Charge air configurations for propulsion diesel engines aboard fast naval combatants

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Synopsis

The Royal Netherlands Navy (RNLN) wants to reduce the fossil fuel dependency of its fleet significantly, to decrease logistic efforts and environmental impact (Defensie, 2015). One of the methods to reduce fossil fuel dependency of ships is to reduce their energy requirement. The operational profile of fast naval combatants for the RNLN requires that the ships operate on the diesel engines for 90 percent of the time, often in part load where the turbocharger cannot supply the engine with the right amount of charge air. This results in a limited operating envelope for the diesel engine, and a decreased efficiency in part load. However, in part load, advanced charge air configurations can potentially resolve this. In this study a Mean Value First Principle (MVFP) diesel engine was used to investigate the effects of advanced charge air configurations on the efficiency and acceleration performance of diesel engines in hybrid configurations aboard fast naval combatants. It was concluded that the application of advanced charge air configurations can significantly improve the engine efficiency in part load. For example, in a diesel hybrid propulsion configuration with power take-off this can lead to an efficiency increase of almost 10 percent at 20 percent load in comparison with a single charged engine. Furthermore, hybrid turbocharging enables extending the operating envelope at low engine speed due to a better air excess ratio. With these concepts therefore, both improved efficiency and improved acceleration performance can be achieved.

Keywords: Turbocharger; hybrid; Operating envelope; Acceleration performance

1 Introduction

The time a naval combatant can be deployed for a mission is limited by its dependency on supplies. Therefore, in future the Royal Netherlands Navy wants to reduce the fossile fuel dependency of its fleet significantly, as seen in Figure 1. The aim is to improve the logistic dependency of the ships, but also to reduce environmental impact. The motivation is further elaborated in the Operationele Energie Strategie (OES) Defensie (2015). The objective of decreasing the energy requirements can be addressed many ways. One could change the operational profile, use alternative fuels or improve the design. For example, if the ship is fitted with very efficient engines the logistic dependency on fuel reduces, as do the emissions. The advantage of increasing the efficiency is that this approach is often complementary to the other approaches.

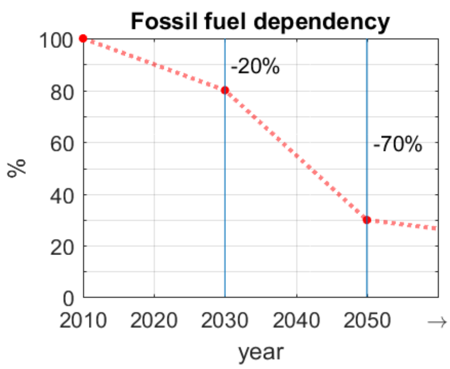

Figure 1: OES Targets (Defensie, 2015)

Currently, the RNLN is planning to replace of the Karel Doorman class frigates. The main task of this ship is anti submarine warfare. The envisioned propulsion plant consists of a hybrid diesel configuration, which enables the ship to use a silent propulsion mode for submarine hunting. A problem frequently encountered on diesel driven naval combatants is overloading of the diesel engines. In the past, the Karel Doorman class frigates have experienced problems with their diesel engines. This resulted in poor acceleration performance as the control strategy was designed very conservatively to protect the engine from overloading as described by van Spronsen and Tousain (2002). Geertsma et al. (2017a) studied the acceleration performance of a diesel hybrid configuration aboard a fast naval combatant and compared it to a combined diesel or gas turbine configuration. It was concluded that the acceleration performance of diesel hybrid configurations can be comparable and in specific cases even better than combined diesel or gas turbine configurations. This is important, since generally diesel engines are more efficient than gas turbines. Therefore, the choice for diesel engines instead of gas turbines would directly contribute to the goal of improving the efficiency of the ship.

The operational profile of a fast naval combatant is depicted in Figure 2. It can be seen that more than 90% of the operation time the ship sails at speeds lower than 21 knots, and operates on the diesel engines. Also, during a substantial part of this 90% the diesel engines are used in part load. Especially in part load, diesel engines are prone to thermal overloading. Thermal overloading of the diesel engine limits the acceleration performance of diesel driven naval combatants as shown by Vollbrandt (2016) and Geertsma et al. (2017b). In both studies the air excess ratio equation (10) was used as an indicator for the thermal loading of the diesel engine.

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The air excess ratio depends on the charge air pressure, provided by the turbocharger. Therefore, the turbocharger has a significant influence on the efficiency, operating envelope and acceleration performance of the ship. The turbocharger configuration and matching is an integral part of the engine design. The matching of a turbocharger is a compromise between high efficiency in the design point, and off design performance. Away from the design point, the turbocharger cannot supply the engine with the right amount of charge air. This results in a limited operating envelope for the diesel engine and a decreased efficiency in part load. Advanced charge air configurations can potentially resolve this and improve the results of Vollbrandt (2016) and Geertsma et al. (2017b) even further.

