Electric power generation from the hybrid turbocharger of a marine propulsion diesel engine

Treball Final de Grau

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Abstract

This thesis analyses the concept of using a hybrid turbocharger as an energy recovery system to generate electric power from the exhaust gases of a marine propulsion Diesel engine. The study has been based in the use of a previously tested numerical model developed in Matlab® and Simulink® software to perform a series of simulations of a MAN V12 51/60 DF turbocharged engine, which originally does not count with this electricity generation technology. For this reason, the study has been notably focused on the effects of this design on the turbocharger, especially on its compressor, but considerable attention also has been paid to the engine fuel-air mixture characteristics and fuel consumption behavior.

Nevertheless, the main objective on the development of this study has been to ascertain the power generation potential that this technology offers when it's utilized in a Diesel engine. In the first instance, this has been done by a verifying the accuracy of the model compared to the real engine. This has been followed by an optimization process of the turbocharger characteristics, the charge air blow-off valve setting and the fuel flow, to bring the turbocompressor to its maximum performance while operating the engine with the minimum intake air for a given power rate as possible. Finally, this intake air conditions have been replicated with the use of the resistance torque offered by the electric generator of the hybrid turbocharger instead of the charge air blow-off valve, consequently generating electrical power, which has proven to be satisfactory.

However, as the analyzed engine has dual fuel characteristics, and by these means, the capability to operate with different types of fuel under different thermodynamic cycles, it has been possible to study the performance discrepancies of this concept when used in the same engine, but contrasting natural gas and Diesel fueled operation. The information of the system operating in natural gas for this comparison, has been obtained from a previous investigation published in [15] .

Finally, after obtaining the main power generation results, the possibility of using this technology as a method to deal with MARPOL Annex VI requirements for the *Energy Efficiency Design Index (EEDI)* fulfillment on a real ship has been considered too, showing satisfactory results.

Resum

En aquest projecte final de grau s'ha portat a terme l'anàlisi d'un turbocompressor híbrid com a sistema pel reaprofitament de l'energia dels gasos d'escapament d'un motor Dièsel marí per tal d'obtenir energia elèctrica. L'estudi està basat en l'ús d'un model numèric prèviament verificat que fou desenvolupat amb els softwares Matlab® i Simulink®, que ha permès portar a terme una sèrie de simulacions sobre un motor MAN V12 51/60 DF sobrealimentat amb turbocompressor, el qual originalment no compta amb aquesta tecnologia de generació d'energia elèctrica. Per aquest motiu, l'estudi ha estat centrat de manera notòria en els efectes d'aquest disseny en el sobrealimentador, sobretot en la part del compressor, encara que també s'ha parat una atenció a les característiques de la mescla de combustible-aire i de l'evolució del consum de combustible.

Tanmateix, l'objectiu principal del desenvolupament d'aquest estudi ha estat esbrinar el potencial de generació d'energia elèctrica que aquesta tecnologia pot oferir al ser utilitzada en un motor Dièsel. Per arribar a aqests resultats, el primer pas ha estat corroborar la precisió del model numeric en comparació al motor real. Seuidament, sha portat a terme un procés 'optimització tant de les característiques del turbocompressor com de la vàlbula de regulaciío de l'aire de sobrealimentació així com de l'aportació de combustible, per tal de opder obtindre les màximes prestacions del turbocompressor mentre el motor opera amb el mínim aire d'admissió per a una potencia determinada. Finalment aquestes característiques de l'aire d'admisió han estat reproduides mitjançant el parell resistent ofert pel generador elèctric del turbocompressor hibrid, generant en consequència energia elèctrica, obtenint resultats satisfactoris.

Tot i això, com que el motor estudiat presenta la capacitat d'operar amb combustibles de doble naturalesa, utilitzant cicles termodinàmics diferents, també ha estat possible estudiar les diferencies en el funcionament d'aquest concepte al ser utilitzat amb el mateix motor però funcionant amb gas natural o Dièsel. La informació d'aquest sistema a l'operar amb el motor a gas natural ha estat obtinguda a partir d'investigacions prèvies les quals es troben publicades a [15] .

Finalment, després d'obtenir els resultats principals sobre la generació d'energia elèctrica, també s'ha considerat l'ús d'aquesta tecnologia com a mètode per assolir els requisits de l'Annex VI del MARPOL en referència a l'Índex de Disseny sobre Eficiència Energètica (EEDI) en un vaixell real, mostrant bons resultats.

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1. Introduction

Deriving from previous investigations carried out with a numerical model of a marine dual fuel engine, in this paper the study for the enhancement of the total energy efficiency of a vessel power-plant is developed. This is attempted by the use of a hybrid turbocharger to recover the maximum power from the exhaust gases of the engine in order to obtain electrical energy. In this case, the study is focused on the Diesel operation mode of the engine to reckon the energy that can be potentially recovered under this operating condition.

The development of this study is motivated by the increasing need to deal with the environmental and economic challenges related to the use of fossil fuels that the maritime industry is facing nowadays and which will become more compromising in the future. Due to the necessity to take action against the harmful consequences derived from the use of fossil fuel powered machines, the regulations implemented during the last years by the International Maritime Organization, most of them hosted in the Annex VI of the MARPOL convention for the prevention of air pollution from ships, have set a demanding horizon that requires new solutions to minimize the footprint of the maritime industry on the environment. The implementation by the industry of the new technologies required to cope with these incoming limitations is consequently motivated by the economic penalties derived from the non-compliance of the regulations and also the rise of fuel prizes which makes interesting and profitable to look for systems with higher energy efficiency and lower consumption rates, even though they become more complex and thus, more expensive to acquire and maintain.

The Diesel engine is the most prevalent power source for the marine transport due to it's higher efficiency when compared to Otto engines and its capability to run with lower quality fuels that make it cheaper to operate. However, burning these kinds of fuels generates high amounts of atmospheric pollution so in the last years the industry has been focusing on the future conversion to other fuels like liquefied natural gas as they reduce significantly the polluting emissions. Nevertheless, the natural gas engines developed so far, haven't achieved the same power rates offered by the biggest Diesel engines propelling many vessels nowadays, which indicates that this technology will be required for long but it will need improvements to keep itself competitive as a more environmentally friendly technology.

In order to help marine engines to reduce their environmental impact and to improve the total energetic

efficiency of a vessel, waste heat recovery systems are one of the most commonly used methods regardless of the type of engine onboard (turbine or reciprocating). These systems transform the heat and pressure of the exhaust gases which would usually be discharged and not used, into electrical power. This is generally done by an intermediate steam circuit heated by the exhaust gases that allows running a steam turbine group, but the use of independent gas turbines powered directly by the engine's exhaust gases has been widely studied also.

Since their introduction, turbochargers have been the most common way to use the energy from the exhaust gases to improve the engine performance by increasing the air mass-flow into the cylinders. However, as gas turbines and compressors have achieved higher efficiencies, less energy has been required to produce the mass flow needed by the engine, which has forced the exhaust gases to be partially by-passed or boost air to be expelled. It's obvious then, that there's the possibility not only to use the exhaust gases to compress intake air, but also to obtain useful work for other purposes, and with this concept in mind comes the idea of an electric generator driven by the turbocharger shaft.

2. Engine description

The study developed in this document is focused on a MAN V12 51/60 DF power plant. This is a 4 stroke turbocharged engine, with 12 cylinders in “V” distribution fed by fuel injection and designed to operate as a dual fuel power-plant offering a maximum rated power of 12.000 kW. The engine is equipped with variable injection timing *VIT* and variable intake valve timing *VIVT* systems. Its cooling system is composed of two freshwater single stage circuits, one for the cylinders and one for the injectors, and a third fresh water double stage circuit for the charge air. The starting motion is achieved by a standard compressed air system.

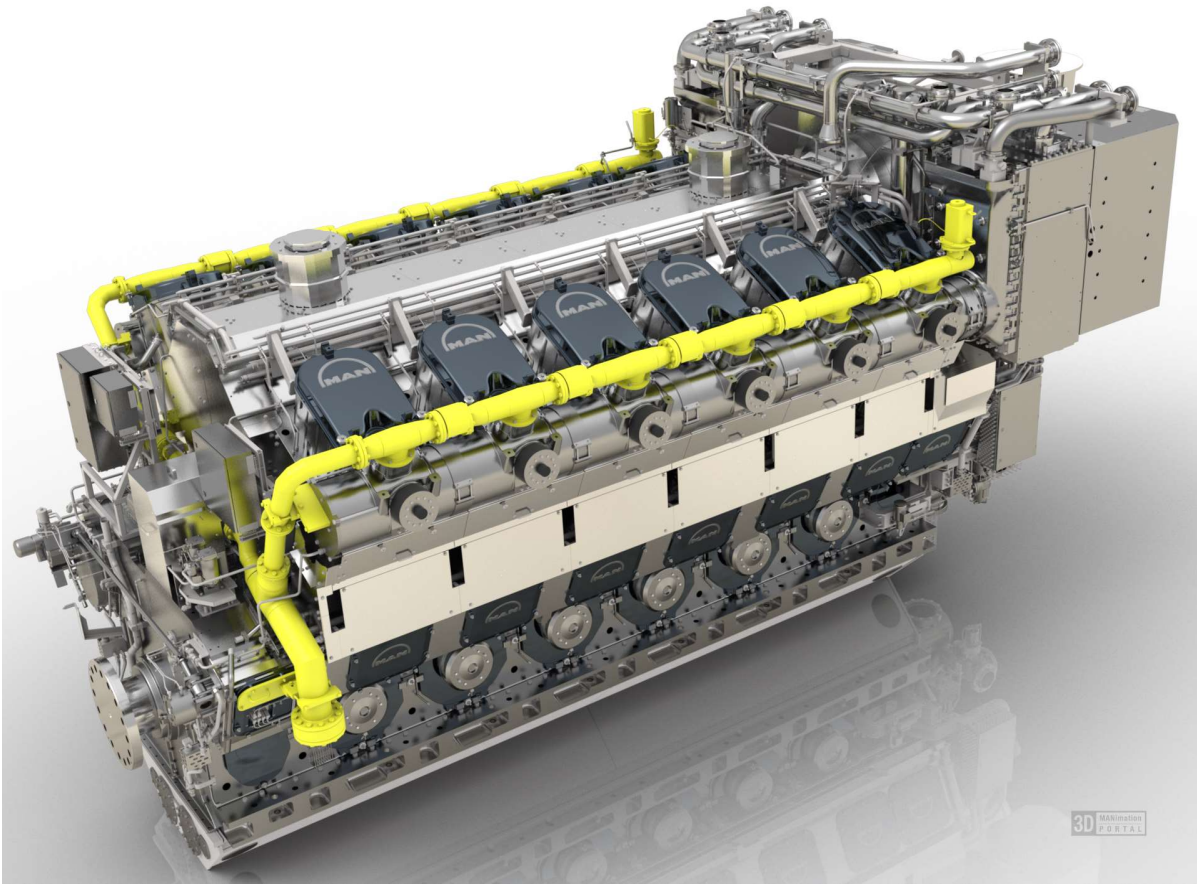

Figure 1: MAN V12 51/60 DF engine overview.

One of the main features of the engine is its already mentioned dual fuel capability, which allows the engine to operate using both gas and liquid fuels, independently or combined. With this, the engine is meant to offer higher efficiencies, lower emissions and more flexibility for future international regulations. When the engine uses gaseous fuel, it relies on Natural Gas (based on methane), and for liquid fuels it can use several types od Diesel fuel, including MGO (DMA, DMZ), MDO (DMB) and HFO (up to a viscosity of 700 mm2/s

(cSt) at 50 °C). Therefore, depending on the requirements, the engine can operate in Natural gas fuel mode or Diesel fuel mode. This adds a considerable complexity to the machine, which has to work under the characteristics of Otto and Diesel cycles and deal with their divergences, like the required injection and valve timing, the ignition method, or the ducting, pumping and temperature treatment of each fuel.

The basics of the engine power regulation procedure differ depending on the fuel. In Diesel mode, the power is exclusively controlled by modulating the fuel quantity injected on the cylinders, and little or no regulation is done to the intake air. In a natural gas engine, like in the majority of Otto engines), the main feature for the power regulation is the intake air, which in this case is adjusted by expelling charge air through a blow off valve in the intake manifold. As the natural gas introduced in the intake manifold is controlled to keep a constant fuel-oxidizer ratio with the intake air, the desired power is obtained.

The design of the engine includes two individual fuel systems, one for each mode. In Diesel mode, the engine uses direct injection to supply the cylinders with the required fuel quantity, whereas in Natural gas mode, the fuel is injected right in the intake manifold. In both modes, the ignition is achieved by controlling the injection timing of the diesel fuel. The combustion in Diesel mode is initiated by the self-ignition of the fuel after being injected directly in the cylinders at the precise moment. In Natural gas mode, a pilot flame is obtained with a short direct injection stream of Diesel fuel (pilot fuel) inside the cylinders, which self-ignites and starts the combustion of the natural gas.

The *VIT* system provides the means for the MAN V12 51/60 DF to modify the injection start and also the ignition start. With this, as the proper pilot fuel injection can be provided, the dual fuel operation becomes possible, and also several benefits can be achieved depending on the system setting. Ignition pressure increment can be obtained if the injection is advanced and thus, a reduction in fuel consumption. In the other hand, a delay in the injection start can cause a reduction in NO_X emissions (see Error: No se encuentra la fuente de referencia).

In order to make the dual fuel operation possible, the engine is also equipped with the *VIVT* system, as it has to be adapted to operate under the characteristics of Diesel or Otto cycles depending on the type of fuel selected. To do so, the engine uses the so called Miller cycle, on which different methods are considered to modify the compression ratio of a supercharged engine. In the case of the MAN V12 51/60 DF

engine, this is done by anticipating or delaying the intake valve closing. On natural gas mode, the intake valve is configured to a have significant advance on its closing, in order to allow the intake air to suffer an in-cylinder expansion before the compression stroke. This advance is set around 40° before bottom dead centre, so the intake valve closes when the piston still has to complete approximately a 35% of the intake stroke. During the interval between the intake valve closing and the bottom dead centre, the charge air will partially expand inside the cylinder, so the pressure at the start of the compression stroke will be lower compared to not anticipated valve closing engine, reducing the temperature achieved at the end of the compression stroke and preventing the self-ignition of the fuel. In Diesel mode, the intake valve timing is delayed to obtain higher temperatures for the correct ignition of the fuel.

The main feature numbers of the engine are indicated in table *1*:

Cylinder bore	510 [mm]
Piston stroke	600 [mm]
Displacement per cylinder	122.5 [dm^3]
Number of cylinders	12
Engine length	9088 [mm]
Engine length with turbocharger	10254 [mm]
Weight without fly wheel	187 [Tons]
Power at 500 rpm (ISO standard conditions)	11700 [kW]
Power at 514 rpm (ISO standard conditions)	12000 [kW]

Table 1: Engine design characteristics

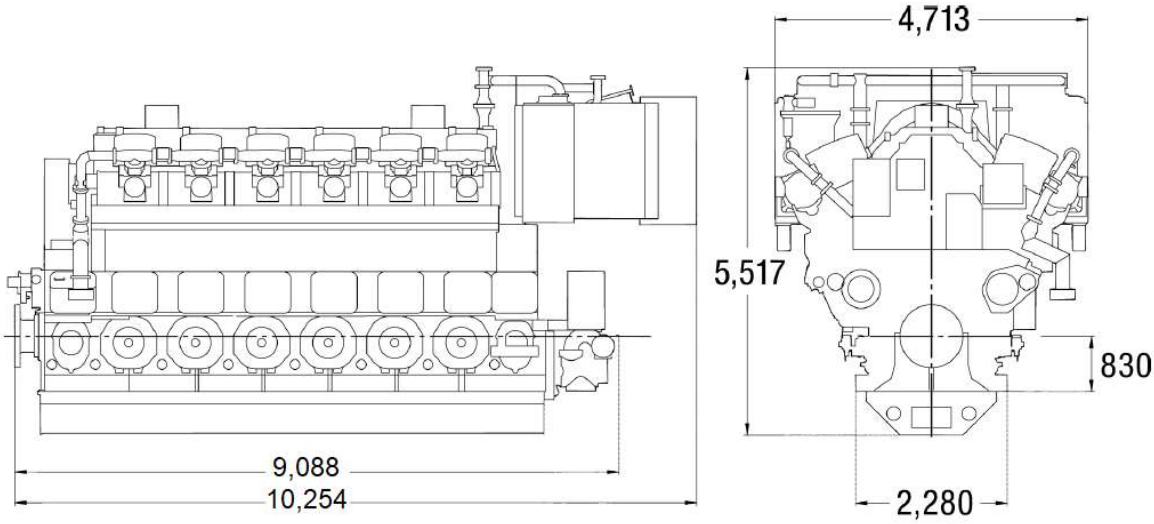

Figure 2: Engine main dimensions

Table 2 indicates the estimated values of fuel oil consumption for electric propulsion/auxiliary GenSet (speed = constant) and mechanical propulsion with CPP:

		HFO/MDO					MGO				
Load %		100	85	75	50	25	100	85	75	50	25
a) Main fuel	[g/kWh]	178,1	176,8	182,5	185,9	197,5	179,6	177,8	183,3	186,4	197,5
b) Pilot fuel	[g/kWh]	1,9	2,2	2,5	4,1	7,5	1,9	2,2	2,5	4,1	7,5
	[kJ/kWh]	81	94	107	175	320	81	94	107	175	320
Total a+b	[g/kWh]	180,0	179,0	185,0	190,0	205,0	181,5	180,0	185,8	190,5	205,0
	[kJ/kWh]	7686	7643	7900	8113	8754	7750	7686	7934	8134	8754

Table 2: Engine fuel consumption for mechanical and electric propulsion/ auxiliary

The MAN V12 51/60 DF is a multifunction drive, projected for marine electrical and mechanical propulsion, including controllable pitch propeller CPP, and also for power generation onboard or in ground settled facilities. As a marine engine, it has been approved by all main classification societies (ABS, BV, CCS, ClassNK, DNV, GL, KR, LR, RINA, RS) as marine main engine, i.e., engine for propulsion purpose, and as a main auxiliary engine. Depending on the role adopted by the engine, the output power limit can be established at 100% of the rated output in the case of driving a propeller, and 110% of the rated output when driving an electrical generator.

The engine is designed to work in constant speed or variable speed conditions independently of the fuel mode. In constant speed, the engine is maintained at 514 min^{-1}, corresponding to the maximum rpm's at contentious service. In variable speed, the rpm differ depending on the engine brake horse power and correspond to the values indicated in *Table 3*.

Power rate	100	85	75	50	25	%
Speed	514	514	510	462	402	min^{-1}

Table 3: Engine power-speed equivalence for variable speed condition.

The next diagrams show the allowed operation conditions for the engine as a function of the output power and engine speed:

Operating range for generator operation/electric propulsion:

- **MCR**: Maximum continuous rating.
- **Range I**: Operating range for continuous service.
- **Range II**: No continuous operation allowed. Maximum operating time less than 2 minutes.
- **Range III**: According to DIN ISO 8528-1 load > 100 % of the rated output is permissible only for a short time to provide additional engine power for governing purposes only (e.g. transient load conditions and suddenly applied load). This additional power shall not be used for the supply of electrical consumers.

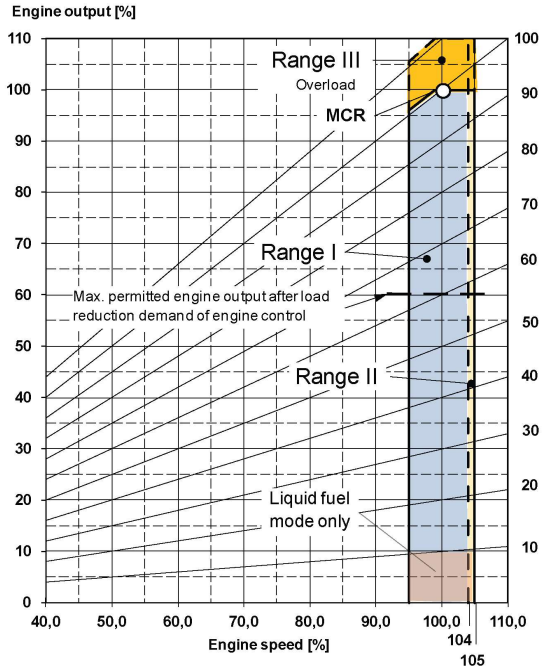

Figure 3: Operating range for generator operation/electric propulsion

Operating range for controllable pitch propeller (CPP):

- **MCR:** Maximum continuous rating.
- **Range I:** Operating range for continuous operation.
- **Range II:** Operating range which is temporarily admissible e. g. during acceleration and manoeuvring.

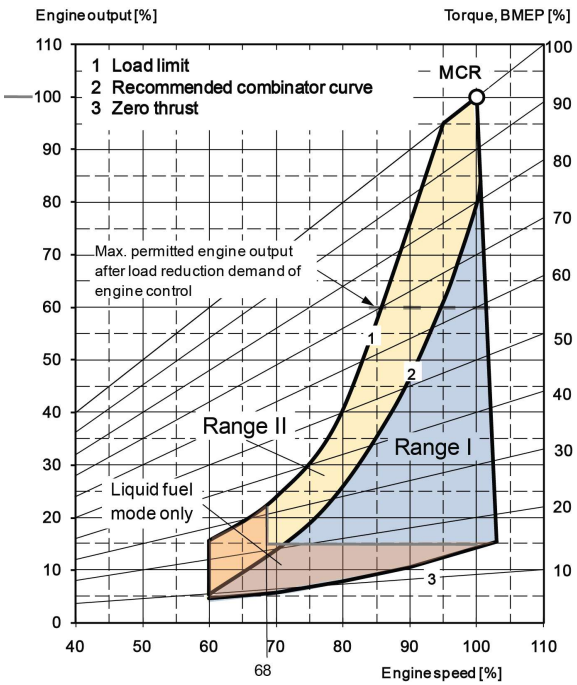

Figure 4: Operating range for controllable pitch propeller (CPP)

The engine characteristics described are defined according to ISO 15550: 2002; ISO 3046-1: 2002 standard operation conditions for engine rating:

Air temperature before turbocharger	298/25 [K/°C]
Total barometric pressure	100 [kPa]
Relative humidity	30 [%]
Cooling water temperature inlet intercooler (LT stage)	298/25 [K/°C]

Table 4: ISO standard conditions for engine rating

The engine is equipped with a turbocharger MAN TCA66-42 (TCA77-42), described later in *Chapter 3*, which provides the charge air. The charge air is conducted through a two-stage air cooler, a low-temperature stage LT and high-temperature stage HT, previous to be sent to the engine intake manifold. The charge air temperature can be regulated by shutting down the LT stage in order to avoid the accumulation of condensation water in low rate operation of the engine.

To control the charge air characteristics, a by-pass is located in the compressor, linking the outlet and inlet of the same in order to preheat the intake air. Also, between the air cooler and the intake manifold, a blow-off valve is located, which is used to control the fuel-air ratio in Diesel mode and as a throttle regulator in natural gas mode. The amount of air discharged by the blow-off valve depends on the working conditions and can differ considerably, but in *Table 5* an approximation is provided.

Power rate	100	85	75	50	25	%
$Q_{\text{blow-off air per cylinder}}$	1000	1400	1650	1200	300	kg/h
$Q_{\text{blow-off air engine total}}$	3,33	4,67	5,5	4	1	kg/s

Table 5: Blow-off air reference

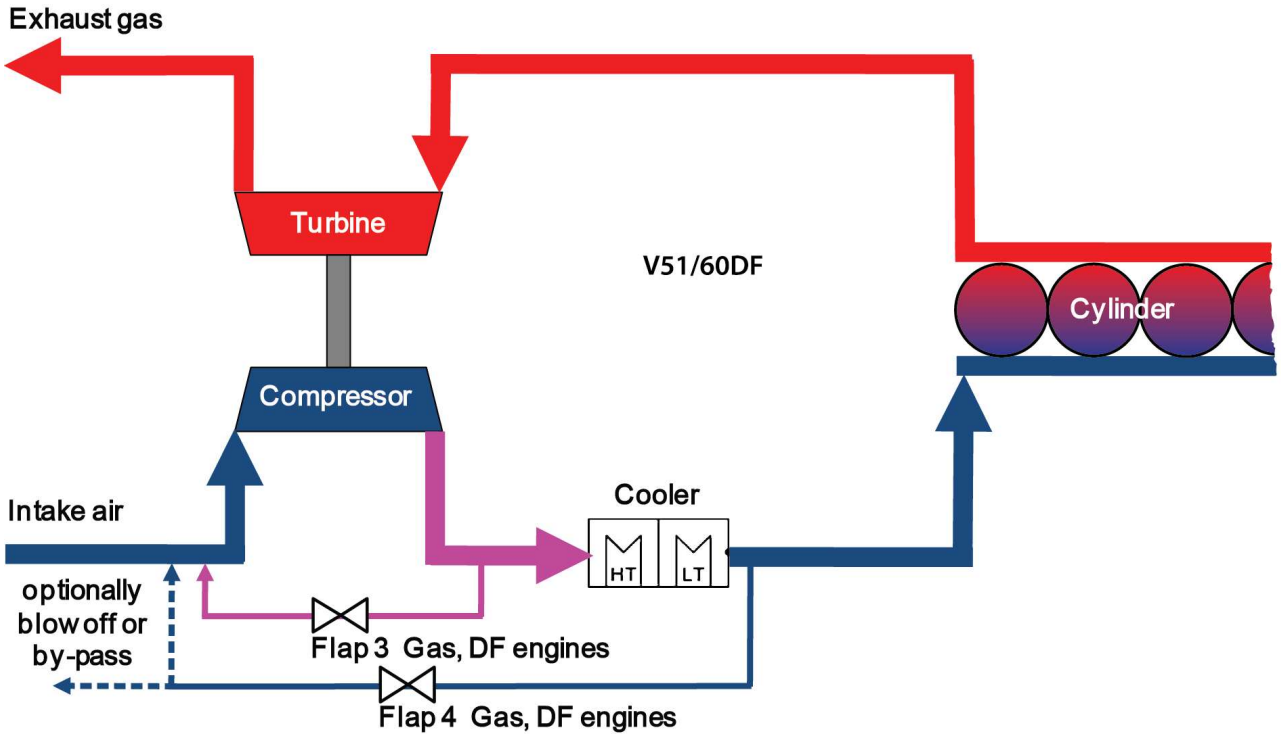

Figure 5: Charge air valves diagram

The next table shows the reference air and exhaust gas data at ISO conditions:

Power rate	**100**	**85**	**75**	**50**	**%**
Air data					
Temp. at compressor outlet	220	196	188	133	°C
Temp. at air cooler outlet	43	43	43	43	°C
Air flow rate	7,11	7,64	7,87	7,75	kg/kWh
Air flow rate (514 rpm)	23,7	21,65	19,45	12,92	kg/s
Charge air pressure (abs.)	4,29	3,76	3,6	2,34	Bar
Exhaust gas data					
Exhaust gas flow rate	7,3	7,83	8,06	7,95	kg/kWh
Exhaust gas flow rate (514 rpm)	24,33	22,19	20,15	13,25	kg/s
Temp. at turbine outlet	323	313	313	359	°C

Table 6: Air/exhaust gas values at ISO conditions - Electric propulsion/Auxiliary GenSet – Constant speed.

Power rate	**100**	**85**	**75**	**50**	**%**
Air data					
Temp. at compressor outlet	220	196	188	133	°C
Temp. at air cooler outlet	43	43	43	43	°C
Air flow rate	7,11	7,64	7,87	7,75	kg/kWh
Air flow rate (514 rpm)	23,7	21,65	19,45	12,92	kg/s
Charge air pressure (abs.)	4,29	3,76	3,6	2,34	Bar
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Exhaust gas flow rate (514 rpm)	24,33	22,19	20,15	13,25	kg/s
Temp. at turbine outlet	323	313	313	359	°C

Table 7: Air/exhaust gas values at ISO conditions - CPP propulsion. Constant speed

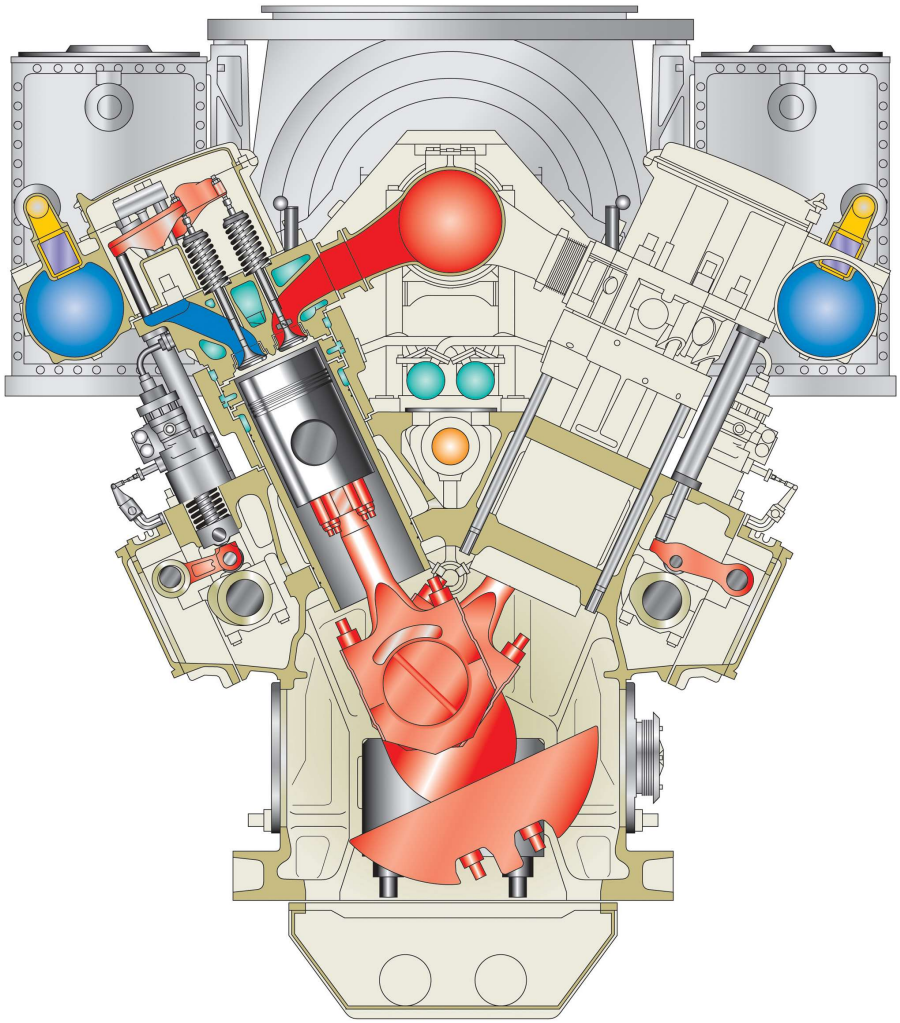

Figure 6: Engine cross section

3. The hybrid turbocharger

The basic design of the turbocharger consists on a single stage compressor coupled to a single stage turbine, where the gases coming from the engine exhaust manifold deliver power to the turbine and consequently drive the compressor to raise the intake air pressure. Thus, a hybrid turbocharger keeps the same elements but includes a third unit mounted in the same shaft, an electric generator which can be run thanks to the positive difference of the work obtained from the turbine and the work required by the compressor. In this section, the operating principals of the turbocharger used in this study will be described.

To suit the requirements of the MAN V12 51/60 DF engine, a MAN TCA66-42T turbocharger is used, a device composed by a single stage centrifugal compressor and a single stage axial turbine, designed to operate in 2-stroke (up to 11600 kW) or 4-stroke engines (up to 14800 kW). The turbocharger in marine applications will be equipped with a silencer in the intake of the compressor, inside of which the electric generator is assembled. This device works as an air filter too, and it generates a pressure loss on the intake air of approximately 200 Pa. The general view of the turbocharger and manifold connection can be seen in the next images:

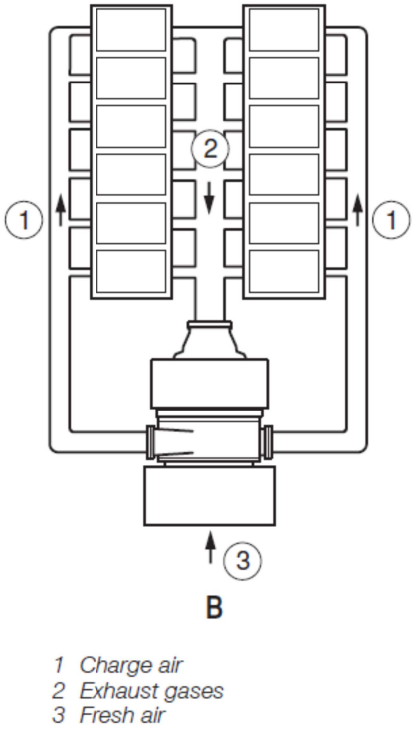

Figure 7: Turbocharger air and gas flow scheme

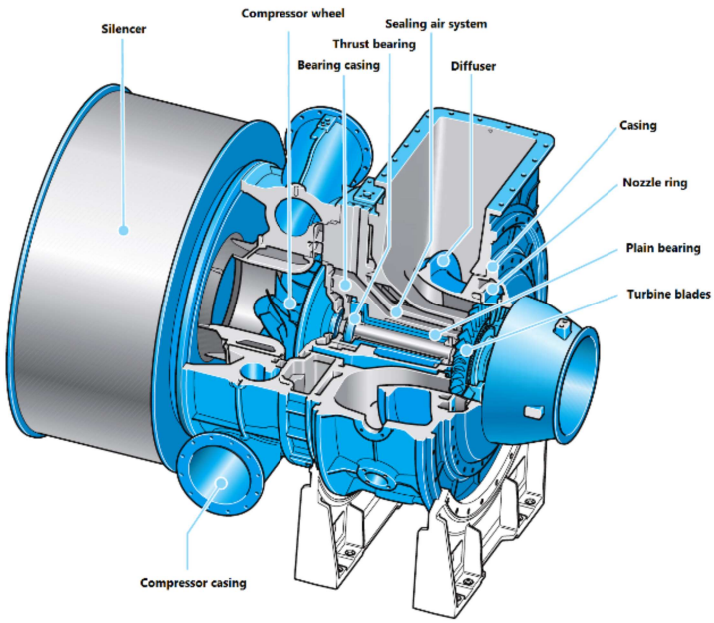

Figure 8: MAN TCA66-42T turbocharger

In any turbocharger, the main characteristics of both compressor and turbine are generally provided in the compressor and turbine maps, which will be described in the corresponding sections. However, as these diagrams for the TCA66-42T turbocharger are not supplied by MAN, the diagrams of a BBC (nowadays ABB) VTR 401 turbocharger has been used in the numerical model introduced in this study. The VTR 401 has the same compressor and turbine types and distribution but has a slightly smaller size than the original MAN model, so several scale factors and corrections will be set in the numerical model introduced in this study to adapt its performance to the MAN system.

3.1. Compressor

This is a single stage centrifugal compressor and is responsible for providing the charge air to the engine cylinders, rising the intake air pressure by a continuous conversion of the angular momentum transmitted to the gas into static pressure. The air enters the compressor in an axial direction through the center of the rotor which spins at high speed, and as it travels through the passages created by the rotor blades, it gains angular momentum and also static pressure due to the compression forced by centrifugal action. In the stator, the gas is decelerated reducing its dynamic pressure and increasing its static pressure again. The pressure increase is characterized by the compression ratio β, which is defined as the absolute outlet pressure divided by the absolute inlet pressure.

$$\beta = \frac{p_{outlet}}{p_{inlet}} = \frac{p_{2\text{tot}}}{p_{1\text{tot}}} \qquad (1)$$

The compression process is achieved by delivering work and will also release heat. The following expressions show the energy balance of the compressor, where Q is the heat transferred through the walls, m is the mass, v is the gas velocity, g the gravitational acceleration, z the height increase from a certain datum and h the gas enthalpy. Assuming a negligible variation on v, z and no Q transfer (adiabatic process), the work input in the compression is given by:

$$-Q + m \cdot \left(h_1 + \frac{v_1^2}{2} + g \cdot z_1 \right) = -W + m \cdot \left(h_2 + \frac{v_2^2}{2} + g \cdot z_2 \right)$$

$$-Q + m \cdot h_1 = -W + m \cdot h_2$$

$$W = m \cdot (h_2 - h_1) \qquad (2)$$

This last equation provides the required work to carry out the compression when it's considered isentropic (adiabatic and reversible) by counting the enthalpy *h* drop. However, a real compression process is irreversible and presents heat dissipation, because the gas is taken to high speeds and thus, it gets affected by viscous shear stress at the interface between the gas and the rotor blades, leakage through seals and heat transfer. To take this phenomenon into account and obtain the actual work required, an isentropic efficiency η_c is used. This efficiency is the ratio of the isentropic total enthalpy change to the polytropic or actual enthalpy change.

$$\eta_C = \frac{c_P \cdot (T_2 - T_1)}{c_P \cdot (T_{2'} - T_1)} = \frac{h_2 - h_1}{h_{2'} - h_1} \qquad (3)$$

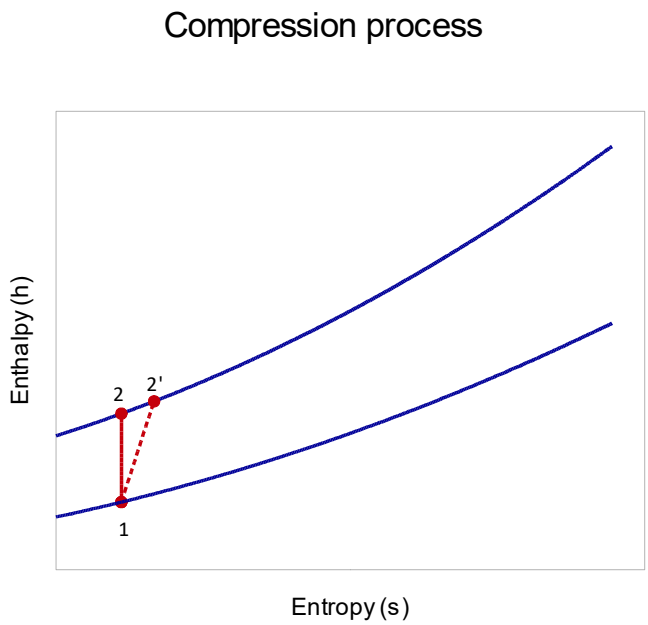

Figure 9: Compression process Entropy vs. Enthalpy diagram

By means of the thermodynamic equations for the isentropic process, the isentropic efficiency η_C can also be described using the pressure ratio and temperatures at the inlet and outlet of the compressor.

$$\eta_C = \frac{\beta^{\left(\frac{n-1}{n}\right)} - 1}{\left(\frac{T_{2'}}{T_1}\right) - 1} \qquad (4)$$

As this last expression defines, the efficiency is not a constant value on any given working condition, but it depends on several variables and characteristics of the compressor. To show how these variables affect the performance of the compressor, the compressor map or diagram (figure 10) is used, giving information about the efficiency, mass flow, compression ratio, and angular velocity. The vertical axis shows the compression ratio β and on the horizontal axis figures the air flow rate; i.e., the volume of air flowing through the compressor in a given period of time expressed in [m^3/s].

The map, defines several characteristics of the compressor. On its left boundary, the surge line is traced and represents the working conditions where surge will appear. Surge is given when counter-pressure is too high compared to the airflow, and the air downstream of the compressor inverts its direction. Then as the pressure in the outlet decreases, the flow goes back to its proper direction until the phenomenon is repeated, causing a continuous fluctuation in the airflow that can lead to damage in the compressor. To reduce the chances for this situation to happen, the by-pass in the compressor can be used, which drives some partially compressed air to the intake of the compressor, increasing the pressure in this zone.

