-14	-1	1,000	n/a	n/a	1	0,344	312,4	12988	0	0,000	n/a	n/a
14-16	-1	1,000	n/a	n/a	1	0,466	454,4	14515	0	0,000	n/a	n/a
16-18	-1	1,000	n/a	n/a	1	0,629	538,7	12988	0	0,000	n/a	n/a
18-20	-1	1,000	n/a	n/a	1	0,863	261,9	4584	0	0,000	n/a	n/a
20-22	-2	0,647	n/a	n/a	2	0,562	170,8	4584	0	0,000	n/a	n/a
22-24	-2	0,647	n/a	n/a	2	0,733	221,3	4584	0	0,000	n/a	n/a
24-26	-2	0,647	n/a	n/a	2	0,963	392,1	6112	0	0,000	n/a	n/a
>26	2	1,000	n/a	n/a	4	0,940	572,9	9168	0	0,000	n/a	n/a

Speed (kts)	Dieselgenerator (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Dieselgenerator (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	1	0,651	26,8	1237	0	0,000	n/a	0
6-8	1	0,755	46,6	1856	0	0,000	n/a	0
8-10	1	0,941	77,8	2474	0	0,000	n/a	0
10-12	2	0,783	160,9	6186	0	0,000	n/a	0
12-14	0	0,000	n/a	0	1	0,483	50,8	3927
14-16	0	0,000	n/a	0	1	0,483	56,8	4389
16-18	0	0,000	n/a	0	1	0,241	29,3	3927
18-20	0	0,000	n/a	0	1	0,241	10,3	1386
20-22	0	0,000	n/a	0	0	0,000	n/a	0
22-24	0	0,000	n/a	0	0	0,000	n/a	0
24-26	0	0,000	n/a	0	0	0,000	n/a	0
>26	2	0,841	51,9	1856	0	0,000	n/a	0

Generator mode

ANNUAL FUEL BURN:	**3435,8**	ton
Fuel capacity 5000 nm at 18 kts:	**309,6**	ton
	3,64E+05	m^3
ANNUAL FUEL COSTS:	**3,03**	M€
ANNUAL MAINTENANCE COSTS:	**0,10**	M€

Name: **Concept 7/CODLADOG**

							INPUT		DIMENSIONS						WEIGHT	COST	EFFICIENCY @ PARTLOAD				SIGNATURES			RELIABILITY		
# of different types	Component		Extra info			#	Power	Speed	Length	Width	Height	Volume	Inlet	Outlet	Weight	IPC	Nominal eff	80%	50%	30%	Noise	EM	IR	MTBF (Reliability)	MTTR	Availability
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	No (propulsion)	Medium speed (300-1000 rpm)	Vee	4-stroke	2	2,1	1000	3,17	1,72	2,39	13,03	0,170	0,160	14,12	1042,21	0,4255	0,4178	0,3820	0,3286				10000	5	0,99950025
2	2	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	4	1,21	900	2,20	1,37	1,86	5,57	0,098	0,092	8,81		0,4286	0,4210	0,3851	0,3318				10000	5	0,99950025
					Total:	6	9,0					48,3	0,7	0,7	63,5	2084,4										
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
	1	No (propulsion)	Simple cycle			2	17,5	5600	7,24	2,51	3,10	56,42	3,378	2,888	19,30	6090,00	0,3699							4500	8	0,99822538
1																										
					Total:	2	35,0					112,8	6,8	5,8	38,6	12180,0										
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
	1		Synchronous	Conventional AC	1,6	2	0,9	90	2,80	1,75	3,44	16,84	-	-	20,21	1078,05	0,9715	0,9634	0,9562	0,9423				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	2	1,8					33,7	-	-	40,4	2156,1										
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,4	4	1,3	900	1,45	0,96	1,89	2,63	-	-	3,16	1454,45	0,9720	0,9640	0,9567	0,9428				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	4	5,2					10,5	-	-	12,6	5817,8										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
	1		Horizontal	Twin input gear	Input #1	2	2,1	130	1,51	1,78	2,08	14,27	-	-	52,00		0,9800	0,9790	0,9760	0,9707				100000	5	0,99995
					Input #2		17,5	212		2,77			-	-												
1					Subtotal:	2			1,51	4,55	2,08	14,27			52,00	2135,80										
													-	-												
													-	-												
					Subtotal:																					
					Total:	2						28,55	-	-	104,01	4271,59										
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		Low voltage (<1 kV)	6	10	2	1,6		1,00	5,53	2,20	12,16	-	-	4,13	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
1													-	-												
													-	-												
					Total:	2	3,2					24,3	-	-	8,3	457,6										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
	1		AC PWM-converter	GTO/IGCT		2	1,1		0,77	4,69	2,30	8,27	-	-	2,33	145,20	0,9748	0,9740	0,9757	0,9745						#DEEL/0!
1													-	-												
													-	-												
					Total:	2	2,2					16,5	-	-	4,7	290,4										
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!			*Width=diameter																
	1		Controllable pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	13,76	513,68										
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	27,52	1027,36										
	Waterjet																									
						18					Total:	274,79			299,58	28,29										
												m^3			ton	M€										