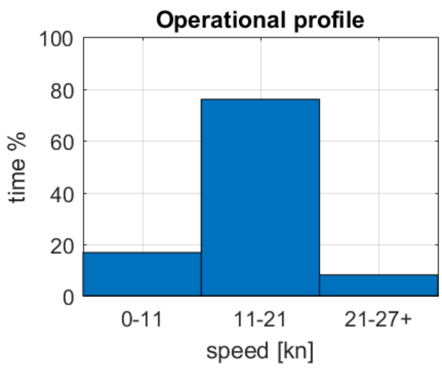


Figure 2: Operational profile (Vollbrandt, 2016)

The aim of this study is to investigate the effect of advanced charge air configurations on the efficiency and acceleration performance of diesel hybrid configurations aboard fast naval combatants. This study will contribute in three parts to this goal. First, a MVFP model with a power balance based turbocharger is explained. Second, a general classification of charge air systems will be presented. Finally, the model will be used to investigate the application of advanced charge air configurations to show the effect on the energy efficiency, acceleration performance, design, and emissions of fast naval combatants.

2 System description

Fast naval combatants with a diesel mechanical or diesel hybrid propulsion plant are typically fitted with two shaft lines consisting of one or more propulsion diesel engines, a gearbox connecting the diesel engine to the shaft and a propeller. This paper focusses on the effects of various charge air configurations on the efficiency of the diesel engine and the dynamic performance of the diesel engine. We therefore assume a theretical quadratic propeller curve, as proposed in Klein woud and Stapersma (2002). The correlation between the diesel engine and the ship's performance was not investigated deeply. The system model as considered in this paper is illustrated in Figure 3. The details of the models are described in the sections below.

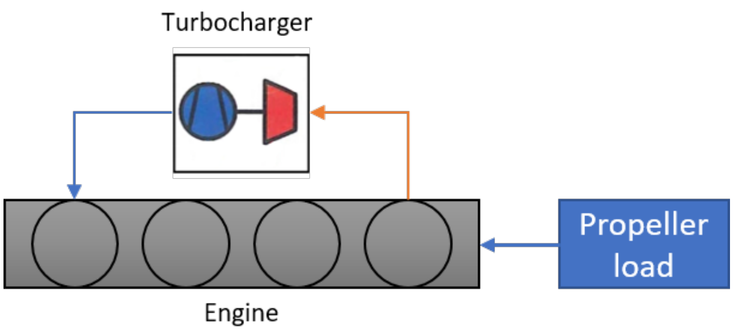


Figure 3: Overview of the overall model

3 Diesel engine model

Selecting the right modelling method is always a trade off between accuracy and computational power, therefore different models have been developed. Geertsma et al. (2017b) categorised diesel engine models according the level of dynamics and physical detail. This gives a convenient indication for accuracy and computational power. This work consists of a comparative study of the effects of different charge air configurations on the diesel engine performance. Therefore, an accurate prediction of the fuel consumption, charge air pressure and temperatures of the engine are required. Geertsma et al. (2017b) used a MVFP diesel engine model to investigate the performance of a single charged diesel engine. Validation showed that the MVFP diesel engine used by Geertsma et al. was able to predict fuel consumption, charge air pressure and temperature of a single charged diesel engine within $\pm 5\%$ accuracy. For comparative studies into propulsion concepts, this is satisfactory. The model as proposed by Geertsma et al. (2017b) was further improved by Loonstijn (2017a) who added a more detailed gas exchange. The model consists of four submodels: A compressor (COM) which also functions as an inlet receiver, a cylinder (CYL) sub model, an outlet receiver (OR) and a turbine (TUR) as depicted in Figure 4.

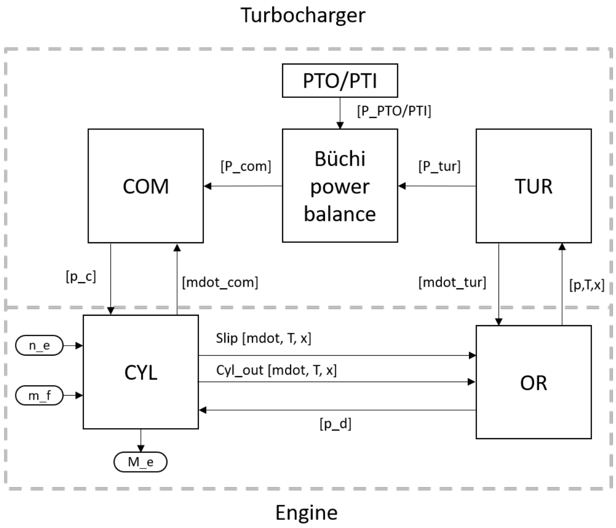


Figure 4: A schematic representation of a simple MVFP diesel engine model with a turbocharger Loonstijn (2017a).