The right boundary of the diagram shows the choking limit, which means that the compressor is working at very low discharge pressure and too high flow rates due to a lack of resistance. As there is very low resistance and the air flow is high, the air stream can reach transonic speeds. As can be seen on the map this zone has lower efficiency values but also can be dangerous for systems with multistage compressors due to the shock waves originated. This can be prevented by using anti-choke valves that restrict the discharge flow.

The speed lines represent constant speed condition points. As higher pressure rates are demanded, the speed lines have higher values, and the same happens in the case of flow rates. It can be also seen how the speed lines are closer to each other in the choke zone, which means that in this condition a flow increase will easily lead to over-speed. The isentropic efficiency of the compressor is shown on the diagram as concentric islands, regions in the map that allow knowing the efficiency for any point.

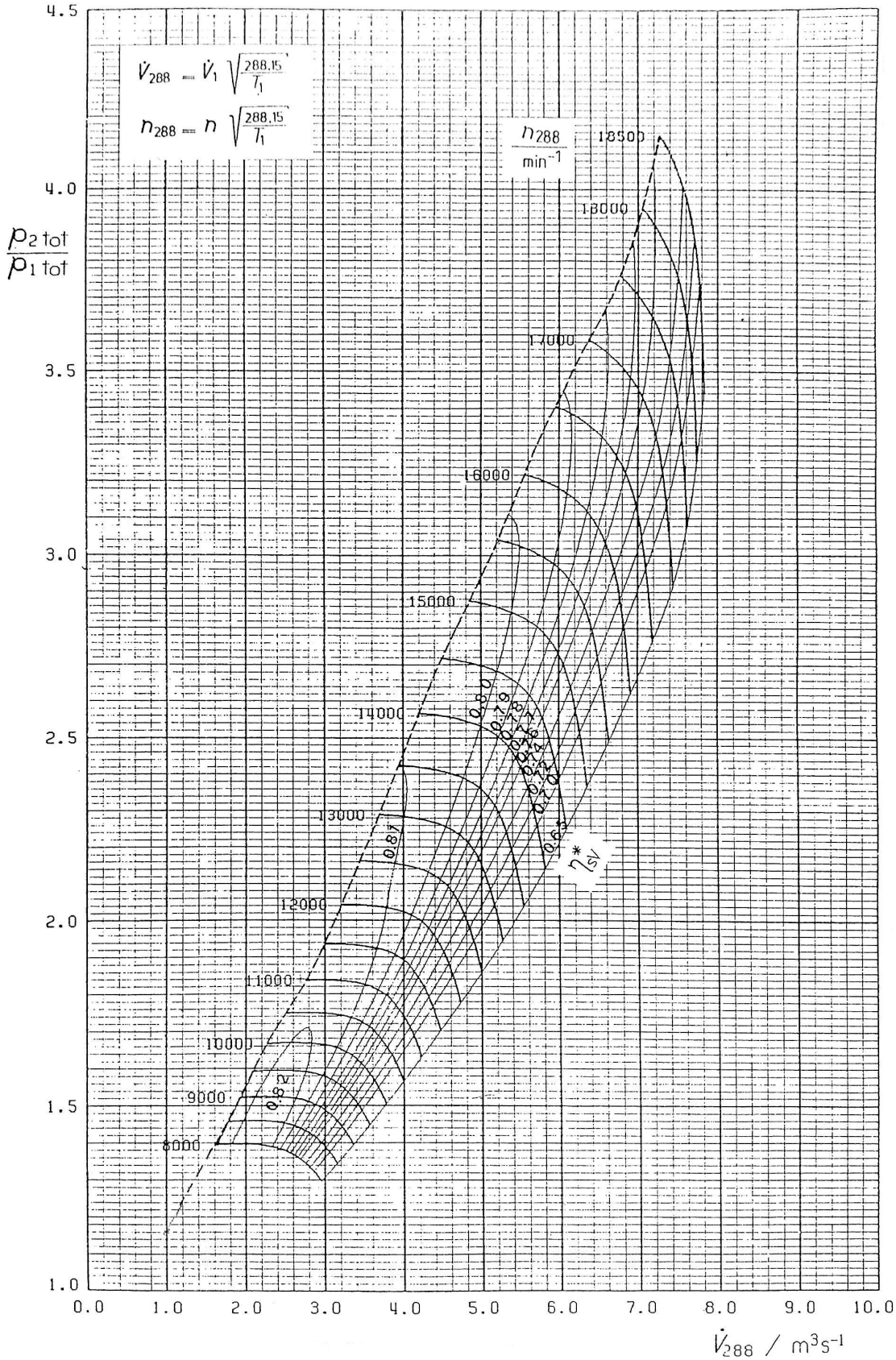

Figure 10: VTR 401 turbocharger compressor map

3.2. Turbine

The turbine is a device with the purpose of converting the energy of a fluid into useful work, in this case from the exhaust gases of the engine. Unlike the compressor, the turbine used for this study is axial, so the fluid keeps an axial direction from the inlet to the outlet of the turbine.

Contrasting to the operation of the compressor, the turbine obtains angular momentum from the kinetic energy and pressure of the fluid. From the Euler equations, it's known that when a particle with a certain mass *m* suffers a variation on its velocity *v* or on its rotation radius *r* during its movement around an axis, an angular momentum *L* is generated. If there is a mass flow suffering this variation, a torque *τ* is generated.

$$L = m \cdot \left(r_a \cdot v_a - r_b \cdot v_b \right)$$

$$\tau = \dot{m} \cdot \left(r_a \cdot V_a - r_b \cdot V_b \right) \qquad (5)$$

A single stage turbine is composed of two rings of blades, a stationary one known as the stator, and another one mounted on a free spinning shaft, known as the rotor. The incidence of the gas on the turbine is primarily in an axial direction, passing in the first instance through the stator blades, where it gets expanded and redirected towards the rotor blades, so its absolute velocity obtains a tangential component. Through the rotor, the flow is forced to turn again and its tangential velocity is modified. As there's a difference on the tangential velocity of the gas before and after each ring, a torque is produced, which is transmitted to the elements causing this variation, the blades. In the case of the rotor, this torque makes it spin at a given angular velocity *ω*, which produces a power *P*.

$$P = \omega \cdot \tau = \omega \cdot \dot{m} \cdot \left(r_a \cdot v_a - r_b \cdot v_b \right)$$

If the expression for the conservation of the energy and the conservation of angular momentum are equated, then the power can be expressed as follows:

$$\omega \cdot \left(r_a \cdot v_a - r_b \cdot v_b \right) = h_a - h_b$$

$$P = \dot{m} \cdot \left(h_a - h_b \right) \qquad (6)$$

The degree of reaction of a turbine, or a turbine stage, is an important factor which defines the ratio of total enthalpy drop developed exclusively in the rotor. For an impulse turbine, all the expansion and acceleration

of the gas are given exclusively in the stator passages, so the degree of reaction is zero and the rotor only produces a redirection of the flow at a constant thermodynamic state. If the rotor also produces expansion and acceleration of the gas, the reaction degree is different from zero and a reaction turbine is obtained. When the velocity triangles are properly compared between impulse and reaction turbines, it's possible to see that an impulse turbine rotor offers a higher variation on the tangential component, producing more power per stage. However, as the velocity is higher the viscous losses are larger and as there is no acceleration through the blade passages, the flow is more prone to boundary layer separation, reducing the efficiency of the stage. This can be prevented by adding some reaction degree, making the stage more efficient but less powerful. On the design of a turbine, there's always a certain degree of reaction, but its value can differ depending on the power and efficiency expected on the stage.

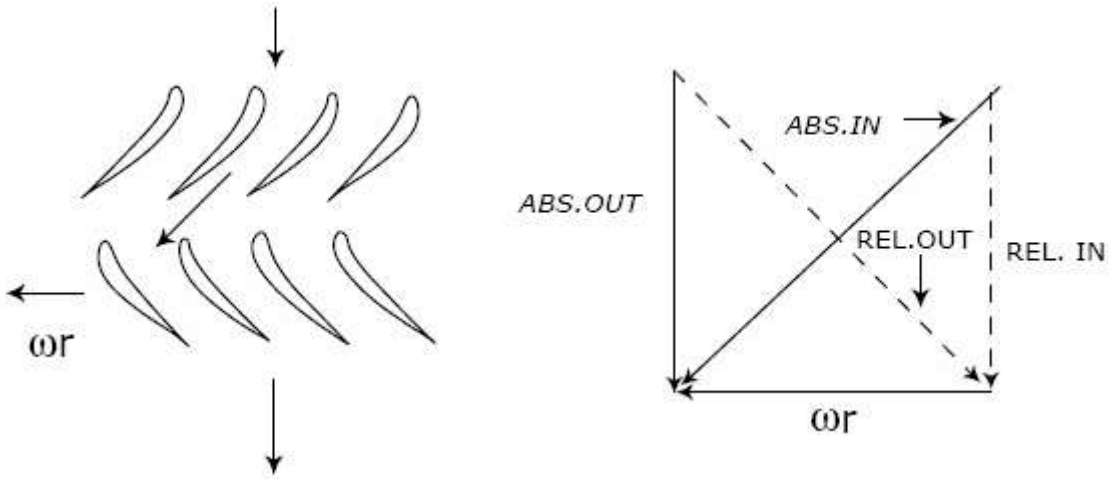

Figure 11: Reaction (50%) turbine velocities triangle.

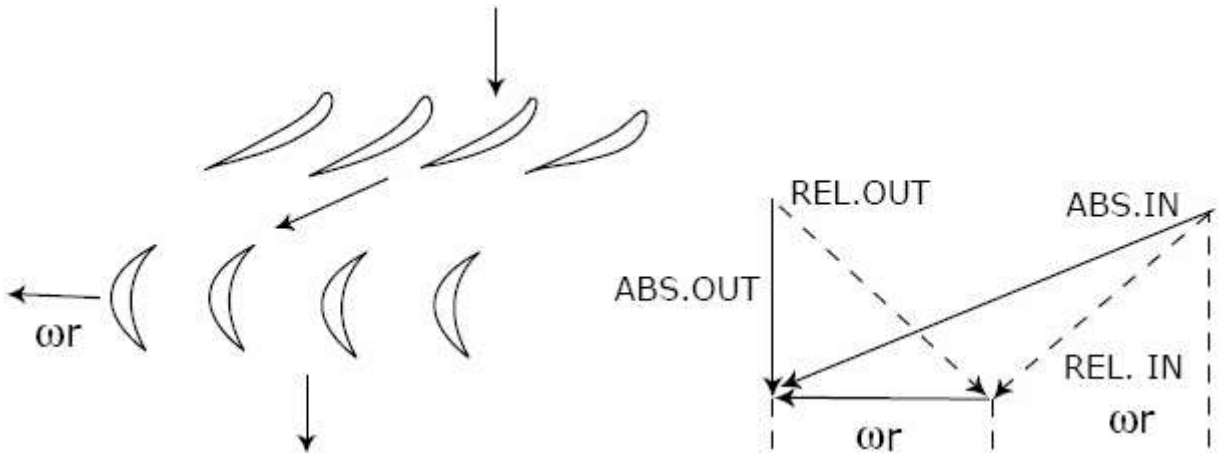

Figure 12: Impulse turbine velocities triangle.

Like it's been commented on the compressor, the enthalpy drop given through the turbine cannot define the real work produced by the turbine due to irreversibility, viscous friction, and heat dissipation. The ratio between the actual enthalpy change and the isentropic enthalpy change is given by the isentropic efficiency.

$$\eta_T = \frac{c_P \cdot (T_3 - T_{4'})}{c_P \cdot (T_3 - T_4)} = \frac{h_3 - h_{4'}}{h_3 - h_4} \qquad (7)$$

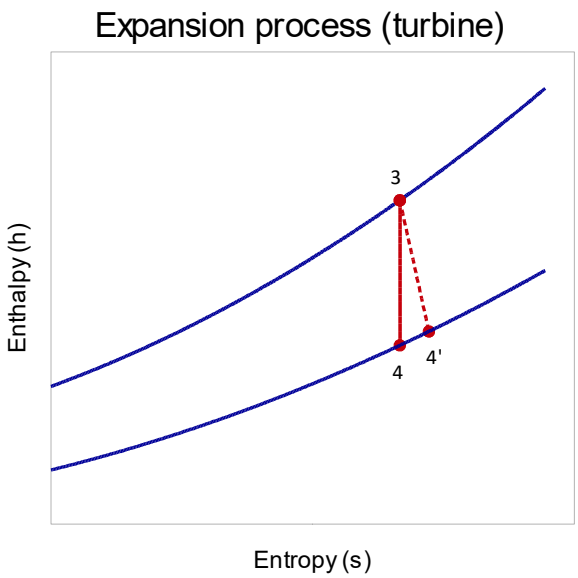

Figure 13: Expansion process (turbine) Entropy vs. Enthalpy diagram

The performance of the turbine is defined in the turbine map, a diagram showing the flow coefficient for friction and flow contraction α_T and turbine isentropic efficiency η_T as a function of the turbine expansion ratio π_T. The corrected mass flow rate is chosen as a reference in order to eliminate the influence of the turbine inlet pressure p_{T1} and temperature T_{T1}.

$$\pi_T = \frac{p_{inlet}}{p_{outlet}} = \frac{p_3}{p_4} \qquad (8)$$

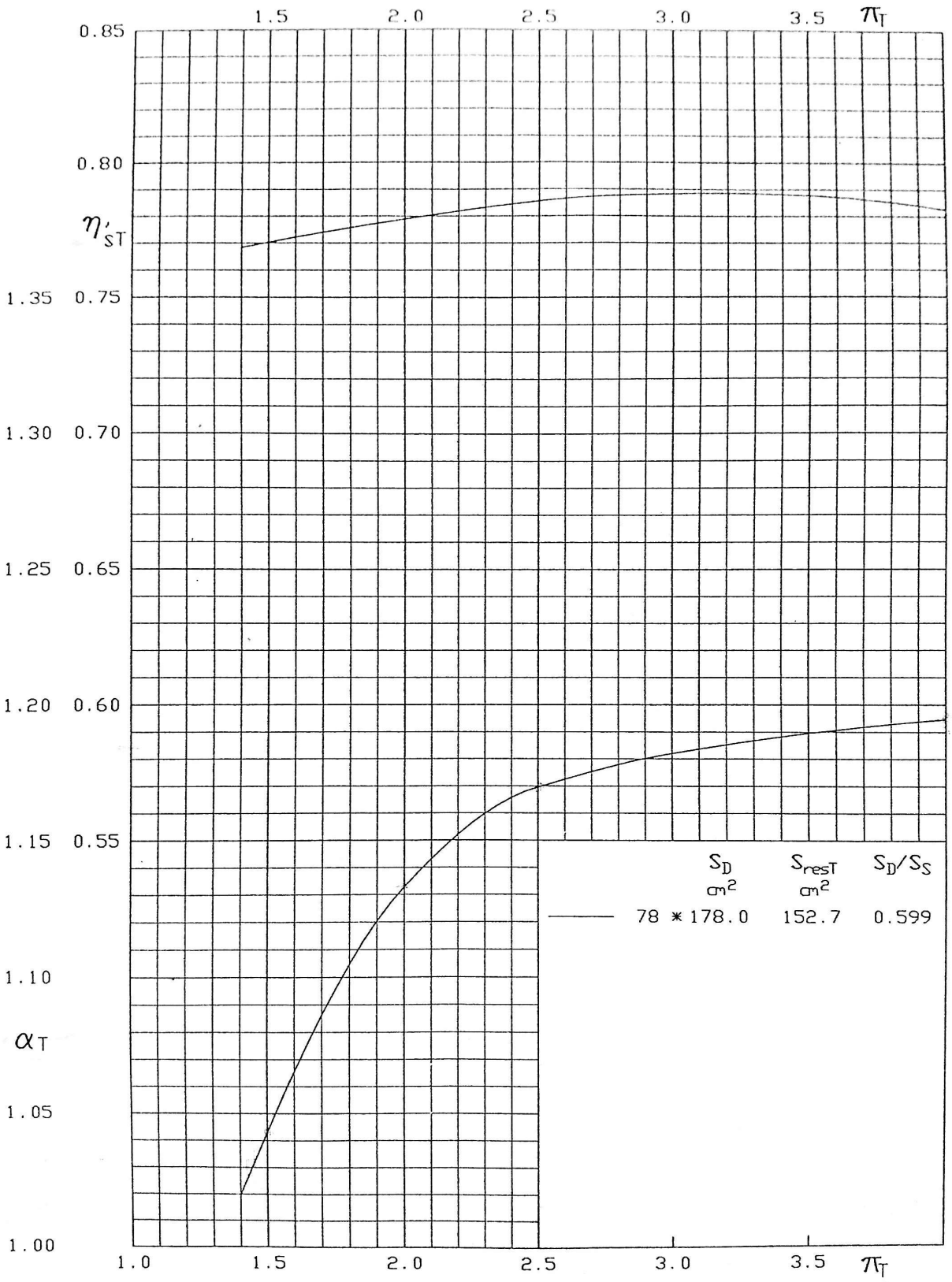

Figure 14: VTR 401 turbocharger turbine map

3.3. Generator

Due to the improvement in the efficiency of both the turbine and compressors along the years, the amount of work required to be extracted from the exhaust gases to obtain the necessary boost air has been reduced. This indicates that considerable quantities of energy remains in the hot gases and that it can be converted into useful work.

As it's been seen in the previously, the turbine delivers torque to the shaft which can provide the necessary work to perform the compression process. If a turbine supplying a higher torque is designed, the shaft will be able to move other components apart from the compressor, like for example the electric generator. The expected working conditions, imply that the hybrid turbocharger heat recovery system will require a generator capable of delivering a high power output and operate with high efficiency at high speeds.

The concept for the generator is based on the unit developed by Mitsubishi Heavy Industries, the MET83MAG, a three-phase permanent magnet synchronous generator (PMSG) with oil lubrication system and water refrigeration, although the ultimate characteristics of the generator on the numerical model do not correspond strictly to the real device on which it is based. In a PMSG, the excitation field is obtained not from a coil, but from a neodymium-boron-iron permanent magnet, presenting higher efficiency and higher torque to volume ratio compared to asynchronous machines. In this suggested design, the generator is placed in the center of the air filtering and silencer unit in order to avoid the high temperatures of the turbine shaft end and is connected to the turbocharger by means of a flexible joint to minimize bearing stresses due to misalignment. The magnets are assembled on a steel shaft conforming a four pole distribution and are secured by carbon fiber windings. These windings are surrounded by an aluminium water jacket to satisfy cooling requirements.

The output of the generator offers different voltage and a frequency depending on the rotating speed, so there must be a proper rectification previous to the connection to the ship electrical system. In the first instance, the three-phase AC current is converted into DC by an IGBT (Insulated Gate Bipolar Transistor) rectifier and later the corresponding alternating current with the required frequency and voltage is obtained with an inverter. This configuration allows to operate in reverse and use the generator as a motor to assist the turbocharger, although no further investigation will be developed about motor operation mode of the device on this document.

To describe the performance of the generator, it can be simplified in the equivalent circuit shown in *Figure 15*, so the power per phase *P* will correspond to expression (9), where *E* is the electromotive force, *k* the machine constant and *ω* is the rotational speed.

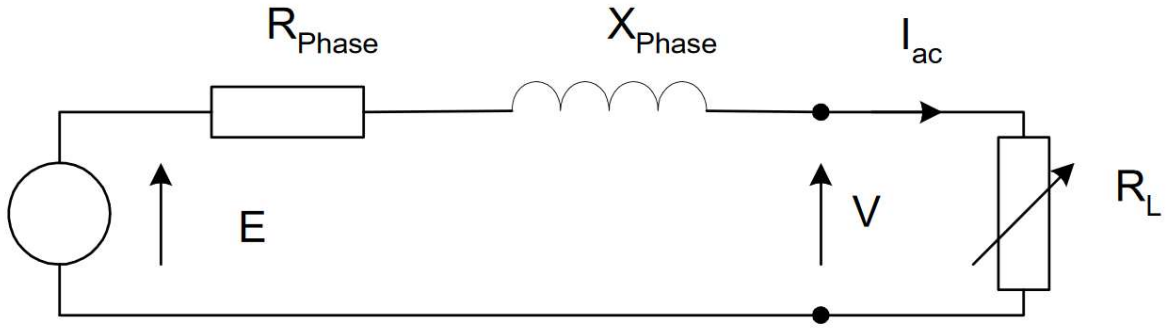

Figure 15: Generator equivalent circuit

$$P=\frac{E^2}{(R_{ph}+R_L)^2+X_{ph}^2}\cdot R_L=\frac{k^2\cdot\omega^2}{(R_{ph}+R_L)^2+X_{ph}^2}\cdot R_L \qquad (9)$$

Considering this expression, in a circuit where the load is kept constant, it is possible to announce that the power delivered by the generator will be a square function of the rotational speed *ω*. Also, it's well known that the power is the product of the rotational speed and the torque, so using a similar expression, the torque is directly proportional to the rotational speed:

$$P=\tau\cdot\omega$$

$$\tau=\frac{k^2\cdot\omega}{(R_{ph}+R_L)^2+X_{ph}^2}\cdot R_L \qquad (10)$$

By these means, the expected torque-speed and power-speed curves of a permanent magnet generator should be linear and parabolic lines respectively. The following images show an example of the torque-speed and power-speed lines of a medium speed commercial permanent magnet generator:

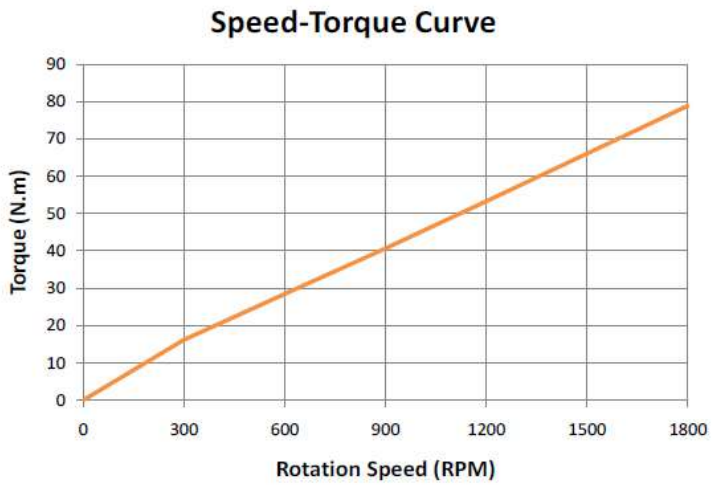

Figure 16: Permanent magnet synchronous generator speed-torque curve.

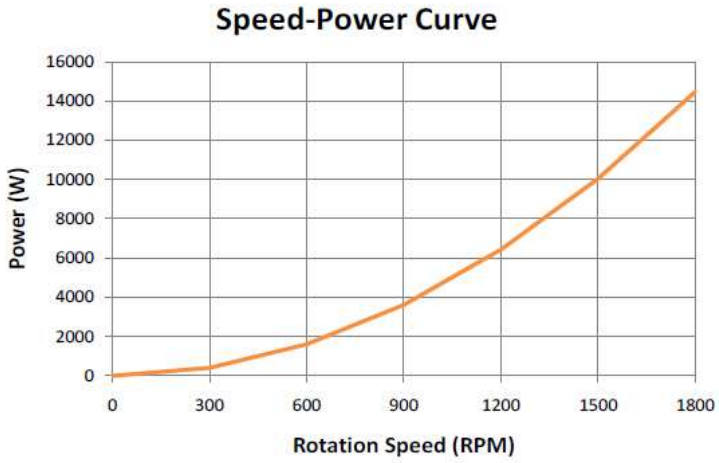

Figure 17: Permanent magnet synchronous generator speed-power curve.

Nevertheless, by changing the electrical load R_L felt by the generator, the resistive torque on the generator can be controlled. This load can be modulated by means of electronic control systems, allowing to increase or reduce the torque regardless of the rotation speed.

Finally, *Figure 18* shows the cross-section of the hybrid turbocharger with the generator assembled inside the filter-silencer unit.

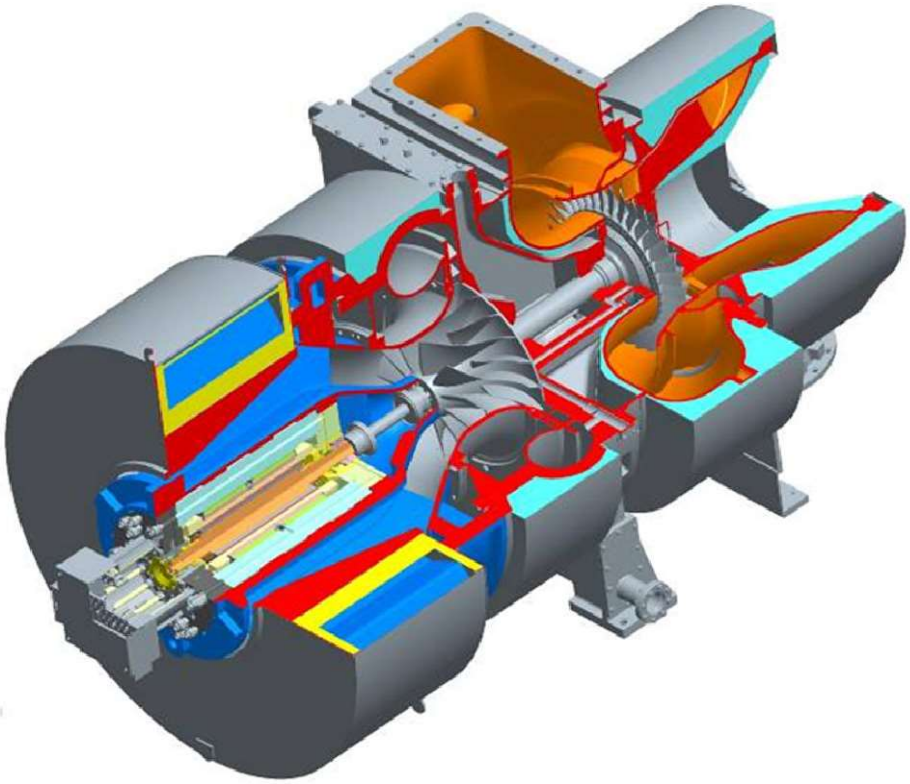

Figure 18: Cross-section of the hybrid turbocharger developed by Mitsubishi Heavy Industries

4. IMO regulations for Diesel engines pollution emissions at sea

The Internation Maritime organization (IMO) is the specialized agency of the United Nations that takes the role as the authority to set the global standards in safety, security and environmental matters for international shipping. The main international regulation about pollution in the marine environment is the "International Convention for the Prevention of Pollution from Ships", from 1973 and modified by the Protocol of 1978 (MARPOL 73/78). This Convention was signed on 17 February 1973 although it didn't have time to come effective. Following the great incidents regarding tankers occurred in 1976 and 1977, the 1978 Protocol absorbed the Convention, and this combined document entered into force on 2 October 1983.

The MARPOL is divided into 6 Annexes, each of them meant to establish the standards to prevent pollution from different sources:

- Annex I: Prevention of Pollution by Oil.
- Annex II: Prevention of Pollution by Noxious Liquid Substances in Bulk.
- Annex III: Prevention of Pollution by Harmful Substances Carried by Sea in Packaged Form.
- Annex IV: Prevention of Pollution by Sewage from Ships.
- Annex V: Prevention of Pollution by Garbage from Ships.
- Annex VI: Prevention of Air Pollution from Ships.

After setting a protocol for amending the Convention, in 1997 Annex VI was added, and it came effective on 19 May 2005 (the last Annex to come into force). This standard limits the emissions caused by the exhaust gases, pollution generated by incineration, emissions from volatile organic compounds and ozone depleting substances. Consequently, the design and installation of any engine or power plant for marine purposes will be strictly linked to the standards defined by this rule, which is divided into several Regulations in order to deal with the limitations in each specific field. The rules included in Annex VI are related to the following emissions:

- Ozone-depleting substances.
- Nitrogen oxides (NO_X).
- Sulphur oxides(SO_X) and particulate matter (PM).
- Volatile organic compounds (VOCs).
- Shipboard incineration.

Regarding the aspects that compromise a diesel engine in a ship, the pertinent Regulations belonging to Annex VI are listed and commented as follows:

Regulation 13: Nitrogen Oxides (NO_x): Nitrogen and oxygen are the two main elements that can be found in the intake air, being the primary sources for the generation of nitrogen oxides. Most of the oxygen is consumed during the combustion, but excess oxygen can be found depending on the fuel-air ratio. In the other hand, the nitrogen normally doesn't react during the combustion, although it can get oxidized if temperatures are high enough, forming nitric oxide (NO) nitric dioxide (NO_2), dinitrogen tetroxide (N_2O_4), nitric and nitrous acid mist, the quantities of which can differ depending on the temperature and also the nitrogen levels in the fuel. Generally, the origination of NO_X is more prone in low-speed engines, so diesel marine engines are a considerably important cause of the emission of these pollutants.

This Regulation, sets the limitations for the emission of the NO_X caused by ships, restrictions that every engine with a power output of more than 130 kW has to comply, although there are several distinctions depending on the date of construction of the ship on which the engine is installed and the rated engine speed in rpm's (*n*). These distinctions are divided into Tier I, II and III.

- Tier I: Sets the limits for those engines installed in ships constructed during 1st January 2000 and 1st January 2001, and they can not exceed:
    - 17.0 g/kWh when *n* is less than 130 rpm;
    - $45 \cdot n^{(-0.2)}$ g/kWh when *n* is 130 or more but less than 2000 rpm;
    - 9.8 g/kWh when *n* is 2000 rpm or more.

- Tier II: Sets the limits for those engines installed in ships constructed after 1st January 2001, and they can not exceed:
    - 14.4 g/kWh when *n* is less than 130 rpm;
    - $44 \cdot n^{(-0.23)}$ g/kWh when *n* is 130 or more but less than 2000 rpm;
    - 7.7 g/kWh when *n* is 2000 rpm or more.

- Tier III: Defines a limit for those ships operating in the areas defined as NO_X Emission Control Areas (NO_X ECA). These restrictions have to be assumed by engines installed on ships constructed after 1st January 2016 operating in the zones defined as the North American Emission Control Area and the United States Caribbean Sea Emission Control Area. From 1st of January 2021 the Baltic Sea Emission Control Area and the North Sea Emission Control Area will be included as NO_X ECA, so for those ships constructed after this date operating in these areas, it will be prohibited to exceed the Tier III limits. The ships operating in these areas can not exceed:
    - 3.4 g/kWh when *n* is less than 130 rpm;
    - $9 \cdot n^{(-0.2)}$ g/kWh when *n* is 130 or more but less than 2,000 rpm;
    - 2.0 g/kWh when *n* is 2,000 rpm or more.

<u>Regulation 14: Sulphur oxides (SO_x):</u> Sulphur oxides are a group of gases that include the sulphur dioxide SO_2 and sulphur trioxide SO_3. During the combustion process, the sulphur existing in the fuel suffers fast oxidation that generates SO_2, and some of it can oxidize as it goes through the combustion chamber and the exhaust manifold, causing the generation of SO_3. The amount of emissions depends on the combustion temperature, the pressure, the fuel ratio and the sulphur quantity in the fuel. Regulation 14 in Annex VI sets the restrictions for the mass to mass sulphur content ratio in the fuel used by the ship engine. These are:

- 4.50% m/m prior to 1 January 2012;
- 3.50% m/m on and after 1 January 2012;
- 0.50% m/m on and after 1 January 2020.

However, for the SO_X Emission Control Areas (SO_X ECA) of the North American Emission Control Area, the United States Caribbean Sea Emission Control Area, the Baltic Sea Emission Control Area and the North Sea Emission Control Area, the mass to mass sulphur content ratio limits are:

- 1.50% m/m prior to 1 July 2010;
- 1.00% m/m on and after 1 July 2010;
- 0.10% m/m on and after 1 January 2015.

Regulation 20: Attained Energy Efficiency Design Index (EEDI): In the combustion process of hydrocarbons like diesel fuels, the main products of the oxidation reaction are H_2O and CO_2. Carbon dioxide has a notorious effect on global warming, and although there are other gases like methane or nitrous oxide with higher global warming potential, carbon dioxide is the main substance used to trace the greenhouse effect caused by human activities due to the big quantities of this gas released to the atmosphere during decades. This is the reason why the IMO has created a system to know, and thus set a limit, to the relative amount of CO_2 released to the atmosphere by a vessel. The attained EEDI is the technical measure that defines an energy efficiency level for a given ship, and it's expressed as a comparison of the amount of CO_2 emitted per each tonne transported and mile.

$$EEDI = \frac{grams\ \ of\ \ CO_2}{Tons\ \cdot\ mile} \qquad (11)$$

The attended EEDI then is the actual index calculated for the given ship. The index is calculated as indicated in (11), which will be more extensively commented in *Chapter 8.* of this document:

$$EEDI = \frac{\left(\prod_{j=1}^{n} f_i\right)\cdot\left(\sum_{i=1}^{nME} P_{ME}\cdot C_{FME}\cdot SFC_{ME}\right)+\left(P_{AE}\cdot C_{FAE}\cdot SFC_{AE}\right)+\left(\left(\prod_{j=1}^{n} f_j\cdot\sum_{i=1}^{nPTI} P_{PTI} - \sum_{i=1}^{neff} f_{eff}\cdot P_{AEeff}\right)\cdot C_{FAE}\cdot SFC_{AE}\right)-\left(\sum_{i=1}^{neff} f_{eff}\cdot P_{eff}\cdot C_{FME}\cdot SFC_{ME}\right)}{f_i\ \cdot\ f_c\ \cdot\ f_j\ \cdot\ Capacity\ \cdot\ f_w\ \cdot\ V_{ref}} \qquad (12)$$

Regulation 21: Required Energy Efficiency Design Index: This index is the one used as a regulatory limit for any ship for which the EEDI has been calculated. On the calculation of this index, a reference line will be obtained to represent the reference EEDI as a function of the ship type and size. Then a reduction factor will be defined to reduce the EEDI line during future years in order to make the limit more restrictive. The method for its calculation is detailed in *Chapter 8.* of this document.

5. Model description

On the development of engineering projects, the numerical analysis method allows to generate a virtual reproduction of the projected systems, in order to obtain results capable of predicting the behaviour of such systems prior to the realization of a physical prototype. This is done by creating a sequence of algorithms that reflect the physical laws that rule the conduct of the developed system (in this case, the hybrid turbocharged diesel engine) known as numerical model.

The model in first instance is programmed to simulate the operation of the MAN V12 51/60 DF engine in it's original setup as defined by the manufacturer, but simultaneously includes the necessary elements required to extract the desired results for this study, simulating the implementation of the blow-off valve for Diesel mode (not contemplated by the manufacturer) and the hybrid turbocharger.

The numerical model used during the development of this study is a combination of algorithms and block diagrams generated in Matlab® and Simulink® softwares, which offer a dynamic simulation of the power-plant and its subsystems, providing a complete set of data referred to the characteristics of the fluids and the working setup of the engine as results. This includes shaft speeds, air, fuel and gas flows, pressures and temperatures or valve timing among others. In order to do so, it takes the selected engine rotational speed and fuel flow as main external input data. Nevertheless, the model also depends on several data tables and other constants, and its performance will be modified also by variations on the blow-off valve or the torque applied on the turbocharger shaft by the electric generator, depending on the simulator configuration being used.

The code lines of the numerical model are included in ANNEX II, but in this chapter, the algebraic and differential equations that define the model are commented. The model is capable of describing the performance of the several elements that belong to the power plant, and which are distributed in modules:

- Cylinder.
- Turbocharger compressor.

- Turbocharger turbine.
- Generator.
- Turbocharger shaft dynamics.
- Intercooler.
- Intake manifold.
- Exhaust manifold.

5.1. Cylinder

The cylinder is modelled using a conditional loop that carries out the calculations in an iterative process. The whole 4 stroke cycle is divided into 720 degrees and all the characteristics of the air and gases are calculated as a function of the rotation angle of the crankshaft so, for each step of one degree, the characteristics are recalculated. At the same time, it shows 5 different stages for each cycle, where some parts of the algorithm used to carry out the calculations are changed in each phase. These stages are intake, compression, fuel injection, combustion and exhaust. However, previous to calculate the cycle phases, several physical and geometric conditions are also defined.

The displacement of one cylinder V_{cyl} is calculated from the piston bore or diameter D and piston stroke C provided by the manufacturer:

$$V_{cyl} = \pi \cdot \frac{D^2}{4} \cdot C \qquad (13)$$

The volume of the combustion chamber V_{CC} is calculated, being CR engine compression ratio:

$$V_{CC} = \frac{V_{cyl}}{CR - 1} \qquad (14)$$

Provided that N is the engine speed in min^{-1}, the mean speed of the piston u_P is:

$$u_P = \frac{2 \cdot N \cdot C}{60} \qquad (15)$$

The initialization exhaust pressure p_0 depends on the engine intake pressure p_I, although after the first

iteration it will respond to another calculation and will be designated as p_3.

$$p_o = \frac{p_I}{1,2} \qquad (16)$$

To calculate the volume gained by the cylinder in the moment of the intake valve closing (IVC), the distance from the TDC to the piston has to be known. Being θ_{IVC} the angle in radians for the complete intake valve closure, R the crankshaft radius at the center of the crank pin and L_r the connecting rod length, this is:

$$X_{IVC} = R \cdot \left(\left(1 - \cos\theta_{IVC}\right) + \frac{L_r}{R} \cdot \left(1 - \sqrt{1 - \left(\frac{L_r}{R}\right)^2 \cdot \sin\theta_{IVC}} \right) \right) \qquad (17)$$

$$R = \frac{C}{2} \qquad (18)$$

So the volume of the cylinder at the moment of the intake valve closure is:

$$V_{IVC} = \pi \cdot \frac{D^2}{4} \cdot X_{IVC} \qquad (19)$$

Based on the formula developed by *Hardenberg and Hase* [7] to define the time delay for the mixture to ignite, the angle of combustion delay θ_{CD} in degrees is determined by the pressure and temperature at the end of the compression in the theoretical cycle approximation, so the angle is:

$$\theta_{CD} = \left(0,36 + 0,22 \cdot u_p\right) \cdot e^Z \qquad (20)$$

$$Z = \left[\frac{61,884}{NC + 25} \cdot \left(\frac{1200}{T_{II}} - 0.582 \right) + \frac{6,85}{\left(10 \cdot p_{II} - 12,4\right)^{0,63}} \right] \qquad (21)$$

The compression is simulated as an adiabatic process, so considering an adiabatic coefficient γ, the air temperature and air pressure after the compression T_{II} and p_{II} are defined as follows, where T_I and p_I are the temperature and pressure at the beginning of the intake stroke, which correspond to the conditions in the intake manifold.

$$T_{II} = T_I \cdot CR^{(\gamma - 1)} \qquad (22)$$

$$p_{II} = p_I \cdot CR^{\gamma} \qquad (23)$$

Then an injection start angle θ_{IS} is obtained from a table as a function of the engine rpm's, so the combustion start angle is:

$$\theta_{CS}=\theta_{IS}+\theta_{CD} \qquad (24)$$

The angle where the combustion ends θ_{CE} is set by adding the angle that defines the length of the combustion θ_{CL} to the combustion start angle:

$$\theta_{CE}=\theta_{CS}+\theta_{CL} \qquad (25)$$

Another angle for the fuel injection duration is defined as θ_{IL}. Every angle involved is described as follows:

- Fuel Injection start θ_{IS};
- Combustion delay θ_{CD};
- Combustion start θ_{CS};
- Fuel injection length θ_{IL}.
- Combustion length θ_{CL};
- Combustion end θ_{CE};
- Intake valve closing θ_{IVC}.