Concept 7

#								
2	Electric motor	0,9	MW	90	rpm	η_{nom}	0,972	
2	Diesel	2,1	MW	1000	rpm	η_{nom}	0,425	
2	Gasturbine	17,5	MW	5600	rpm	η_{nom}	0,370	

Gearbox?	Yes	η_{nom}	0,98
Cross-connect gearbox?	No	η_{nom}	1

4	Dieselgenerator	1,15	Mwe	900	rpm	$\eta_{m,nom}$	0,429	$\eta_{e,nom}$	0,972
			Mwe		rpm	$\eta_{m,nom}$		$\eta_{e,nom}$	

Speed (kts)	Time (hrs)	Aux. Power (MWe)	P_D (MW)	P_D trailing shaft (MW)	Trailing shaft Y/N	Geared Y/N	Cross-connect gear Y/N	Delivered mechanical (MW)	Delivered electrical (MWe)
4-6	120	0,800	0,277	0,288	N	N	N	0,000	1,109
6-8	180	0,800	0,449	0,470	N	N	N	0,000	1,285
8-10	240	0,800	0,755	0,794	N	N	N	0,000	1,600
10-12	300	1,350	1,250	1,323	N	N	N	0,000	2,663
12-14	510	1,350	1,959	2,082	Y	Y	N	1,257	2,282
14-16	570	1,350	2,936	3,129	N	Y	N	3,084	1,350
16-18	510	1,100	4,238	4,525	N	Y	N	2,543	2,964
18-20	180	1,100	6,114	6,538	Y	Y	N	6,785	1,100
20-22	90	1,100	8,532	9,134	Y	Y	N	9,450	1,100
22-24	90	1,100	11,459	12,277	Y	Y	N	12,676	1,100
24-26	120	1,100	15,410	16,520	Y	Y	N	17,032	1,100
>26	90	1,100	33,854	36,339	N	Y	N	32,969	2,964

Speed (kts)	Electric motor (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Diesel (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Gasturbine (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	2	0,156	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	0
6-8	2	0,252	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	0
8-10	2	0,424	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	0
10-12	2	0,701	n/a	n/a	0	0,000	n/a	0	0	0,000	n/a	0
12-14	1	1,000	n/a	n/a	1	0,625	131,1	5906	0	0,000	n/a	0
14-16	0	0,000	n/a	n/a	2	0,734	343,7	13201	0	0,000	n/a	0
16-18	2	1,000	n/a	n/a	2	0,624	261,6	11812	0	0,000	n/a	0
18-20	0	0,000	n/a	n/a	0	0,000	n/a	0	1	0,388	333,6	20475
20-22	0	0,000	n/a	n/a	0	0,000	n/a	0	1	0,540	213,4	10238
22-24	0	0,000	n/a	n/a	0	0,000	n/a	0	1	0,724	270,9	10238
24-26	0	0,000	n/a	n/a	0	0,000	n/a	0	1	0,973	467,1	13650
>26	2	1,000	n/a	n/a	0	0,000	n/a	0	2	0,943	681,4	20475

Speed (kts)	Dieselgenerator (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	1	0,964	27,0	998	0	0,000	n/a	n/a
6-8	2	0,559	47,1	2993	0	0,000	n/a	n/a
8-10	2	0,696	77,5	3991	0	0,000	n/a	n/a
10-12	3	0,772	161,0	7484	0	0,000	n/a	n/a
12-14	2	0,992	236,9	8482	0	0,000	n/a	n/a
14-16	2	0,587	156,2	9479	0	0,000	n/a	n/a
16-18	3	0,859	305,4	12722	0	0,000	n/a	n/a
18-20	1	0,957	40,2	1497	0	0,000	n/a	n/a
20-22	1	0,957	20,1	748	0	0,000	n/a	n/a
22-24	1	0,957	20,1	748	0	0,000	n/a	n/a
24-26	1	0,957	26,8	998	0	0,000	n/a	n/a
>26	3	0,859	53,9	2245	0	0,000	n/a	n/a