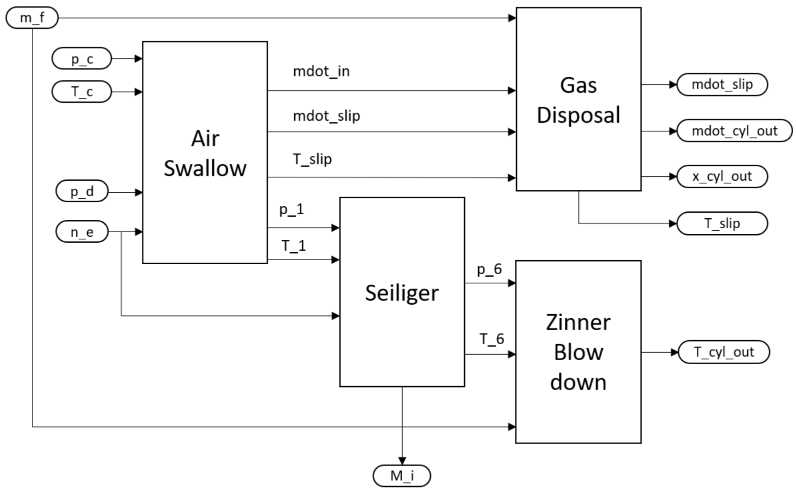


Figure 5: Schematic overview of the cylinder model

3.1 Cylinder and Outlet receiver model

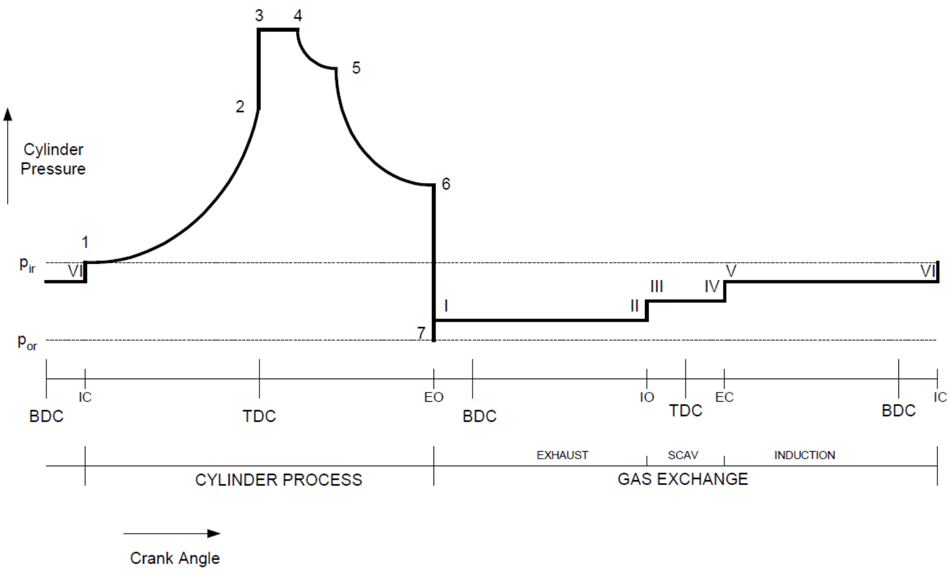


Figure 6: Diesel process as proposed by Schulten et al. Schulten and Stapersma (2003)

The cylinder model consists of the cylinder process and the gas exchange as depicted in Figure 6. The cylinder process is represented by a six point seiliger cycle and is modelled in the seiliger sub model. The gas is treated as a perfect gas with a homogeneous composition. The gas exchange is modelled in the air swallow, gas disposal and Zinner blow down sub model. A schematic overview of the cylinder model is depicted in Figure 5.

3.1.1 Air swallow

The air swallow sub model determines the mass flow of fresh air which goes into the cylinder. The air swallow sub model is extensivly discussed in the work of Loonstijn (2017a) with the theoretical background from Stapersma (2010a) and Stapersma (2010b).

The total mass flow into the engine $\dot{m}_{in}$ consist of the induction mass flow and the scavenge mass flow, as follows:

$$\dot{m}_{in} = \dot{m}_{ind} + \dot{m}_{sc} \tag{1}$$

where $\dot{m}_{ind}$ is the induction mass flow [kg/s] and $\dot{m}_{sc}$ is the scavange massflow in [kg/s].

$$\dot{m}_{ind} = \frac{p_c \cdot (V_{IC} - V_{EC})}{R_{air} \cdot T_{ind}} \tag{2}$$

In order to determine the scavenge mass flow, isentropic flow through a nozzle is assumed as proposed in Stapersma (2010b) [6.120].

$$\dot{m}_{sc} = i \cdot A_{sc,eff} \cdot \left(\frac{p_c}{\sqrt{R_c \cdot T_c}} \right) \cdot \Psi(\pi_{sc}) \tag{3}$$

where Ψ is the non dimensional flow coefficient and $A_{sc,eff}$ the effective scavenge area of the cylinder in $[m^2]$. In the model of the scavange flow, some important simplifications were made as explained by Loonstijn (2017b). First the scavenge efficiency was constant and second the trapped scavenge temperature was equal to the induction temperature. Also, the scavenge volume was not represented by a volume element based on the first law of thermodynamics, but as a combination of two flow restrictions as seen in Figure 7.