RPM	θ_{IS}
154.2	-13
205.6	-14
257.0	-15
308.4	-16
359.8	-18
411.2	-19
462.6	-19
514.0	-20

Table 8: Fuel injection start depending on engine speed.

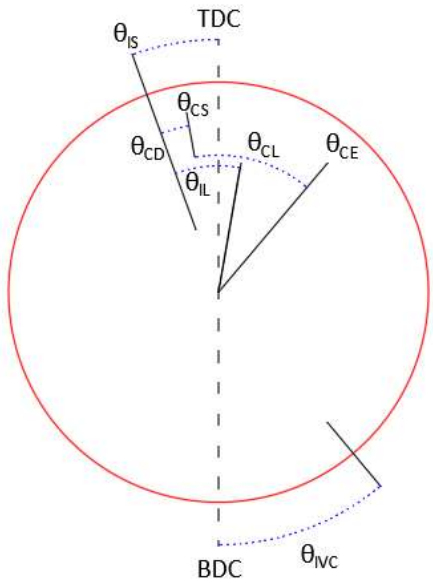

Figure 19: Injection and intake valve timing.

From the general gas equation, the air density ρ in the intake manifold is obtained.

$$\rho = \frac{p_I}{R_{gas} \cdot T_I} \qquad (26)$$

The amount of fuel consumed by the engine and used in the combustion process m_F, is calculated in (27), where m_{Fproj} is the maximum fuel mass injected per cycle as designated in the engine project guide (0,1476 kg), and m_{Fper} is the input corresponding to the selected fuel percentage to use.

$$m_F = \frac{m_{F\,proj}}{100} \cdot m_{F\,per} \qquad (27)$$

The variable $m_{F\,ciycle}$ defines the actual fuel quantity injected in each cylinder. To calculate the amount of air that will be used in the combustion, the following method is used.

$$m_{Fcycle} = \frac{m_F}{12} \qquad (28)$$

$$m_{A\,proj\,cycle} = \lambda_V \cdot \rho \cdot \eta_V \cdot V_{cyl} \cdot V_r \qquad (29)$$

$$\lambda_V = \frac{real\ air\ mass}{ideal\ air\ mass} = \frac{\rho \cdot V_{apparent}}{\rho \cdot V_{cylinder}} \qquad (30)$$

$$V_r = \frac{V_{IVC}}{V_{cyl}} \qquad (31)$$

In this equation, the cylinder filling coefficient λ_V, relates the theoretical amount of air that fits in the cylinder and the real air that actually enters in it due to the intake conditions. The η_V indicates the air renewal efficiency, the amount of real fresh air that will be in the cylinder at the end of the intake stroke due to the presence of previous combustion gases. Finally, V_r expresses the ratio between the displacement and the volume at the intake valve closure point.

To obtain the total mass of the exhaust gases for each cycle:

$$m_{GAS\,cycle} = m_{F\,cycle} + m_{A\,proj\,cycle} \qquad (32)$$

In the next lines, the method used by the simulator to reproduce the thermodynamic cycle in the engine is described:

5.1.1. Intake

This stage calculates the air conditions in the cylinder from the top dead center until the point where the intake valve closes. Right at the beginning of the intake stroke, the mass of the remaining combustion gas from the previous cycle m_{rem} is calculated, being p_{IV} and T_{IV} the exhaust pressure and temperature :

$$m_{rem}=\frac{p_{IV}\cdot V_C}{R_{gas}\cdot T_{IV}} \qquad (33)$$

As long as the piston keeps moving the volume of the cylinder will vary, so the volume is calculated as follows:

$$X(\theta)=R\cdot\left((1-\cos\theta)+\frac{L_r}{R}\cdot\left(1-\sqrt{1-\left(\frac{L_r}{R}\right)^2\cdot\sin\theta}\right)\right) \qquad (34)$$

$$V(\theta)=\frac{D^2}{4}\cdot\pi\cdot X(\theta)+V_{CC} \qquad (35)$$

While the piston moves along the liner, air with equal density as the air in the intake manifold fills the cylinder, which makes possible to calculate the mass and temperature:

$$m(\theta)=\frac{V(\theta)-V_{CC}}{V_{IVC}}\cdot\lambda_V\cdot\rho\cdot\eta_V\cdot V_{cyl}\cdot V_r \qquad (36)$$

In order to simulate the gas conditions that make possible to predict the performance of the engine, a thermal model is required. In this case, the calculations are based on the ideal gas law, and the process is considered as non-adiabatic, where the heat exchange is estimated by several methods, described in the following paragraphs, allowing to obtain the instantaneous heat flow.

$$T_g(\theta)=\frac{p(\theta)\cdot V(\theta)}{\left(m_{rem}+m(\theta)\right)\cdot R_{gas}} \qquad (37)$$

The instantaneous heat flow makes possible to know the heat transferred from the gas to the cylinder walls as a function of time, or as in the case of the model, as a function of the angle rotated by the crankshaft.

To obtain the heat transfer during the stroke, the method defined by *Woschni-Annand* is used, where the instantaneous heat flow through the cylinder liner, piston and cylinder head is calculated only as a forced convection process, although it's properly incremented to take into account the radiation process. In the first instance, the heat transfer coefficient h_i is defined as:

$$h_i(\theta)=C_1\cdot D^{-0,2}\cdot\left(p(\theta)\right)^{0,8}\cdot T_g^{-0,53}\cdot u(\theta)^{0,8} \qquad (38)$$

And the *u* number that defines the in-cylinder gas velocity, is considered to be proportional to the piston speed during the compression and exhaust stroke.

$$u(\theta)=C_2\cdot u_p \qquad (39)$$

The C_1 and C_2 coefficients depend on the stroke that is being considered so during the intake stroke, their values are 3,26 and 6,18 respectively. Once the heat transfer coefficient is obtained, the heat flow $\dot{q}$ can be calculated following *Newton*'s heating law.

$$\dot{q}=h_i\cdot\left(T_g-T_i\right) \qquad (40)$$

Then, the whole heat transferred loss from the gas to the cylinder walls is:

$$\frac{dQ_r}{d\theta}=\frac{\frac{\pi\cdot D^2}{2}+\pi\cdot D\cdot\frac{X(\theta)+C}{CR-1}}{2\cdot\pi\cdot N\cdot h_i(\theta)\cdot\left(T_g-T_l\right)} \qquad (41)$$

As the piston moves along the cylinder, the increment on the volume as a function of the rotated angle is not constant, and is defined by the following expression:

$$\frac{dV}{d\theta}=\frac{V_{cyl}}{2}\cdot\frac{\sin\theta+\frac{R\cdot\sin(2\cdot\theta)}{L_r}}{2\cdot\sqrt{1-\left(\frac{R}{L_r}\right)^2\cdot\sin^2\theta}} \qquad (42)$$

To establish the pressure rise as a function of the rotated angle, the calculation is based on the 1st law of thermodynamics, where the convective heat transferred also affects to the pressure increment. It has to be considered that the heat capacity ratio γ is not constant, but it can be calculated using the experimentally obtained expression (45).

$$dQ-dW=dU \qquad (43)$$

$$dQ - p \cdot dV = m \cdot c_v \cdot dT$$

$$dQ - p \cdot dV = \frac{c_v}{R} \cdot (p \cdot dV - V \cdot dp)$$

$$\frac{dQ}{d\theta} - \left(1 + \frac{c_v}{R}\right) \cdot p \cdot \frac{dV}{d}\theta = \frac{c_v}{R} \cdot V \cdot \frac{dp}{d\theta}$$

$$\frac{dp}{d\theta} = \frac{\gamma - 1}{V} \cdot \frac{dQ}{d\theta} - \gamma \cdot \frac{p}{V} \cdot \frac{dV}{d\theta}$$

$$\frac{dp}{d\theta} = \frac{-(\gamma(\theta) - 1)}{V(\theta)} \cdot \frac{dQ_r(\theta)}{d(\theta)} - \frac{\gamma(\theta)}{V(\theta)} \cdot p(\theta) \cdot \frac{dV(\theta)}{d(\theta)} + \frac{\gamma(\theta) - 1}{V(\theta)} \cdot \lambda_V \cdot \rho \cdot \eta_V \cdot c_p \cdot T_l \qquad (44)$$

$$\gamma(\theta) = 1{,}4 - \frac{T_g(\theta) - 300}{12000} \qquad (45)$$

To obtain the pressure that will be used in the next step, the increment is added to the previous iteration pressure:

$$p(\theta + 1) = p(\theta) + \frac{dp(\theta)}{d(\theta)} \cdot \frac{\pi}{180} \qquad (46)$$

5.1.2. Compression

For the simulation of the compression process in the cylinder, the angle considered ranges from the intake valve closing θ_{IVC} to the injection start angle θ_{IS}, so only air is considered to be compressed during this stage. The calculations are performed using the same process as in the intake stage, although as it is defined in the *Woschni-Annand* method, the C_2 coefficient in *(39)* is 2,28. In the compression stage, expression *(44)* is replaced by (47) to calculate the pressure increase, whereas expressions the procedures described on *(34)*, *(35)*, *(38)*, *(41)*, *(42)*, *(45)*, *(46)* are kept.

$$\frac{dp}{d\theta} = \frac{dQ_r(\theta)}{d(\theta)} \cdot \frac{\gamma(\theta) - 1}{V(\theta)} - p(\theta) \cdot \frac{dV(\theta)}{d(\theta)} + \frac{\gamma(\theta)}{V(\theta)} \qquad (47)$$

5.1.3. Injection

At the injection stage, the remaining exhaust gases of the previous cycle and the fuel are present at the same time, so the mass of both have to be considered. The fuel is injected during a period defined by the angle θ_{IL} which starts at θ_{IS}. However, the stage considered in this section only takes into account that part of the injection where the combustion process has not started yet, so it will develop the calculation during the combustion delay angle θ_{CD} (until the combustion start angle θ_{CS} is reached). In the calculations, variable *m* is utilized now for the instant fuel mass injected, and although the fuel is liquid, is treated as a gas.

$$m(\theta)=\frac{\theta-(361+\theta_{IS})}{\theta_{IL}}\cdot m_{Fcycle} \qquad (48)$$

$$T_g(\theta)=\frac{p(\theta)\cdot V(\theta)}{\left(m_{rem}+m_{A\,projcycle}+m(\theta)\right)\cdot R_{gas}} \qquad (49)$$

$$\frac{dp}{d\theta}=\frac{-\left(\gamma(\theta)-1\right)}{V(\theta)}\cdot\frac{dQ_r(\theta)}{d(\theta)}-\frac{\gamma(\theta)}{V(\theta)}\cdot p(\theta)\cdot\frac{dV(\theta)}{d(\theta)}+\frac{\gamma(\theta)-1}{V(\theta)}\cdot\frac{m_{Fcycle}\cdot H_{injected\,fuel}}{\theta_{IL}\cdot\frac{\pi}{180}} \qquad (50)$$

Expressions *(34)*, *(35)*, *(38)*, *(41)*, *(42)*, *(45)*, *(46)* and *Error: No se encuentra la fuente de referencia* are kept for the calculation of the rest of parameters during this stage.

5.1.4. Combustion

This stage is extended from the combustion start point to the exhaust valve opening point, and includes two sub-stages; combustion while the injection is maintained, and combustion when the injection has ceased. The parameters calculated with expressions *(34)*, *(35)*, *(38)*, *(42)*, *(45)* and *(46)* are obtained with the same procedure. In this stage, the heat increment coming from the fuel combustion is introduced, and the behaviour of the process is modelled following the *Wiebe* method. The equation that *Wiebe* defined, provides the burnt fraction *f* to obtain the fuel heat release as a function of the main combustion parameters. These are combustion start θ_{CS} [rad], combustion length θ_{CL} [rad], the form factor m_w and the efficiency factor *a*. These two factors have a very significant influence on the burnt fraction, and several studies offer different values obtained from experimental investigations, but for this model 1,2 and 6,9 are chosen for m_w and *a* respectively.

$$f = 1 - e^{-a \cdot y^{m_w+1}} \qquad (51)$$

$$y = \left(\frac{\theta - \theta_{CS}}{\theta_{CL}} \right) \qquad (52)$$

Now if the expression is multiplied by the lower calorific value H_i (42700 kJ/kg) and by the amount of fuel introduced in the cylinder each cycle, the formula that defines the released heat as a function of the crankshaft angle is obtained. If such expression is derived, it is possible to calculate the heat release differential:

$$\frac{dQ_b}{d\theta} = a \cdot (m_w + 1) \cdot y^{m_w} \cdot \exp\left[- a\, y^{m_w+1} \right] \cdot m_{F\,cycle} \cdot H_i \qquad (53)$$

In this stage, so as to define the heat loss, the radiative heat transmission takes considerably higher importance compared to the previous phases of the cycle. This means that not only the convective phenomenon has to be taken into account, but also the radiative. As it's been done in the previous stages, to calculate the loses, the method defined by *Woschni-Annand* is used, but the formula that expresses the in-cylinder gas velocity *u* is no longer proportional to the piston speed and is treated as follows:

$$u(\theta) = C_2 \cdot u_p + C_{3.} \frac{V(\theta) \cdot T_0}{p_0 \cdot V_0} \cdot \left(p(\theta) - p_{tr} \right) \qquad (54)$$

$$p_{tr}(\theta) = p_0 \cdot \left(\frac{V_0}{V(\theta)} \right)^n \qquad (55)$$

Where p_{tr} is the instantaneous pressure in the cylinder in case no combustion is given, and for *n*, a mean value of 1,32 is set. Also, T_0, p_0 and V_0 are the temperature, pressure and volume at the reference conditions, in this case, at the intake valve closing point.

Now the heat transfer coefficient can be calculated as defined by *Wiebe*, and thus the convective heat flow. However, in the combustion stage, the loss by heat transfer is considerably affected by the radiative component generated, so such component has to be added. In order to do so, the model calculates the radiative heat using the *Stefan–Boltzmann* law.

$$\dot{q} = \varepsilon \cdot \sigma_0 \cdot \left(T_g^4 - T_i^4\right) \qquad (56)$$

The empiric coefficient ε is the non-dimensional emissivity coefficient: 0,6; σ_0 is the Stefan–Boltzmann constant: 5,6703·10^{-8} [W/m^{2}K^4], T_g is the combustion gas temperature, and T_i is the cylinder walls or liner temperature. So once the radiative heat transfer is known, the total transmitted heat flow can be described as a function of the rotated angle increment in the following expression:

$$\frac{dQ_r}{d\theta} = \frac{\frac{\pi \cdot D^2}{2} + \pi \cdot D \cdot \frac{X(\theta) + C}{CR - 1}}{2 \cdot \pi \cdot N \cdot \left(h_i(\theta) \cdot \left(T_g - T_l\right) + \varepsilon \cdot \sigma_0 \cdot \left(T_g^4 - T_i^4\right)\right)} \qquad (57)$$

<u>Combustion during injection</u>

This is the first sub-stage and defines the phenomena from the combustion start point until the injection endpoint, so it lasts all along the angle that describes the combustion length, this is θ_{IL}. Consequently, the total fuel mass found in the cylinder as a function of the crankshaft rotated angle is:

$$m(\theta) = \frac{\theta - \left(361 + \theta_{IS} + \theta_{ID}\right)}{\theta_{IL}} \cdot m_{F\ cycle} \qquad (58)$$

And during this sub-stage, the temperature is calculated as described in (49), and the pressure increment is:

$$\frac{dp}{d\theta} = \frac{\gamma(\theta) - 1}{V(\theta)} \cdot \left(\frac{dQ_b(\theta)}{d(\theta)} - \frac{dQ_r(\theta)}{d(\theta)} - \frac{\gamma(\theta)}{V(\theta)} \cdot p(\theta) \cdot \frac{dV(\theta)}{d(\theta)}\right) + \frac{\gamma(\theta) - 1}{V(\theta)} \cdot \frac{m_{F\ cycle} \cdot H_{injected\ fuel}}{\theta_{IL} \cdot \frac{\pi}{180}} \qquad (59)$$

Combustion after injection has finished

From the moment the injection endpoint is reached until the crankshaft reaches the exhaust valve opening angle θ_{EVC}, the model takes into consideration the next algorithms.

$$T_g(\theta) = \frac{p(\theta) \cdot V(\theta)}{\left(m_{rem} + m_{GAS\ cycle}\right) \cdot R_{gas}} \qquad (60)$$

$$\frac{dp}{d\theta} = \frac{\gamma(\theta) - 1}{V(\theta)} \cdot \left(\frac{dQ_b(\theta)}{d(\theta)} - \frac{dQ_r(\theta)}{d(\theta)} - \frac{\gamma(\theta)}{V(\theta)} \cdot p(\theta) \cdot \frac{dV(\theta)}{d(\theta)} \right) \qquad (61)$$

5.1.5. Exhaust

This is the last stage of the cycle in the engine and goes from the exhaust valve opening point to the top dead center. The parameters calculated with expressions *(34)* and *(35)* are obtained with the same procedure. As the exhaust valve is open, the pressure drops to a value given by p_3, which is defined on the exhaust manifold model (*section 5.6.*). Then the exhaust gas temperature is calculated considering an atmospheric temperature of 25 °C

$$T_g(\theta) = \left[T_g(\theta_{EVC}) \cdot \left(1 - \frac{\gamma(\theta) - 1}{\gamma(\theta)} \right) \cdot \left(1 - \frac{p_3}{p(\theta_{EVC})} \right) \right] + 25 \qquad (62)$$

Now, with all the data obtained, it is possible to extract all the output values that characterize the performance of the engine. With the information obtained in each iteration, it is possible to obtain diagrams for the instant cylinder pressure during the whole cycle, expressed as a function of the crankshaft rotation angle and the instant volume of the cylinder.

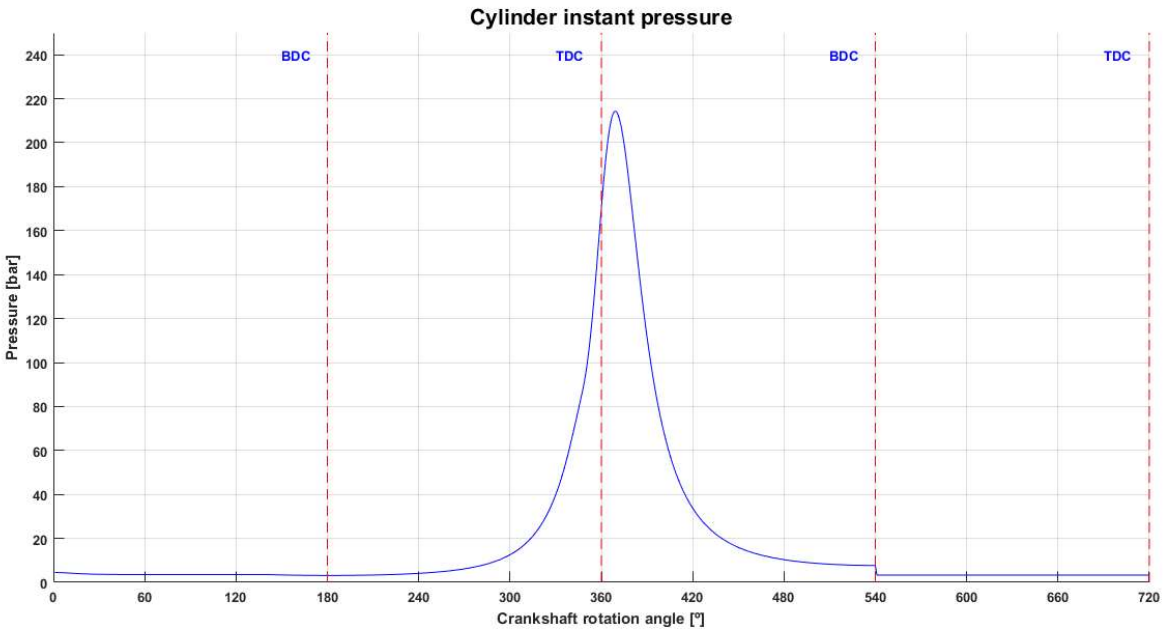

Figure 20: Cylinder pressure diagram

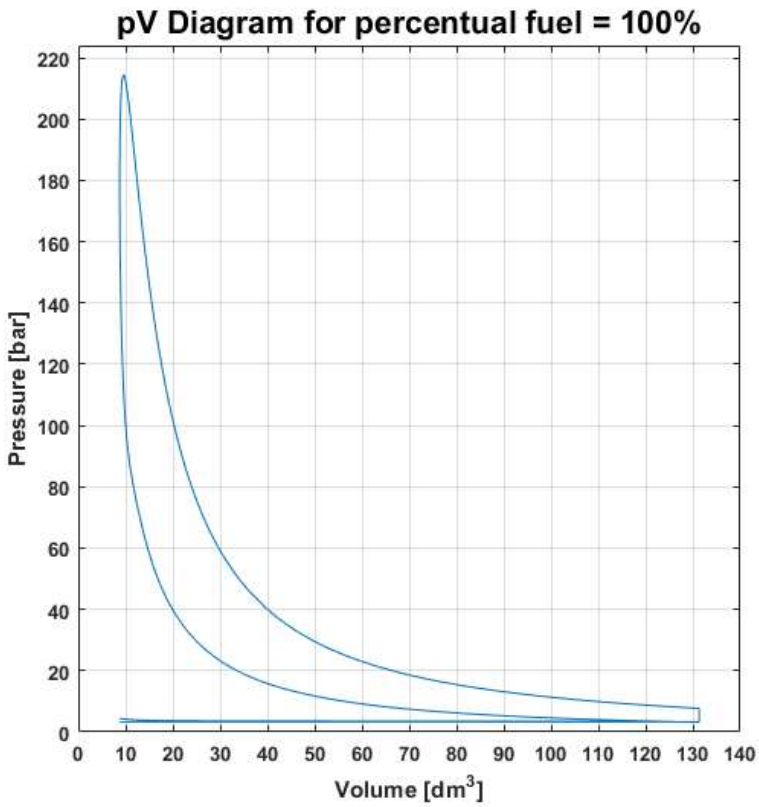

Figure 21: Cylinder pressure-volume diagram

The power is deduced from the pressure-volume diagram that displays the pressure inside the cylinder as a function of the instantaneous volume. Integrating the expression given by the diagram, the work produced by the engine is obtained:

$$W = \oint p \, dv \qquad (63)$$

In accordance with this, so as to compute the developed work in each one-degree step or work increment, the model executes the following calculation:

$$\frac{dW}{d\theta} = \frac{p(\theta) + p(\theta+1)}{2} \cdot \left[V(\theta+1) - V(\theta)\right] \qquad (64)$$

The sum of these increments equals the total work developed in a cycle, so their sum divided by the cylinder displacement in the model is equivalent to the indicated mean effective pressure in a real engine.

$$imep = \frac{\sum dW}{V_{cyl}} \qquad (65)$$

The friction mean effective pressure is the value that allows to deduce the energy loss due to the motion of the mechanism, and is expressed as a pressure that has to be subtracted from the mean indicated pressure. To make an estimation of the value that this deduction can achieve, a combination of the *Chen-Flynn* model and the *Winterborne* model is used.

The *Chen-Flynn* method is an empirical model based on experimentally obtained data and suggests that the *fmep* can be calculated as a function of the maximum pressure in the cylinder and the mean piston speed. The expression that defines the model is:

$$fmep = p_B + \frac{p_{max}}{200} + 0{,}162 \cdot u_P \qquad (66)$$

The first factor p_B represents a base pressure that has to be overcome, so it is taken as a constant in the expression with a value of 0,137. The term depending on u_P , reflects the friction losses due to the interaction of the piston rings and cylinder walls, and the maximum cycle pressure p_{max} characterizes the losses in the mechanism piston-rod-crankshaft.

The *Winterbone-Tennant* method makes a similar prediction with the following expression:

$$fmep = 0{,}061 + \frac{p_{max}}{60} + 0{,}294 \cdot \frac{n}{1000} \qquad (67)$$

After these two estimations are carried out, the model generates a combined result using the two methods. To do so, the α_F coefficient is introduced to adjust the weight of each one of the results of both methods. The value for the α_F coefficient is 0,78.

$$fmep = \alpha_F \cdot fmep_{Chen-Flinn} + (1 - \alpha_F) \cdot fmep_{Winterbone} \qquad (68)$$

This last result is the one that the model uses to compensate the simulated indicated mean pressure and to obtain the brake mean effective pressure.

$$bmep = imep - fmep \qquad (69)$$

Now the model can easily give a result for the brake effective power of the engine, and thus, for the torque, the brake specific fuel consumption and the total power-plant efficiency η_{TPP}:

$$P_E = bmep \cdot V_{cyl} \cdot \frac{n}{2} \cdot N_{cyl} \qquad (70)$$

$$\tau = \frac{bemp \cdot V_{cyl} \cdot N_{cyl}}{4 \cdot \pi} \qquad (71)$$

$$BSFC = \frac{m_F \cdot 3600}{bemp \cdot N_{cyl} \cdot V_{cyl}} \qquad (72)$$

$$\eta_{TPP} = \frac{P_E + P_{EG}}{P_E \cdot BSFC \cdot \frac{3600}{1000} \cdot H_i} \qquad (73)$$

Where H_i is the heat value of marine diesel oil (42700 kJ/kg) and P_{EG} is the electrical power obtained from the electric generator of the hybrid turbocharger system, which will be explained in following sections of this document. Also, the exhaust gas mass-flow rate $\dot{m}_{GAS}$ is calculated:

$$\dot{m}_{GAS} = m_{Gas\ cycle} \cdot \frac{n}{2} \cdot N_{cyl} \qquad (74)$$

By now it's been possible to see how the model simulates the phenomena occurred in the same engine, but not how the air and exhaust gases characteristics are modified as they pass through the several subsystems of the power plant. These are described in the following paragraphs.

5.2. Turbocharger compressor

So as to obtain the final conditions of the intake air that will be used in the engine, the model has to simulate the variations in pressure, density and temperature suffered by the air as it is driven through the compressor. This is made by following the data provided by the compressor map, that enables to relate the different properties of the air and the compressor performance, and which in the model is introduced by two matrices. One matrix provides the compressor ratio and the other provides the compressor efficiency, both giving these values as a function of the volumetric airflow and the shaft speed.

In order to obtain the results, the main variables are the mass-flow requested by the engine and the turbocharger shaft speed, which are $\dot{m}_A$ and n_{TC} respectively. Then, the general gas equation leads from the mass flow to the volumetric flow, the main input for the compressor map.

$$\dot{m}_A = \lambda_v \cdot V_{cil} \cdot \rho \cdot \frac{n}{2} \cdot N_{cil} \qquad (75)$$

$$\dot{V}_C = \dot{m}_A \cdot \frac{T_1 \cdot R_{gas}}{p_1} \qquad (76)$$

Where T_1 and p_1 are the temperature and pressure at the compressor inlet and correspond to the atmospheric conditions. However, the volumetric flow has to be adapted so as it can be used correctly as a reference for the compressor map. It has to be considered that as the compressor map for the TCA66-42 turbocharger is not provided by MAN, a compressor map from another turbocharger has been selected for the simulator. The map used is from a Brown Boveri & Cie. VTR 401 turbocharger, a device with a smaller air flow capacity, so a scale factor k_{mass} is introduced in the model to adapt the BBC compressor to the MAN

engine demands. Then, also the square root of the reference to atmospheric temperatures ratio is added to compensate the effect of the temperature to the compressor performance as the compressor map is defined under specific air conditions, specifically those at a temperature of 288,15 ^{O}K.

$$\dot{V}_{288} = \dot{V}_C \cdot \sqrt{\frac{T_{288}}{T_1}} \cdot k_{mass} \qquad (77)$$

With the second main input variable taken by the compressor model, the shaft speed n_{TC}, the same philosophy is used to tune this variable before entering the matrices of the compressor map.

$$n_{288} = n_{TC} \cdot \sqrt{\frac{T_{288}}{T_1}} \cdot k_n \qquad (78)$$

Once the shaft speed and volumetric flow are known, the model can provide the compression ratio β and isentropic efficiency η_C interpolating in the BBC compressor map matrices. Later, the dimension factors k_n are included and also a correction factor δ_η depending on the engine brake effective power for the compressor efficiency.

$$\beta_{MAN} = k_\beta \cdot \beta_{BBC} \qquad (79)$$

$$\eta_{MAN} = k_\eta \cdot \delta_\eta \cdot \eta_{BBC} \qquad (80)$$

By theses means, first the theoretical T_2 and then the real $T_{2'}$ air temperature and pressure p_2 can be calculated as an isentropic process where the efficiency is added:

$$T_2 = T_1 \cdot \beta_{MAN}^{\left(\frac{\gamma_A - 1}{\gamma_A}\right)} \qquad (81)$$

$$T_{2'} = T_1 + \frac{T_2 - T_1}{\eta_{MAN}} \qquad (82)$$

In the case of the compression ratio, no efficiency has to be taken into account as the ratio obtained from the map matrix is the real instead of the theoretical.

$$p_2 = \beta_{MAN} \cdot p_1 \qquad (83)$$

After this, the power requested by the compressor is calculated, which at the same time allows to obtain the torque:

$$P = c_P \cdot \dot{m}_A \cdot (T_{2'} - T_1) \qquad (84)$$

$$\tau_C = n_{TC} \cdot 2\pi \cdot P \qquad (85)$$

5.3. Turbocharger turbine

The turbine is the element responsible for extracting the energy from the exhaust gas and provide the necessary power to the compressor and the electric generator. The performance of the turbine is defined in the turbine map, a diagram showing the flow coefficient for friction and flow contraction α_T and turbine isentropic efficiency η_T as a function of the turbine expansion ratio π_T.

The blades of the turbine rotor and stator act as a nozzle, driving the gas from an inlet high pressure point (p_{inlet}) to a lower outlet pressure environment (p_{outlet}), so the fluid suffers a pressure reduction, measured by the expansion ratio π_T, and also an increment on its velocity.

$$\pi_T = \frac{p_{inlet}}{p_{outlet}} = \frac{p_3}{p_4} \qquad (86)$$

The expansion ratio and the mass flow rate are linked. Let us consider a nozzle where $p_{inlet} = p_{outlet}$, in this case, no flow will exist, but from the moment when the outlet pressure is reduced, a subsonic flow in an inlet to outlet direction will appear. Then, as the difference between the inlet and outlet pressures increases, the flow rate will also rise, and as the continuity equation defines, the velocity of the gas will increase too. This is described in the following paragraphs.

In a nozzle or a turbine, the mass-flow can be calculated as described in *(87)*, where A_{Eff} stands for the effective turbine area, or the throat area in the case of a nozzle. However, the corrected mass flow rate is used as a reference in order to eliminate the influence of the turbine inlet pressure p_3 and temperature T_3.

$$\dot{m}_T = \alpha_T \cdot \frac{A_{Eff} \cdot p_3}{\sqrt{R \cdot T_3}} \cdot \sqrt{\frac{2 \cdot \gamma}{\gamma - 1} \cdot \left[\left(\frac{1}{\pi_T} \right)^{\frac{2}{\gamma}} - \left(\frac{1}{\pi_T} \right)^{\frac{\gamma+1}{\gamma}} \right]} ; \dot{m}_{T\,cor} = \alpha_T \cdot \frac{A_{Eff}}{\sqrt{R}} \cdot \sqrt{\frac{2 \cdot \gamma}{\gamma - 1} \cdot \left[\left(\frac{1}{\pi_T} \right)^{\frac{2}{\gamma}} - \left(\frac{1}{\pi_T} \right)^{\frac{\gamma+1}{\gamma}} \right]} \quad (87)$$

It is possible to announce that the corrected mass flow rate does not depend on a given inlet or outlet pressure, but only on the expansion ratio. However, there is always a velocity associated with the mass flow rate, so in case this is high enough, the local speed of sound *c* will be reached. In a subsonic condition, if the pressure in the nozzle outlet is reduced, the wave caused by the pressure drop travels upstream transmitted at a relative velocity (regarding to the flowing gas) equal to the speed of sound, so it can be perceived in the inlet and thus, cause an increase in the mass flow rate and flow velocity. In the other hand, in a supersonic condition, as the gas travels downstream at the speed of sound relative to the nozzle walls, the wave generated by the pressure drop cannot reach the nozzle inlet, and the mass flow rate will not increase. This phenomenon is known as nozzle choking, and the condition where it shows up is characteristic of the gas. The supersonic choking condition will be given when the critical expansion ratio is reached, which is defined by the following expression:

$$\pi_{critical} = \left(\frac{\gamma + 1}{2} \right)^{\frac{\gamma}{\gamma - 1}} \quad (88)$$

The turbine mass-flow rate will increase as the expansion ratio gets higher, and from the moment where the critical expansion ratio is reached, the flow rate will stabilize and won't rise beyond. This means that concerning to the calculation of the mass-flow rate, the next criteria has to be used:

$$\pi_T < \pi_{critical} \quad \rightarrow \quad \dot{m}_T = \alpha_T \cdot \frac{A_{Eff} \cdot p_3}{\sqrt{R \cdot T_3}} \cdot \sqrt{\frac{\gamma}{\gamma - 1} \cdot \left[\left(\frac{1}{\pi_T} \right)^{\frac{2}{\gamma}} - \left(\frac{1}{\pi_T} \right)^{\frac{\gamma+1}{\gamma}} \right]} \quad (89)$$

$$\pi_T \geqslant \pi_{critical} \quad \rightarrow \quad \dot{m}_T = \alpha_T \cdot \frac{A_{Eff} \cdot p_3}{\sqrt{R \cdot T_3}} \cdot \sqrt{\frac{\gamma}{\gamma - 1} \cdot \left[\left(\frac{1}{\pi_{critical}} \right)^{\frac{2}{\gamma}} - \left(\frac{1}{\pi_{critical}} \right)^{\frac{\gamma+1}{\gamma}} \right]} \quad (90)$$

The model then obtains the turbine inlet temperature and pressure from the engine exhaust ports, and using the commented equations is capable to provide the mass-flow rate that will pass through, but although in the compressor model the heat capacity ratio γ was considered constant, in the case of the

exhaust gases a correction depending on the temperature has to be taken into account as this factor causes a considerable variation on γ_{gas}. For this reason, *Langen* equations are implemented, which by means of several factors obtained from experimental data are capable of calculating the value for the constant pressure and constant volume heat capacities c_P and c_V, where a_a = 992,1; a_{ap} = 703,2 and b_a = 0,136.

$$c_P = \left(a_a + b_a \cdot T_3\right) \cdot 1{,}06676 \qquad (91)$$

$$c_V = \left(a_{ap} + b_a \cdot T_3\right) \cdot 1{,}08646 \qquad (92)$$

$$\gamma_{gas} = \frac{c_P}{c_V} \qquad (93)$$

In the turbine mass flow rate equation the equivalent turbine area A_{Eff} is requested, but the available data comes from the values provided by ABB regarding the VTR 401 turbocharger so a scale factor is used to tune the flow rate.

The outlet pressure p_4 has a value close to the atmospheric pressure, and is given by the following equation:

$$p_4 = p_{atm} + 9901{,}3443 \cdot 10^{-8} + 9865{,}5315 \cdot 10^{-8} \cdot \dot{m}_T \qquad (94)$$

Once the outlet pressure is defined, also the theoretical temperature of the gas after the expansion T_4 can be obtained.

$$T_4 = T_3 \cdot \left(\frac{p_3}{p_4}\right)^{\frac{1-\gamma_{gas}}{\gamma_{gas}}} \qquad (95)$$

Using the expansion ratio π_T as input data, the turbine isentropic efficiency η_T can be deduced from the turbine map. Differing from the method used in the compressor to extract the data from its respective diagram where the input data are the shaft speed and volume flow, in the case of the turbine, the input π_T is non-dimensional, so there is no need to introduce a scale factor. With the efficiency η_T defined it's possible to obtain the real outlet temperature $T_{4'}$.

$$T_{4'} = T_3 - \left(T_3 - T_4\right) \cdot \eta_T \qquad (96)$$

Finally, the torque developed by the turbine can be described by means of the enthalpy drop:

$$\tau_T = \dot{m}_T \cdot c_P \cdot \frac{T_3 - T_{4'}}{2\,\pi \cdot n_{TC}} \qquad (97)$$

5.4. Electric generator

The calculations for the power output of the generator are quite simplified due to the lack of technical data from the manufacturer, and they depend upon the shaft speed and the torque delivered by the magnetic field on the PMSG. This means that the torque-speed curve of the generator has been adapted (as described in chapter 6.2.2.) to suit the performance of the turbocharger. As it's recommended in most of the literature, an estimated value for the efficiency of the generator η_{EG}, 0,95 will be taken into account, so the power obtained from the electric generator is:

$$P = \tau \cdot \omega$$

$$P_{EG} = \tau_T \cdot n_{TC} \cdot \frac{2\pi}{60} \cdot \eta_{EG} \qquad (98)$$

5.5. Turbocharger dynamics

As it's been possible to see in the previous paragraphs, both compressor and turbine react to the input of several variables, which are in fact air or gas characteristics and depend on the engine performance or the environment. However, there's another variable that defines the turbocharger performance and is the shaft speed of the same turbocharger. This makes it necessary to create a loop on the model cable of providing the compressor and the turbine model of a correct value for the rotating speed depending on the joint dynamics of both turbine and compressor.

The calculations for this value are based on Euler's second law of motion making possible to link the angular speed with the torque.

$$\frac{d\omega}{dt} = \frac{1}{J} \cdot \sum \tau \qquad (99)$$

The sum of torques is obtained from the turbine and compressor results, being the turbine positive as it's the one to generate useful work, and being the compressor negative. In this point, the electric generator is added to the model also as a constant negative torque.

The turbocharger speed thus, will depend on the balance between those torques, so as the expansion of the gas in the turbine will make increase the shaft spinning rate, the torque caused by the compressor resistance will also rise, and together with the torque from the generator will lead to a situation where the sum of torques comes too small to make the differential considerable. At this point the shaft speed will be stable.

The moment of inertia of the rotating element is described by *J*, and the shaft angular acceleration will depend on this constant, but will not affect the final shaft speed. Its value is estimated taking into account the mass distribution of the turbine rotor.

5.6. Exhaust manifold

As it's been commented, when the combustion gases leave the cylinder they don't flow freely through the exhaust duct because they are stopped by the turbine as it reaches the nozzle choking condition. Due to this fact, the characteristics of the gas in the turbine inlet will differ from the cylinder exhaust as the gas can suffer compression in the manifold.

The manifold model is based on the general gas equation, considering all the manifold as a control volume V_m and using the difference between the mass flow rate flowing in ($\dot{m}_{GAS\ cycle}$) and out ($\dot{m}_T$) of this volume to obtain the pressure variation.

$$p_3 = \frac{R \cdot T}{V_m} \cdot \int \left(\dot{m}_{GAS} - \dot{m}_T\right) dt \qquad (100)$$

In the simulation, the calculated in and outflow rates tend to equalize and when this condition is reached, there is no mass accumulation in the control volume and the integration does not generate any further pressure increase so the pressure in the exhaust manifold stabilizes.