ANNUAL FUEL BURN:	3875,0	ton
Fuel capacity 5000 nm at 18 kts:	309,1	ton
	3,64E+05	m^3
ANNUAL FUEL COSTS:	3,41	M€
ANNUAL MAINTENANCE COSTS:	0,16	M€

Name: **Concept 8/IFEP**

# of different types	Component		Extra info			#	INPUT Power	Speed	DIMENSIONS Length	Width	Height	Volume	Inlet	Outlet	WEIGHT Weight	COST IPC	EFFICIENCY @ PARTLOAD Nominal eff	80%	50%	15%	SIGNATURES Noise	EM	IR	RELIABILITY MTBF (Reliability)	MTTR	Availability
						(-)	(MW)	(rpm)	(m)	(m)	(m)	(m^3)	(m^2)	(m^2)	(ton)	(k€)	(-)	(-)	(-)	(-)	(+/-)	(+/-)	(+/-)	(hour)	(hour)	(-)
	Diesel engine	For generator?	High/medium/slow speed	Vee/Line	2-stroke/4-stroke																					
	1	Yes (generator)	Medium speed (300-1000 rpm)	Line	4-stroke	3	4,1	720	5,89	2,63	3,25	50,35	0,332	0,312	35,39		0,4353	0,4277	0,3919	0,2867				10000	5	0,99950025
2	2	Yes (generator)	Medium speed (300-1000 rpm)	Vee	4-stroke	1	0,85	900	1,74	1,18	1,58	3,24	0,069	0,065	6,19		0,4286	0,4210	0,3851	0,2800				10000	5	0,99950025
					Total:	4	13,2					154,3	1,1	1,0	112,3											
	Gasturbine	For generator?	Simple cycle/ICR cycle																							
	1	Yes (generator)	Simple cycle			1	29,0	3600	8,34	2,75	3,26	74,86	5,597	4,785	23,74		0,3910							4500	8	0,99822538
1																										
					Total:	1	29,0					74,9	5,6	4,8	23,7											
	Electric motor		Asynchronous/Synchronous	AC/DC/PM/HTS	**L/D ratio**																					
	1		Synchronous	Conventional AC	1,6	2	18,4	212	5,75	3,60	7,06	146,17	-	-	175,41	15544,68	0,9740	0,9659	0,9586	0,9068				201480	120	0,99940476
1													-	-												
													-	-												
					Total:	2	36,8					292,3	-	-	350,8	31089,4										
	Generator		Diesel/Gasturbine	AC/DC/PM/HTS	**L/D ratio**																					
	1		Diesel driven	Conventional AC	1,2	3	4,4	720	2,12	1,63	3,21	11,13	-	-	13,36	2969,07	0,9732	0,9651	0,9579	0,9061				201480	120	0,99940476
3	2		Diesel driven	Conventional AC	1,7	1	0,9	900	1,47	0,80	1,57	1,86	-	-	2,23	1187,99	0,9715	0,9635	0,9562	0,9046				201480	120	0,99940476
	3		Gasturbine driven	Conventional AC	1,6	1	31,2	3600	2,89	1,67	3,28	15,79	-	-	18,95	14196,00	0,9741	0,9660	0,9588	0,9070				201480	120	0,99940476
					Total:	5	45,3					51,0	-	-	61,3	24291,2										
	Gearbox		Horizontal/Vertical offset	Single/Twin input			P_{in}	N_{out}																		
													-	-												
													-	-												
0					Subtotal:																					
													-	-												
													-	-												
					Subtotal:																					
					Total:								-	-												
	Switchboard		Low voltage/Medium voltage	Nr. of incoming fields	Nr. of outgoing fields				*Length=depth																	
	1		High voltage (>1 kV)	6	4	2	39,5		1,70	6,50	2,60	28,73	-	-	6,00	5648,50	0,9950	0,9950	0,9950	0,9950				∞	0	1
2	2		Low voltage (<1 kV)	4	8	2	1,6		1,00	3,90	2,20	8,58	-	-	2,90	228,80	0,9950	0,9950	0,9950	0,9950				∞	0	1
													-	-												
					Total:	4	82,2					74,6	-	-	17,8	11754,6										
	Converter		Chopper/PWM	GTO/IGCT/IGBT			!!!		*Length=depth																	
	1		AC PWM-converter	GTO/IGCT		2	20,0		1,90	4,92	2,30	21,50	-	-	16,50	2640,00	0,9865	0,7468	0,7630	0,6793						#DEEL/0!
1													-	-												
													-	-												
					Total:	2	40,0					43,0	-	-	33,0	5280,0										
	Propeller		Fixed pitch/Controllable pitch	Low/Medium/High loading			!!!		*Width=diameter																	
	1		Fixed pitch	Medium loading (1 MW/m2)		2	36,8		-	4,50	-	-	-	-	10,72	410,89										#DEEL/0!
1									-		-	-	-	-												
									-		-	-	-	-												
					Total:	2	36,8					-	-	-	21,43	821,78										
	Waterjet																									
0																										
						15					Total:	***690,16***			***620,40***	***73,24***										#DEEL/0!
												m^3			ton	M€										