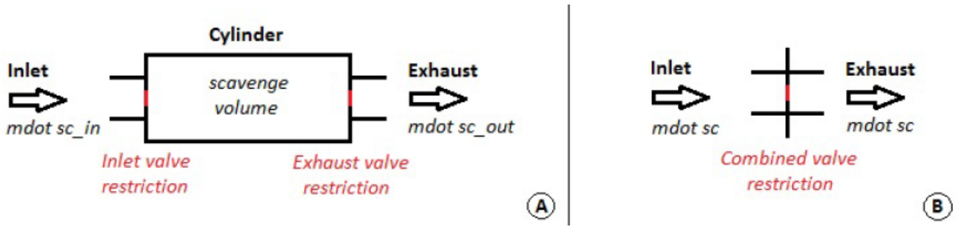


Figure 7: a) Filling and emptying model. b) nozzle equation model (Loonstijn, 2017b)

These simplifications are discussed by Stapersma (2010b) [6.3] and result in equation(3). The induction mass flow and the scavenge mass flow determine the total mass flow for the engine and can be calculated according to equation (1). The trapped mass can be calculated with equation (4) according to Stapersma (2010b).

The trapped mass is the mass that is actually 'trapped' in the cylinder just before combustion and takes part in the combustion. If the simplification $T_{ind} = T_{sc-tr}$ is applied, the trapped mass can be written as:

$$m_1 = \frac{p_c \cdot V_{IC}}{R_{air} \cdot T_{ind}} \tag{4}$$

In order to find the amount of fresh air in the cylinder, the scavenge efficiency has to be taken into account. In the model, this is taken as a constant value of 1.0 since the model only considers 4 stroke diesel engine with significant air slip (Stapersma, 2010a, 2.117).

3.1.2 Seiliger

The Seiliger sub model consist of a 6 point seiliger cycle and an heat release sub model. The six point seiliger is extensivly discussed in the work of Stapersma (2010a) and Geertsma et al. (2017b) and will therefore not be explained here. The heat release is modeled according to the summary found in the work of Geertsma et al. (2017b). The heat release model represents the heat release in kJ/kg during isochoric q_{23} , isobaric q_{34} and isothermal q_{45} combustion as represented by the six point seiliger cycle.

3.1.3 Zinner blowdown

After opening of the exhaust valve the hot exhaust gases are expelled to the outlet receiver. The blowdown was represented according to Zinner (1980) and is also described by Stapersma (2010b). The blow down temperature can be calculated as follows:

$$T_{bld} = \left(\frac{1}{n_{bld}} + \frac{n_{bld} - 1}{n_{bld}} \cdot \frac{p_d}{p_6} \right) \cdot T_6 \tag{5}$$

The polytropic expansion coefficient n_{bld} allows for heat transfer to the cylinder wall, exhaust valve and inlet duct. The resulting temperature for the outlet receiver is a result of the combustion gas flow and the slip flow.

In order to get an estimator for the thermal loading of the diesel engine as explained by Sapra et al. (2017) the temperature of the exhaust valve as proposed by Grimmelius and Stapersma (2000) is also calculated in the Zinner blow down sub model:

$$T_{ev,est} = \frac{T_6 + r \cdot T_1}{1 + r} \tag{6}$$

with:

$$r = s^{0.8} \cdot \left[\frac{T_1}{T_6}\right]^{0.25} \cdot \left[\frac{EC - IO}{IO - EC}\right]^{0.2} \quad with: \quad s = \frac{\dot{m}_{trough}}{\dot{m}_{bld}} \tag{7}$$

Where $T_{ev,est}$ is the estimated exhaust valve temperature [K] and r the heat transfer ratio, s is the scavenge factor. The connection to the Zinner blow down is the polytropic expansion coefficient, which allows for the heat transfer from the blow down gasses to the exhaust valve.

3.2 *Turbocharger model*

The compressor model is based on the Büchi equation (8), essentially stating $w_{com} = w_{tur}$. It is a very convenient way to model the turbocharger because it is simple, the model can easily be adapted to more advanced configurations and the amount of inputs are limited and usually available in the FAT data of an engine as explained by Geertsma et al. (2017b). The Büchi equation represents the charge pressure p1 as follows:

$$\pi_{com} = \left\{1 + \beta \cdot \delta \cdot \chi \cdot \eta_{\mathrm{TC}} \cdot \tau \cdot \left(1 - \frac{1}{\pi_{tur}^{\frac{\gamma_{de}-1}{\gamma_{de}}}}\right)\right\}^{\frac{\gamma_{air}}{\gamma_{air}-1}} \quad with: \quad \pi_{com} = \frac{p_c}{p_b} \tag{8}$$

Where π is the pressure ratio, β is the power correction factor, δ is the fuel addition factor, χ ratio of specific heats of air and exhaust gas, τ is the temperature ratio and γ the specific heat ratio.

In the model, the turbine and the compressor are assumed to be in dynamic equilibrium. Additional power can be added to the power balance in case of hybrid turbocharging.