5.7. Intercooler

Prior to being introduced in the cylinder, the air is driven to the intercooler, where the air temperature is reduced, allowing to raise the density and the engine efficiency. The working principle of this module is the heat conduction, transmitting the heat from the air to the cooling liquid by means of a heat conducting wall that separates both fluids. The intercooler is modeled using the intercooler efficiency definition η_{IC}.

$$\eta_{IC} = \frac{T_{air\ inlet} - T_{air\ outlet}}{T_{water\ inlet} - T_{water\ outlet}} \qquad (101)$$

The value for the efficiency is estimated, and the inlet air and water outlet temperature are considered identical so the formula becomes:

$$T_I = T_2 - \left(T_2 - T_{water\ inlet}\right) \cdot \eta_{IC} \qquad (102)$$

5.8. Blow-off valve

This element is responsible for adjusting the mass-flow rate perceived by the compressor and is modelled by simulating a discharge valve that behaves like a nozzle and which is located in the intake manifold. Some of the charge air compressed by the turbocharger will be expelled to the atmosphere, so in the same way, as it's been described in the turbine section 5.3. , there will exist a critical expansion ratio that will limit the mass-flow rate being ejected. The method for the calculation of the mass-flow expelled through the valve $\dot{m}_{blow\text{-}off}$ is the same used in *(88)*, *(89)* and *(90)*. After obtaining the blow-off mass-flow, the value sent to the compressor block is:

$$\dot{m}_c = \dot{m}_{blow-\ off} + \dot{m}_A \qquad (103)$$

The valve model receives the charge air conditions provided by the intercooler and calculates the mass flow rate expelled through the valve nozzle. Then this flow is added to the flow rate requested by the engine, and it is sent back to the compressor settling the flow rate at which it has to operate.

6. Engine Diesel mode results analysis

The first aim of this study consists in obtaining the most accurate numerical model of a verified operative engine to make sure the results provide a realistic approximation of the performance of a power-plant. The second aim is to verify the correct operation and the capabilities of a hybrid turbocompressor when it's added to the numerical model of the engine. Considering these two premises, in this chapter the accuracy of the model compared to the real engine will be discussed, and the results of the model when the HyTC is included will be studied. During the analysis described in this chapter, the results obtained with the developed numerical model on Diesel mode, are arranged in three operation modes:

- Diesel standard operation. This mode makes reference to the engine configuration set by the manufacturer, which means that neither the blow-off valve nor the turbocharger generator are used.
- Diesel blow-off operation. This configuration includes the blow-off valve system to find out the amount of charge air that can be rejected without compromising the intake air that the engine needs to deliver the required power.
- Diesel hybrid turbocharger operation (HyTC): The same intake air characteristics than in blow-off operation are reproduced, but the torque delivered by the electric generator is used to modulate the turbocharger performance and control the airflow instead, obtaining electric power at the same time.

6.1. Verification of the model

The verification of the model is based on the results obtained from the numerical model on standard operation mode, which are contrasted with the manufacturer's project guide data. To obtain an accurate simulation of the engine, several variables have to be modulated and because of this, two settings of the model are presented. The initial one is an attempt to obtain the most accurate results compared to the project guide whereas the final one deals with the corrections required to obtain a more proper and stable simulation.

In order to procure a reference for the reliability of the calculations performed by the model, the results generated by the simulator must be compared to the reference data described on the manufacturer specifications. This reference data regards to the operation of the MAN V51-60 DF engine under *ISO 15550: 2002/ISO 3046-1: 2002* conditions for constant speed controllable pitch propulsion (514 min^{-1}) included in the engine project guide, which contains information for six different fields:

- Specific fuel consumption.
- Compressor outlet temperature.
- Charge air pressure.
- Compressor air flow rate.
- Exhaust mass flow.
- Turbine outlet temperature.

Four values are given for each field depending on the engine power, corresponding to different percentages of the maximum continuous rate MCR (100%, 85%, 75% and 50%). Only the specific fuel consumption information can be verified for both constant and variable speed operation, including values regarding 25% MCR.

6.1.1. Standard operation (initial setting)

In the following tables, the values from the engine project guide and the ones obtained with the numerical model with its initial settings can be confronted. For the major part of the data, the manufacturer indicates that all the values provided on the project guide are within an allowance of ±5%, except the temperatures, where the limits are set at ±20 ^{0}C. The accuracy ratio or the difference of the information provided by the model and the manufacturer is calculated from each reference value in the project guide, so if the ratio remains within the allowance limits, the reliability of the model regarding that field at that certain power rate remains proven. Other discrepancies further from the allowance limits shall be discussed.

Load	100	85	75	50	%
Compressor outlet temp.					
Project guide	220,000	196,000	192,000	141,000	ºC
	493,150	469,150	465,150	414,150	K
Numerical Model	486,084	466,176	457,738	391,497	K
Difference	-7,066	-2,974	-7,412	-22,653	K
Charge air pressure					
Project guide	4,290	3,760	2,910	1,620	Bar
Numerical Model	4,242	3,730	3,100	1,807	Bar
Ratio	-1,12%	-0,80%	6,53%	11,54%	
Compressor air flow rate					
Project guide	23,700	21,647	20,800	14,167	kg/s
Numerical Model	23,769	21,087	19,790	12,671	kg/s
Ratio	0,29%	-2,59%	-4,86%	-10,56%	
Exhaust mass flow					
Project guide	24,333	22,185	21,275	14,500	kg/s
Numerical Model	24,405	21,628	20,272	13,009	kg/s
Ratio	0,29%	-2,51%	-4,71%	-10,28%	
Turbine outlet temp.					
Project guide	323,000	313,000	297,000	338,000	ºC
	596,150	586,150	570,150	611,150	K
Numerical Model	593,494	594,211	583,499	624,241	K
Difference	2,656	-8,061	-13,349	-13,091	K

Table 9: Project guide data vs. Diesel standard numerical model comparison. Initial setting.

Load	100	85	75	50	25	%
Project guide	181,500	180,000	185,800	190,500	205,500	g/kWh
Numerical Model	189,525	191,776	194,050	201,822	230,061	g/kWh
Ratio	-4,421	-6,542	-4,440	-5,943	-11,952	%

Table 10: BSFC. Project guide vs. Diesel standard numerical model. Constant speed. Initial setting.

Project guide	181,500	180,000	183,300	187,000	187,000	g/kWh
Numerical Model	189,525	191,776	192,642	199,040	211,391	g/kWh
Ratio	-4,421	-6,542	-5,097	-6,439	-13,043	%

Table 11: BSFC. Project guide vs. Diesel standard numerical model. Variable speed. Initial setting.

The values obtained show similar tendencies on each field, where higher accuracy is achieved at higher power rates. At 100% MCR the results in all fields remain within the limits, also at 85% MCR except for the specific fuel consumption (surpassing the 5% limit by 1,54%). Similar conditions can be observed at 75% MCR, although the discrepancy affects the charge air pressure, with an overshoot of 1,53% over the limit. Finally, a major error can be noticed at 50% MCR, where all the values exceed the allowance. Thus, it's obvious that the accuracy of the model varies depending on the different power rates, and this is a consequence of two main reasons which are detailed on the following lines.

The first one is the fact that the compressor and turbine maps used on the model do not stand for the original turbocompressor used on the MAN V12 51/60 DF engine, but for a BBC VTR 401 turbocharger. In the case of the compressor, the data from the maps provide the corresponding efficiency η_c and compression ratio β, depending on the volume flow and turbocharger speed. For the turbine, the flow coefficient α_T and the turbine isentropic efficiency η_T are obtained from the expansion ratio π_T, as it's been commented on *Chapter 5.* . These output data from the maps have been corrected with variable coefficients depending on the air flow rate to provide the most similar performance of the simulator to the manufacturer specifications.

The other factor compromising the accuracy of the model is the fact that the code developed to simulate the physics of the engine, does not include the necessary tools to reflect the fluid dynamics on the combustion chamber, the intake manifold and on the exhaust. On the model, the air mass flowing through the compressor depends on the engine mass-flow $\dot{m}_A$ and is defined as a function *(75)* of the volume of the cylinders, the engine speed, the density, and the volumetric efficiency or filling coefficient λ_V. During constant speed operation, all these values remain constant except for the density, which is obtained from the pressure increase of the previous iteration of the simulation. This means that differing to a real engine, the only variable affecting the mass flow entering the cylinders in the model, is the air density obtained after the compression on the turbocharger. However, it's known that the unsteady flows of reciprocating internal combustion engines cause inertial and wave effects which can have considerably important consequences on the volumetric efficiency.

The inertial phenomenon is usually studied by means of the *Helmholtz* [8] resonance equation, which defines the resonance frequency of a volume of gas enclosed in a pipe when it's disturbed by an external

force, i.e., the piston. This oscillation can force or reduce the final amount of mass going into the cylinder, generating a variation of the volumetric efficiency. The filling coefficient λ_v is modified following a damped harmonic function depending on the disturbance frequency, i.e., the piston frequency *n*. In the following diagram, it's possible to appreciate the ratio of the modified volumetric efficiency λ'_v and the volumetric efficiency when no disturbance occurs λ_v as a function of the ratio of the resonance frequency f_0 and the disturbance frequency f_m .

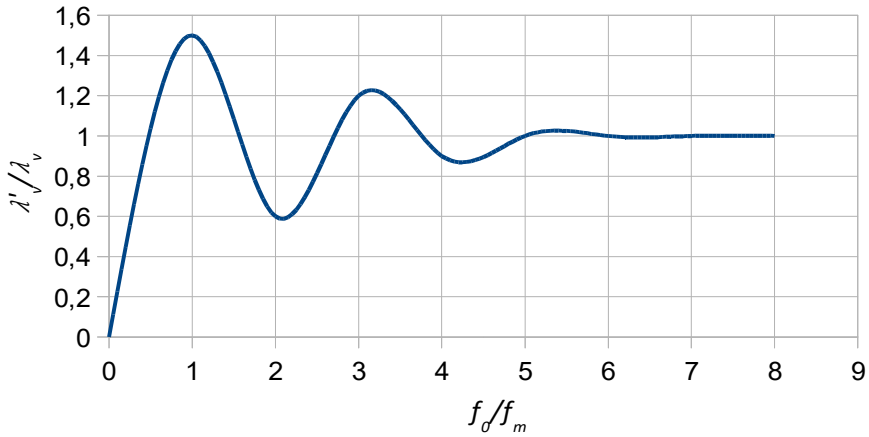

Figure 22: Volumetric efficiency vs. oscillation frequency

The behaviour of this function depends on the geometries of the intake duct and the cylinder, and allows to predict for example the resonant peak, where the volumetric efficiency increase becomes higher. This point can be expressed as the optimum engine speed n_{opt} (in revolutions per second) and is defined by the following expression:

$$n_{opt} = \frac{c}{4\pi}\sqrt{\frac{S_m}{L_m \cdot \frac{V_{cyl}}{2}}} \qquad (104)$$

Where S_m and L_m are the section an length of the manifold, *c* stands for the speed of sound of the air in the intake manifold, and V_{cyl} is the displacement of the cylinder used to calculate the mean volume of the cylinder during the disturbance action. However, these dimensions are not provided by the manufacturer, so it's not possible to predict how the volumetric efficiency will vary during the operation at each engine speed, and in order to obtain the most reliable model of the engine, λ_v has been set to suit the engine requirements depending on the air flow rate.

To evaluate the correct operation of the engine, it is also completely necessary to know the conditions where the compressor is operating. In order to identify its proper operation points, a plot of the compressor map showing the volumetric airflow, compression ratio, efficiency, and speed is traced. This information is primarily calculated considering the information from the BBC VTR 401 turbocharger, not from the MAN TCA66. For this reason, in order to offer an appropriate visualization, the values are not provided in absolute values but in relative or specific numbers, making it possible to refer to both turbochargers, and to make an easier contrast of the different operating conditions. This method allows to visualize if the operating point of the compressor is close to the surge or choke limits, situations that should be avoided, as well as the speed and the efficiency progression.

The next plot shows the operating points for the model tuned to provide the closest results to the specifications in the project guide. The line describes a pronounced drop in the compression ratio that leads to a dangerous approximation to the choke boundary in variable speed, and surpassing the line in constant speed operation.

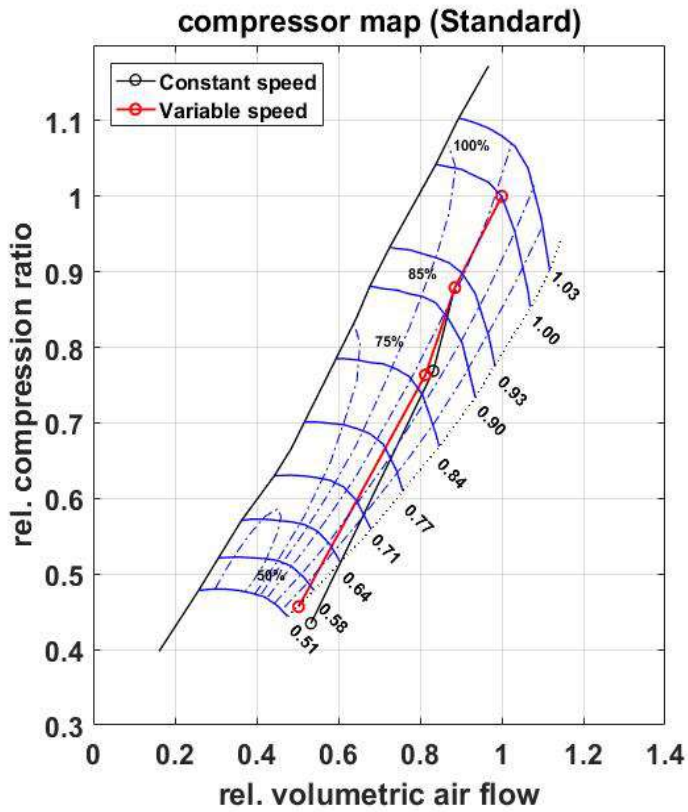

Figure 23: Compressor map. Diesel – Standard. Initial setting.

This indicates a discrepancy between an accurate operation of the engine in accordance to the manufacturer specifications and the proper conditions in the compressor described in the map, which forces to modify the criteria to obtain a correct simulation of the power-plant. It is necessary to use a model with a more stable tendency on the compressor operating points and no choking conditions.

To do so, the previously commented coefficients which were used to modify the volumetric efficiency λ_V of the cylinders, the compressor efficiency η_c and compression ratio β, as well as α_T and isentropic efficiency η_T, are required to be adjusted again.

6.1.2. Standard operation (final setting)

As it has been seen in the previous section, the model has required several adjustments to obtain better stability, so the final results for the simulation on standard operation are now available. At this point, the model has been set to provide an approximated simulation to the actual engine but keeping a proper and stable self performance especially on the compressor, so it is still important to contrast the results achieved with this final configuration and the manufacturer's project guide.

These results indicate how the simulator is less factually accurate when compared to the project guide, however, this divergence enables a better behaviour of the turbocharger. It can be seen on the compressor map, how the slope of both lines remains almost permanent in both constant and variable speed operation, providing more stability. The map also confirms that under this configuration, no choking condition is reached. Only an approximation in constant speed shows a tendency to the choking boundary when the engine is running on the lowest power rates, but the line is not surpassed. Additionally, as the map indicates, under these low volumetric air flow and compression ratio conditions, the choking phenomenon starts to fade. Tables *15* and *16* indicate the actual values recorded on the simulations referred to the MAN TCA66 turbocompressor.

Load	100	85	75	50	%
Compressor outlet temp.					
Project guide	220,000	196,000	192,000	141,000	°C
	493,150	469,150	465,150	414,150	K
Numerical Model	486,084	466,047	450,788	391,618	K
Difference	-7,066	-3,103	-14,362	-22,532	K
Charge air pressure					
Project guide	4,290	3,760	2,910	1,620	Bar
Numerical Model	4,242	3,778	3,424	2,254	Bar
Ratio	-1,12%	0,48%	17,66%	39,14%	%
Compressor air flow rate					
Project guide	23,700	21,647	20,800	14,167	kg/s
Numerical Model	23,769	21,047	19,193	12,945	kg/s
Ratio	0,29%	-2,77%	-7,73%	-8,63%	%
Exhaust mass flow					
Project guide	24,333	22,185	21,275	14,500	kg/s
Numerical Model	24,401	21,590	19,677	13,280	kg/s
Ratio	0,28%	-2,68%	-7,51%	-8,41%	%
Turbine outlet temp.					
Project guide	323,000	313,000	297,000	338,000	°C
	596,150	586,150	570,150	611,150	K
Numerical Model	593,494	594,970	597,988	614,285	K
Difference	-2,656	8,820	27,838	3,135	K

Table 12: Project guide data vs. Diesel standard numerical model comparison. Final setting.

Load	100	85	75	50	25	%
Project guide	181,500	180,000	185,800	190,500	205,500	g/kWh
Numerical Model	189,525	191,83	193,46	200,843	230,061	g/kWh
Ratio	4,421%	6,572%	4,123%	5,429%	11,952%	%

Table 13: BSFC. Project guide vs. Diesel standard numerical model. Constant speed. Final setting.

Project guide	181,500	180,000	183,300	187,000	187,000	g/kWh
Numerical Model	189,525	191,83	192,278	197,482	211,434	g/kWh
Ratio	4,421%	6,572%	4,898%	5,605%	13,066%	%

Table 14: BSFC. Project guide vs. Diesel standard numerical model. Variable speed. Final setting.

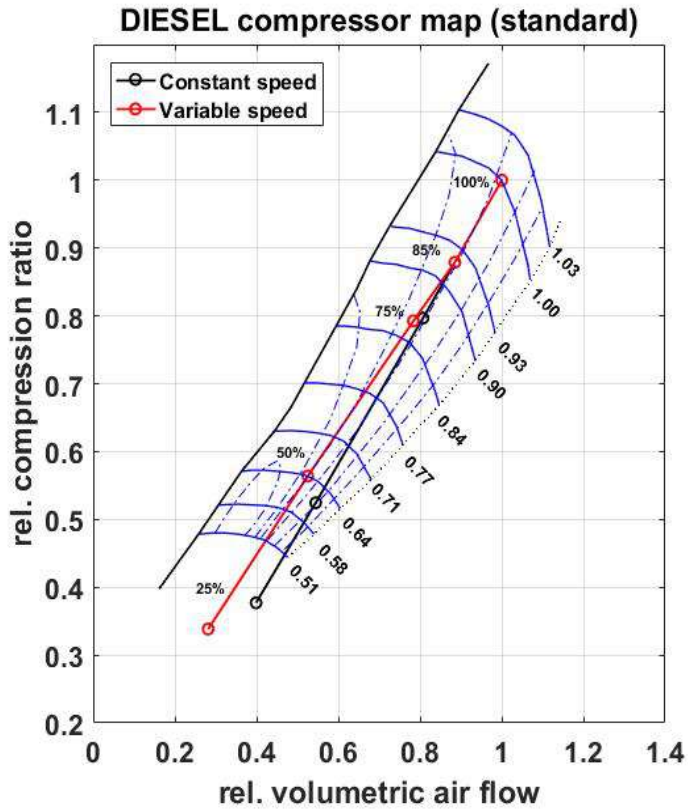

Figure 24: Compressor map. Diesel – Standard.

Power rate	100,075	85,030	75,000	50,051	25,062	%
Efficiency	0,810	0,809	0,812	0,817	0,849	-
Volume flow	7,616	7,357	7,159	6,371	5,770	m^3/s
Speed	22.061,924	20.367,902	19.125,151	13.658,454	8.481,625	min^{-1}
Pressure ratio	4,242	3,729	3,380	2,225	1,600	-
Compressor power	4.470,593	3.536,905	2.932,472	1.211,826	477,504	kW

Table 15: Compressor operation conditions. Diesel standard. Constant speed.

Power rate	100,075	85,030	74,994	49,983	24,976	%
Efficiency	0,810	0,809	0,818	0,861	0,893	-
Volume flow	7,616	7,357	6,957	5,779	4,356	m3/s
Speed	22.061,924	20.367,902	18.952,797	14.267,475	6.315,511	min-1
Pressure ratio	4,242	3,729	3,362	2,392	1,435	-
Compressor power	4.470,593	3.536,905	2.811,000	1.222,282	242,733	kW

Table 16: Compressor operation conditions. Diesel standard. Variable speed.

Power rate	100,075	85,030	75,000	50,051	25,062	%
Exhaust mass flow	24,4011	21,5905	19,6770	13,2799	9,6675	kg/s
Temperature drop	157,096	140,714	128,058	78,797	43,009	K
Cp	1.167,245	1.165,083	1.163,684	1.158,902	1.148,205	J/kgK
Turbine power	4.511,812	3.575,827	2.965,801	1.231,632	489,384	kW

Table 17: Turbine operation conditions. Diesel - Standard. Constant speed.

Power rate	100,08	85,030	74,994	49,983	24,976	%
Exhaust mass flow	24,4011	21,5905	19,1145	12,8152	6,8642	kg/s
Temperature drop	157,096	140,714	126,264	81,899	30,531	K
Cp	1.167,245	1.165,083	1.164,624	1.164,439	1.158,397	J/kgK
Turbine power	4.511,812	3.575,827	2.840,658	1.235,323	246,665	kW

Table 18: Turbine operation conditions. Diesel - Standard. Variable speed.

On the next paragraphs, the air-fuel ratio data is detailed, which is a delicate part of the engine operation as it has an important influence in the performance and anti-pollution setup of the engine. For this reason, the following lines add an overview of what the air-fuel mixture is, how is it measured, and which are its effects on the engine.

The air-fuel mixture is the combination of two components, the intake air (approximately with a 21% of oxygen and 78% of nitrogen) and the fuel. The oxidation reaction of the fuel and the oxygen provides the combustion, and the amounts of each component in the mixture are defined by the air-fuel ratio AFR or α.

$$AFR = \frac{m_{air}}{m_{fuel}} = \alpha \qquad (105)$$

To obtain a correct reaction, there has to be enough oxygen to combust the fuel, so the stoichiometric ratio indicates the mass proportion of air required for the total combustion of the fuel, which will take different values depending on the fuel type. However, other ratios can be used, and they will be considered lean if there is less fuel, and rich if there is more fuel when compared to the stoichiometric value. Although the stoichiometric ratio should offer a correct combustion, the physical and chemical characteristics of the fuel cause inefficiencies in the correct blending of the mixture. This is solved by adding excess air, so in real

conditions, a lean mixture is used for a proper combustion. To measure the characteristics of the mixture, the fuel-air equivalence ratio ϕ and the air-fuel equivalence ratio λ are commonly used. The first one is defined as the coefficient obtained dividing the stoichiometric ratio by the actual air-fuel ratio of the engine.

$$\phi = \frac{\left(oxidizer / fuel\ ratio\right)_{stoichiometric}}{oxidizer / fuel\ ratio\left(AFR\right)} = \frac{1}{\lambda} \qquad (106)$$

Using lean or rich mixtures has significant effects on the engine, and an inappropriate fuel-air ratio can compromise its performance. When the engine uses a rich mixture, the non combusted fuel absorbs heat by evaporation and cools the combustion chamber, which can be desirable depending on the situation but will obviously cause higher fuel consumptions. As the proportion of air is increased, this cooling effect is reduced, and higher temperatures are achieved. Once the air-fuel ratio reaches the value at which the temperature is maximum, a further increase in the AFR will have a temperature absorption effect on the combustion chamber.

The control of the combustion temperature is crucial and has direct effects related to the mechanical behaviour of the materials and on the generation of nitrogen oxides NO_X. When oxygen and hydrogen are combined in a high temperature environment, they combine to generate the different gases known as NO_X. In an internal combustion engine, this is mainly caused by the reaction of the nitrogen and oxygen from the intake air when they are exposed to the temperatures of the combustion chamber. The higher the temperature, the more easily these two elements will react.

Nevertheless, the use of excessively lean mixtures has also counterparts. In a naturally aspirated engine, considering a constant rotational speed the volumetric airflow doesn't vary, and to obtain a leaner mixture, the fuel flow has to be reduced, so the power will decrease. This can be corrected with charge air to have a lean mixture with the same amount of fuel, but it will also increase the pressure in the cylinder, which can achieve undesired values. Using lean AFR can also cause uneven distribution of the fuel inside the cylinder, and in the case of non self-ignited fuels, the combustion propagation can be interrupted. The characteristics of the mixture will vary depending on the type of engine, and there can be considerable differences between compression ignition engines (Diesel) and spark or pilot fuel ignition engines.

The stoichiometric ratio of Diesel fuel, although it can differ depending on the exact composition, is considered to be 14,7:1. However, on Diesel engines, lean mixtures are generally used to obtain a better performance due to several reasons.

On Diesel engines, the power is controlled by the amount of fuel injected in the cylinder at the end of the compression stroke. At the moment the fuel is injected, each particle will almost immediately combust after being mixed with the air in the cylinder, so although the initially sprayed particles are able to combust, if excess air is not added, it's more difficult for the particles sprayed at the final part of the injection stream to obtain the required air to combust. Additionally, Diesel fuel offers reduced volatilization and atomization performances due to its viscosity, both necessary to obtain an even and complete combustion, and it requires high compression temperatures to self-ignite. By adding excess air on the cylinder it is possible to achieve higher pressures during the compression stroke, which also leads to higher temperatures. This helps to get a proper viscosity, allowing the correct atomization, and the self-ignition of the fuel. Also, due to the fact that Diesel engines do not regulate the power with a throttle valve, these kinds of engines have tendency to run with excess air if speed is not reduced, especially when loads come to lower rates.

On the following lines, the results obtained in the simulations regarding the characteristics of the air-fuel mixture are detailed. The diagram displays fuel-air equivalence ratio ϕ as a function of the engine power rate, including two lines, one for each constant and variable speed modes. It clearly shows the use of lean mixture in all power conditions and the predicted tendency for leaner ratios as power decreases. This happens because, in order to reduce the engine power, there is a higher reduction of the fuel flow on the injection system rather than in the airflow being sent to the cylinders. This phenomenon can be observed with ease on the constant speed line. However, for those operation points where the engine speed is reduced, the variable speed line shows higher equivalence ratios when compared to constant speed. This fact is induced by the larger decrease in the airflow compared to the fuel flow, as the engine speed slows down when power is cut.

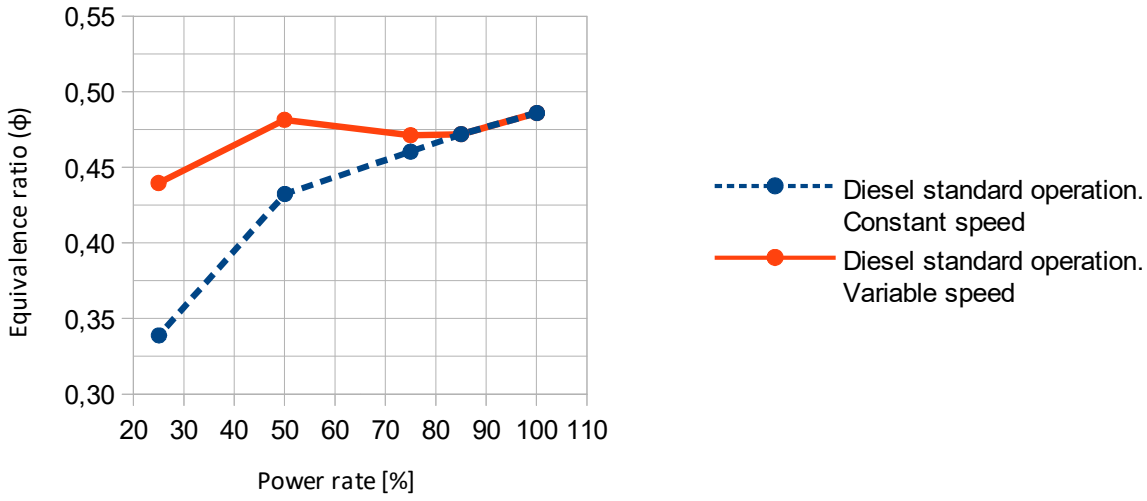

Figure 25: Equivalence ratio diagram. Diesel – Standard.

Power rate	100,075	85,030	75,000	50,051	25,062	%
Intake mass flow	23,769	21,047	19,193	12,945	9,475	kg/s
Alpha	30,249	31,145	31,930	33,996	43,384	-
Equivalence ratio (ϕ)	0,486	0,472	0,460	0,432	0,339	-

Table 19: Fuel-air mixture data. Diesel - Standard. Constant speed.

Power rate	100,075	85,030	74,994	49,983	24,976	%
Intake mass flow	23,769	21,047	18,634	12,486	6,688	kg/s
Alpha	30,249	31,145	31,192	30,533	33,435	-
Equivalence ratio (ϕ)	0,486	0,472	0,471	0,481	0,440	-

Table 20: Fuel-air mixture data. Diesel - Standard. Variable speed.

6.2. Blow-off and HyTC operating points setting:

The final goal of this process is to identify which is the highest torque that can be delivered to the turbocharger shaft, without reducing the air supplied by the compressor to a limit where it becomes inappropriate to obtain the desired power on the engine. As this torque will be used to produce electric power, it will be intended to use the highest torque as possible.

The analysis is performed in two steps, where the first is done by an optimization process that starts by increasing the efficiency of both compressor and turbine. By these means, it is intended to enhance the compressor performance to compensate for the extra work required and maximize the airflow of the turbocharger without reaching critical performance. After that, by simulating a blow-off valve on the intake manifold to study which amount of air can be expelled, the cylinder can be fed with the precisely controlled quantity of air. This will give the information about which is the lowest airflow that the engine needs in order to operate in a given power rate. In the second phase, the engine is fed with the same airflow obtained previously, but this time such airflow is not obtained by expelling excess air through the blow-off valve but by reducing the turbocharger speed as an extra resistant torque is applied to simulate the operation with an electric generator. To compensate for the reduction of the airflow in the intake manifold caused by the delivered torque, the same compressor and turbine efficiency enhancement from the operation in the blow-off valve are kept.

The optimization process using the blow-off valve is carried out considering two criteria. The first is that the mass flow of air used by the engine to develop a specific power has to be the minimum as possible. The second one is to maximize the blow-off valve throat diameter so a larger part of the charge air can be rejected, leading to the highest operating condition of the compressor that at the same time allows the first criteria. This means that the fuel flow ratio has to be regulated and also the diameter of the blow-off valve to achieve the highest performance of the compressor and see how many charge air produced by the compressor can be rejected without compromising the power requested to the engine. This process will cause the brake specific fuel consumption to increase.

However, there are several restrictions on the acceptable operating conditions of the compressor. The first of them is the AFR which cannot be lower than 28. Another limitation is imposed by the compressor map as the operating points must remain inside its limits. The same simulator also represents a limitation because it will start to produce irregular results for the compressor if a certain amount of air is blown from the intake manifold. This happens when the air flow of the compressor is out of the range included on the data tables extracted from the compressor map, and which the simulator uses to develop the required calculations. This last restriction can be visualized, during the simulations on the plotted compressor map, when the operating point is located far outside the map limits.

This optimization process is carried out for five power conditions, each one corresponding to a percentage of the maximum continuous rate MCR (100%, 85%, 75%, 50%, and 25%). At the same time, the analysis takes into account the operation of the engine in constant and variable speed mode.

6.2.1. Blow-off valve operation:

This section includes the results of the optimization process developed in blow-off valve operation. Observing in the first instance the compressor data, the map indicates higher volume flows and lower pressure ratios when compared to the engine on standard operation, and it's also possible to see how on variable speed it works with lower volume flow and pressure ratios than in constant speed. On blow-off operation, it would be expected to operate the engine with the same air mass-flow in the cylinder intake ports than in standard operation, however, during the optimization process to expel the maximum air, the cylinder intake airflow has been reduced, and consequently the gas flow in the turbine. For this reason, the compressor map indicates a lower performance of the turbocompressor. To maintain the engine power output it has been necessary to increase the fuel flow. Now the operating points are closer to the choke limit, which is slightly surpassed when the engine operates at the constant speed lower power rates.

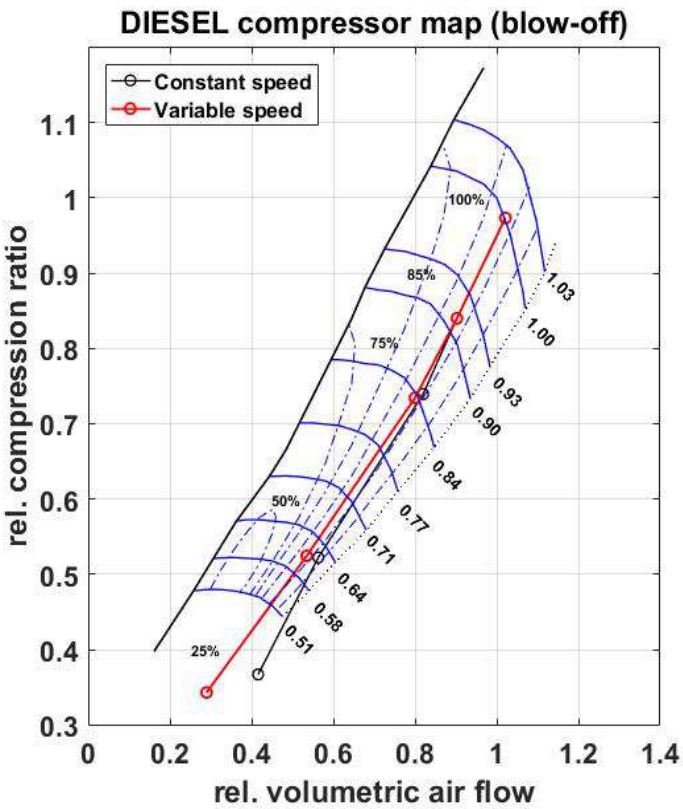

Figure 26: Compressor map. Diesel – Blow-off.

Power rate	99,837	85,076	74,967	50,008	25,018	%
Efficiency	0,808	0,802	0,804	0,803	0,849	-
Volume flow	7,930	7,754	7,646	6,777	6,031	m^3/s
Speed	22.045,178	20.210,152	18.794,741	13.325,244	8.657,980	min^{-1}
Pressure ratio	4,129	3,563	3,160	2,115	1,579	-
Compressor power	4.472,285	3.487,272	2.807,877	1.195,918	475,238	kW

Table 21: Compressor operation conditions. Diesel - Blow-off. Constant speed.

Power rate	99,837	85,076	74,995	49,995	25,036	%
Efficiency	0,808	0,802	0,809	0,843	0,914	-
Volume flow	7,930	7,754	7,502	6,211	4,433	m^3/s
Speed	22.045,178	20.210,152	18.425,648	13.597,246	6.515,584	min^{-1}
Pressure ratio	4,129	3,563	3,115	2,225	1,455	-
Compressor power	4.472,285	3.487,272	2.687,454	1.154,222	254,178	kW

Table 22: Compressor operation conditions. Diesel - Blow-off. Variable speed.

Power	99,837	85,076	74,967	50,008	25,018	%
Exhaust mass flow	23,7564	20,6561	18,4736	12,7023	9,5519	kg/s
Temperature drop	161,338	144,746	130,546	79,525	43,782	K
Cp	1.168,871	1.168,233	1.167,692	1.161,585	1.148,769	J/kgK
Turbine power	4.511,217	3.519,098	2.838,517	1.188,947	492,223	kW

Table 23: Turbine operation conditions. Diesel - Blow-off. Constant speed.

Power rate	99,837	85,076	74,995	49,995	25,036	%
Exhaust mass flow	23,7564	20,6561	17,7573	11,9841	6,9548	kg/s
Temperature drop	161,338	144,746	129,038	82,568	31,556	K
Cp	1.168,871	1.168,233	1.169,576	1.169,036	1.157,663	J/kgK
Turbine power	4.511,217	3.519,098	2.696,938	1.164,645	258,311	kW

Table 24: Turbine operation conditions. Diesel - Blow-off. Constant speed.

The equivalence ratio line follows a similar tendency to the engine in standard operation, with the fuel-air mixture permanently in lean conditions and reducing the fuel proportion as the power rate decreases.

Likewise, it can be noticed a rise in the equivalence ratio on the variable speed operation due to the airflow reduction derived from the engine speed decrease. Differing to the engine standard operation, on the 100% to 75% MCR range, the equivalence ratio stays almost constant as the ratio of air rejected through the blow-off valve increases when power is reduced, leading to a stable equivalence ratio close to 0,50.

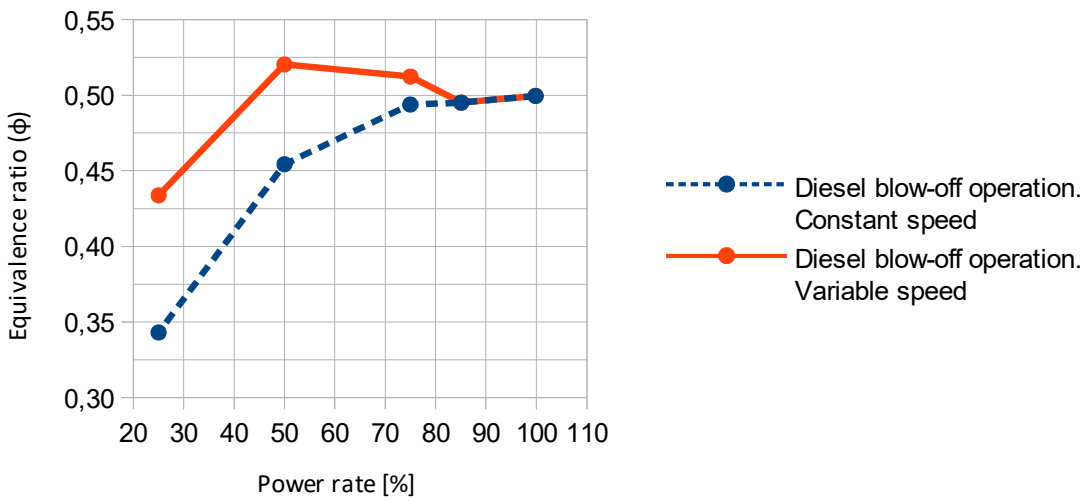

Figure 27: Equivalence ratio diagram. Diesel - Blow-off.

Power rate	99,837	85,076	74,967	50,008	25,018	%
Intake mass flow	23,124	20,111	17,987	12,366	9,360	kg/s
Alpha	29,429	29,691	29,768	32,354	42,855	-
Equivalence ratio (ϕ)	0,500	0,495	0,494	0,454	0,343	-

Table 25: Fuel-air mixture data. Diesel - Standard. Constant speed.

Power rate	99,837	85,076	74,995	49,995	25,036	%
Intake mass flow	23,124	20,111	17,279	11,652	6,779	kg/s
Alpha	29,429	29,691	28,693	28,250	33,888	-
Equivalence ratio (ϕ)	0,500	0,495	0,512	0,520	0,434	-

Table 26: Fuel-air mixture data. Diesel - Standard. Constant speed.

Here, the information regarding the specific fuel consumption of the engine is documented. Compared to standard operation, an increase can be noticed:

Power rate	99,837	85,076	74,967	50,008	25,018	%
BSFC	189,976	192,172	194,556	201,445	230,467	g/kWh

Table 27: Brake specific fuel consumption. Diesel - Blow-off. Constant speed.

Power rate	99,837	85,076	74,995	49,995	25,036	%
BSFC	189,976	192,172	193,758	199,140	210,931	g/kWh

Table 28: Brake specific fuel consumption. Diesel - Blow-off. Variable speed.

The next tables indicate the air expelled through the blow-off valve on each power condition, as well as the ratio of air rejected compared to the compressor mass flow. The tables are complemented with the blow-off air versus power rate diagram.