Concept 8

#							
2	Electric motor	18,4	MW	212	rpm	η_{nom}	0,974
			MW		rpm	η_{nom}	
			MW		rpm	η_{nom}	

Gearbox?	No	η_{nom}	1
Cross-connect gearbox?	No	η_{nom}	1

3	Dieselgenerator	3,9	Mwe	720	rpm	$\eta_{m,nom}$	0,435	$\eta_{e,nom}$	0,973
1	Gasturbine genera	27,8	Mwe	3600	rpm	$\eta_{m,nom}$	0,389	$\eta_{e,nom}$	0,974

Speed (kts)	Time (hrs)	Aux. Power (MWe)	P_D (MW)	P_D trailing shaft (MW)	Trailing shaft Y/N	Geared Y/N	Cross-connect gear Y/N	Delivered mechanical (MW)	Delivered electrical (MWe)
4-6	120	0,800	0,277	0,288	N	N	N	0,000	1,601
6-8	180	0,800	0,449	0,470	N	N	N	0,000	2,097
8-10	240	0,800	0,755	0,794	N	N	N	0,000	2,868
10-12	300	1,350	1,250	1,323	N	N	N	0,000	3,377
12-14	510	1,350	1,959	2,082	N	N	N	0,000	3,956
14-16	570	1,350	2,936	3,129	N	N	N	0,000	4,892
16-18	510	1,100	4,238	4,525	N	N	N	0,000	5,943
18-20	180	1,100	6,114	6,538	N	N	N	0,000	7,849
20-22	90	1,100	8,532	9,134	N	N	N	0,000	10,324
22-24	90	1,100	11,459	12,277	N	N	N	0,000	13,332
24-26	120	1,100	15,410	16,520	N	N	N	0,000	17,401
>26	90	1,100	33,854	36,339	N	N	N	0,000	36,447

Speed (kts)	Electric motor (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	(-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
4-6	2	0,008	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
6-8	2	0,012	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
8-10	2	0,021	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
10-12	2	0,034	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
12-14	2	0,054	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
14-16	2	0,081	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
16-18	2	0,116	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
18-20	2	0,168	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
20-22	2	0,234	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
22-24	2	0,315	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
24-26	2	0,423	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a
>26	2	0,929	n/a	n/a	0	0,000	n/a	n/a	0	0,000	n/a	n/a

Dieselgenerator (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)	Gasturbine generato (-)	Loading per engine (%)	Fuel consumption (ton)	Maintenance cost (€)
1	0,410	39,8	1953	0	0,000	n/a	0
1	0,538	75,8	2930	0	0,000	n/a	0
1	0,735	136,4	3906	0	0,000	n/a	0
1	0,866	201,3	4883	0	0,000	n/a	0
2	0,507	407,3	16603	0	0,000	n/a	0
2	0,627	554,7	18556	0	0,000	n/a	0
2	0,762	600,6	16603	0	0,000	n/a	0
3	0,671	280,4	8790	0	0,000	n/a	0
3	0,882	184,7	4395	0	0,000	n/a	0
0	0,000	n/a	0	1	0,480	243,4	16263
0	0,000	n/a	0	1	0,626	415,0	21684
3	1,000	210,8	4395	1	0,890	442,5	16263

ANNUAL FUEL BURN:	**3792,8**	ton
Fuel capacity 5000 nm at 18 kts:	**327,4**	ton
	3,85E+05	m^3
ANNUAL FUEL COSTS:	**3,34**	M€
ANNUAL MAINTENANCE COSTS:	**0,14**	M€