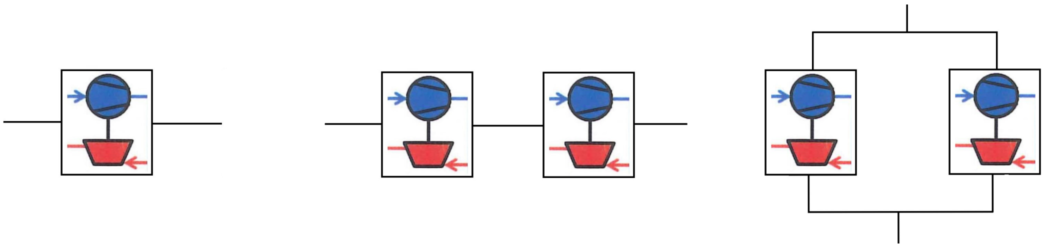
Scavenging

The mass flow through the diesel engine as described by equation (1) is highly depended on the charge pressure in the inlet receiver since both the induction mass flow equation (2) and the scavenge mass flow equation (3) are depended on p_c. The massflow trough the engine has a connection to the thermal loading via the scavenge factor as seen in Equation (6). The amount of air flowing trough the engine between the power strokes determines the cooling and therefore influences the thermal loading of the engine.

4 Turbocharger

4.1 *Turbocharger classification*

In order to avoid confusion about the turbocharger configuration a straightforward classification is required. When classifying turbocharging concepts, as in many other engineering subjects, the resemblance with an electric circuit can be made. This paper proposes a mechanical-electrical analogy, as depicted in Figure 8. The aim of the classification is that its simple, unambiguous and also clear for people who are not familiar with the classification.



(a) Single turbocharging (b) Series turbocharging (c) Parallel turbocharging

Figure 8: Different turbocharger configurations

First the position of the chargers with respect to the other chargers in the system is determined, in principle this can be a *single*, *series* or *parallel* configuration. Then, in order to indicate the number of chargers a prefix is added. For example, if two chargers are connected in series the configuration is called: two-series turbocharging, now often indicated as two-stage turbocharging. Furthermore, for each type of configuration an additional series of *number-configuration-technologie* is added to the name of the complete charger configuration. In order to indicate whether the chargers are switched sequentially, are supported by an electric motor or fitted with variable turbine geometries a suffix is added. Although the suffixes as proposed here are limited to *sequential*, *VTG* and *hybrid* other suffixes are possible. If a turbocharger is fitted with more then one additional technology all are mentioned in the name. In summary, this leads to the following classification:

$$\underbrace{number}_{\text{1,2,3..}} \quad - \quad \underbrace{configuration}_{\textit{single, series, parallel}} \quad - \quad \underbrace{technology}_{\textit{sequential, hybrid, VTG}} \tag{9}$$

4.2 Hybrid turbocharging

In this work, a hybrid turbocharger is defined as a turbocharger with an electric machine coupled to the main shaft as seen in Figure 9. Hybrid turbochargers can be used for different proposes. The electric machine can work in generator mode, subtracting power from the turbocharger (PTO) or in motor mode, adding power to the turbocharger (PTI). The objective of PTO is to subtract additional energy from the exhaust gases. For example, to supply energy to the electric grid of the ship. The objective of PTI is to diminish the turbo 'lag' to improve transient behaviour and to increase the charge air pressure for the engine.

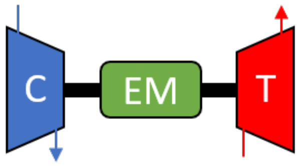


Figure 9: hybrid turbocharger schematic Gerada et al. (2012)

4.2.1 Air excess ratio control

The air excess ratio λ of the engine is defined as the ratio of total fresh air in the cylinder to the minimum amount of fresh air required for combustion (Stapersma, 2010c).

$$\lambda = \frac{m_1}{m_f \cdot \sigma_f} \tag{10}$$

where σ_f is the stoiciometric air to fuel ratio of the fuel (14.5), m_f is the mass of fuel [kg] injected and m_1 is the trapped mass of air [kg] at the start of compression.

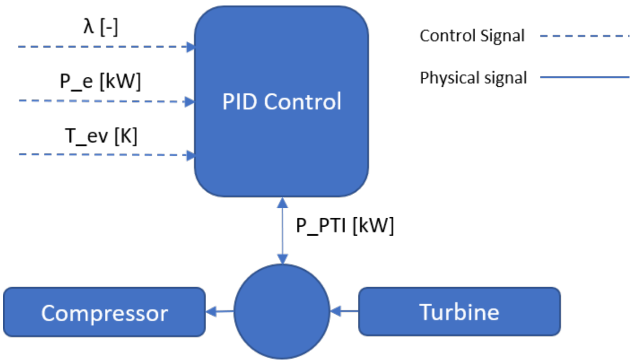


Figure 10: Control system for hybrid turbocharging

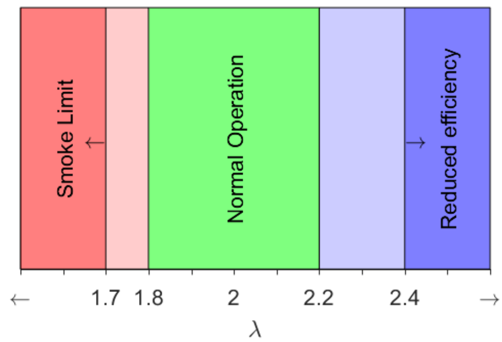


Figure 11: Lambda values as recommended by Stapersma (2010a)