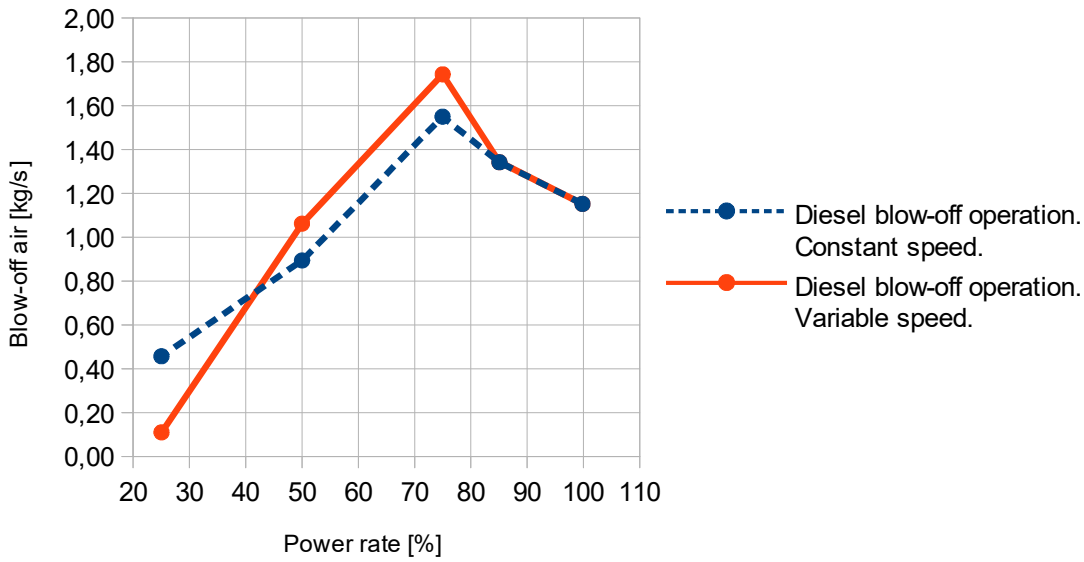

Figure 28: Blow-off air mass-flow. Diesel - Blow-off.

Power rate	99,837	85,076	74,967	50,008	25,018	%
Comp. Massflow	24,277	21,454	19,538	13,26	9,816	kg/s
Blow-off air	1,152	1,342	1,550	0,894	0,457	kg/s
Blow-off ratio	4,747	6,258	7,936	6,744	6,603	%

Table 29: Blow-off air mass-flow. Diesel - Blow-off. Constant speed.

Power rate	99,837	85,076	74,995	49,995	25,036	%
Comp. Massflow	24,277	21,454	19,016	12,715	6,888	kg/s
Blow-off air	1,152	1,342	1,743	1,062	0,110	kg/s
Blow-off ratio	4,474	6,258	9,166	8,355	1,591	%

Table 30: Blow-off air mass-flow. Diesel - Blow-off. Constant speed.

6.2.2. Hybrid turbocharger (HyTC) operation:

As it's been commented before, the results obtained on blow-off operation tests have been used as a reference to set the correct generator torque in order to achieve the same intake air conditions on HyTC than in blow-off operating mode. The results obtained after defining the torque, are commented in the next paragraphs.

The compressor map in this case, differing from the engine on blow-off valve operation mode, shows how the compressor volume flow has decreased when compared to the standard operation mode. This reduction that can be also noticed in the rotational speed, is caused by the supplementary load added to the turbocharger shaft by the electric generator. In HyTC operation, the turbine offers a similar enthalpy drop than in blow-off operation, but in this case, the generator torque limits the acceleration of the shaft, avoiding it from offering the same compressor performance. However, as no air is rejected through the blow-off valve, the cylinders can receive the same airflow they received in blow-off operation. This airflow is also smaller than the airflow from standard operation, so to maintain the same power output, the fuel flow has to be increased. Regarding the compression ratio, the behaviour is considerably similar to the blow-off operation, although the values are slightly lower. It's possible to see how the slopes of the lines are also fairly constant, especially on constant speed conditions, providing as in the case of standard operation, a more stable response of the compressor. No choke conditions are registered.

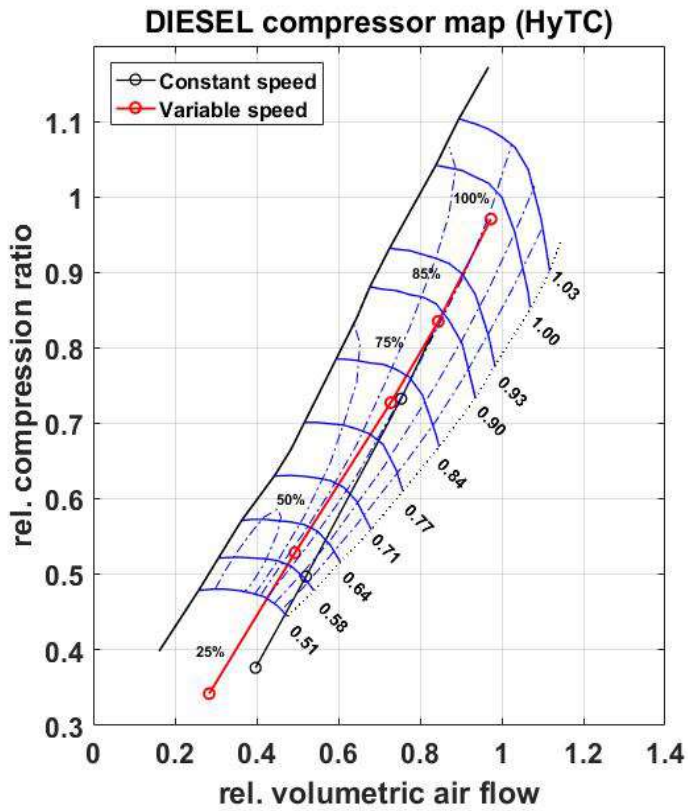

Figure 29: Compressor map. Diesel – HyTC.

Power rate	99,859	84,930	74,953	50,038	25,051	%
Efficiency	0,833	0,835	0,828	0,826	0,871	-
Volume flow	7,483	7,179	7,026	6,271	5,746	m^3/s
Speed	21.657,084	19.720,774	18.160,548	12.757,801	8.434,278	min^{-1}
Pressure ratio	4,118	3,537	3,107	2,081	1,594	-
Compressor power	4.124,241	3.109,053	2.468,029	1.026,969	460,761	kW

Table 31: Compressor operation conditions. Diesel - HyTC. Constant speed.

Power rate	99,859	84,930	74,941	49,963	25,017	%
Efficiency	0,833	0,835	0,835	0,873	0,915	-
Volume flow	7,483	7,179	6,803	5,630	4,357	m^3/s
Speed	21.657,084	19.720,774	17.904,976	13.125,845	6.439,456	min^{-1}
Pressure ratio	4,118	3,537	3,086	2,219	1,449	-
Compressor power	4.124,241	3.109,053	2.346,829	1.012,074	245,701	kW

Table 32: Compressor operation conditions. Diesel - HyTC. Variable speed.

Power rate	99,859	84,930	74,953	50,038	25,051	%
Exhaust mass flow	23,7719	20,5777	18,4038	12,5551	9,6391	kg/s
Temperature drop	161,140	144,690	130,262	79,324	44,011	K
Cp	1.168,752	1.168,348	1.167,864	1.162,339	1.148,332	J/kgK
Turbine power	4.508,621	3.504,385	2.821,641	1.172,199	499,315	kW

Table 33: Turbine operation conditions. Diesel - HyTC. Constant speed.

Power rate	99,859	84,930	74,941	49,963	25,017	%
Exhaust mass flow	23,7719	20,5777	17,8021	11,9288	6,9263	kg/s
Temperature drop	161,140	144,690	128,636	82,431	31,477	K
Cp	1.168,752	1.168,348	1.169,249	1.169,297	1.157,890	J/kgK
Turbine power	4.508,621	3.504,385	2.695,319	1.157,348	256,609	kW

Table 34: Turbine operation conditions. Diesel - HyTC. Variable speed.

During hybrid turbocharger operation, it's possible to see a similar progression of the equivalence ratio to the blow-off operation mode. On constant speed, for the highest power rates (100% to 75% MCR), the equivalence ratio remains almost constant until it starts to decrease as power is reduced. Again, on the power rate range affected by variable speed operation, (75% to 25%), the reduction of air flow caused by the engine speed, increases the equivalence ratio (compared to constant speed operation).

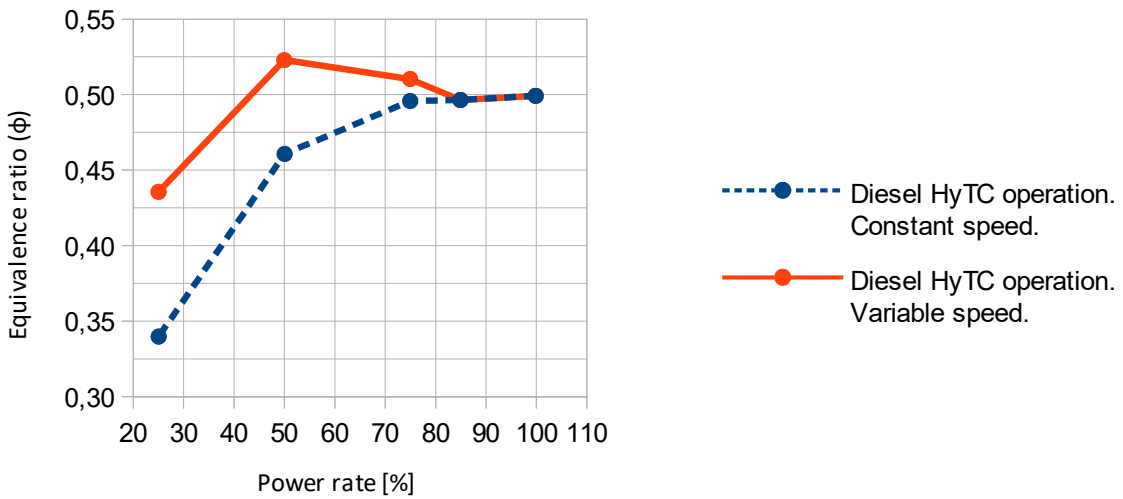

Figure 30: Equivalence ratio diagram. Diesel - HyTC.

Power rate	99,859	84,930	74,953	50,038	25,051	%
Intake mass flow	23,140	20,033	17,917	12,218	9,447	kg/s
Alpha	29,448	29,611	29,652	31,908	43,254	-
Equivalence ratio (ϕ)	0,499	0,496	0,496	0,461	0,340	-

Table 35: Fuel-air mixture data. Diesel - Standard. Constant speed.

Power rate	99,859	84,930	74,941	49,963	25,017	%
Intake mass flow	23,140	20,033	17,318	11,597	6,750	kg/s
Alpha	29,448	29,611	28,805	28,116	33,746	-
Equivalence ratio (ϕ)	0,499	0,496	0,510	0,523	0,436	-

Table 36: Fuel-air mixture data. Diesel - Standard. Constant speed.

The specific fuel consumption of the engine is included in the following tables. It has to be considered these BSFC values are referred to the power output of the engine, and the power obtained by the generator does not compute in this calculation. Like on blow-off operation, the values have increased compared to standard operation.

Power rate	99,859	84,930	74,953	50,038	25,051	%
BSFC	189,93	192,280	194,593	202,032	230,166	g/kWh

Table 37: Brake specific fuel consumption. Diesel - HyTC. Constant speed.

Power rate	99,859	84,930	74,941	49,963	25,017	%
BSFC	189,934	192,280	193,648	199,265	211,089	g/kWh

Table 38: Brake specific fuel consumption. Diesel - HyTC. Variable speed.

After observing the behaviour of the engine and the turbocharger, now the electric generator data can be discussed. The operation points defined for the engine using the HyTC system are achieved by applying the corresponding torque to the turbocharger shaft by means of the electric generator. This is basically the working principal of the heat recovery system, allowing to obtain electric power from the exhaust gases of the engine. The next paragraphs describe the capacities of the generator to control the intake air supply and to produce electric power so as to enhance the overall efficiency of the power plant.

As described in the model description *Chapter 5.* , the model can provide information about the generator power output as a function of the torque and angular velocity. These values have been calculated considering the efficiency of the generator η_{EG}, which is rated at 0,95. It is also interesting to analyse what increment of power represents the electric generator power *EGP* confronted to the engine brake effective power (relative electric power, *REP*). The selected torque values are shown on tables *39, 40* and on the diagram on *Figure 31.*

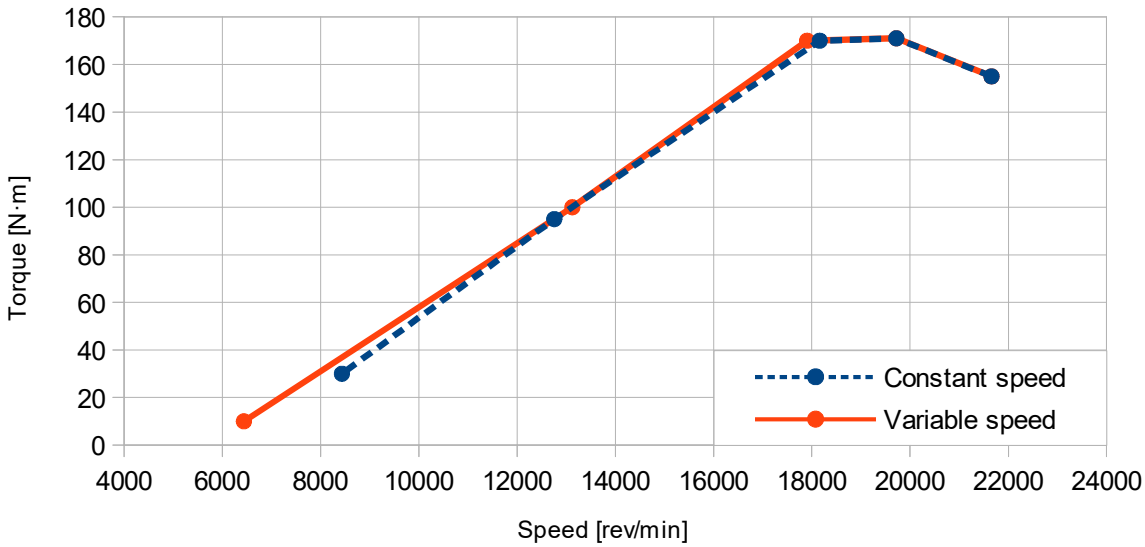

Figure 31: Generator torque-speed diagram. Diesel – HyTC.

Power rate	99,859	84,930	74,953	50,038	25,051	%
Speed	21.657,084	19.720,774	18.160,548	12.757,801	8.434,278	min^{-1}
Torque	155	171	170	95	30	N·m
EGP	333,952	335,484	307,136	120,573	25,172	kW
P_E	11983,107	10191,579	8994,359	6004,521	3006,097	kW
REP	2,787	3,292	3,415	2,008	0,837	%

Table 39: Generator data. Diesel - HyTC. Constant speed.

Power rate	99,859	84,930	74,941	49,963	25,017	%
Speed	21657	19720	17904	13125	6439	min^{-1}
Torque	155	171	170	100	10	N·m
EGP	333,951	335,471	302,797	130,572	6,406	kW
P_E	11983,107	10191,579	8992,941	6005,678	3002,052	kW
REP	2,787	3,292	3,367	2,174	0,213	%

Table 40: Generator data. Diesel - HyTC. Variable speed.

The torque-speed line is adjusted to match in both constant and variable speed conditions, so it can be considered as a proper torque-speed line for the generator. It is possible to appreciate a similar behaviour with the blow-off air when diagrams for blown air (*Figure 28*) and torque *(Figure 31)* are confronted.

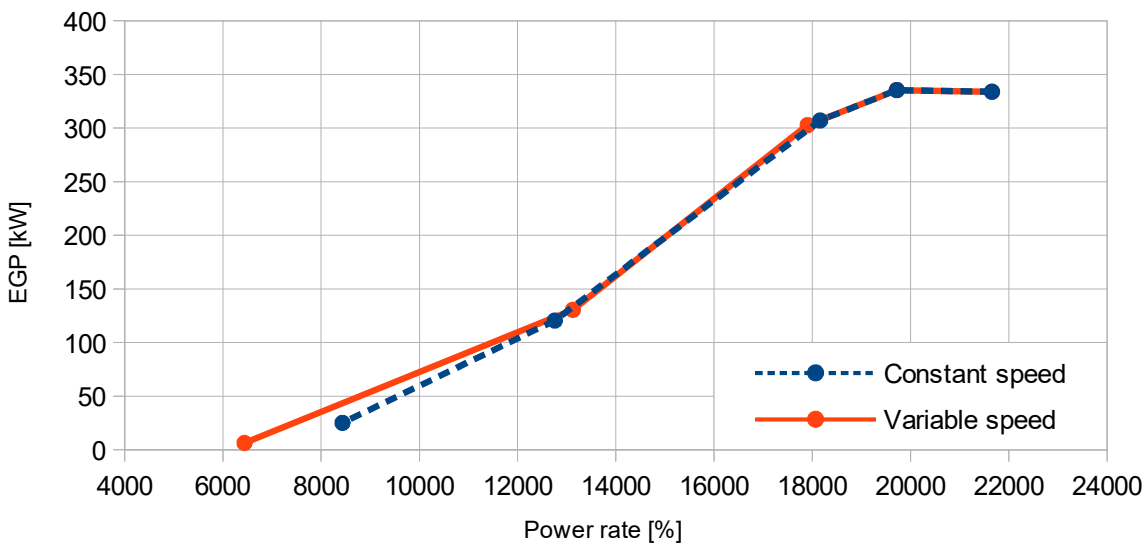

Figure 32: Generator power-speed diagram. Diesel – HyTC.

The torque line traced by the generator allows to optimize the power output at the highest engine power rates, from 75% to 100%. As described in the hybrid turbocharger chapter, the torque is a function of the rotation speed, but also from the load, so if the load is reduced while the rotation speed rises, the torque can be kept constant or reduced as the speed increases, in order to obtain a stable power output.

The obtained data regarding the generator power indicates a considerably stable output during the 100% to 85% MCR range, corresponding to the most frequently required power settings of the engine, although the highest relative electric power is registered for both, constant and variable speed conditions, at 75% MCR. After this, as the engine power rate decreases, both HyTC generator power output and REP evolve in the same way.

Now it is possible to assume a plausible generator power output with a considerably stable value around 330 kW, which can be delivered when the engine power rates correspond to those required for most of the existing ships to navigate from cruise speed to maximum speed (100% to 85% MCR).

Now that the power obtained with the generator is defined, it is possible to analyse how does it affect the overall efficiency of the power-plant η_{TPP}. The method for the calculation of the power-plant efficiency is commented on the model description *Chapter 5.* , where the electrical power of the hybrid turbocharger, properly adjusted with its respective efficiency, is taken into account. The results obtained for the efficiencies with the HyTC system are contrasted with the values for the engine on standard operation mode so as to identify the increase *Δ*.

Power rate	100	85	75	50	25	%
Standard efficiency	44,485	43,950	43,580	41,978	36,646	%
HyTC efficiency	45,626	45,290	44,805	42,569	36,936	%
Efficiency Δ	2,501	2,959	2,734	1,388	0,785	%

Table 41: Power-plant total efficiency. Diesel standard vs. HyTC. Constant speed.

Power rate	100	85	75	50	25	%
Standard efficiency	44,485	43,950	43,847	42,692	39,875	%
HyTC efficiency	45,626	45,290	45,003	43,232	40,025	%
Efficiency Δ	2,501	2,959	2,569	1,249	0,375	%

Table 42: Power-plant total efficiency. Diesel standard vs. HyTC. Variable speed.

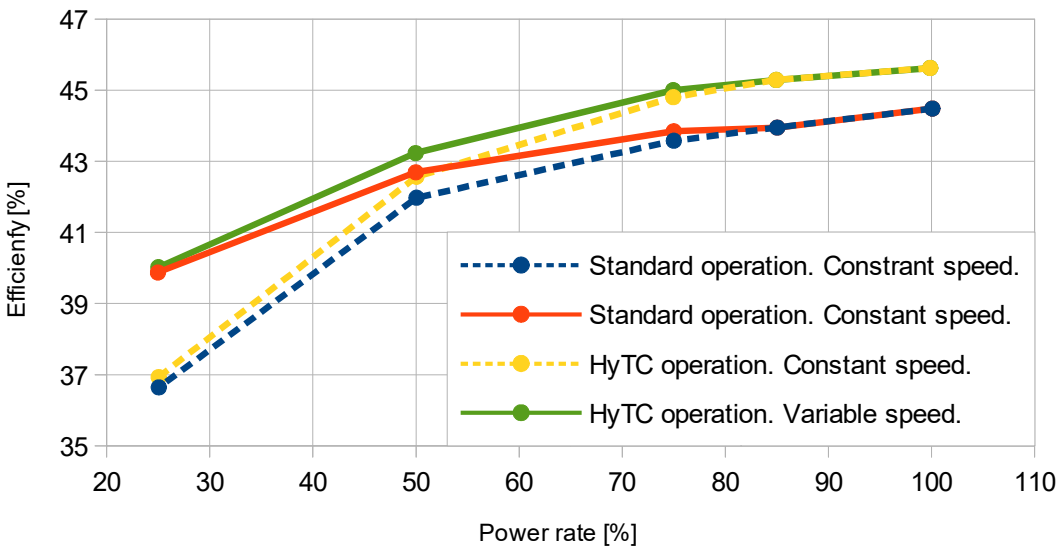

Figure 33: Power-plant total efficiency. Diesel standard vs. HyTC.

It can be observed that on standard operation, the efficiency of the power-plant differs substantially between constant and variable speed lines, where a significant drop on the efficiency is perceived on constant speed, in contrast to variable speed mode for the same load conditions. Once the hybrid turbocharger is active, the increase in the total efficiency varies depending on each load condition, but the highest enhancements are achieved on the higher power rates, so that it can match the usual power requirements of the ship. This increase reaches 3% around 85% MCR, corresponding to the most common power setting for cruise speed.

7. Diesel and Gas fuel operation comparison

This chapter details the comparison of the results obtained during the simulations of the MAN V12 51/60 DF engine on Diesel operation and the natural gas tests obtained from [15] . As it's been commented in previous chapters, the engine intake valve timing is adjustable, so it can operate with both Diesel and Otto characteristics depending on the fuel. However, each cycle will require different air quantities, inducing to different performances of the engine and turbocompressor which will be discussed. In the following paragraphs, the discussion will be focused on the operating modes considered by the manufacturer and the enhanced variations developed only in the numerical models for this study:

- Natural gas fuel with blow-off valve, including constant and variable speed.
- Natural gas fuel with HyTC, including constant and variable speed.
- Diesel fuel standard operation (no blow-off valve), including constant and variable speed.
- Diesel fuel blow-off valve, including constant and variable speed.
- Diesel fuel with HyTC, including constant and variable speed.

7.1. Manufacturer based models

This section is focused on the comparison of the numerical models based on the configurations developed by the engine manufacturer, i.e., natural gas fuel with blow-off valve and Diesel fuel standard operation. All Diesel standard operation data corresponds to the values included in *Chapter 6.1.2.*

7.1.1. Compressor data:

In this section, the information regarding to the compressor maps for natural gas fuel with blow-off valve and Diesel fuel standard operation are contrasted. Like in previous chapters, the diagrams are provided in relative numbers including information for the volumetric airflow, compression ratio, efficiency, and speed.

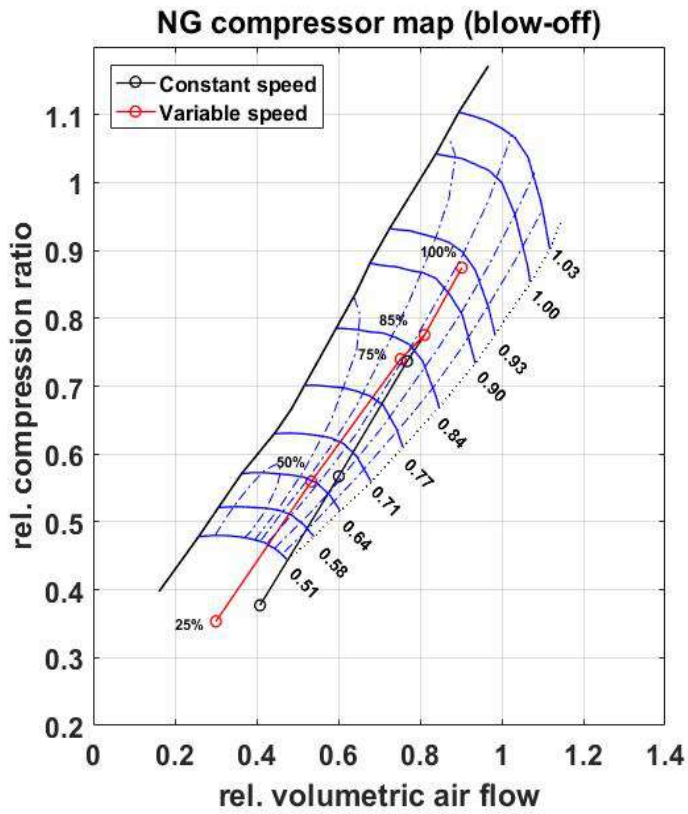

Figure 34: Compressor map. Natural gas – Blow-off.

Power rate	100,035	85,023	75,009	50,019	25,044	%
Efficiency	0,797	0,802	0,804	0,825	0,838	-
Volume flow	7,565	7,352	7,202	6,661	5,921	m^3/s
Speed	16781,467	15484,125	14957,718	12182,931	7090,396	min^{-1}
Pressure ratio	3,710	3,290	3,125	2,405	1,600	-
Compressor power	3645,089	2904,786	2607,025	1471,712	495,731	kW

Table 43: Compressor operation conditions. Natural Gas - Blow-off. Constant speed

Power rate	100,035	85,023	75,015	49,970	25,016	%
Efficiency	0,797	0,802	0,812	0,846	0,889	-
Volume flow	7,565	7,352	7,009	5,931	4,518	m^3/s
Speed	16781,467	15484,125	14908,289	11633,364	5662,944	min^{-1}
Pressure ratio	3,710	3,290	3,138	2,374	1,499	-
Compressor power	3645,089	2904,786	2536,098	1252,305	293,729	kW

Table 44: Compressor operation conditions. Natural Gas - Blow-off. Variable speed.

Power rate	100,035	85,023	75,009	50,019	25,044	%
Exhaust mass flow	20,2207	17,5858	15,9194	11,5002	7,0134	kg/s
Temperature drop	153,177	140,354	139,046	108,601	59,902	K
Cp	1.176,818	1.176,899	1.177,725	1.178,479	1.180,384	J/kgK
Turbine power	3.645,012	2.904,665	2.604,920	1.469,767	494,401	kW

Table 45: Turbine operation conditions. Natural Gas - Blow-off. Constant speed.

Power rate	100,035	85,023	75,015	49,970	25,016	%
Exhaust mass flow	20,2207	17,5858	15,3146	10,8492	5,9511	kg/s
Temperature drop	153,177	140,354	136,98	98,234	42,069	K
Cp	1.176,818	1.176,899	1.176,931	1.175,036	1.172,707	J/kgK
Turbine power	3.645,012	2.904,665	2.468,721	1.254,206	294,624	kW

Table 46: Turbine operation conditions. Natural Gas - Blow-off. Variable speed.

Tables *43 44* and *Figure 34* indicate a considerable reduction of all the parameters of the compressor on Natural Gas mode compared to Diesel standard mode, as a consequence of the lower air consumption of the engine operating in this fuel, which causes the turbine to receive less gas-flow and reduce the performance of the turbocharger.

7.1.2. Fuel-air mixture data:

As it has been commented in previous sections of this document, the adjustment of the fuel-oxidizer mixture in an engine defines many characteristics of its performance. The influence of the mixture in the diesel engine has been analysed in *Chapter 6.* , but in this section, it's important to identify the differences between the required mixture in a Diesel and a natural gas engine.

Although it contains other components depending on the source, natural gas is essentially formed by methane CH_4, so generally, the stoichiometric ratio of 17,4:1 is taken for NG. In natural gas engines (like others subjected to the Otto thermodynamic cycle), the main feature for the power regulation is the intake

air, and this regulation is generally achieved with a throttle valve, although in the MAN V12 51/60 DF engine this is done with the blow-off system in order to obtain the control of the turbocompressor charge air. After this, the air receives the required and properly atomized fuel to create the mixture. The most common method to add the fuel is in the intake manifold before entering the cylinders, like it's done in the engine studied in this document, but in some designs, the fuel is directly injected in the combustion chamber.

In turbocharged engines, the correct amount of air has to be controlled, because if an excessive quantity is introduced and pressure is too high, after adding the fuel, the mixture can self-ignite during the compression stroke, previous to the expected ignition timing. This is especially notable in petrol engines, but thanks to the higher octane number and wider flammability range of the natural gas, NG engines are less prone to this phenomenon. By these means, these engines can operate in higher compression ratios, increasing its efficiency, and use leaner mixtures to reduce the temperature and the generation of NO_X.

Also, to achieve the highest efficiencies, natural gas engines use ultra-lean air-fuel ratios, with fuel-air equivalence ratios reaching values around ϕ=0.5 (λ=2). To make it possible to operate with these ratios, the partially stratified combustion method based on the use of pilot fuels, offer a flame kernel hot enough to obtain the combustion and its proper propagation.

The next tables and diagram, indicate the values for the intake air mass flow, alpha coefficient and equivalence ratio for the engine in natural gas operation.

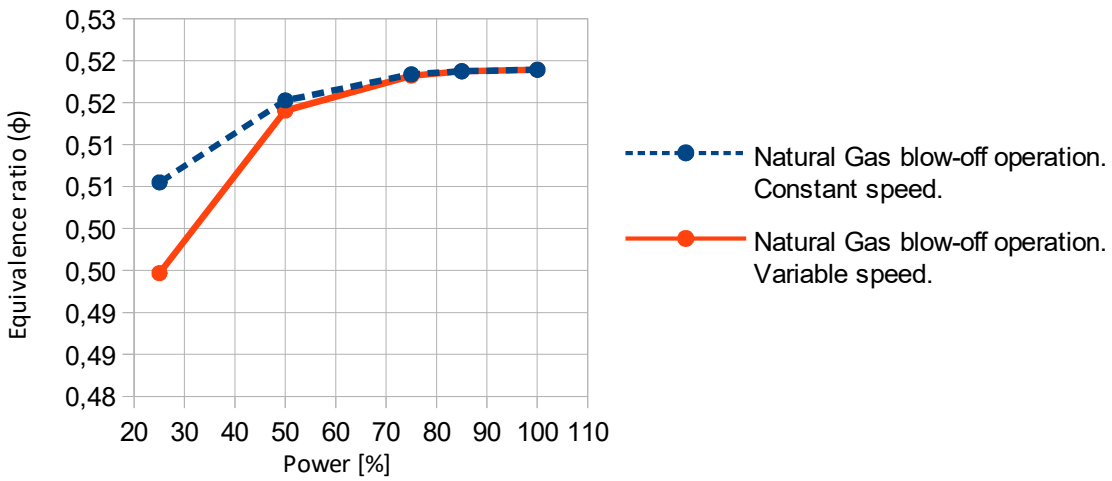

Figure 35: Equivalence ratio diagram. Natural gas – Blow-off.

Power rate	100,035	85,023	75,009	50,019	25,044	%
Intake mass flow	19,685	17,120	15,498	11,196	6,828	kg/s
Alpha	33,532	33,543	33,567	33,769	34,422	-
Equivalence ratio (ϕ)	0,519	0,519	0,518	0,515	0,505	-

Table 47: Fuel-air mixture data. Natural Gas - Blow-off. Constant speed.

Power rate	100,035	85,023	75,015	49,970	25,016	%
Intake mass flow	19,685	17,120	15,315	10,562	5,794	kg/s
Alpha	33,532	33,543	33,577	33,850	35,174	-
Equivalence ratio (ϕ)	0,519	0,519	0,518	0,514	0,495	-

Table 48: Fuel-air mixture data. Natural Gas - Blow-off. Variable speed.

In the results displayed in tables *47, 48* and *Figure 35*, it is possible to see a considerable difference in the intake airflow values between NG and Diesel operation to obtain the same power output. As it's known, the stoichiometric ratios of NG and Diesel are 17,4:1 and 14,7 respectively, but the heating value of the fuel has to be considered too. The net calorific values of both fuels can be different depending on their exact composition, but in this study 49 MJ/kg is considered for NG and 42,7 MJ/kg for Diesel fuel. This means that Diesel offers around 12,8 % less heat per weight unit than natural gas, but as the stoichiometric ratio indicates, it requires less air to combust and release that heat. Consequently, in an ideal situation, air quantities should be similar in both fuel types, but on each fuel mode, different equivalence ratios and airflows are recorded as a consequence of several factors.

The use of the Miller cycle to change the intake valve timing and modify the compression ratio, which modifies the fresh air admission in the cylinder (to allow a relative expansion before the compression stroke) depending on which fuel is being used. By this method (Miller cycle), on NG operation the intake air in the cylinders is reduced. This is required to avoid the self-ignition in natural gas without compromising the higher amounts of air that Diesel operation requires to obtain the correct pressure and temperature in the combustion chamber to obtain the proper atomization of the fuel.

Another of the main variations in the fuel-air mixture results between each fuel mode is their stability. It can be observed that in NG mode, the equivalence ratio is maintained in a more narrow range than in Diesel, which is caused by the different power control method used in each fuel mode. In natural gas, the

fuel is added to obtain a correct mixture depending on the intake air, which is precisely controlled, so the mixture can be kept more constant in all power rates. In the other hand, in Diesel there is no such control.

The next tables *49* and *50* include the specific fuel consumption results, indicating lower values for NG operation (see tables *13* and *14* for Diesel standard reference). This is a consequence of the different net calorific values (NCV) of each fuel. As natural gas NCV is higher, it produces lower BSFC values.

Power rate	100,035	85,023	75,009	50,019	25,044	%
BSFC	158,453	162,082	166,193	178,961	213,849	g/kWh

Table 49: Brake specific fuel consumption. Natural gas – Blow-off. Constant speed.

Power rate	100,035	85,023	75,015	49,970	25,016	%
BSFC	158,453	162,082	164,164	168,595	177,776	g/kWh

Table 50: Brake specific fuel consumption. Natural gas - Blow-off. Variable speed.

7.2. Enhanced models

This section is focused on the analysis of the models developed in order to finally obtain the hybrid turbocharger results. For the study in NG mode, the blow-off system is replaced by the turbocompressor shaft torque modulation method provided by the HyTC generator, so the airflow reduction in the engine cylinders is achieved by adding the resistive torque on the turbocompressor shaft instead of the blow-off valve.

7.2.1. Compressor data

On Diesel operation, the blow-off valve and also the HyTC system, which are not conceived in the manufacturer's design, are implemented. However, as it's been seen on the previous section, on Diesel mode, the engine has difficulties to provide a certain output power without the correct airflow. So since the compressor has to supply enough air to the cylinders with the presence of the blow-off valve or the

generator, the efficiency on both compressor and turbine must be increased modifying the output data of the compressor and turbine map tables. As the efficiencies of the turbocharger components are improved, the turbocharger is led to its maximum performance trying not to cause surge or choke conditions. Then, first with the blow-off valve and after with the HyTC, the intake air is reduced according to the requirements of the corresponding power rate. The next diagrams and tables indicate the compressor and turbine performance on NG mode:

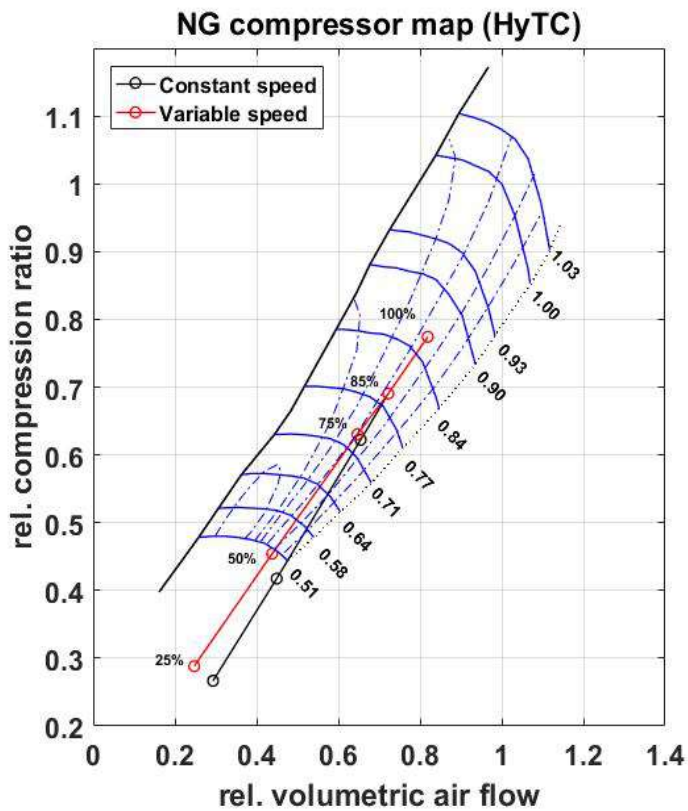

Figure 36: Natural gas compressor map for HyTC operation

Power rate	98,609	85,024	74,954	46,391	25,055	%
Efficiency	0,798	0,797	0,788	0,800	0,898	-
Volume flow	7,443	7,088	6,903	6,137	5,343	m^3/s
Speed	15564,683	14245,048	13186,793	8612,261	3118,165	min^{-1}
Pressure ratio	3,283	2,929	2,640	1,771	1,131	-
Compressor power	2939,965	2306,127	1880,480	705,814	82,921	kW

Table 51: Compressor operation conditions. Natural Gas - HyTC. Constant speed

Power rate	98,609	85,024	74,937	48,607	25,016	%
Efficiency	0,798	0,797	0,801	0,831	0,906	-
Volume flow	7,443	7,088	6,729	5,618	4,266	m^3/s
Speed	15564,683	14245,048	13201,796	9210,976	3523,633	min^{-1}
Pressure ratio	3,283	2,929	2,675	1,927	1,223	-
Compressor power	2939,965	2306,127	1856,073	769,617	114,507	kW

Table 52: Compressor operation conditions. Natural Gas - HyTC. Variable speed

Power rate	98,609	85,024	74,954	46,391	25,055	%
Exhaust mass flow	19,992	17,640	15,985	10,967	7,165	kg/s
Temperature drop	150,216	140,228	138,822	103,540	60,078	K
Cp	1.176,396	1.176,559	1.177,214	1.177,723	1.178,331	J/kgK
Turbine power	3.534,034	2.910,936	2.611,457	1.336,243	506,557	kW

Table 53: Turbine operation conditions. Natural gas - HyTC. Constant speed

Power rate	98,609	85,024	74,937	48,607	25,016	%
Exhaust mass flow	19,992	17,640	15,789	10,694	6,038	kg/s
Temperature drop	150,216	140,228	136,773	96,219	42,154	K
Cp	1.176,396	1.176,559	1.176,445	1.174,343	1.171,350	J/kgK
Turbine power	3.534,034	2.910,936	2.541,268	1.210,894	299,550	kW

Table 54: Turbine operation conditions. Natural gas - HyTC. Variable speed.

As the results from tables *43, 44, 51, 52* and diagrams *34* and *36* indicate, there's an important difference between the compressor performance in blow-off and HyTC operation. Although there's certain variation, in both conditions intake air to the cylinders can be considered similar, and also the exhaust gas mass-flow. For this reason, the turbine can provide similar amounts of power in both conditions. In blow-off operation, the compressor uses all the power delivered by the turbine to provide certain airflow, which is higher than the required by the engine and consequently the excessive charge air has to be expelled. In HyTC operation, the excessive power provided by the turbine is absorbed by the electric generator, preventing the compressor from delivering excessive charge air to the cylinders.

Observing the behaviour of the compressor in Diesel blow-off and Diesel HyTC, higher values than in NG can be noticed in all the parameters, but differing to NG, the difference between blow-off and HyTC results is smaller. The variation on this difference in each fuel mode is caused by the characteristics of the exhaust gases, because in an equal turbine power production situation among NG and Diesel operation, there's a significant variation on how the energy is delivered to the turbine in each fuel mode. In Diesel operation, there's higher exhaust gas mass-flow, but lower exhaust gas temperature and thus, the gas has a lower constant pressure heat capacity c_P (see *Langen* equations (91)and (92)) than in NG mode. As the temperature drop in the turbine is also lower, this means that the energy extracted by the turbine on Diesel mode is more dependant on the mass-flow. Consequently, a reduction in the charge air with the blow-off valve will decrease the mass flow supply, which will be reflected as a more notable power reduction in the turbine. If this happens, the compressor is not capable to sustain the airflow and the turbocharger will suffer critical performance. For this reason, the air rejected cannot be as high as in NG. In the other hand, as the required intake air is lower in NG, more air has to be rejected on blow-off operation, and thus more power is available to be absorbed in HyTC operation.