ABSOLUTE SCORES

		Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6	Concept 7	Concept 8
Operation characteristics	→ Maneuverability	ref	0	--	+++	+++	+	++	+++
	→ Signature profile	ref	0	-	++	++	+	+	+++
	→ Redundancy	2	2	2	3	3	3	3	2
Integration in ship	→ Nr of components	14	14	14	18	16	17	18	15
	→ Space consumption	286,3	280,6	715,0	450,3	459,3	677,2	274,8	690,2
	→ Total weight	255,9	256,1	549,4	334,1	368,4	487,7	299,6	620,4
	→ Fuel capacity	302,4	306,5	301,7	312,4	312,4	309,6	309,1	327,4
Availability	→ Reliability	ref	-	++	+	+	+	-	+++
	→ Maintainability	ref	+	--	++	+++	--	-	+++
	→ Shock resistance	ref	0	--	-	-	--	-	+
Costs	→ Purchase costs	26,14	26,37	21,93	32,04	33,17	23,93	28,29	73,24
	→ Fuel consumption	3886,9	4024,5	3434,3	3903,1	3950,0	3435,8	3875,0	3792,8
	→ Annual fuel costs	3,42	3,55	3,03	3,44	3,48	3,03	3,41	3,34
	→ Maintenance costs	0,168	0,225	0,119	0,140	0,176	0,097	0,158	0,137

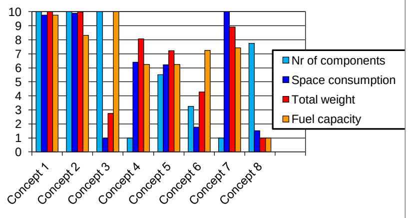
SCORES ON SCALE [1-10]

	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6	Concept 7	Concept 8
Maneuverability	5,0	5,0	2,1	9,3	9,3	6,4	7,9	9,3
Signature profile	5,0	5,0	3,6	7,9	7,9	6,4	6,4	9,3
Redundancy	1,0	1,0	1,0	10,0	10,0	10,0	10,0	1,0
Nr of components	10,0	10,0	10,0	1,0	5,5	3,3	1,0	7,8
Space consumption	9,8	9,9	1,0	6,4	6,2	1,8	10,0	1,5
Total weight	10,0	10,0	2,8	8,1	7,2	4,3	8,9	1,0
Fuel capacity	9,8	8,3	10,0	6,2	6,2	7,2	7,4	1,0
Reliability	5,0	3,6	7,9	6,4	6,4	6,4	3,6	9,3
Maintainability	5,0	6,4	2,1	7,9	9,3	2,1	3,6	9,3
Shock resistance	5,0	5,0	2,1	3,6	3,6	2,1	3,6	6,4
Purchase costs	9,3	9,2	10,0	8,2	8,0	9,6	8,9	1,0
Fuel consumption	3,1	1,0	10,0	2,9	2,1	10,0	3,3	4,5
Annual fuel costs	3,1	1,0	10,0	2,9	2,1	10,0	3,3	4,5
Maintenance costs	5,0	1,0	8,5	7,0	4,5	10,0	5,7	7,2

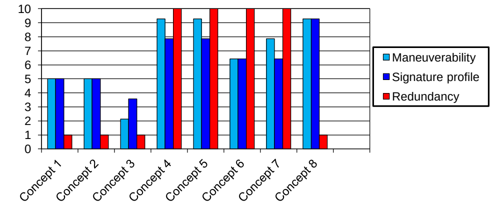
RELATIVE SCORES COMPARED TO REFERENCE

	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6	Concept 7	Concept 8	
Maneuverability	0	0	-2	3	3	1	2	3	0
Signature profile	0	0	-1	2	2	1	1	3	0
Redundancy									
Nr of components	ref	0%	0%	29%	14%	21%	29%	7%	
Space consumption	ref	-2%	150%	57%	60%	137%	-4%	141%	
Total weight	ref	0%	115%	31%	44%	91%	17%	142%	
Fuel capacity	ref	1%	0%	3%	3%	2%	2%	8%	
Reliability	0	-1	2	1	1	1	-1	3	0
Maintainability	0	1	-2	2	3	-2	-1	3	0
Shock resistance	0	0	-2	-1	-1	-2	-1	1	0
Purchase costs	ref	1%	-16%	23%	27%	-8%	8%	180%	
Fuel consumption	ref	4%	-12%	0%	2%	-12%	0%	-2%	
Annual fuel costs	ref	4%	-12%	0%	2%	-12%	0%	-2%	
Maintenance costs	ref	34%	-29%	-17%	5%	-42%	-6%	-18%	

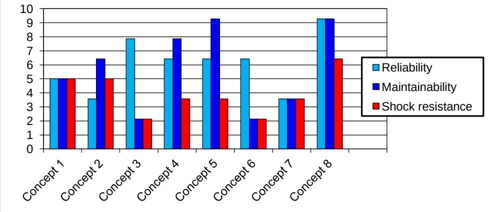
Scorecard "Ship integration"



Scorecard "Operational characteristics"



Scorecard "Availability"



Scorecard "Costs"