The air excess ratio is an important indicator for the thermal loading of the engine as discussed in Sapra et al. (2017). If the air excess ratio drops below a certain level, the temperatures in the cylinder rises. As a consequence, the temperatures in the combustion chamber and exhaust system significantly exceed the permissible range which can lead to piston burn-off, cracks in the cylinder head, valve and turbocharger damage (von Drathen, 2014). Therefore, the leading control parameter is the air excess ratio. In order to control the air excess ratio via the turbocharger a control loop was implemented in the model as depicted in Figure 10. Normal operating values for the air excess ratio of a diesel engine are between 1.8 and 2.2 as seen in Figure 11. Therefore, the objective value for the air excess ratio was set to $\lambda = 2$. If the air excess ratio is above $\lambda = 2$, the system will take off power (PTO) from the turbocharger. If the value is under $\lambda = 2$ the system will supply power to the turbocharger (PTI).

5 Results

Four turbocharger configurations were investigated: single turbocharging, single-hybrid turbocharging, 2-parallel-sequential turbocharging, and 2-parallel-sequential,hybrid turbocharging. The models discussed here were based on the Wartsilä W26 diesel engine as installed aboard the Zeven Provincin class frigates of the RNLN. In order to make a quantitative comparison between the various charge air concepts, the nominal engine power and turbocharger size were kept constant. Inevitably, this leads to unfavourable matching. For the non sequentially turbocharged engines the mismatch is quite severe, the area where the engine is normally operating on one turbocharger is now operated on two turbochargers. The 2-parallel-sequential,hybrid turbocharging resulted in a low exhaust valve temperature and an almost constant air excess ratio. Therefore, the operating envelope was enlarged following a constant torque line.

5.1 Simulation experiments

5.1.1 Static working points over complete operating envelope of the engine

The first experiment investigates the behaviour of the diesel engine in static working points, from 20% power to 100% power, in steps of 2%. The experiment is used to determine whether the engine is overloaded in static working points and to investigate the effect of PTO and PTI on fuel consumption and thermal loading.

5.2 Brake specific fuel consumption

In Figure 12 till Figure 15 the brake specific fuel consumption for each turbocharging configuration is shown over the complete operating envelope. In general it can be seen that the lowest specific fuel consumption was achieved at the boundary of the operating envelope around 700 [rpm]. At this point the engine already delivers a big part of its rated power while the engine speed and mechanical losses are still relatively low.

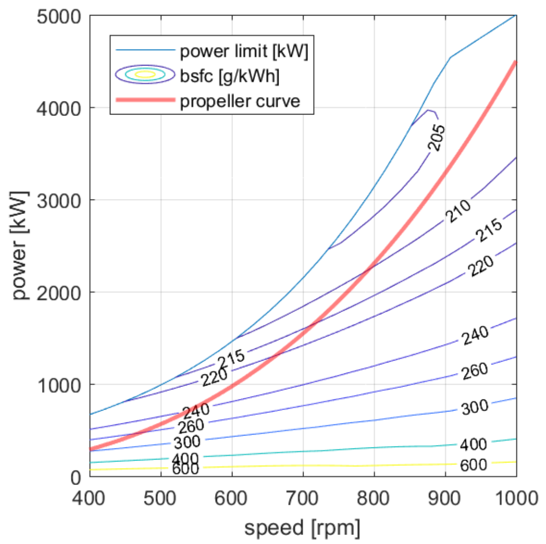


Figure 12: Single (a)

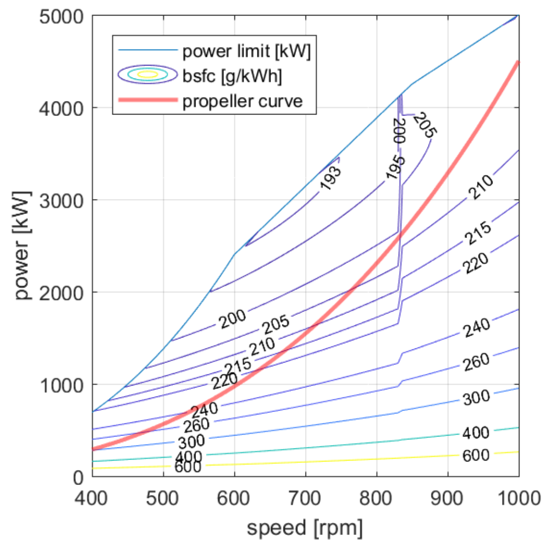


Figure 13: 2-parallel-sequential (b)