The turbine data for blow-off operation condition in both NG and Diesel fuel modes can be confronted in the following tables *23, 24, 45* and *46*. If the information for similar mass-flows is compared, it's possible to identify a higher temperature drop and higher heat capacity on NG operation. This indicates the lower dependency on the mass-flow to obtain the turbine power on natural gas mode.

In the following tables and diagram, it can be seen how the air is regulated by the blow-off on natural gas operation. To compensate the excess air in constant speed, the blow-off valve releases a considerable higher amount of air in this condition than in variable speed.

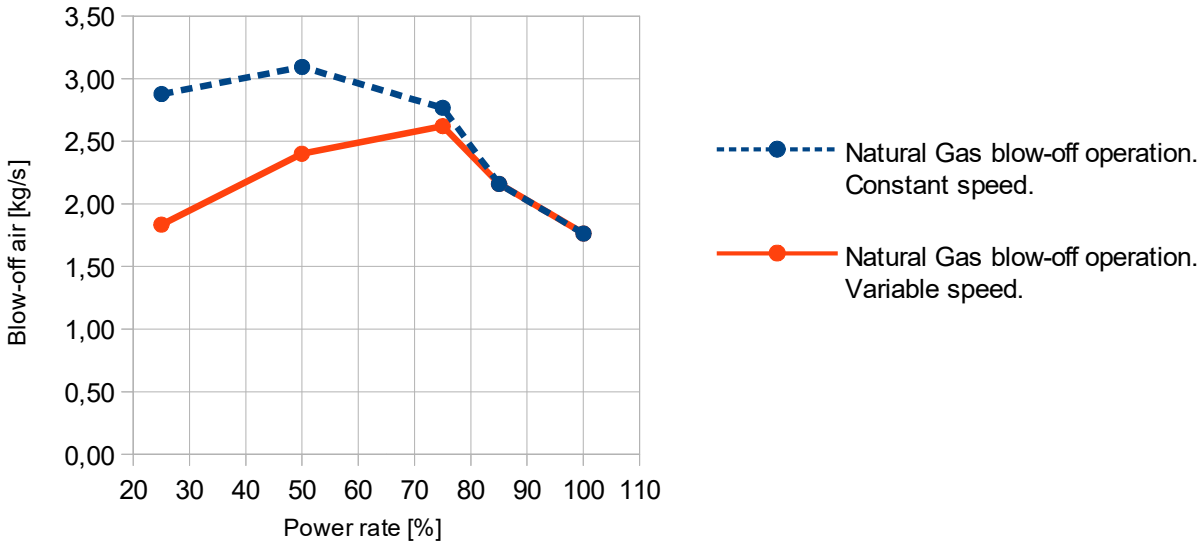

Figure 37: Blow-off air mass-flow. Natural gas - Blow-off.

Power rate	100,035	85,023	75,009	50,019	25,044	%
Comp. Mass-flow	21,448	19,280	18,265	14,289	9,705	kg/s
Blow-off air	1,763	2,159	2,767	3,094	2,878	kg/s
Blow-off ratio	8,220	11,200	15,150	21,650	29,650	%

Table 55: Blow-off air mass-flow. Natural gas - Blow-off. Constant speed.

Power rate	100,035	85,023	75,015	49,970	25,016	%
Comp. Mass-flow	21,448	19,280	18,265	14,289	9,705	kg/s
Blow-off air	1,763	2,159	2,621	2,401	1,834	kg/s
Blow-off ratio	8,220	11,200	14,350	16,800	18,900	%

Table 56: Blow-off air mass-flow. Natural gas - Blow-off. Constant speed.

7.2.2. Fuel-air mixture data

The fuel-air mixture characteristics for the simulations of the enhanced versions of the engine can be observed in this section. The next diagrams and tables include the evolution of the equivalence ratio depending on the power rate for NG mode on HyTC operation:

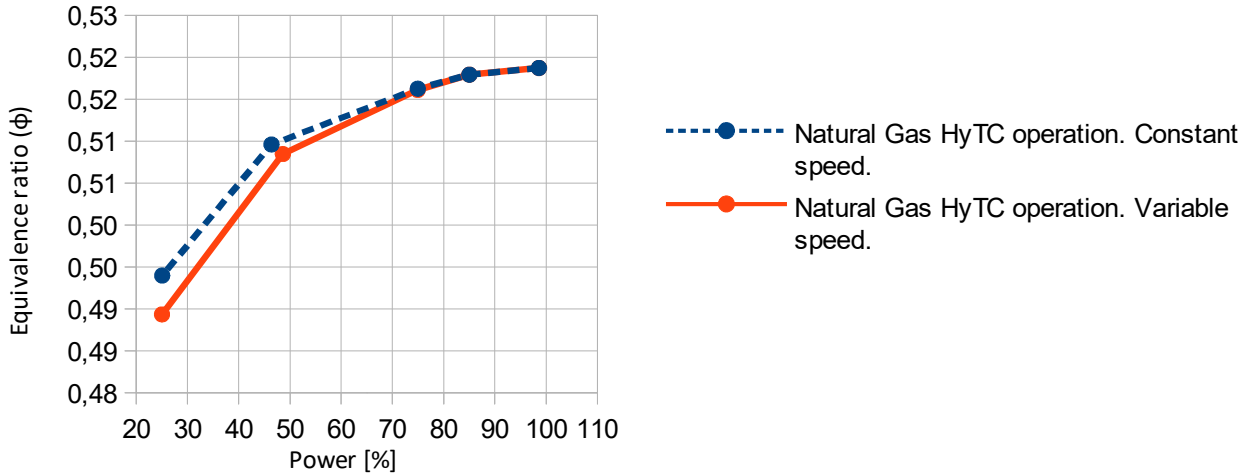

Figure 38: Equivalence ratio diagram. Natural gas – HyTC.

Power rate	98,609	85,024	74,954	46,391	25,055	%
Intake mass flow	19,462	17,173	15,562	10,676	6,975	kg/s
Alpha	33,544	33,596	33,705	34,145	35,225	-
Equivalence ratio (ϕ)	0,519	0,518	0,516	0,510	0,494	-

Table 57: Equivalence ratio diagram. Natural gas – HyTC. Constant speed.

Power rate	98,609	85,024	74,937	48,607	25,016	%
Intake mass flow	19,462	17,173	15,371	10,411	5,879	kg/s
Alpha	33,544	33,596	33,714	34,221	35,558	-
Equivalence ratio (ϕ)	0,519	0,518	0,516	0,508	0,489	-

Table 58: Equivalence ratio diagram. Natural gas – HyTC. Variable speed.

As it's been commented for the manufacturer based models, the enhanced versions also show how the equivalence ratio on NG operation is more constant due to the engine power regulation method.

Similar behaviour on the specific fuel consumption values than in the manufacturer based models can be appreciated now, showing lower values in NG than in Diesel mode due to the different net calorific values.

Power rate	98,609	85,024	74,954	46,391	25,055	%
BSFC	158,863	162,321	166,319	181,980	213,395	g/kWh

Table 59: Brake specific fuel consumption. Natural gas - HyTC. Constant speed.

Power rate	98,609	85,024	74,937	48,607	25,016	%
BSFC	158,863	162,321	164,263	168,987	178,434	g/kWh

Table 60: Brake specific fuel consumption. Natural gas - HyTC. Variable speed.

In Diesel mode, when standard and blow-off/HyTC are contrasted, BSFC values reach slightly higher quotes. This is a consequence of the rejection of charge air (blow-off) or the delivery of torque (HyTC), where lower air mass-flow is sent to the cylinders. To achieve the same engine power output, and as in Diesel mode it can be modified independently from the air-flow, the fuel-flow is increased, rising the specific fuel consumption values.

7.2.3. Electric generator data

With all the engine and turbocharger main data properly contrasted, it is possible to discuss the different performances of the HyTC electric generator on each fuel mode. The tables *61*, *62* include the torque values delivered to the turbocharger shaft, the electric generator power *EGP* and the relative electric power *REP* for Natural Gas mode, which can be contrasted to tables 39 and 40 from Diesel operation.

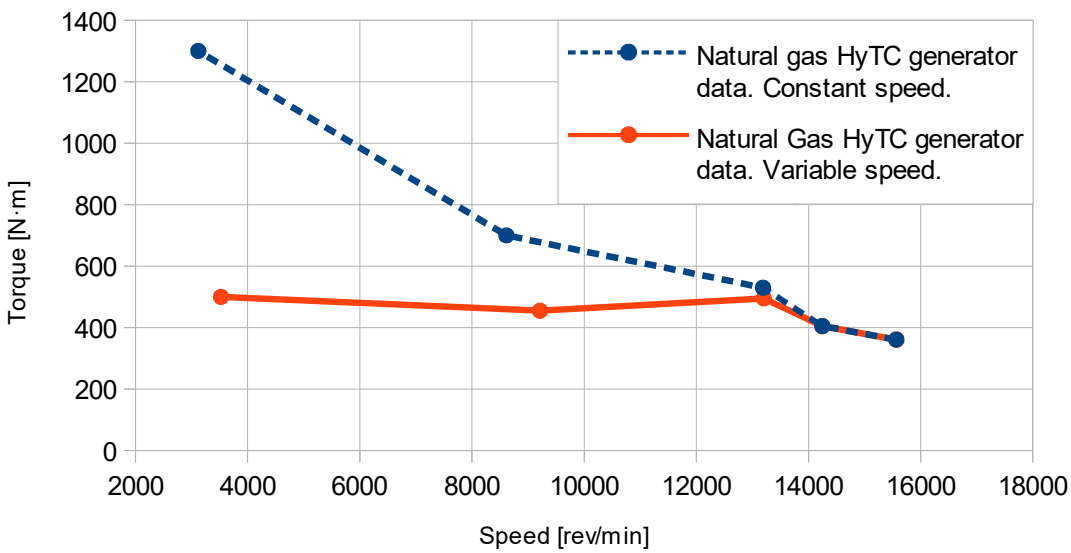

Figure 39: Generator torque-speed diagram. Natural gas – HyTC.

Power rate	98,609	85,024	74,954	46,391	25,055	%
Speed	15.564,683	14.245,048	13.186,793	8.612,261	3.118,165	min^{-1}
Torque	361	405	530	700	1300	N·m
EGP	558,984	573,946	695,292	599,746	403,269	kW
P_E	11983,107	10191,579	8994,359	6004,521	3006,097	kW
REP	4,665	5,632	7,730	9,988	13,415	%

Table 61: Generator data. Natural gas - HyTC. Constant speed.

Power rate	98,609	85,024	74,937	48,607	25,016	%
Speed	15.564,683	14.245,048	13.201,796	9.210,976	3.523,633	min^{-1}
Torque	361	405	495	455	500	N·m
EGP	558,984	573,946	650,115	416,936	175,272	kW
P_E	11983,107	10191,579	8992,941	6005,678	3002,052	kW
REP	4,665	5,632	7,229	6,942	5,838	%

Table 62: Generator data. Natural gas - HyTC. Variable speed.

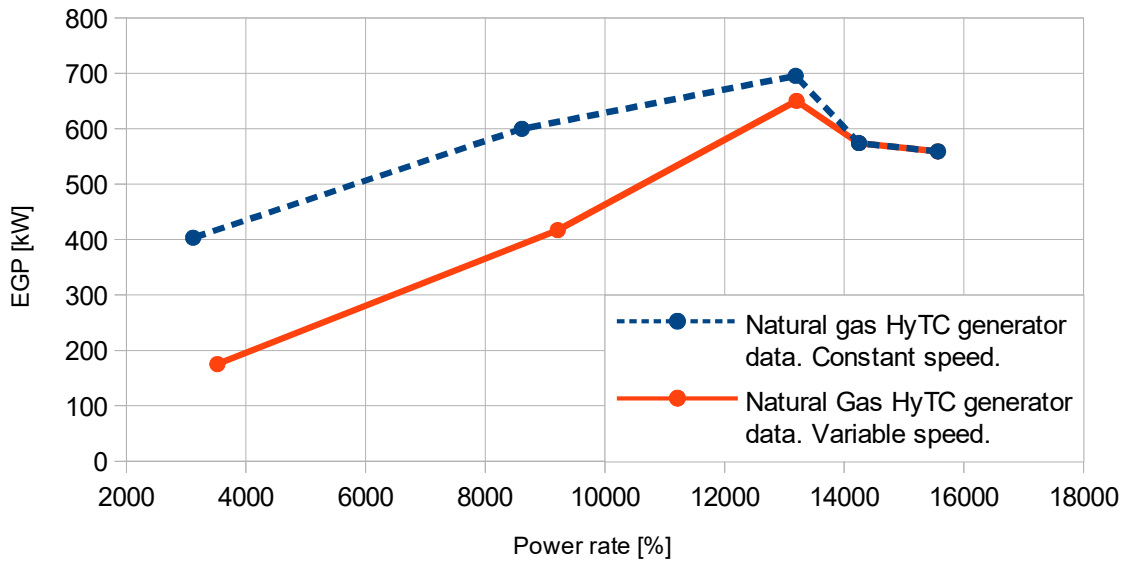

Figure 40: Generator power-speed diagram. Natural gas – HyTC.

The results indicate a considerable difference between the power output in both fuel modes. It can be appreciated how the generation capabilities of the HyTC on natural gas operation are higher than on Diesel,

as a consequence of the lower dependency of the turbine on the gas mass-flow to produce power. As it's been commented, the use of methane (with higher heating value) provides higher temperatures and higher constant pressure heat capacity on the exhaust gas, reducing the mass-flow required for the turbine to produce the same power than in Diesel operation. For this reason, the reduction of charge air due to the delivered torque is less critical for the turbocharger, and more energy originally invested in producing charge air can be derived to the electricity production process.

After obtaining the generator power output data, the total efficiency of the power-plant can be confronted to the engine efficiency in both fuel modes. This information is included in the following tables:

Power rate	100	85	75	50	25	%
Blow-off efficiency	46,272	45,236	44,117	40,970	34,286	%
HyTC efficiency	48,333	47,710	47,492	44,631	38,967	%
Efficiency Δ	4,264	5,185	7,106	8,203	12,013	%

Table 63: Power-plant total efficiency. Natural gas blow-off vs. HyTC. Constant speed.

Power rate	100	85	75	50	25	%
Blow-off efficiency	46,272	45,236	44,663	43,489	41,243	%
HyTC efficiency	48,333	47,710	47,863	46,489	43,490	%
Efficiency Δ	4,264	5,185	6,686	6,453	5,167	%

Table 64: Power-plant total efficiency. Natural gas blow-off vs. HyTC. Variable speed.

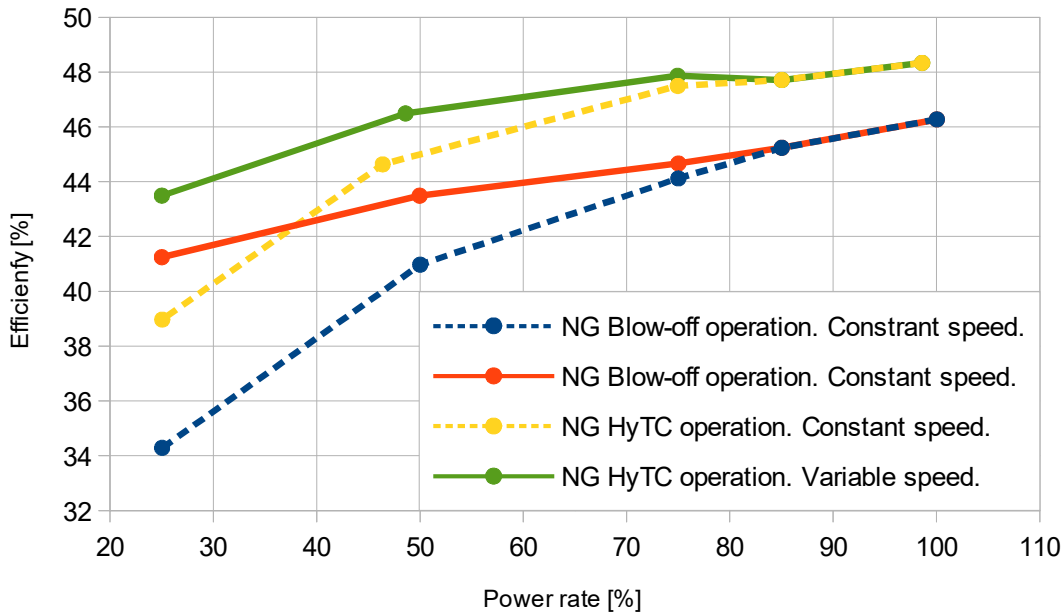

Figure 41: Power-plant total efficiency. Natural gas blow-off vs. HyTC.

The initial efficiency of the engine in NG is noticeably higher, and with it's larger electrical generator power output, there's a further enhancement of the efficiency in Natural Gas fuel mode.

8. EEDI Calculation

The EEDI is a CO_2 emission index referred to a given vessel, so in this chapter an already existing ship will be selected and its engine hypothetically replaced by the MAN V12 51/60 DF engine to observe the effects of the application of the HyTC on the value of this index. In the calculation for the Energy Efficiency Design Index (expression (12)), several systems are considered and counted in different blocks. Each block can behave as a direct CO_2 emissions source, or in the other hand as a way to avoid such emissions. These main blocks are detailed in the next paragraphs, and all the factors in the formulae are detailed in *Table 68.*:

- Main engine. In most of the vessels operating nowadays, it consists of an internal combustion engine or steam turbine, and it's the main source of propulsive power and the main producer of CO_2 emissions. Only the power delivered mechanically to the propulsion system is considered (electric propulsion contributes to other blocks) and it is represented by the following expression:

$$EEDI_{ME} = \frac{\left(\prod_{j=1}^{n} f_i\right) \cdot \left(\sum_{i=1}^{nME} P_{ME} \cdot C_{FME} \cdot SFC_{ME}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} \qquad (107)$$

- Auxiliary engine. The systems computed in this block are the ones that provide the power consumed in normal maximum sea load. The consumers can be both mechanical and electrical, but the source to obtain the power to operate them is always considered as a heat engine which will increase the CO_2 emissions. They are represented by the following expression:

$$EEDI_{AE} = \frac{\left(P_{AE} \cdot C_{FAE} \cdot SFC_{AE}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} \qquad (108)$$

- Electric power. This factor takes into account the emissions caused by propulsion electric motors, as the power required for this purpose is excluded from P_{ME} , and those systems which allow to save emissions by converting wasted power into electrical power. It's focused on the emissions caused by the use of electric shaft generators, and simultaneously on the implementation of Innovative

Energy Efficiency Technologies (IEET) for electrical power generation, which by means of energy recovery systems or alternative energies can reduce the necessity of obtaining power from the combustion of fuels. The power generated with those systems considered in this block replaces the power obtained from the auxiliary engines, hence the necessity to calculate it using the auxiliary engine's coefficients.

$$EEDI_{EP} = \frac{\left(\left(\prod_{j=1}^{n} f_j \cdot \sum_{i=1}^{nPTI} P_{PTI} - \sum_{i=1}^{neff} f_{eff} \cdot P_{AEeff}\right) \cdot C_{FAE} \cdot SFC_{AE}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} \quad (109)$$

- Innovative Energy Efficiency Technologies for propulsion. In the last block, those systems that provide mechanical power for ship propulsion without emitting CO_2 such as wind power (sails, kites, etc.) are taken into account. These systems are considered as a method to reduce the power provided by the main engine, so their contribution is computed with the use of the main engine coefficients.

$$EEDI_{IEET\ for\ propulsion} = \frac{-\left(\sum_{i=1}^{neff} f_{eff} \cdot P_{eff} \cdot C_{FME} \cdot SFC_{ME}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} \quad (110)$$

In the case an Energy Efficiency Technology is used, the deduction in CO_2 emissions is to be calculated following the methods described in the document “Guidance on treatment of innovative energy efficiency technologies for calculation and verification of the EEDI”. On this guidance, first a categorization of the technologies is done:

- Category A: Technologies that shift the power curve, which results in the change of combination of propulsive power P_p and V_{ref}. e.g: If V_{ref} is kept constant, P_p will be reduced and if P_p is kept constant, V_{ref} will increase.
- Category B: Technologies that reduce the propulsion power P_p at V_{ref}, but do not generate electricity. The saved power is counted as P_{eff}. If the technology can be used at any moment, then is considered category (B-1) and f_{eff}= 1,00. If the Technology can provide full output power only under limited conditions, is considered category (B-2) and its f_{eff} < 1,00.

- Category C: Technologies that generate electricity, where the saved power is counted as P_{AEeff}. If the technology can be used at any moment, is considered category (C-1) and f_{eff}= 1,00. If the technology can provide full output power only under limited conditions, is classified under category (C-2) and its f_{eff}< 1,00.

8.1. Attained EEDI calculation

For the study of the EEDI, it is necessary to make reference to a certain ship, and with this purpose, the JS INEOS INSIGHT has been used. This ship is a single-screw gas tanker originally equipped with two engines, providing 5850 kW each (11700 kW total propulsive power) configured for the transportation of ethane, LPG or LNG among others with the option of being powered by ethane, LNG and conventional diesel power. The original ship's data has been obtained from the ship's GAS FORM-C, and is shown in table *67*. All the values extracted from the simulations correspond to the Diesel HyTC mode numerical model, and more precisely to the results on constant speed.

For the calculations, first the non-dimensional coefficient C_F is needed. As it may be that the main engine and the auxiliary engine are using different fuels, the suffix *ME* and *AE* are used to distinguish both. The carbon factor is the main coefficient that allows to calculate the amount of CO_2 that will be released to the atmosphere. It's a ratio that relates the mass of CO_2 emitted to the mass of fuel consumed deduced from the carbon content in the fuel. This means that each fuel has a different factor and some of them are listed on *Table 65*.:

Type of fuel	Lower calorific value [kJ/ kg]	Carbon content	C_F [t-CO_2 / t-Fuel]
Heavy fuel oil	40200	0,8493	3,114
Diesel / Gas oil	42700	0,8744	3,206
LNG	48000	0,7500	2,750

Table 65: Properties for fuels.

As described in the regulations, the corresponding fuel used in the engine defines the C_{FME} coefficient which for diesel fuel is 3,206 [t-CO_2 / t-Fuel]. If the operation of the engine were considered as dual fuel (LNG

combined with Diesel), the C_F coefficient should be modified and the C_F for each fuel should apply, with an influence on the EEDI depending on the relative contribution of each fuel. However, as this study is only focused on the diesel operation, this correction is not considered.

The definition for *capacity* varies depending on the type of vessel, and for gas carriers the selected value should correspond to the deadweight, this is 20.917,9 tonnes.

The main engine power P_{ME} corresponds to the 75% of the MCR of the installed main propulsion power-plant.

$$EEDI_{ME} = \frac{\left(\prod_{j=1}^{n} f_i\right) \cdot \left(\sum_{i=1}^{nME} P_{ME} \cdot C_{FME} \cdot SFC_{ME}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} \qquad (111)$$

$$P_{ME} = Total\ propulsion\ power \cdot 75\ \% \qquad (112)$$

For the auxiliary engines power P_{AE}, it's not possible to take the rated power output of the generator, but is necessary to adjust it to another value depending on the vessel's total MCR power. The regulations establish two formulae to calculate this value depending on weather the vessel's total MCR power is above or below 10.000 kW. As in this analysis MCR_{ME} is higher than 10.000 kW, the procedure defines the following formula, where P_{PTI}, the power consumed by propulsion electric motors, is neglected as the ship is not equipped with these devices.

$$P_{AE} = \left[0{,}025 \cdot \left(\sum_{i=1}^{nME} MCR_{ME(i)} + \frac{\sum_{i=1}^{nPTI} P_{PTI(i)}}{0{,}75}\right)\right] + 250 \qquad if: \left(\sum MCR_{ME(i)} + \frac{\sum P_{PTI(i)}}{0{,}75}\right) \geqslant 10.000 \qquad (113)$$

For LNG carriers, a considerable amount of power is invested in the treatment of the boil-off gas (BOG) from the cargo tanks. In order to keep the pressure in the tanks below the limit, the boil-off gas can be reliquefied and redirected to the tanks or otherwise it can be used as fuel for DF engines if installed. As a diesel-only operation is being considered, only the reliquefaction and storage of the boil-off gas is

considered, so following the regulations, the power used for this purpose has to be added in the P_{AE} formula by means of the following estimation:

$$P_{AE}\ addition = Cargo\ tank\ capacity + BOR + COP_{reliquefy} + R_{reliquefy} \qquad (114)$$

BOR is the design rate of boil-off gas for the entire ship per day, which is usually specified in the vessel's data sheet, nonetheless this value is not provided by the available data, so reference values are taken. The values for this rate can oscillate from 0.10 to 0.15% for loaded condition voyages and from 0.06 to 0.10 % for ballast voyage (Głomski and Michalski, 2011 Error: No se encuentra la fuente de referencia). $COP_{Reliquefy}$ is the coefficient of design power performance for reliquefying boil-off gas per unit volume, obtained as follows.

$$COP_{reliquefy} = \frac{gas\ density\ at\ 110\ K \cdot heat\ of\ vaporization}{24\,[h] \cdot 3600\,[s] \cdot COP_{cooling}} = \frac{425\,[kg/m^3] \cdot 511\,[kJ/kg]}{24\,[h] \cdot 3600\,[s] \cdot COP_{cooling}} \qquad (115)$$

The coefficient of design performance of reliquefaction $COP_{Cooling}$ corresponds to 0.166. $R_{Reliquefy}$ is the ratio of boil-off gas to be reliquefied to entire BOG, calculated as shown in *(116)*. The vessel analysed is equipped with a VOC recovery system to recover the BOG from the cargo tanks, so all the boil-off gas generated is processed, so this factor is set as 1.

$$R_{Reliquefy} = \frac{BOG_{Reliquefy}}{BOG_{Total}} \qquad (116)$$

In this vessel, two shaft generators are installed, contemplated as P_{PTO} and computed using the sum of the 75% of the rated power of each shaft generator. The value of P_{PTO} is subtracted from the sum of P_{ME} in the power balance, however, the regulations define that in case the power of the shaft generator exceeds the power defined by P_{AE}, the value of this last one is the maximum that should be subtracted from P_{ME}.

$$P_{PTO} = Shaft\ generator\ electric\ power\ output \cdot 75\ \% \qquad (117)$$

$$\sum_{i=1}^{nME} P_{ME(i)} = 0{,}75 \cdot \left(\sum MCR_{ME(i)} - \sum P_{PTO(i)}\right) \qquad if: \quad 0{,}75 \cdot \sum P_{PTO} \leqslant P_{AE} \qquad (118)$$

The f_j factor is defined on the regulations as a correction of the power consumption depending on the type of vessel. For Liquefied gas carriers this coefficient is 1.

The f_w factor takes into account the decrease of speed due to sea conditions and weather. In no wind conditions and flat sea, 1 is taken as value for this coefficient.

The f_{eff} factor establishes the availability for the innovative energy efficiency technologies installed onboard, which is the ratio between the navigation time that this technologies can be operating and the total navigation time. The hybrid turbocharger generator does not depend on any external condition to operate, so during normal navigation this factor can be considered 1.

The f_i factor stands for a correction as a result of the technical/regulatory limitations regarding to the vessel's capacity. As there is no limitation derived from the capacity for the ship analysed in this study, a value of 1 is set .

The f_c factor is a cubic capacity correction that depends on the type of vessel. It's described on the regulations that for liquefied gas carriers it depends on *(119)*, where *R* is the ratio of the deadweight of the ship (tonnes) divided by the total cubic capacity of the cargo tanks of the ship (m^3):

$$f_C = R^{-0{,}56} \qquad (119)$$

The f_l factor considers the power addition from cargo handling systems like cranes, ramps or side-loaders, and it applies to the ship for which this study is carried on as it has one crane for manifold handling. The coefficient is estimated by the use of *(120)* provided in the regulations where *SWL* is the maximum save working load allowed to lift, and *Reach* is the distance [m] at which the load can be manipulated.

$$f_l = 1 + \frac{\sum_{n=1}^{n} \left(0{,}0519 \cdot SWL_n \cdot Reach_n + 31{,}11\right)}{Capacity} \tag{120}$$

Term	Value	Unit
Capacity	20.917,900	[Tonne]
C_{FAE}	3,206	[t-CO_2 / t-Fuel]
C_{FME}	3,206	[t-CO_2 / t-Fuel]
f_{eff}	1	[-]
f_i	1	[-]
f_c	1,167	[-]
f_j	1	[-]
f_w	1	[-]
f_l	1,002	[-]
n_{eff}	1	[-]
n_{ME}	1	[-]
n_{PTI}	0	[-]
P_{ME}	9000	[kW]
P_{AE}	1.176,115	[kW]
P_{AEeff}	326,577	[kW]
P_{eff}	0	[kW]
P_{PTO}	2.812,500	[kW]
SFC_{AE}	196	[g/kWh]
SFC_{ME}	194,750	[g/kWh]
V_{ref}	16,86	[knot]

Table 66: EEDI factors and coefficients

As the engine is being modified and the MARPOL regulations require to calculate the EEDI using a power equal to the 75% of the MCR, it's not possible to use the same speeds and powers provided in the vessel's documents. This makes necessary to know the ship's speed for the power provided by the new engine, but the power-speed line for this ship is not available. Hence it is necessary to use an estimation method to obtain the speed at the given power, so the Admiralty coefficient is used since the power variation is small. From the data provided in the FORM-C for the normal service speed and power (75% MCR) using the

original engine, the speed at the same power percentage for the new engine can be obtained. The admiralty coefficient is a number which remains constant for a given ship, and depends on the speed, displacement Δ and power. As the displacement does not change, if one operating point in the power-speed line is available it is possible to obtain the new speed as the power is varied.

$$Admiralty\ coefficient = \frac{\Delta^{2/3} \cdot v^3}{P} \qquad (121)$$

$$\frac{\Delta_0^{2/3} \cdot v_0^3}{P_0} = \frac{\Delta_1^{2/3} \cdot v_1^3}{P_1}; \qquad \Delta_0 = \Delta_1;$$

$$v_1 = \left(\frac{P_1 \cdot v_0^3}{P_0} \right)^{(1/3)} = V_{ref}$$

The influence of the IEET in the total index results in a subtraction proportional to the power produced by the system. In the case of a hybrid turbocharger, the regulations classify these systems as “waste heat recovery system for generation of electricity”, included in category (C-1) so the power reduction P_{AEeff} is obtained by the following procedure:

$$P_{AEeff} = P'_{AEeff} - P_{AEeff\ Loss} \qquad (122)$$

$$P'_{AEeff} = \frac{W_E}{\eta_G} \qquad (123)$$

The calculations for the IEET are based on the results obtained in the simulations in constant speed operation, so the generator power output corresponds to the value obtained in table *39 “Generator data. Diesel - HyTC. Constant speed”* for a power rate of 75% (as the P_{ME} corresponds to 75% of MCR). On *(122)* and *(123)* W_e is the electrical power generated as calculated in the model, η_G is the efficiency of the generator, and $P_{AEeff\ Loss}$ is the power consumed by all the mechanical or electrical devices required to drive the waste heat recovery system, like pumps, electrical rectifiers etc. The turbocharger generator is connected to a frequency converter to deliver the generated power at the correct voltage, current and frequency for the ship's consumers, which at the same time consumes a certain amount of power. Generally for frequency converters an efficiency of >93% is assumed.

Main ship's particulars	
Name of vessel	JS INEOS INSIGHT
LR/IMO number	9685425
Flag	Denmark
Type of vessel	Liquefied gas carrier
Registered owner	SNC Multigas
Commercial operator	EverGas Management A/S
Builder	Nantong Sino Pacific Offshore & Engineering
Classification Society	Bureau Veritas
Class Notation	BV I, +HULL, +MACH, Liquefied Gas Carrier, Type 2G - Dualfuel, Unrestricted Navigation, CPS (WBT), +VeriSTAR - HULL DFL 25 Years, +AUT-UMS, +SYS-NEQ, MON-SHAFT, CLEAN PASSPORT, GREENSHIP
Net registered tonnage	6866
Gross registered tonnage	22887
Hull particulars	
Length overall (LOA)	180,3 [m]
Length between perpendiculars (LBP)	170,8 [m]
Extreme breadth	26,6 [m]
Extreme depth	17,8 [m]
Summer draught	9,4 [m]
Light displacement	11170,0 [Tonnes]
Loaded displacement (with summer Deadweight)	32087,9 [Tonnes]
Corresponding summer deadweight	20917,9 [Tonnes]
Normal service speed	16,7 [knots]
Main engine particulars	
Make and type	Wärtsilä SL50DF tire-II
Number of units	2
Maximum continuous rating (MCR)	5850 [kW]
Auxiliary plants particulars	
Make and type	Wärtsilä SL20DF
Number of units	2
Maximum generator output per unit	1056 [kW]
Shaft generator (two units are installed)	2 x 1875 [kW]
Emergency generator	150 [kW]

Table 67: Reference vessel data (JS INEOS INSIGHT)

Term	Unit	Description
Capacity	[Tonne]	Ship capacity in deadweight or gross tonnage at summer load line draught (for container and passenger ships other values apply)
C_{FAE}	[t-CO_2 / t-Fuel]	Carbon factor for fuel for auxiliary engines
C_{FME}	[t-CO_2 / t-Fuel]	Carbon factor for fuel for main engines
f_{eff}	[-]	Correction factor for availability of innovative technologies
f_i	[-]	Correction factor for capacity of ships with technical/regulatory elements that influence ship capacity
f_c	[-]	Correction factor for capacity of ships with technical/regulatory elements that influence ship capacity
f_j	[-]	Correction factor for ship specific design features (e.g. Ice-class ship)
f_w	[-]	Correction factor for speed reduction due to representative sea conditions
n_{eff}	[-]	Number of innovative technologies
n_{ME}	[-]	Number of main engines
n_{PTI}	[-]	Number of power take-in technologies (e.g. shaft motors)
P_{ME}	[kW]	Ship propulsion power that is 75% of main engine Maximum Continuous Rating (MCR) or shaft motor (where applicable); also taking into account the shaft generator. This will be influenced by alternative propulsion configurations.
P_{AE}	[kW]	Ship auxiliary power requirements at normal sea going conditions.
P_{AEeff}	[kW]	Auxiliary power reduction due to use of innovative electric power generation technologies.
P_{eff}	[kW]	75% of installed power for each innovative technology that contributes to ship propulsion.
P_{PTI}	[kW]	75% of installed power for each power take-in system (e.g. propulsion shaft motors).
P_{PTO}	[kW]	75% of installed power for each power take-out system (e.g. shaft generators).
SFC_{AE}	[g/kWh]	Specific fuel consumption for auxiliary engines as per NO_x certification values.
SFC_{ME}	[g/kWh]	Specific fuel consumption for main engines as per NO_x certification values.
V_{ref}	[knot]	Reference ship speed attained at propulsion power equal to P_{ME} and under calm sea and deep water operation at summer load line draught.

Table 68: EEDI calculation factors.

8.2. Attained EEDI calculation detailed

Relevant data:

Vessel type: Liquefied gas carrier.

Deadweight: 20917,9 [Tonnes]

MCR_{ME}: 12000 [kW]

Number of main engines: 1

Main engine fuel: Diesel/Gas oil

Engine speed condition: constant 514 rpm.

Main engine specific fuel consumption at 75% MCR_{ME} with HyTC: 194,750 [g/kWh]

Main engine specific fuel consumption at 75% MCR_{ME} without HyTC: 194,729 [g/kWh]

MCR_{AE}: 1056 [kW]

Number of auxiliary engines: 2

Auxiliary engine fuel: Diesel-oil

Auxiliary engine specific fuel consumption at 75% MCR_{AE}: 196 [g/kWh]

Shaft generator power output: 1875 kW

Number of shaft generators: 2

Hybrid turbocharger output power at 75% MCR_{ME}: 307,136 [kW]

$$Capacity = DWT = 20.917{,}9\,[Tonnes]$$

$$P_{ME} = Total\ propulsion\ power \cdot 0{,}75 = 12000 \cdot 0{,}75 = 9.000 \quad kW$$

$$P_{PTO} = Shaft\ generator\ electric\ power\ output \cdot 75\ \% = 2 \cdot 1875 \cdot 0{,}75 = 2.812{,}5\ kW$$

The calculation for the reference ship speed is based on the Admiralty coefficient. From the vessel's FORM-C it is known that 8775 kW correspond to 16,7 knots (32,46 m/s):

$$v_1 = \left(\frac{P_1 \cdot v_0^3}{P_0}\right)^{(1/3)} = \left(\frac{9000 \cdot 16{,}7^3}{8775}\right)^{(1/3)} = V_{ref} = 16{,}842 \text{ knots}$$

Addition for liquefied gas carriers P_{AE} calculation:

$$COP_{reliquefy} = \frac{425\,[kg/m^3] \cdot 511\,[kJ/kg]}{24\,[h] \cdot 3600\,[s] \cdot COP_{cooling}} = \frac{425 \cdot 511}{24 \cdot 3600 \cdot 0{,}166} = 15{,}142$$

$$\text{Cargo tank capacity} \cdot BOR \cdot COP_{reliquefy} \cdot R_{reliquefy} = 27566{,}079 \cdot 0{,}0015 \cdot 15{,}142 \cdot 1 = 626{,}115 \text{ kW}$$

$$P_{AE} = \left[0{,}025 \cdot \left(\sum_{i=1}^{nME} MCR_{ME(i)} + \frac{\sum_{i=1}^{nPTI} P_{PTI(i)}}{0{,}75}\right)\right] + 250 + COP_{reliquefy} = \left[0{,}025 \cdot (12000 + 0)\right] + 250 + 626{,}155$$

$$P_{AE} = 1176{,}115 \text{ kW}$$

$$\sum_{i=1}^{nME} P_{ME(i)} = 0{,}75 \cdot \left(\sum MCR_{ME(i)} - \sum P_{PTO(i)}\right)$$

As $\Sigma P_{PTO} > \Sigma P_{AE}$, then:

$$\sum_{i=1}^{nME} P_{ME(i)} = 0{,}75 \cdot \left(\sum MCR_{ME(i)} - P_{AE(i)}\right) = 0{,}75 \cdot (12000 - 1176{,}115) = 8.117{,}914 \text{ kW}$$

$$P'_{AEeff} = \frac{W_E}{\eta_G} = \frac{307{,}126}{0{,}95} = 323{,}301 \text{ kW}$$

$$P_{AEeff} = P'_{AEeff} - P_{AEeff\ Loss} = 323{,}291 - 323{,}291 \cdot (1 - 0{,}93) = 300{,}670 \text{ kW}$$

$$f_C = R^{-0{,}56} = \left(\frac{DWT}{\text{Total tank volume}}\right)^{-0{,}56} = \left(\frac{20917{,}9}{27566{,}079}\right)^{-0{,}56} = 1{,}167$$

$$f_I = 1 + \frac{\sum_{n=1}^{n} \left(0{,}0519 \cdot SWL_n \cdot Reach_n + 31{,}11\right)}{Capacity} = 1 + \frac{(0{,}0519 \cdot 6 \cdot 11{,}7 + 31{,}11)}{20917{,}9} = 1{,}002$$

$$EEDI_{ME} = \frac{\left(\prod_{j=1}^{n} f_i\right) \cdot \left(\sum_{i=1}^{nME} P_{ME} \cdot C_{FME} \cdot SFC_{ME}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} = \frac{1 \cdot (8117{,}914 \cdot 3{,}206 \cdot 194{,}75)}{1 \cdot 1{,}167 \cdot 1 \cdot 20917{,}9 \cdot 1 \cdot 16{,}842} = 12{,}306$$

$$EEDI_{AE} = \frac{\left(P_{AE} \cdot C_{FAE} \cdot SFC_{AE}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}} = \frac{\left(1176{,}115 \cdot 3{,}206 \cdot 196\right)}{1 \cdot 1{,}167 \cdot 1 \cdot 20917{,}9 \cdot 1 \cdot 16{,}842} = 1{,}794$$

$$EEDI_{Secondary\ electric\ generation} = \frac{\left(\left(\prod_{j=1}^{n} f_j \cdot \sum_{i=1}^{nPTI} P_{PTI} - \sum_{i=1}^{neff} f_{eff} \cdot P_{AEeff}\right) \cdot C_{FAE} \cdot SFC_{AE}\right)}{f_i \cdot f_c \cdot f_j \cdot Capacity \cdot f_w \cdot V_{ref}}$$

$$EEDI_{Secondary\ electric\ generation} = \frac{\left(\left(1 \cdot 0 - 1 \cdot 300{,}660\right) \cdot 3{,}206 \cdot 196\right)}{1 \cdot 1{,}167 \cdot 1 \cdot 20917{,}9 \cdot 1 \cdot 16{,}842} = -0{,}459$$

If the HyTC is not operative, the EEDI is:

$$Ateined\ EEDI = 12{,}306 + 1{,}794 = 14{,}101 \left[\frac{grams\ CO_2}{Tonnes \cdot nm}\right]$$

In the other hand, if the HyTC is operative, the EEDI obtained is:

$$Attained\ EEDI_{HyTC} = 12{,}306 + 1{,}794 - 0{,}459 = 13{,}642 \left[\frac{grams\ CO_2}{Tonnes \cdot nm}\right]$$

8.3. Required EEDI calculation detailed

The required EEDI sets the maximum acceptable value for the attained EEDI of the given ship. This required index is obtained from a base line and then is modified during calendar scheduled phases using the corresponding reduction factor *X*.

$$Attained\ EEDI \leqslant Required\ EEDI = \left(\frac{100 - X}{100}\right) \cdot reference\ line\ value \qquad (124)$$

The base line is calculated from an exponential function which depends on the deadweight and two statistical factors, *a* and *c,* which are specific for each type of vessel . The expression that defines the line is shown together with some examples of the statistical factors:

Ship type	*a*	*c*
Bulk carrier	961,79	0,477
Gas carrier	1.120,00	0,456
LNG carrier	2.253,70	0,474

Table 69: Reference line factors.

$$reference\ line\ value = a \cdot DWT^{-c} \quad (125)$$

$$reference\ line\ value = 2.253{,}7 \cdot 20.917{,}9^{-0{,}474} = 20{,}182$$

The reduction factors are provided in a table where depending on the vessel type, vessel's deadweight and time period, it's possible to select the applicable factor for the given ship. For each type of vessel, the factors are presented as a range or as a single value, so in case that the factor embraces a range of values it will have to be interpolated by means of the corresponding deadweight. In the other hand, if the factor is provided as a single value, the factor obtention is direct. The time periods classified in phases reflect the tendency for the efficiency regulations to tighten during years to come. As it's shown in the table, there are 4 phases scheduled in the calendar that define the reduction factor that should be applied on each year. In table 70 some of the values for the reduction factor are detailed:

Ship type	Size	Phase 0	Phase 1	Phase 2	Phase 3
Bulk Carrier	20.000 DWT and above	0	10	20	30
	10.000 – 20.000 DWT	n/a	0 – 10	0 – 20	0 – 30
Gas Carrier	10.000 DWT and above	0	10	20	30
	20.00 – 10000 DWT	n/a	0 – 10	0 – 20	0 – 30
LNG carrier	10.000 DWT and above	0	10	20	30

Table 70: Reduction factors.