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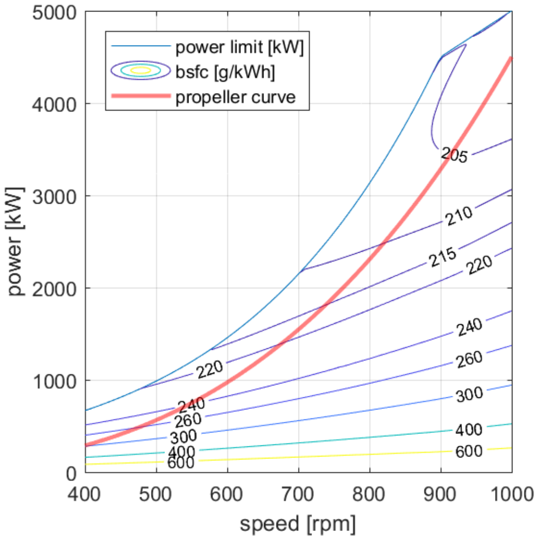

Figure 14: Single-hybrid (c)

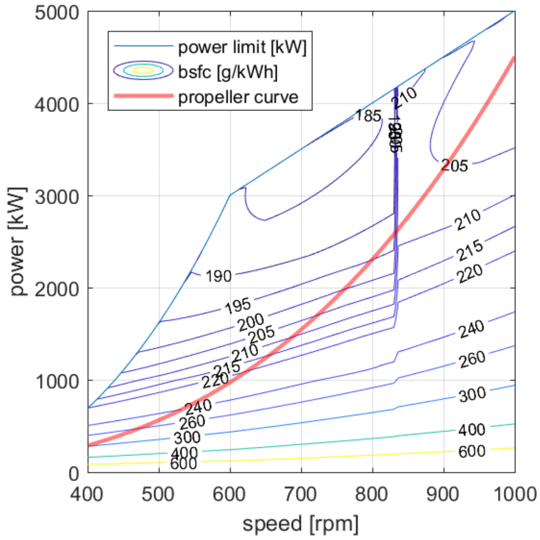

Figure 15: 2-parallel-sequential,hybrid (d)

5.2.1 *Direct drive*

The results are compared on the propeller curve as depicted in Table 1. Table 1 shows that on the propeller curve, the effect of parallel-sequential turbocharging with respect to single turbocharging lead to an maximal bsfc decrease of approximately 3%. In hybrid turbocharging configurations, power is added to the turbocharger to keep the air excess ratio at the required value of 2, which will increase the efficiency of the engine, but comes at the cost of the power supplied to the turbocharger. Table 1 shows that on the propeller curve, the effect of 2-parallel-sequential,hybrid turbocharging with respect to single turbocharging resulted into an maximal bsfc decrease of approximately 7%.

Table 1: The bsfc [g/kWh] of the four charge air configurations on the propeller curve and compared to the single charged concept

rpm	500	600	700	800	900	1000		500	600	700	800	900	1000
Concept	bsfc [g/kWh]							change ratio to concept a in %					
a	250,9	229	216,3	209,2	205,9	207,2		-	-	-	-	-	-
b	248,3	225,9	210,9	202,5	206,2	207,2		-1 %	-1 %	-2 %	-3 %	0 %	0 %
c	250,8	229,3	217,5	210,3	204,6	201,2		0 %	0 %	1 %	1 %	-1 %	-3 %
d	250,1	223,4	207,8	194,7	205	201,2		0 %	-2 %	-4 %	-7 %	0 %	-3 %

5.2.2 *Hybrid propulsion*

First, a direct drive configuration without electric machines on the propulsion shaft is investigated. It is clear that the full potential of the turbocharging configurations is not utilized yet. In order to improve the efficiency further, the operating points of engine are shifted in the direction of the most efficient part of the operating envelope as depicted in Figure 18 and Figure 19. Therefore, a power take off on the propeller shaft as explained by Klein woud and Stapersma (2002) and seen in Figure 16 was used. The power subtracted from the propeller shaft is fed to the electric grid of the ship. Another option is to transfer power from one shaft to a second shaft via electric machines, as depicted in Figure 17. In the latter case, two shafts will be driven in parallel by one diesel engine.