Phase 0: 1st January 2013 – 31st December 2014.

Phase 1: 1st January 2015 – 31st December 2019.

Phase 2: 1st January 2020 – 31st December 2024.

Phase 3: 1st January 2025 and onwards.

The obtained results for the required EEDI base line and following phases are:

	Base line	Phase 0	Phase 1	Phase 2	Phase 3
Reduction factor	N/A	0	10	20	30
Required EEDI	20,182	20,182	18,164	16,146	14,128

Table 71: Required EEDI phase progression.

Based on the mentioned calculations, the *Figure 42* tells the required and attained EEDI for the studied vessel, with and without the implementation of the IEET. On the diagram it can be seen that thanks to the use of the hybrid turbocharger, the ship is capable to satisfy the requirements regarding to the emissions of CO_2 for all the planed calendar, even for Phase 3. The critical condition is met when the Phase 3 applies and the required EEDI has a value of 14,128, which although it is less demanding than the attained EEDI obtained if the ship does not operate with the HyTC, it is considerably close for a projected ship estimation, and by using such IEET, the attained index can be reduced from 14,101 to 13,642, providing a satisfactory margin in case of for example, any modification in the *Capacity* happens during the operative live of the ship.

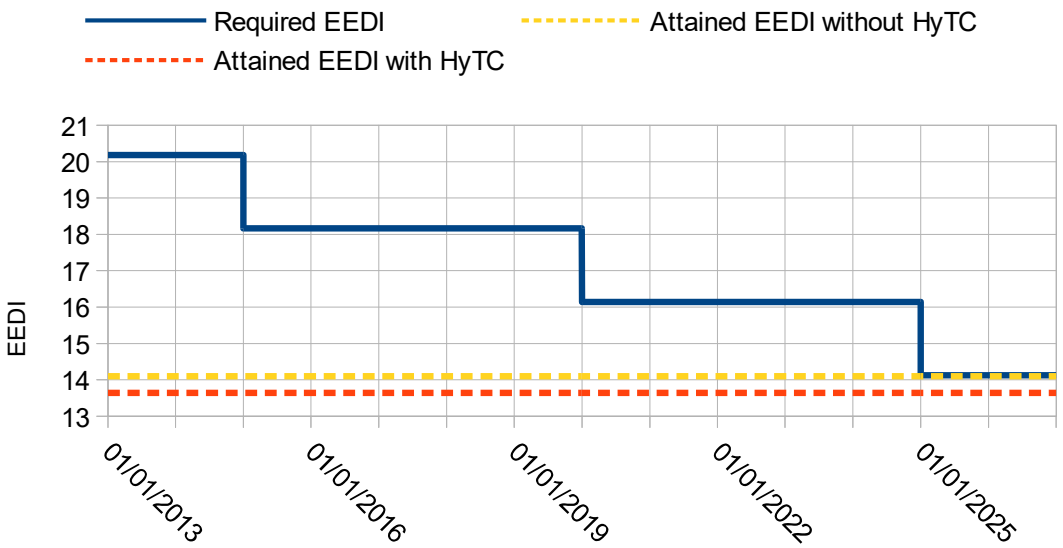

Figure 42: Influence of the HyTC on the vessel EEDI.

9. Conclusions

During the development of this study, the analysis of a hybrid turbocharger on a marine dual fuel engine focused on its operation on Diesel fuel has been carried out by using a numerical model. With the preliminary validation of the model, it's been possible to identify its accuracy and limitations, and discuss the causes for its discrepancy with the real engine on which it is based. After that, it has been possible to estimate the capabilities of the HyTC design as a method to enhance the total efficiency of a power-plant in a vessel and as a method to control the performance of the engine and the turbocompressor. At the same time, these performance characteristics have been contrasted to those offered by the same power-plant operating in natural gas. Finally, the benefits of using this system to help to cope with the international maritime pollution emission regulations have been also analysed.

The amount of energy that can be extracted from the engine exhaust in order to be converted to electric power, is found to be considerably high, achieving the levels of a medium power generator. If this power is used to supply specific loads on the ship, the requirements for power sources directly driven by the engine or by other auxiliary power sources can be reduced. By these means, the combustion of considerable amounts of fuel can be saved, leading to a reduction on the operational costs and pollution.

As a thermic machine, the turbocharger is optimized to operate under certain conditions in a limited range of pressures, temperatures, rotating speeds, and mass flows. The addition of a third device arranged to absorb the power of the turbine, like the electric generator analysed in this study, will inherently modify the capability of the turbocompressor to operate under these conditions. However, by delivering a variable and controlled resistive torque to its shaft, the performance of the turbocharger can be precisely adjusted to the desired conditions, with the benefits of wasting no energy on that process.

By contrasting the performance of the system operating in Diesel fuel or natural gas, it's been possible to identify the differences between both conditions and the reasons for these variations. The most notable is the lower power production of the hybrid turbocharger on Diesel mode, caused by the difference in how the energy is delivered to the turbine. As on Diesel mode, the exhaust gas has higher mass-flow and lower temperature for the same delivered power compared to natural gas mode, the reduction on the intake air mass-flow (primary contributor to the formation of exhaust gas) derived from the implementation of an

additional resistance on the turbocharger, leads to an earlier decrease on the power generation capacities of the turbine. This not only prevents the HyTC to generate lower power but also compromises the compressor performance and the engine operation.

Finally, to equip a ship with the hybrid turbocharger since its initial design or to add it to an existing vessel in order to satisfy the requirements of Marpol Annex IV regulations, is proven to be satisfactory. The use of this heat recovery system allows to reduce the amount of carbon dioxide per distance travelled and tone transported, setting it under the specified levels required by this regulation nowadays and on the future years.

10. Future lines of study

As a result of the concepts analysed and developed during this study, several areas and topics have shown potential for further investigation. This is a list of some of these subjects that could lead to a more accurate resolution of the ideas attended in this thesis:

- Numerical model enhancement. Study of the missing geometries of the engine, improvement of the thermodynamic performance of the model by adding air and gas tables, and study of the correction in the filling coefficient λ_V by Helmholtz.
- Study of the hybrid turbocharger in positive torque operation, to assist the engine start and as a method to control the turbocharger performance.
- Study of the instantaneous response of the generator to improve the turbocharger performance in the fluctuating flow situation imposed by the intake air and exhaust gas circulations.
- Study of the hybrid turbocharger generator electric power distribution on the ship. Which systems to supply and how to adjust the current characteristics and deliver the power to supply these systems.

  Economic advantages and disadvantages of the HyTC system. Estimation of the acquisition and maintenance costs as well as fuel saving.

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ANNEX I: Simulations data recordings

Diesel – Standard. Constant speed

```
Power rate: 100%
Displayed simulation time =    80.0000 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    355.723 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.770
Compr. efficiency MAN   EtaC1 =      0.810
Shaft torque          Mc  =   1935.054 [N*m]
Charge air pressure     p2  =      4.298 [bar]
Charge air temp.   T2  =    486.084 [K]
Air flow   Vc =      7.616 [m3/s]
Reduced air flow   Vcrid =      6.200 [m3/s]
Shaft speed       ntg =  22061.924 [rev/min]
Reduced shaft speed       nCrid =  15543.336 [rev/min]
Beta compressor MAN           BetaC1 =      4.242
Beta compressor Brown Boveri  BetaC  =      2.917
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     23.769 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.214 [bar]
Exhaust temp.    T3 =    750.590 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.835
Alpha coeff.              alpha =      1.185 [N*m]
Gas mass flow   m3  =    24.4046 [kg/s]
Epsilon Brown Boveri  =      3.165 [N*m]
Epsilon MAN   =      3.165 [N*m]
Shaft torque          Mt  =   1934.900 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    593.494 [K]
Turbine shaft speed    ntg =  22061.924 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     22.151 [bar]
Exhaust temp.   T4c =    750.590 [K]
Exhaust pressure     p4c =      3.214 [bar]
Exhaust gas mass flow m4c =    24.4011 [kg/s]
Intake air mass flow  m1c =    23.7689 [kg/s]
Mixture coefficient alfa=    30.2491
Rho =     4.2107 [kg/m^3]
Mean effective pressure.         mep =     19.062 [bar]
Mechanical losses fmep=      3.089 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  12008.979 [kW]
Fuel consumption   BSFC =    189.525 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    223.108 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =    100.075  [%]
Total plant efficiency    =     44.485  [%]
Displayed simulation time =    79.9208 [s]
```

```
Power rate: 85%
Fuel flow percentual   Mfperc =     86.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    352.924 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.770
Compr. efficiency MAN   EtaC1 =      0.809
Shaft torque          Mc  =   1658.244 [N*m]
Charge air pressure     p2  =      3.778 [bar]
Charge air temp.   T2  =    466.047 [K]
Air flow   Vc =      7.357 [m3/s]
Reduced air flow   Vcrid =      5.490 [m3/s]
Shaft speed       ntg =  20367.902 [rev/min]
Reduced shaft speed       nCrid =  14349.843 [rev/min]
Beta compressor MAN           BetaC1 =      3.729
Beta compressor Brown Boveri  BetaC  =      2.564
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     21.047 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.835 [bar]
Exhaust temp.    T3 =    735.684 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.842
Alpha coeff.              alpha =      1.177 [N*m]
Gas mass flow   m3  =    21.6063 [kg/s]
Epsilon Brown Boveri  =      2.792 [N*m]
Epsilon MAN   =      2.792 [N*m]
Shaft torque          Mt  =   1661.905 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    594.970 [K]
Turbine shaft speed    ntg =  20367.902 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     19.063 [bar]
Exhaust temp.   T4c =    735.684 [K]
Exhaust pressure     p4c =      2.835 [bar]
Exhaust gas mass flow m4c =    21.5905 [kg/s]
Intake air mass flow  m1c =    21.0468 [kg/s]
Mixture coefficient alfa=    31.1452
Rho =     3.7285 [kg/m^3]
Mean effective pressure.         mep =     16.196 [bar]
Mechanical losses fmep=      2.867 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10203.628 [kW]
Fuel consumption   BSFC =    191.830 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.567 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.030  [%]
Total plant efficiency    =     43.950  [%]
Displayed simulation time =    80.0000 [s]
```

```
Power rate: 75%
Fuel flow percentual   Mfperc =     76.500 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.789 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.772
Compr. efficiency MAN   EtaC1 =      0.812
Shaft torque          Mc  =   1464.200 [N*m]
Charge air pressure     p2  =      3.424 [bar]
Charge air temp.   T2  =    450.788 [K]
Air flow   Vc =      7.159 [m3/s]
Reduced air flow   Vcrid =      5.006 [m3/s]
Shaft speed       ntg =  19125.151 [rev/min]
Reduced shaft speed       nCrid =  13474.285 [rev/min]
Beta compressor MAN           BetaC1 =      3.380
Beta compressor Brown Boveri  BetaC  =      2.324
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     19.193 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.579 [bar]
Exhaust temp.    T3 =    726.046 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.845
Alpha coeff.              alpha =      1.171 [N*m]
Gas mass flow   m3  =    19.6857 [kg/s]
Epsilon Brown Boveri  =      2.540 [N*m]
Epsilon MAN   =      2.540 [N*m]
Shaft torque          Mt  =   1466.291 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    597.988 [K]
Turbine shaft speed    ntg =  19125.151 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     16.965 [bar]
Exhaust temp.   T4c =    726.046 [K]
Exhaust pressure     p4c =      2.579 [bar]
Exhaust gas mass flow m4c =    19.6770 [kg/s]
Intake air mass flow  m1c =    19.1934 [kg/s]
Mixture coefficient alfa=    31.9296
Rho =     3.4002 [kg/m^3]
Mean effective pressure.         mep =     14.286 [bar]
Mechanical losses fmep=      2.680 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   9000.024 [kW]
Fuel consumption   BSFC =    193.460 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    167.206 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     75.000  [%]
Total plant efficiency    =     43.580  [%]
Displayed simulation time =    79.7069 [s]
```

```
Power rate: 50%
Fuel flow percentual   Mfperc =     53.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    342.507 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.752
Compr. efficiency MAN   EtaC1 =      0.817
Shaft torque          Mc  =    847.274 [N*m]
Charge air pressure     p2  =      2.254 [bar]
Charge air temp.   T2  =    391.618 [K]
Air flow   Vc =      6.371 [m3/s]
Reduced air flow   Vcrid =      3.377 [m3/s]
Shaft speed       ntg =  13658.454 [rev/min]
Reduced shaft speed       nCrid =   9622.821 [rev/min]
Beta compressor MAN           BetaC1 =      2.225
Beta compressor Brown Boveri  BetaC  =      1.530
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     12.945 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.818 [bar]
Exhaust temp.    T3 =    693.082 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.831
Alpha coeff.              alpha =      1.095 [N*m]
Gas mass flow   m3  =    13.2816 [kg/s]
Epsilon Brown Boveri  =      1.791 [N*m]
Epsilon MAN   =      1.791 [N*m]
Shaft torque          Mt  =    848.964 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    614.285 [K]
Turbine shaft speed    ntg =  13658.454 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     11.805 [bar]
Exhaust temp.   T4c =    693.082 [K]
Exhaust pressure     p4c =      1.818 [bar]
Exhaust gas mass flow m4c =    13.2799 [kg/s]
Intake air mass flow  m1c =    12.9448 [kg/s]
Mixture coefficient alfa=    33.9964
Rho =     2.2932 [kg/m^3]
Mean effective pressure.         mep =      9.533 [bar]
Mechanical losses fmep=      2.272 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   6006.078 [kW]
Fuel consumption   BSFC =    200.843 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    111.583 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     50.051  [%]
Total plant efficiency    =     41.978  [%]
Displayed simulation time =    79.9800 [s]
```

```
Power rate: 25%
Fuel flow percentual   Mfperc =     30.400 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    336.455 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.792
Compr. efficiency MAN   EtaC1 =      0.849
Shaft torque          Mc  =    537.613 [N*m]
Charge air pressure     p2  =      1.621 [bar]
Charge air temp.   T2  =    348.395 [K]
Air flow   Vc =      5.770 [m3/s]
Reduced air flow   Vcrid =      2.472 [m3/s]
Shaft speed       ntg =   8481.625 [rev/min]
Reduced shaft speed       nCrid =   5975.578 [rev/min]
Beta compressor MAN           BetaC1 =      1.600
Beta compressor Brown Boveri  BetaC  =      1.100
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =      9.475 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.435 [bar]
Exhaust temp.    T3 =    619.352 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.821
Alpha coeff.              alpha =      1.022 [N*m]
Gas mass flow   m3  =     9.6670 [kg/s]
Epsilon Brown Boveri  =      1.415 [N*m]
Epsilon MAN   =      1.415 [N*m]
Shaft torque          Mt  =    536.899 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    576.343 [K]
Turbine shaft speed    ntg =   8481.625 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =      6.717 [bar]
Exhaust temp.   T4c =    619.352 [K]
Exhaust pressure     p4c =      1.435 [bar]
Exhaust gas mass flow m4c =     9.6675 [kg/s]
Intake air mass flow  m1c =     9.4753 [kg/s]
Mixture coefficient alfa=    43.3843
Rho =     1.6786 [kg/m^3]
Mean effective pressure.         mep =      4.774 [bar]
Mechanical losses fmep=      1.944 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   3007.468 [kW]
Fuel consumption   BSFC =    230.061 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =     55.874 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     25.062  [%]
Total plant efficiency    =     36.646  [%]
```

Diesel – Standard. Variable speed

```
Power rate: 100%
Displayed simulation time =    80.0000 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    355.723 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.770
Compr. efficiency MAN   EtaC1 =      0.810
Shaft torque          Mc  =   1935.054 [N*m]
Charge air pressure     p2  =      4.298 [bar]
Charge air temp.   T2  =    486.084 [K]
Air flow   Vc =      7.616 [m**3/s]
Reduced air flow   Vcrid =      6.200 [m3/s]
Shaft speed       ntg =  22061.924 [rev/min]
Reduced shaft speed       nCrid =  15543.336 [rev/min]
Beta compressor MAN           BetaC1 =      4.242
Beta compressor Brown Boveri  BetaC  =      2.917
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     23.769 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.214 [bar]
Exhaust temp.    T3 =    750.590 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.835
Alpha coeff.              alpha =      1.185 [N*m]
Gas mass flow   m3  =    24.4046 [kg/s]
Epsilon Brown Boveri  =      3.165 [N*m]
Epsilon MAN   =      3.165 [N*m]
Shaft torque          Mt  =   1934.900 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    593.494 [K]
Turbine shaft speed    ntg =  22061.924 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     22.151 [bar]
Exhaust temp.   T4c =    750.590 [K]
Exhaust pressure     p4c =      3.214 [bar]
Exhaust gas mass flow m4c =    24.4011 [kg/s]
Intake air mass flow  m1c =    23.7689 [kg/s]
Mixture coefficient alfa=    30.2491
Rho =     4.2107 [kg/m^3]
Mean effective pressure.         mep =     19.062 [bar]
Mechanical losses fmep=      3.089 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  12008.979 [kW]
Fuel consumption   BSFC =    189.525 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    223.108 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =    100.075  [%]
Total plant efficiency    =     44.485  [%]
Displayed simulation time =    79.9208 [s]
```

```
Power rate: 85%
Fuel flow percentual   Mfperc =     86.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    352.924 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.770
Compr. efficiency MAN   EtaC1 =      0.809
Shaft torque          Mc  =   1658.244 [N*m]
Charge air pressure     p2  =      3.778 [bar]
Charge air temp.   T2  =    466.047 [K]
Air flow   Vc =      7.357 [m**3/s]
Reduced air flow   Vcrid =      5.490 [m3/s]
Shaft speed       ntg =  20367.902 [rev/min]
Reduced shaft speed       nCrid =  14349.843 [rev/min]
Beta compressor MAN           BetaC1 =      3.729
Beta compressor Brown Boveri  BetaC  =      2.564
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     21.047 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.835 [bar]
Exhaust temp.    T3 =    735.684 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.842
Alpha coeff.              alpha =      1.177 [N*m]
Gas mass flow   m3  =    21.6063 [kg/s]
Epsilon Brown Boveri  =      2.792 [N*m]
Epsilon MAN   =      2.792 [N*m]
Shaft torque          Mt  =   1661.905 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    594.970 [K]
Turbine shaft speed    ntg =  20367.902 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     19.063 [bar]
Exhaust temp.   T4c =    735.684 [K]
Exhaust pressure     p4c =      2.835 [bar]
Exhaust gas mass flow m4c =    21.5905 [kg/s]
Intake air mass flow  m1c =    21.0468 [kg/s]
Mixture coefficient alfa=    31.1452
Rho =     3.7285 [kg/m^3]
Mean effective pressure.         mep =     16.196 [bar]
Mechanical losses fmep=      2.867 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10203.628 [kW]
Fuel consumption   BSFC =    191.830 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.567 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.030  [%]
Total plant efficiency    =     43.950  [%]
Displayed simulation time =    79.7603 [s]
```

```
Power rate: 75%
Fuel flow percentual   Mfperc =     78.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.521 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.777
Compr. efficiency MAN   EtaC1 =      0.818
Shaft torque          Mc  =   1416.312 [N*m]
Charge air pressure     p2  =      3.406 [bar]
Charge air temp.   T2  =    448.854 [K]
Air flow   Vc =      6.957 [m**3/s]
Reduced air flow   Vcrid =      4.860 [m3/s]
Shaft speed       ntg =  18952.797 [rev/min]
Reduced shaft speed       nCrid =  13352.856 [rev/min]
Beta compressor MAN           BetaC1 =      3.362
Beta compressor Brown Boveri  BetaC  =      2.312
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     18.634 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.520 [bar]
Exhaust temp.    T3 =    732.522 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.845
Alpha coeff.              alpha =      1.169 [N*m]
Gas mass flow   m3  =    19.1109 [kg/s]
Epsilon Brown Boveri  =      2.482 [N*m]
Epsilon MAN   =      2.482 [N*m]
Shaft torque          Mt  =   1414.799 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    606.258 [K]
Turbine shaft speed    ntg =  18952.797 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     17.311 [bar]
Exhaust temp.   T4c =    732.522 [K]
Exhaust pressure     p4c =      2.520 [bar]
Exhaust gas mass flow m4c =    19.1145 [kg/s]
Intake air mass flow  m1c =    18.6338 [kg/s]
Mixture coefficient alfa=    31.1915
Rho =     3.3867 [kg/m^3]
Mean effective pressure.         mep =     14.655 [bar]
Mechanical losses fmep=      2.656 [bar]
Teta injection    TETAIN =    -19.747 [°]
Power              Pb  =   8999.332 [kW]
Fuel consumption   BSFC =    192.278 [g/kWh]
Shaft speed   =    501.000 [rpm]
Torque   Qm  =    171.532 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     74.994  [%]
Total plant efficiency    =     43.847  [%]
Displayed simulation time =    79.8372 [s]
```

```
Power rate: 50%
Fuel flow percentual   Mfperc =     57.900 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    343.103 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.793
Compr. efficiency MAN   EtaC1 =      0.861
Shaft torque          Mc  =    818.080 [N*m]
Charge air pressure     p2  =      2.424 [bar]
Charge air temp.   T2  =    395.891 [K]
Air flow   Vc =      5.779 [m**3/s]
Reduced air flow   Vcrid =      3.257 [m3/s]
Shaft speed       ntg =  14267.475 [rev/min]
Reduced shaft speed       nCrid =  10051.896 [rev/min]
Beta compressor MAN           BetaC1 =      2.392
Beta compressor Brown Boveri  BetaC  =      1.645
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     12.486 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.806 [bar]
Exhaust temp.    T3 =    731.245 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.831
Alpha coeff.              alpha =      1.093 [N*m]
Gas mass flow   m3  =    12.8146 [kg/s]
Epsilon Brown Boveri  =      1.780 [N*m]
Epsilon MAN   =      1.780 [N*m]
Shaft torque          Mt  =    817.621 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    649.346 [K]
Turbine shaft speed    ntg =  14267.475 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =     12.766 [bar]
Exhaust temp.   T4c =    731.245 [K]
Exhaust pressure     p4c =      1.806 [bar]
Exhaust gas mass flow m4c =    12.8152 [kg/s]
Intake air mass flow  m1c =    12.4862 [kg/s]
Mixture coefficient alfa=    30.5334
Rho =     2.4609 [kg/m^3]
Mean effective pressure.         mep =     10.592 [bar]
Mechanical losses fmep=      2.174 [bar]
Teta injection    TETAIN =    -19.000 [°]
Power              Pb  =   5997.907 [kW]
Fuel consumption   BSFC =    197.482 [g/kWh]
Shaft speed   =    462.000 [rpm]
Torque   Qm  =    123.974 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     49.983  [%]
Total plant efficiency    =     42.692  [%]
Displayed simulation time =    79.7350 [s]
```

```
Power rate: 25%
Fuel flow percentual   Mfperc =     35.600 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    334.481 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.832
Compr. efficiency MAN   EtaC1 =      0.893
Shaft torque          Mc  =    367.021 [N*m]
Charge air pressure     p2  =      1.454 [bar]
Charge air temp.   T2  =    334.293 [K]
Air flow   Vc =      4.356 [m**3/s]
Reduced air flow   Vcrid =      1.745 [m3/s]
Shaft speed       ntg =   6315.511 [rev/min]
Reduced shaft speed       nCrid =   4449.481 [rev/min]
Beta compressor MAN           BetaC1 =      1.435
Beta compressor Brown Boveri  BetaC  =      0.987
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =      6.688 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.052

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.266 [bar]
Exhaust temp.    T3 =    689.604 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.812
Alpha coeff.              alpha =      0.981 [N*m]
Gas mass flow   m3  =     6.8642 [kg/s]
Epsilon Brown Boveri  =      1.248 [N*m]
Epsilon MAN   =      1.248 [N*m]
Shaft torque          Mt  =    366.972 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    659.073 [K]
Turbine shaft speed    ntg =   6315.511 [rev/min]
kEtaT               =      1.000

ENGINE DATA:
Indicated mean pressure         imp =      7.756 [bar]
Exhaust temp.   T4c =    689.604 [K]
Exhaust pressure     p4c =      1.266 [bar]
Exhaust gas mass flow m4c =     6.8642 [kg/s]
Intake air mass flow  m1c =     6.6881 [kg/s]
Mixture coefficient alfa=    33.4354
Rho =     1.5149 [kg/m^3]
Mean effective pressure.         mep =      6.083 [bar]
Mechanical losses fmep=      1.673 [bar]
Teta injection    TETAIN =    -18.821 [°]
Power              Pb  =   2997.158 [kW]
Fuel consumption   BSFC =    211.434 [g/kWh]
Shaft speed   =    402.000 [rpm]
Torque   Qm  =     71.196 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     24.976  [%]
Total plant efficiency    =     39.875  [%]
```

Diesel – Blow-off. Constant speed

```
Power rate: 100%
Displayed simulation time =     9.9730 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    355.188 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.749
Compr. efficiency MAN   EtaC1 =      0.808
Shaft torque          Mc  =   1937.257 [N*m]
Charge air pressure     p2  =      4.184 [bar]
Charge air temp.   T2  =    482.216 [K]
Air flow   Vc =      7.930 [m3/s]
Reduced air flow   Vcrid =      6.332 [m3/s]
Shaft speed       ntg =  22045.178 [rev/min]
Reduced shaft speed       nCrid =  15531.538 [rev/min]
Beta compressor MAN           BetaC1 =      4.129
Beta compressor Brown Boveri  BetaC  =      2.839
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.025
Blow off mass flow  =      1.152 [kg/s]
Compressor mass flow  =     24.277 [kg/s]
Extracted air ratio  =      4.747 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.157 [bar]
Exhaust temp.    T3 =    761.794 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.858
Alpha coeff.              alpha =      1.184 [N*m]
Gas mass flow   m3  =    23.7689 [kg/s]
Epsilon Brown Boveri  =      3.108 [N*m]
Epsilon MAN   =      3.108 [N*m]
Shaft torque          Mt  =   1946.679 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    600.456 [K]
Turbine shaft speed    ntg =  22045.178 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     22.079 [bar]
Exhaust temp.   T4c =    761.794 [K]
Exhaust pressure     p4c =      3.157 [bar]
Exhaust gas mass flow m4c =    23.7564 [kg/s]
Intake air mass flow  m1c =    23.1242 [kg/s]
Mixture coefficient alfa=    29.4287
Rho =     4.0965 [kg/m^3]
Mean effective pressure.         mep =     19.016 [bar]
Mechanical losses fmep=      3.063 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  11980.498 [kW]
Fuel consumption   BSFC =    189.976 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    222.578 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     99.837  [%]
Total plant efficiency    =     44.379  [%]
```

```
Power rate: 85%
Displayed simulation time =     9.7545 [s]

Fuel flow percentual   Mfperc =     86.200 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    352.183 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.744
Compr. efficiency MAN   EtaC1 =      0.802
Shaft torque          Mc  =   1647.736 [N*m]
Charge air pressure     p2  =      3.610 [bar]
Charge air temp.   T2  =    460.549 [K]
Air flow   Vc =      7.754 [m3/s]
Reduced air flow   Vcrid =      5.596 [m3/s]
Shaft speed       ntg =  20210.152 [rev/min]
Reduced shaft speed       nCrid =  14238.703 [rev/min]
Beta compressor MAN           BetaC1 =      3.563
Beta compressor Brown Boveri  BetaC  =      2.450
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.028
Blow off mass flow  =      1.342 [kg/s]
Compressor mass flow  =     21.454 [kg/s]
Extracted air ratio  =      6.258 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.758 [bar]
Exhaust temp.    T3 =    757.396 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    20.6749 [kg/s]
Epsilon Brown Boveri  =      2.716 [N*m]
Epsilon MAN   =      2.716 [N*m]
Shaft torque          Mt  =   1662.516 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    612.650 [K]
Turbine shaft speed    ntg =  20210.152 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     18.996 [bar]
Exhaust temp.   T4c =    757.396 [K]
Exhaust pressure     p4c =      2.758 [bar]
Exhaust gas mass flow m4c =    20.6561 [kg/s]
Intake air mass flow  m1c =    20.1111 [kg/s]
Mixture coefficient alfa=    29.6914
Rho =     3.5627 [kg/m^3]
Mean effective pressure.         mep =     16.205 [bar]
Mechanical losses fmep=      2.791 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10209.127 [kW]
Fuel consumption   BSFC =    192.172 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.669 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.076  [%]
Total plant efficiency    =     43.872  [%]
```

```
Power rate: 75%
Displayed simulation time =     9.9148 [s]

Fuel flow percentual   Mfperc =     76.900 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    349.579 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.746
Compr. efficiency MAN   EtaC1 =      0.804
Shaft torque          Mc  =   1431.789 [N*m]
Charge air pressure     p2  =      3.202 [bar]
Charge air temp.   T2  =    442.235 [K]
Air flow   Vc =      7.646 [m3/s]
Reduced air flow   Vcrid =      5.096 [m3/s]
Shaft speed       ntg =  18794.741 [rev/min]
Reduced shaft speed       nCrid =  13241.501 [rev/min]
Beta compressor MAN           BetaC1 =      3.160
Beta compressor Brown Boveri  BetaC  =      2.173
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.031
Blow off mass flow  =      1.550 [kg/s]
Compressor mass flow  =     19.538 [kg/s]
Extracted air ratio  =      7.936 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.479 [bar]
Exhaust temp.    T3 =    753.669 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.865
Alpha coeff.              alpha =      1.166 [N*m]
Gas mass flow   m3  =    18.4779 [kg/s]
Epsilon Brown Boveri  =      2.442 [N*m]
Epsilon MAN   =      2.442 [N*m]
Shaft torque          Mt  =   1433.408 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    623.123 [K]
Turbine shaft speed    ntg =  18794.741 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     16.914 [bar]
Exhaust temp.   T4c =    753.669 [K]
Exhaust pressure     p4c =      2.479 [bar]
Exhaust gas mass flow m4c =    18.4736 [kg/s]
Intake air mass flow  m1c =    17.9874 [kg/s]
Mixture coefficient alfa=    29.7677
Rho =     3.1865 [kg/m^3]
Mean effective pressure.         mep =     14.279 [bar]
Mechanical losses fmep=      2.635 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   8996.056 [kW]
Fuel consumption   BSFC =    194.556 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    167.132 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     74.967  [%]
Total plant efficiency    =     43.334  [%]
```

```
Power rate: 50%
Displayed simulation time =     9.7612 [s]

Fuel flow percentual   Mfperc =     53.200 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    341.820 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.721
Compr. efficiency MAN   EtaC1 =      0.803
Shaft torque          Mc  =    841.071 [N*m]
Charge air pressure     p2  =      2.143 [bar]
Charge air temp.   T2  =    386.509 [K]
Air flow   Vc =      6.777 [m3/s]
Reduced air flow   Vcrid =      3.459 [m3/s]
Shaft speed       ntg =  13325.244 [rev/min]
Reduced shaft speed       nCrid =   9388.064 [rev/min]
Beta compressor MAN           BetaC1 =      2.115
Beta compressor Brown Boveri  BetaC  =      1.454
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.026
Blow off mass flow  =      0.894 [kg/s]
Compressor mass flow  =     13.260 [kg/s]
Extracted air ratio  =      6.744 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.775 [bar]
Exhaust temp.    T3 =    711.574 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.088 [N*m]
Gas mass flow   m3  =    12.6894 [kg/s]
Epsilon Brown Boveri  =      1.750 [N*m]
Epsilon MAN   =      1.750 [N*m]
Shaft torque          Mt  =    838.842 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    632.049 [K]
Turbine shaft speed    ntg =  13325.244 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     11.775 [bar]
Exhaust temp.   T4c =    711.574 [K]
Exhaust pressure     p4c =      1.775 [bar]
Exhaust gas mass flow m4c =    12.7023 [kg/s]
Intake air mass flow  m1c =    12.3660 [kg/s]
Mixture coefficient alfa=    32.3543
Rho =     2.1907 [kg/m^3]
Mean effective pressure.         mep =      9.525 [bar]
Mechanical losses fmep=      2.249 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   6000.923 [kW]
Fuel consumption   BSFC =    201.774 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    111.488 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     50.008  [%]
Total plant efficiency    =     41.784  [%]
```

```
Power rate: 25%
Displayed simulation time =     9.8380 [s]

Fuel flow percentual   Mfperc =     30.400 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    336.251 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.772
Compr. efficiency MAN   EtaC1 =      0.849
Shaft torque          Mc  =    529.666 [N*m]
Charge air pressure     p2  =      1.600 [bar]
Charge air temp.   T2  =    346.922 [K]
Air flow   Vc =      6.031 [m3/s]
Reduced air flow   Vcrid =      2.561 [m3/s]
Shaft speed       ntg =   8657.980 [rev/min]
Reduced shaft speed       nCrid =   6099.826 [rev/min]
Beta compressor MAN           BetaC1 =      1.579
Beta compressor Brown Boveri  BetaC  =      1.086
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.020
Blow off mass flow  =      0.457 [kg/s]
Compressor mass flow  =      9.816 [kg/s]
Extracted air ratio  =      4.651 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.428 [bar]
Exhaust temp.    T3 =    623.237 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.841
Alpha coeff.              alpha =      1.020 [N*m]
Gas mass flow   m3  =     9.5524 [kg/s]
Epsilon Brown Boveri  =      1.408 [N*m]
Epsilon MAN   =      1.408 [N*m]
Shaft torque          Mt  =    529.950 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    579.455 [K]
Turbine shaft speed    ntg =   8657.980 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =      6.704 [bar]
Exhaust temp.   T4c =    623.237 [K]
Exhaust pressure     p4c =      1.428 [bar]
Exhaust gas mass flow m4c =     9.5519 [kg/s]
Intake air mass flow  m1c =     9.3597 [kg/s]
Mixture coefficient alfa=    42.8552
Rho =     1.6581 [kg/m^3]
Mean effective pressure.         mep =      4.765 [bar]
Mechanical losses fmep=      1.939 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   3002.176 [kW]
Fuel consumption   BSFC =    230.467 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =     55.776 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     25.018  [%]
Total plant efficiency    =     36.582  [%]
```

Diesel – Blow-off. Variable speed

```
Power rate: 100%
Displayed simulation time =     9.9730 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    355.188 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.749
Compr. efficiency MAN   EtaC1 =      0.808
Shaft torque          Mc  =   1937.257 [N*m]
Charge air pressure     p2  =      4.184 [bar]
Charge air temp.   T2  =    482.216 [K]
Air flow   Vc =      7.930 [m3/s]
Reduced air flow   Vcrid =      6.332 [m3/s]
Shaft speed       ntg =  22045.178 [rev/min]
Reduced shaft speed       nCrid =  15531.538 [rev/min]
Beta compressor MAN           BetaC1 =      4.129
Beta compressor Brown Boveri  BetaC  =      2.839
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.025
Blow off mass flow  =      1.152 [kg/s]
Compressor mass flow  =     24.277 [kg/s]
Extracted air ratio  =      4.747 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.157 [bar]
Exhaust temp.    T3 =    761.794 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.858
Alpha coeff.              alpha =      1.184 [N*m]
Gas mass flow   m3  =    23.7689 [kg/s]
Epsilon Brown Boveri  =      3.108 [N*m]
Epsilon MAN   =      3.108 [N*m]
Shaft torque          Mt  =   1946.679 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    600.456 [K]
Turbine shaft speed    ntg =  22045.178 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     22.079 [bar]
Exhaust temp.   T4c =    761.794 [K]
Exhaust pressure     p4c =      3.157 [bar]
Exhaust gas mass flow m4c =    23.7564 [kg/s]
Intake air mass flow  m1c =    23.1242 [kg/s]
Mixture coefficient alfa=    29.4287
Rho =     4.0965 [kg/m^3]
Mean effective pressure.         mep =     19.016 [bar]
Mechanical losses fmep=      3.063 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  11980.498 [kW]
Fuel consumption   BSFC =    189.976 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    222.578 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     99.837  [%]
Total plant efficiency    =     44.379  [%]
```

```
Power rate: 85%
Displayed simulation time =     9.7545 [s]

Fuel flow percentual   Mfperc =     86.200 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    352.183 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.744
Compr. efficiency MAN   EtaC1 =      0.802
Shaft torque          Mc  =   1647.736 [N*m]
Charge air pressure     p2  =      3.610 [bar]
Charge air temp.   T2  =    460.549 [K]
Air flow   Vc =      7.754 [m3/s]
Reduced air flow   Vcrid =      5.596 [m3/s]
Shaft speed       ntg =  20210.152 [rev/min]
Reduced shaft speed       nCrid =  14238.703 [rev/min]
Beta compressor MAN           BetaC1 =      3.563
Beta compressor Brown Boveri  BetaC  =      2.450
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.028
Blow off mass flow  =      1.342 [kg/s]
Compressor mass flow  =     21.454 [kg/s]
Extracted air ratio  =      6.258 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.758 [bar]
Exhaust temp.    T3 =    757.396 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    20.6749 [kg/s]
Epsilon Brown Boveri  =      2.716 [N*m]
Epsilon MAN   =      2.716 [N*m]
Shaft torque          Mt  =   1662.516 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    612.650 [K]
Turbine shaft speed    ntg =  20210.152 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     18.996 [bar]
Exhaust temp.   T4c =    757.396 [K]
Exhaust pressure     p4c =      2.758 [bar]
Exhaust gas mass flow m4c =    20.6561 [kg/s]
Intake air mass flow  m1c =    20.1111 [kg/s]
Mixture coefficient alfa=    29.6914
Rho =     3.5627 [kg/m^3]
Mean effective pressure.         mep =     16.205 [bar]
Mechanical losses fmep=      2.791 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10209.127 [kW]
Fuel consumption   BSFC =    192.172 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.669 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.076  [%]
Total plant efficiency    =     43.872  [%]
```