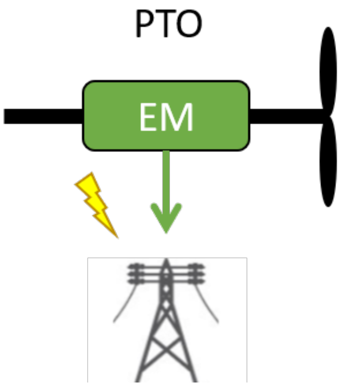

Figure 16: PTO on propeller shaft

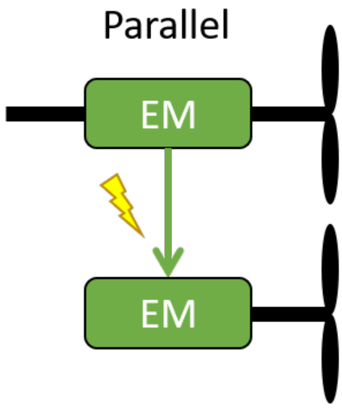

Figure 17: Drive two shafts parallel with one diesel engine

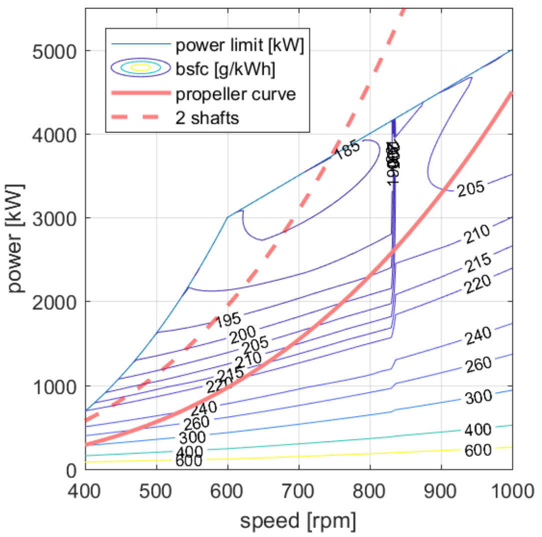

Figure 18: Bsfc [g/kWh] contours with two shafts driven parallel

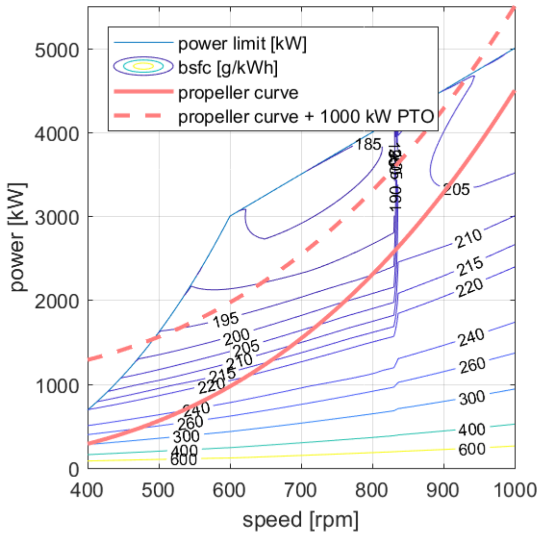

Figure 19: Bsfc [g/kWh] contours with PTO

The results of the altered propeller curve are given in Table 2 and Table 3, and show that the effect of the charge air configurations increases significantly if the propeller curve is altered. In comparison with a single charged diesel engine driving one shaft, without PTO, the bsfc decreases with more than 15% in part load.

Table 2: Two shafts driven by one diesel engine, compared to concept(a) with one shaft

rpm	500	600	700	800	900	rpm	500	600	700	800	900
Concept	bsfc [g/kWh]					Concept	change ratio to concept a in %				
a	250,9	229,0	216,3	-	-	a	-	-	-	-	-
b	207,5	196,7	192,8	-	-	b	-17 %	-14 %	-11 %	-	-
d	206,1	192,3	184,4	-	-	d	-18 %	-16 %	-15 %	-	-

Table 3: Propeller curve + 1000 kW PTO on shaft, compared to concept(a) without PTO

rpm	600	700	800	900	1000	rpm	600	700	800	900	1000
Concept	bsfc [g/kWh]					Concept	change ratio to concept a in %				
a	229,0	216,3	209,2	205,9	-	a	-	-	-	-	-
b	196,4	195	194,3	206,1	-	b	-14 %	-10 %	-7 %	0%	-
d	191,8	187,1	185,6	207,8	-	d	-16 %	-13 %	-11 %	-1 %	-

6 Conclusions and further research

The effect of the charge air concepts on the efficiency of the diesel engine is significant. 2-parallel-sequential and 2-parallel-sequential,hybrid turbocharging both show an increase of efficiency on the propeller curve of 3% and 7% respectively. In hybrid configuration the effect of advanced charge air configurations on the efficiency can be increased by more than 15% or 36,7 g/kWh. Further, the results for the static working points suggest that thermal loading of the engine can be significantly lowered during acceleration, improving the acceleration performance. To investigate the impact on the interaction of engine, ship and propeller it is recommended to set up a dynamic simulation of a complete ship propulsion model to predict acceleration performance and thermal loading. Then the exact reduction in fossil fuel dependency can be determined, and the societal benefits estimated.

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Nomenclature

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Greek Symbols

ε_{INL}	heat exchange effectiveness
π_{sc}	Pressure difference between inlet and outlet of the cylinder [Pa]
Ψ	non dimensional flow coefficient

Roman Symbols

$A_{sc,eff}$	effective scavenge area of the cylinder [m^2]
p_c	inlet receiver pressure [Pa]
i	number of cylinders

Subscripts

COM	Ccompressor
CYL	Cylinder
eng	engine
IC	inlet valve closure
ind	induction
INL	inlet
OR	Outlet receiver
sc	scavenge
TUR	Turbine

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