Power rate: 75%

```
Displayed simulation time =     9.9256 [s]

Fuel flow percentual   Mfperc =     78.600 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    349.181 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.750
Compr. efficiency MAN   EtaC1 =      0.809
Shaft torque          Mc  =   1392.803 [N*m]
Charge air pressure     p2  =      3.156 [bar]
Charge air temp.   T2  =    439.328 [K]
Air flow   Vc =      7.502 [m3/s]
Reduced air flow   Vcrid =      4.960 [m3/s]
Shaft speed       ntg =  18425.648 [rev/min]
Reduced shaft speed       nCrid =  12981.463 [rev/min]
Beta compressor MAN           BetaC1 =      3.115
Beta compressor Brown Boveri  BetaC  =      2.142
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.033
Blow off mass flow  =      1.743 [kg/s]
Compressor mass flow  =     19.016 [kg/s]
Extracted air ratio  =      9.166 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.417 [bar]
Exhaust temp.    T3 =    766.657 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.161 [N*m]
Gas mass flow   m3  =    17.7897 [kg/s]
Epsilon Brown Boveri  =      2.381 [N*m]
Epsilon MAN   =      2.381 [N*m]
Shaft torque          Mt  =   1411.599 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    637.619 [K]
Turbine shaft speed    ntg =  18425.648 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     17.261 [bar]
Exhaust temp.   T4c =    766.657 [K]
Exhaust pressure     p4c =      2.417 [bar]
Exhaust gas mass flow m4c =    17.7573 [kg/s]
Intake air mass flow  m1c =    17.2729 [kg/s]
Mixture coefficient alfa=    28.6927
Rho =     3.1393 [kg/m^3]
Mean effective pressure.         mep =     14.655 [bar]
Mechanical losses fmep=      2.606 [bar]
Teta injection    TETAIN =    -19.747 [°]
Power              Pb  =   8999.394 [kW]
Fuel consumption   BSFC =    193.758 [g/kWh]
Shaft speed   =    501.000 [rpm]
Torque   Qm  =    171.533 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     74.995  [%]
Total plant efficiency    =     43.513  [%]
```

Power rate: 50%

```
Displayed simulation time =     9.7412 [s]

Fuel flow percentual   Mfperc =     58.400 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    342.120 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.757
Compr. efficiency MAN   EtaC1 =      0.843
Shaft torque          Mc  =    810.606 [N*m]
Charge air pressure     p2  =      2.255 [bar]
Charge air temp.   T2  =    388.780 [K]
Air flow   Vc =      6.211 [m3/s]
Reduced air flow   Vcrid =      3.316 [m3/s]
Shaft speed       ntg =  13597.246 [rev/min]
Reduced shaft speed       nCrid =   9579.698 [rev/min]
Beta compressor MAN           BetaC1 =      2.225
Beta compressor Brown Boveri  BetaC  =      1.530
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.028
Blow off mass flow  =      1.062 [kg/s]
Compressor mass flow  =     12.715 [kg/s]
Extracted air ratio  =      8.355 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.749 [bar]
Exhaust temp.    T3 =    762.936 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.850
Alpha coeff.              alpha =      1.083 [N*m]
Gas mass flow   m3  =    11.9958 [kg/s]
Epsilon Brown Boveri  =      1.724 [N*m]
Epsilon MAN   =      1.724 [N*m]
Shaft torque          Mt  =    823.590 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    680.368 [K]
Turbine shaft speed    ntg =  13597.246 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     12.737 [bar]
Exhaust temp.   T4c =    762.936 [K]
Exhaust pressure     p4c =      1.749 [bar]
Exhaust gas mass flow m4c =    11.9841 [kg/s]
Intake air mass flow  m1c =    11.6522 [kg/s]
Mixture coefficient alfa=    28.2501
Rho =     2.2966 [kg/m^3]
Mean effective pressure.         mep =     10.595 [bar]
Mechanical losses fmep=      2.142 [bar]
Teta injection    TETAIN =    -19.000 [°]
Power              Pb  =   5999.346 [kW]
Fuel consumption   BSFC =    199.140 [g/kWh]
Shaft speed   =    462.000 [rpm]
Torque   Qm  =    124.003 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     49.995  [%]
Total plant efficiency    =     42.337  [%]
```

Power rate: 25%

```
Displayed simulation time =     9.5828 [s]

Fuel flow percentual   Mfperc =     35.600 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    334.565 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.831
Compr. efficiency MAN   EtaC1 =      0.914
Shaft torque          Mc  =    372.525 [N*m]
Charge air pressure     p2  =      1.474 [bar]
Charge air temp.   T2  =    334.900 [K]
Air flow   Vc =      4.433 [m3/s]
Reduced air flow   Vcrid =      1.797 [m3/s]
Shaft speed       ntg =   6515.584 [rev/min]
Reduced shaft speed       nCrid =   4590.439 [rev/min]
Beta compressor MAN           BetaC1 =      1.455
Beta compressor Brown Boveri  BetaC  =      1.001
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.010
Blow off mass flow  =      0.110 [kg/s]
Compressor mass flow  =      6.888 [kg/s]
Extracted air ratio  =      1.591 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.270 [bar]
Exhaust temp.    T3 =    684.541 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.833
Alpha coeff.              alpha =      0.982 [N*m]
Gas mass flow   m3  =     6.9543 [kg/s]
Epsilon Brown Boveri  =      1.253 [N*m]
Epsilon MAN   =      1.253 [N*m]
Shaft torque          Mt  =    371.365 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    652.985 [K]
Turbine shaft speed    ntg =   6515.584 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =      7.775 [bar]
Exhaust temp.   T4c =    684.541 [K]
Exhaust pressure     p4c =      1.270 [bar]
Exhaust gas mass flow m4c =     6.9548 [kg/s]
Intake air mass flow  m1c =     6.7787 [kg/s]
Mixture coefficient alfa=    33.8883
Rho =     1.5354 [kg/m^3]
Mean effective pressure.         mep =      6.097 [bar]
Mechanical losses fmep=      1.678 [bar]
Teta injection    TETAIN =    -18.821 [°]
Power              Pb  =   3004.294 [kW]
Fuel consumption   BSFC =    210.931 [g/kWh]
Shaft speed   =    402.000 [rpm]
Torque   Qm  =     71.365 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     25.036  [%]
Total plant efficiency    =     39.970  [%]
```

Diesel – HyTC. Constant speed

```
Power rate: 100%
Displayed simulation time =     9.8321 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    354.316 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.773
Compr. efficiency MAN   EtaC1 =      0.833
Shaft torque          Mc  =   1818.509 [N*m]
Charge air pressure     p2  =      4.173 [bar]
Charge air temp.   T2  =    476.231 [K]
Air flow   Vc =      7.483 [m3/s]
Reduced air flow   Vcrid =      6.036 [m3/s]
Shaft speed       ntg =  21657.084 [rev/min]
Reduced shaft speed       nCrid =  15258.113 [rev/min]
Beta compressor MAN           BetaC1 =      4.118
Beta compressor Brown Boveri  BetaC  =      2.832
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     23.140 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.156 [bar]
Exhaust temp.    T3 =    760.978 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.858
Alpha coeff.              alpha =      1.184 [N*m]
Gas mass flow   m3  =    23.7720 [kg/s]
Epsilon Brown Boveri  =      3.107 [N*m]
Epsilon MAN   =      3.107 [N*m]
Shaft torque          Mt  =   1968.110 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    599.838 [K]
Turbine shaft speed    ntg =  21657.084 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     22.083 [bar]
Exhaust temp.   T4c =    760.978 [K]
Exhaust pressure     p4c =      3.156 [bar]
Exhaust gas mass flow m4c =    23.7719 [kg/s]
Intake air mass flow  m1c =    23.1397 [kg/s]
Mixture coefficient alfa=    29.4483
Rho =     4.0993 [kg/m^3]
Mean effective pressure.         mep =     19.021 [bar]
Mechanical losses fmep=      3.062 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  11983.107 [kW]
Fuel consumption   BSFC =    189.934 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    222.627 [kN*m]
Electric generator power   EGP =    333.952  [kW]
EG percentual power % P_engine      =      2.786  [%]
EG torque  QHyTC =    155.000  [N*m]

Percentual power     =     99.859  [%]
Total plant efficiency    =     45.626  [%]
```

Power rate: 85%

```
Displayed simulation time =     9.9086 [s]

Fuel flow percentual   Mfperc =     86.100 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    351.159 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.774
Compr. efficiency MAN   EtaC1 =      0.835
Shaft torque          Mc  =   1505.482 [N*m]
Charge air pressure     p2  =      3.583 [bar]
Charge air temp.   T2  =    453.191 [K]
Air flow   Vc =      7.179 [m3/s]
Reduced air flow   Vcrid =      5.226 [m3/s]
Shaft speed       ntg =  19720.774 [rev/min]
Reduced shaft speed       nCrid =  13893.921 [rev/min]
Beta compressor MAN           BetaC1 =      3.537
Beta compressor Brown Boveri  BetaC  =      2.432
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     20.033 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.753 [bar]
Exhaust temp.    T3 =    758.193 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    20.6296 [kg/s]
Epsilon Brown Boveri  =      2.712 [N*m]
Epsilon MAN   =      2.712 [N*m]
Shaft torque          Mt  =   1718.179 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    613.503 [K]
Turbine shaft speed    ntg =  19720.774 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     18.963 [bar]
Exhaust temp.   T4c =    758.193 [K]
Exhaust pressure     p4c =      2.753 [bar]
Exhaust gas mass flow m4c =    20.5777 [kg/s]
Intake air mass flow  m1c =    20.0334 [kg/s]
Mixture coefficient alfa=    29.6111
Rho =     3.5490 [kg/m^3]
Mean effective pressure.         mep =     16.177 [bar]
Mechanical losses fmep=      2.786 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10191.579 [kW]
Fuel consumption   BSFC =    192.280 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.343 [kN*m]
Electric generator power   EGP =    335.484  [kW]
EG percentual power % P_engine      =      3.293  [%]
EG torque  QHyTC =    171.000  [N*m]

Percentual power     =     84.930  [%]
Total plant efficiency    =     45.290  [%]
```

Power rate: 75%

```
Displayed simulation time =     9.9290 [s]

Fuel flow percentual   Mfperc =     76.900 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    348.823 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.768
Compr. efficiency MAN   EtaC1 =      0.828
Shaft torque          Mc  =   1297.755 [N*m]
Charge air pressure     p2  =      3.148 [bar]
Charge air temp.   T2  =    435.743 [K]
Air flow   Vc =      7.026 [m3/s]
Reduced air flow   Vcrid =      4.674 [m3/s]
Shaft speed       ntg =  18160.548 [rev/min]
Reduced shaft speed       nCrid =  12794.691 [rev/min]
Beta compressor MAN           BetaC1 =      3.107
Beta compressor Brown Boveri  BetaC  =      2.137
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     17.918 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.470 [bar]
Exhaust temp.    T3 =    754.852 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.865
Alpha coeff.              alpha =      1.165 [N*m]
Gas mass flow   m3  =    18.3905 [kg/s]
Epsilon Brown Boveri  =      2.433 [N*m]
Epsilon MAN   =      2.433 [N*m]
Shaft torque          Mt  =   1494.785 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    624.590 [K]
Turbine shaft speed    ntg =  18160.548 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     16.908 [bar]
Exhaust temp.   T4c =    754.852 [K]
Exhaust pressure     p4c =      2.470 [bar]
Exhaust gas mass flow m4c =    18.4038 [kg/s]
Intake air mass flow  m1c =    17.9176 [kg/s]
Mixture coefficient alfa=    29.6521
Rho =     3.1741 [kg/m^3]
Mean effective pressure.         mep =     14.277 [bar]
Mechanical losses fmep=      2.632 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   8994.359 [kW]
Fuel consumption   BSFC =    194.593 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    167.101 [kN*m]
Electric generator power   EGP =    307.136  [kW]
EG percentual power % P_engine      =      3.409  [%]
EG torque  QHyTC =    170.000  [N*m]

Percentual power     =     74.953  [%]
Total plant efficiency    =     44.805  [%]
```

Power rate: 50%

```
Displayed simulation time =     9.7124 [s]

Fuel flow percentual   Mfperc =     53.300 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    341.245 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.742
Compr. efficiency MAN   EtaC1 =      0.826
Shaft torque          Mc  =    768.693 [N*m]
Charge air pressure     p2  =      2.109 [bar]
Charge air temp.   T2  =    382.053 [K]
Air flow   Vc =      6.271 [m3/s]
Reduced air flow   Vcrid =      3.187 [m3/s]
Shaft speed       ntg =  12757.801 [rev/min]
Reduced shaft speed       nCrid =   8988.282 [rev/min]
Beta compressor MAN           BetaC1 =      2.081
Beta compressor Brown Boveri  BetaC  =      1.431
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     12.218 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.766 [bar]
Exhaust temp.    T3 =    716.771 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.086 [N*m]
Gas mass flow   m3  =    12.5495 [kg/s]
Epsilon Brown Boveri  =      1.740 [N*m]
Epsilon MAN   =      1.740 [N*m]
Shaft torque          Mt  =    877.755 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    637.447 [K]
Turbine shaft speed    ntg =  12757.801 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     11.775 [bar]
Exhaust temp.   T4c =    716.771 [K]
Exhaust pressure     p4c =      1.766 [bar]
Exhaust gas mass flow m4c =    12.5551 [kg/s]
Intake air mass flow  m1c =    12.2182 [kg/s]
Mixture coefficient alfa=    31.9075
Rho =     2.1645 [kg/m^3]
Mean effective pressure.         mep =      9.531 [bar]
Mechanical losses fmep=      2.244 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   6004.521 [kW]
Fuel consumption   BSFC =    202.032 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    111.554 [kN*m]
Electric generator power   EGP =    120.573  [kW]
EG percentual power % P_engine      =      2.008  [%]
EG torque  QHyTC =     95.000  [N*m]

Percentual power     =     50.038  [%]
Total plant efficiency    =     42.569  [%]
```

Power rate: 25%

```
Displayed simulation time =     9.7561 [s]

Fuel flow percentual   Mfperc =     30.400 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    336.224 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.792
Compr. efficiency MAN   EtaC1 =      0.871
Shaft torque          Mc  =    521.674 [N*m]
Charge air pressure     p2  =      1.615 [bar]
Charge air temp.   T2  =    346.774 [K]
Air flow   Vc =      5.746 [m3/s]
Reduced air flow   Vcrid =      2.464 [m3/s]
Shaft speed       ntg =   8434.278 [rev/min]
Reduced shaft speed       nCrid =   5942.221 [rev/min]
Beta compressor MAN           BetaC1 =      1.594
Beta compressor Brown Boveri  BetaC  =      1.096
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =      9.447 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.433 [bar]
Exhaust temp.    T3 =    620.225 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.841
Alpha coeff.              alpha =      1.022 [N*m]
Gas mass flow   m3  =     9.6403 [kg/s]
Epsilon Brown Boveri  =      1.413 [N*m]
Epsilon MAN   =      1.413 [N*m]
Shaft torque          Mt  =    551.756 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    576.214 [K]
Turbine shaft speed    ntg =   8434.278 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =      6.714 [bar]
Exhaust temp.   T4c =    620.225 [K]
Exhaust pressure     p4c =      1.433 [bar]
Exhaust gas mass flow m4c =     9.6391 [kg/s]
Intake air mass flow  m1c =     9.4469 [kg/s]
Mixture coefficient alfa=    43.2543
Rho =     1.6735 [kg/m^3]
Mean effective pressure.         mep =      4.772 [bar]
Mechanical losses fmep=      1.942 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =   3006.097 [kW]
Fuel consumption   BSFC =    230.166 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =     55.848 [kN*m]
Electric generator power   EGP =     25.172  [kW]
EG percentual power % P_engine      =      0.838  [%]
EG torque  QHyTC =     30.000  [N*m]

Percentual power     =     25.051  [%]
Total plant efficiency    =     36.936  [%]
```

Diesel – HyTC. Variable speed

Power rate: 100%

```
Displayed simulation time =     9.8321 [s]

Fuel flow percentual   Mfperc =    100.000 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    354.316 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.773
Compr. efficiency MAN   EtaC1 =      0.833
Shaft torque          Mc  =   1818.509 [N*m]
Charge air pressure     p2  =      4.173 [bar]
Charge air temp.   T2  =    476.231 [K]
Air flow   Vc =      7.483 [m3/s]
Reduced air flow   Vcrid =      6.036 [m3/s]
Shaft speed       ntg =  21657.084 [rev/min]
Reduced shaft speed       nCrid =  15258.113 [rev/min]
Beta compressor MAN           BetaC1 =      4.118
Beta compressor Brown Boveri  BetaC  =      2.832
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     23.140 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      3.156 [bar]
Exhaust temp.    T3 =    760.978 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.858
Alpha coeff.              alpha =      1.184 [N*m]
Gas mass flow   m3  =    23.7720 [kg/s]
Epsilon Brown Boveri  =      3.107 [N*m]
Epsilon MAN   =      3.107 [N*m]
Shaft torque          Mt  =   1968.110 [N*m]
Turbine outflow pressure p4  =      1.016 [bar]
Turbine outflow temp.   T4  =    599.838 [K]
Turbine shaft speed    ntg =  21657.084 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     22.083 [bar]
Exhaust temp.   T4c =    760.978 [K]
Exhaust pressure     p4c =      3.156 [bar]
Exhaust gas mass flow m4c =    23.7719 [kg/s]
Intake air mass flow  m1c =    23.1397 [kg/s]
Mixture coefficient alfa=    29.4483
Rho =     4.0993 [kg/m^3]
Mean effective pressure.         mep =     19.021 [bar]
Mechanical losses fmep=      3.062 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  11983.107 [kW]
Fuel consumption   BSFC =    189.934 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    222.627 [kN*m]
Electric generator power   EGP =    333.952  [kW]
EG percentual power % P_engine      =      2.786  [%]
EG torque  QHyTC =    155.000  [N*m]

Percentual power     =     99.859  [%]
Total plant efficiency    =     45.626  [%]
```

Power rate: 85%

```
Displayed simulation time =     9.9086 [s]

Fuel flow percentual   Mfperc =     86.100 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    351.159 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.774
Compr. efficiency MAN   EtaC1 =      0.835
Shaft torque          Mc  =   1505.482 [N*m]
Charge air pressure     p2  =      3.583 [bar]
Charge air temp.   T2  =    453.191 [K]
Air flow   Vc =      7.179 [m3/s]
Reduced air flow   Vcrid =      5.226 [m3/s]
Shaft speed       ntg =  19720.774 [rev/min]
Reduced shaft speed       nCrid =  13893.921 [rev/min]
Beta compressor MAN           BetaC1 =      3.537
Beta compressor Brown Boveri  BetaC  =      2.432
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     20.033 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.753 [bar]
Exhaust temp.    T3 =    758.193 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    20.6296 [kg/s]
Epsilon Brown Boveri  =      2.712 [N*m]
Epsilon MAN   =      2.712 [N*m]
Shaft torque          Mt  =   1718.179 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    613.503 [K]
Turbine shaft speed    ntg =  19720.774 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     18.963 [bar]
Exhaust temp.   T4c =    758.193 [K]
Exhaust pressure     p4c =      2.753 [bar]
Exhaust gas mass flow m4c =    20.5777 [kg/s]
Intake air mass flow  m1c =    20.0334 [kg/s]
Mixture coefficient alfa=    29.6111
Rho =     3.5490 [kg/m^3]
Mean effective pressure.         mep =     16.177 [bar]
Mechanical losses fmep=      2.786 [bar]
Teta injection    TETAIN =    -20.000 [°]
Power              Pb  =  10191.579 [kW]
Fuel consumption   BSFC =    192.280 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.343 [kN*m]
Electric generator power   EGP =    335.484  [kW]
EG percentual power % P_engine      =      3.293  [%]
EG torque  QHyTC =    171.000  [N*m]

Percentual power     =     84.930  [%]
Total plant efficiency    =     45.290  [%]
```

Power rate: 75%

```
Displayed simulation time =     9.7792 [s]

Fuel flow percentual   Mfperc =     78.500 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    348.493 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.775
Compr. efficiency MAN   EtaC1 =      0.835
Shaft torque          Mc  =   1251.639 [N*m]
Charge air pressure     p2  =      3.127 [bar]
Charge air temp.   T2  =    433.511 [K]
Air flow   Vc =      6.803 [m3/s]
Reduced air flow   Vcrid =      4.517 [m3/s]
Shaft speed       ntg =  17904.976 [rev/min]
Reduced shaft speed       nCrid =  12614.632 [rev/min]
Beta compressor MAN           BetaC1 =      3.086
Beta compressor Brown Boveri  BetaC  =      2.122
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     17.318 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.416 [bar]
Exhaust temp.    T3 =    764.402 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.864
Alpha coeff.              alpha =      1.161 [N*m]
Gas mass flow   m3  =    17.8074 [kg/s]
Epsilon Brown Boveri  =      2.380 [N*m]
Epsilon MAN   =      2.380 [N*m]
Shaft torque          Mt  =   1460.858 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    635.766 [K]
Turbine shaft speed    ntg =  17904.976 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     17.251 [bar]
Exhaust temp.   T4c =    764.402 [K]
Exhaust pressure     p4c =      2.416 [bar]
Exhaust gas mass flow m4c =    17.8021 [kg/s]
Intake air mass flow  m1c =    17.3184 [kg/s]
Mixture coefficient alfa=    28.8049
Rho =     3.1476 [kg/m^3]
Mean effective pressure.         mep =     14.645 [bar]
Mechanical losses fmep=      2.606 [bar]
Teta injection    TETAIN =    -19.747 [°]
Power              Pb  =   8992.941 [kW]
Fuel consumption   BSFC =    193.648 [g/kWh]
Shaft speed   =    501.000 [rpm]
Torque   Qm  =    171.410 [kN*m]
Electric generator power   EGP =    302.813  [kW]
EG percentual power % P_engine      =      3.368  [%]
EG torque  QHyTC =    170.000  [N*m]

Percentual power     =     74.941  [%]
Total plant efficiency    =     45.003  [%]
```

Power rate: 50%

```
Displayed simulation time =     9.7345 [s]

Fuel flow percentual   Mfperc =     58.400 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    341.584 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.784
Compr. efficiency MAN   EtaC1 =      0.873
Shaft torque          Mc  =    736.303 [N*m]
Charge air pressure     p2  =      2.248 [bar]
Charge air temp.   T2  =    385.270 [K]
Air flow   Vc =      5.630 [m3/s]
Reduced air flow   Vcrid =      3.025 [m3/s]
Shaft speed       ntg =  13125.845 [rev/min]
Reduced shaft speed       nCrid =   9247.581 [rev/min]
Beta compressor MAN           BetaC1 =      2.219
Beta compressor Brown Boveri  BetaC  =      1.526
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =     11.597 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.745 [bar]
Exhaust temp.    T3 =    764.734 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.850
Alpha coeff.              alpha =      1.083 [N*m]
Gas mass flow   m3  =    11.9453 [kg/s]
Epsilon Brown Boveri  =      1.720 [N*m]
Epsilon MAN   =      1.720 [N*m]
Shaft torque          Mt  =    842.384 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    682.303 [K]
Turbine shaft speed    ntg =  13125.845 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =     12.727 [bar]
Exhaust temp.   T4c =    764.734 [K]
Exhaust pressure     p4c =      1.745 [bar]
Exhaust gas mass flow m4c =    11.9288 [kg/s]
Intake air mass flow  m1c =    11.5970 [kg/s]
Mixture coefficient alfa=    28.1162
Rho =     2.2857 [kg/m^3]
Mean effective pressure.         mep =     10.588 [bar]
Mechanical losses fmep=      2.139 [bar]
Teta injection    TETAIN =    -19.000 [°]
Power              Pb  =   5995.582 [kW]
Fuel consumption   BSFC =    199.265 [g/kWh]
Shaft speed   =    462.000 [rpm]
Torque   Qm  =    123.926 [kN*m]
Electric generator power   EGP =    130.581  [kW]
EG percentual power % P_engine      =      2.178  [%]
EG torque  QHyTC =    100.000  [N*m]

Percentual power     =     49.963  [%]
Total plant efficiency    =     43.232  [%]
```

Power rate: 25%

```
Displayed simulation time =     9.6502 [s]

Fuel flow percentual   Mfperc =     35.600 [perc]

INTERCOOLER DATA:
Outflow temp.   T1c =    334.495 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.832
Compr. efficiency MAN   EtaC1 =      0.915
Shaft torque          Mc  =    364.359 [N*m]
Charge air pressure     p2  =      1.468 [bar]
Charge air temp.   T2  =    334.399 [K]
Air flow   Vc =      4.357 [m3/s]
Reduced air flow   Vcrid =      1.761 [m3/s]
Shaft speed       ntg =   6439.456 [rev/min]
Reduced shaft speed       nCrid =   4536.805 [rev/min]
Beta compressor MAN           BetaC1 =      1.449
Beta compressor Brown Boveri  BetaC  =      0.996
kmassaBlowOff                 =      1.000
Blow off valve diameter       =      0.000
Blow off mass flow  =      0.000 [kg/s]
Compressor mass flow  =      6.750 [kg/s]
Extracted air ratio  =      0.000 [%]
kEtaC               =      1.078

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.269 [bar]
Exhaust temp.    T3 =    686.106 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.833
Alpha coeff.              alpha =      0.982 [N*m]
Gas mass flow   m3  =     6.9265 [kg/s]
Epsilon Brown Boveri  =      1.251 [N*m]
Epsilon MAN   =      1.251 [N*m]
Shaft torque          Mt  =    374.598 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    654.629 [K]
Turbine shaft speed    ntg =   6439.456 [rev/min]
kEtaT               =      1.025

ENGINE DATA:
Indicated mean pressure         imp =      7.769 [bar]
Exhaust temp.   T4c =    686.106 [K]
Exhaust pressure     p4c =      1.269 [bar]
Exhaust gas mass flow m4c =     6.9263 [kg/s]
Intake air mass flow  m1c =     6.7502 [kg/s]
Mixture coefficient alfa=    33.7458
Rho =     1.5290 [kg/m^3]
Mean effective pressure.         mep =      6.093 [bar]
Mechanical losses fmep=      1.676 [bar]
Teta injection    TETAIN =    -18.821 [°]
Power              Pb  =   3002.052 [kW]
Fuel consumption   BSFC =    211.089 [g/kWh]
Shaft speed   =    402.000 [rpm]
Torque   Qm  =     71.312 [kN*m]
Electric generator power   EGP =      6.406  [kW]
EG percentual power % P_engine      =      0.213  [%]
EG torque  QHyTC =     10.000  [N*m]

Percentual power     =     25.017  [%]
Total plant efficiency    =     40.025  [%]
```

Natural gas - Blow-off. Constant speed

```
Displayed simulation time =    99.4996 [s]

Blow-off percentual   =      8.220 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    353.192 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.757
Compr. efficiency MAN   EtaC1 =      0.797
Shaft torque          Mc  =   2074.195 [N*m]
Charge air pressure     p2  =      3.759 [bar]
Charge air temp.   T2  =    467.947 [K]
Air flow   Vc =      7.565 [m3/s]
Reduced air flow   Vcrid =      5.595 [m3/s]
Compressor mass flow  m1c =    21.4484 [kg/s]
Shaft speed       ntg =  16781.467 [rev/min]
Reduced shaft speed       nCrid =  14450.436 [rev/min]
Beta compressor MAN           BetaC1 =      3.710
Beta compressor Brown Boveri  BetaC  =      2.551

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.801 [bar]
Exhaust temp.    T3 =    816.570 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.842
Alpha coeff.              alpha =      1.176 [N*m]
Gas mass flow   m3  =    20.2198 [kg/s]
Epsilon Brown Boveri  =      2.758 [N*m]
Epsilon MAN   =      2.758 [N*m]
Shaft torque          Mt  =   2074.007 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    663.393 [K]
Turbine shaft speed    ntg =  16781.467 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     21.993 [bar]
Exhaust temp.   T4c =    816.570 [K]
Exhaust pressure     p4c =      2.801 [bar]
Exhaust gas mass flow m4c =    20.2207 [kg/s]
Intake air mass flow  m1c =    19.6854 [kg/s]
Mixture coefficient alfa=    33.5316
Rho =     0.5284 [kg/m^3]
Mean effective pressure.         mep =     19.054 [bar]
Mechanical losses fmep=      2.938 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  12004.260 [kW]
Fuel consumption   BSFC =    158.453 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    223.020 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =    100.035  [%]
Total plant efficiency    =     46.272  [%]
```

```
Displayed simulation time =    99.7678 [s]

Blow-off percentual   =     11.200 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.517 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.762
Compr. efficiency MAN   EtaC1 =      0.802
Shaft torque          Mc  =   1791.426 [N*m]
Charge air pressure     p2  =      3.334 [bar]
Charge air temp.   T2  =    448.667 [K]
Air flow   Vc =      7.352 [m3/s]
Reduced air flow   Vcrid =      5.029 [m3/s]
Compressor mass flow  m1c =    19.2796 [kg/s]
Shaft speed       ntg =  15484.125 [rev/min]
Reduced shaft speed       nCrid =  13333.301 [rev/min]
Beta compressor MAN           BetaC1 =      3.290
Beta compressor Brown Boveri  BetaC  =      2.262

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.531 [bar]
Exhaust temp.    T3 =    817.129 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.847
Alpha coeff.              alpha =      1.169 [N*m]
Gas mass flow   m3  =    17.4892 [kg/s]
Epsilon Brown Boveri  =      2.493 [N*m]
Epsilon MAN   =      2.493 [N*m]
Shaft torque          Mt  =   1776.176 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    676.775 [K]
Turbine shaft speed    ntg =  15484.125 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     19.021 [bar]
Exhaust temp.   T4c =    817.129 [K]
Exhaust pressure     p4c =      2.531 [bar]
Exhaust gas mass flow m4c =    17.5858 [kg/s]
Intake air mass flow  m1c =    17.1202 [kg/s]
Mixture coefficient alfa=    33.5431
Rho =     0.4594 [kg/m^3]
Mean effective pressure.         mep =     16.195 [bar]
Mechanical losses fmep=      2.827 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  10202.759 [kW]
Fuel consumption   BSFC =    162.082 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.551 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.023  [%]
Total plant efficiency    =     45.236  [%]
```

```
Displayed simulation time =    99.7867 [s]

Blow-off percentual   =     15.150 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    349.382 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.764
Compr. efficiency MAN   EtaC1 =      0.804
Shaft torque          Mc  =   1664.375 [N*m]
Charge air pressure     p2  =      3.167 [bar]
Charge air temp.   T2  =    440.732 [K]
Air flow   Vc =      7.202 [m3/s]
Reduced air flow   Vcrid =      4.764 [m3/s]
Compressor mass flow  m1c =    18.2651 [kg/s]
Shaft speed       ntg =  14957.718 [rev/min]
Reduced shaft speed       nCrid =  12880.015 [rev/min]
Beta compressor MAN           BetaC1 =      3.125
Beta compressor Brown Boveri  BetaC  =      2.149

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.478 [bar]
Exhaust temp.    T3 =    822.824 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.166 [N*m]
Gas mass flow   m3  =    15.9191 [kg/s]
Epsilon Brown Boveri  =      2.441 [N*m]
Epsilon MAN   =      2.441 [N*m]
Shaft torque          Mt  =   1664.208 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    683.778 [K]
Turbine shaft speed    ntg =  14957.718 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     17.044 [bar]
Exhaust temp.   T4c =    822.824 [K]
Exhaust pressure     p4c =      2.478 [bar]
Exhaust gas mass flow m4c =    15.9194 [kg/s]
Intake air mass flow  m1c =    15.4980 [kg/s]
Mixture coefficient alfa=    33.5669
Rho =     0.4155 [kg/m^3]
Mean effective pressure.         mep =     14.287 [bar]
Mechanical losses fmep=      2.757 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   9001.109 [kW]
Fuel consumption   BSFC =    166.193 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    167.226 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     75.009  [%]
Total plant efficiency    =     44.117  [%]
```

```
Displayed simulation time =    99.4995 [s]

Blow-off percentual   =     21.650 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    343.824 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.767
Compr. efficiency MAN   EtaC1 =      0.825
Shaft torque          Mc  =   1153.566 [N*m]
Charge air pressure     p2  =      2.437 [bar]
Charge air temp.   T2  =    400.994 [K]
Air flow   Vc =      6.661 [m3/s]
Reduced air flow   Vcrid =      3.727 [m3/s]
Compressor mass flow  m1c =    14.2893 [kg/s]
Shaft speed       ntg =  12182.931 [rev/min]
Reduced shaft speed       nCrid =  10490.660 [rev/min]
Beta compressor MAN           BetaC1 =      2.405
Beta compressor Brown Boveri  BetaC  =      1.654

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.000 [bar]
Exhaust temp.    T3 =    828.021 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.847
Alpha coeff.              alpha =      1.128 [N*m]
Gas mass flow   m3  =    11.4969 [kg/s]
Epsilon Brown Boveri  =      1.971 [N*m]
Epsilon MAN   =      1.971 [N*m]
Shaft torque          Mt  =   1152.987 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    719.420 [K]
Turbine shaft speed    ntg =  12182.931 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     12.098 [bar]
Exhaust temp.   T4c =    828.021 [K]
Exhaust pressure     p4c =      2.000 [bar]
Exhaust gas mass flow m4c =    11.5002 [kg/s]
Intake air mass flow  m1c =    11.1957 [kg/s]
Mixture coefficient alfa=    33.7692
Rho =     0.2984 [kg/m^3]
Mean effective pressure.         mep =      9.527 [bar]
Mechanical losses fmep=      2.571 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   6002.293 [kW]
Fuel consumption   BSFC =    178.961 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    111.513 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     50.019  [%]
Total plant efficiency    =     40.970  [%]
```

```
Displayed simulation time =    99.7900 [s]

Blow-off percentual   =     29.650 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    336.551 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.781
Compr. efficiency MAN   EtaC1 =      0.838
Shaft torque          Mc  =    667.647 [N*m]
Charge air pressure     p2  =      1.621 [bar]
Charge air temp.   T2  =    349.078 [K]
Air flow   Vc =      5.921 [m3/s]
Reduced air flow   Vcrid =      2.532 [m3/s]
Compressor mass flow  m1c =     9.7054 [kg/s]
Shaft speed       ntg =   7090.396 [rev/min]
Reduced shaft speed       nCrid =   6105.504 [rev/min]
Beta compressor MAN           BetaC1 =      1.600
Beta compressor Brown Boveri  BetaC  =      1.100

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.459 [bar]
Exhaust temp.    T3 =    841.154 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.827
Alpha coeff.              alpha =      1.028 [N*m]
Gas mass flow   m3  =     7.0139 [kg/s]
Epsilon Brown Boveri  =      1.439 [N*m]
Epsilon MAN   =      1.439 [N*m]
Shaft torque          Mt  =    668.224 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    781.252 [K]
Turbine shaft speed    ntg =   7090.396 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =      7.153 [bar]
Exhaust temp.   T4c =    841.154 [K]
Exhaust pressure     p4c =      1.459 [bar]
Exhaust gas mass flow m4c =     7.0134 [kg/s]
Intake air mass flow  m1c =     6.8277 [kg/s]
Mixture coefficient alfa=    34.4216
Rho =     0.1785 [kg/m^3]
Mean effective pressure.         mep =      4.770 [bar]
Mechanical losses fmep=      2.383 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   3005.267 [kW]
Fuel consumption   BSFC =    213.849 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =     55.833 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     25.044  [%]
Total plant efficiency    =     34.286  [%]
```

Natural gas - Blow-off. Variable speed

```
Displayed simulation time =    99.4996 [s]

Blow-off percentual   =      8.220 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    353.192 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.757
Compr. efficiency MAN   EtaC1 =      0.797
Shaft torque          Mc  =   2074.195 [N*m]
Charge air pressure     p2  =      3.759 [bar]
Charge air temp.   T2  =    467.947 [K]
Air flow   Vc =      7.565 [m3/s]
Reduced air flow   Vcrid =      5.595 [m3/s]
Compressor mass flow  m1c =    21.4484 [kg/s]
Shaft speed       ntg =  16781.467 [rev/min]
Reduced shaft speed       nCrid =  14450.436 [rev/min]
Beta compressor MAN           BetaC1 =      3.710
Beta compressor Brown Boveri  BetaC  =      2.551

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.801 [bar]
Exhaust temp.    T3 =    816.570 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.842
Alpha coeff.              alpha =      1.176 [N*m]
Gas mass flow   m3  =    20.2198 [kg/s]
Epsilon Brown Boveri  =      2.758 [N*m]
Epsilon MAN   =      2.758 [N*m]
Shaft torque          Mt  =   2074.007 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    663.393 [K]
Turbine shaft speed    ntg =  16781.467 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     21.993 [bar]
Exhaust temp.   T4c =    816.570 [K]
Exhaust pressure     p4c =      2.801 [bar]
Exhaust gas mass flow m4c =    20.2207 [kg/s]
Intake air mass flow  m1c =    19.6854 [kg/s]
Mixture coefficient alfa=    33.5316
Rho =     0.5284 [kg/m^3]
Mean effective pressure.         mep =     19.054 [bar]
Mechanical losses fmep=      2.938 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  12004.260 [kW]
Fuel consumption   BSFC =    158.453 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    223.020 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =    100.035  [%]
Total plant efficiency    =     46.272  [%]
```

```
Displayed simulation time =    99.7678 [s]

Blow-off percentual   =     11.200 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.517 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.762
Compr. efficiency MAN   EtaC1 =      0.802
Shaft torque          Mc  =   1791.426 [N*m]
Charge air pressure     p2  =      3.334 [bar]
Charge air temp.   T2  =    448.667 [K]
Air flow   Vc =      7.352 [m3/s]
Reduced air flow   Vcrid =      5.029 [m3/s]
Compressor mass flow  m1c =    19.2796 [kg/s]
Shaft speed       ntg =  15484.125 [rev/min]
Reduced shaft speed       nCrid =  13333.301 [rev/min]
Beta compressor MAN           BetaC1 =      3.290
Beta compressor Brown Boveri  BetaC  =      2.262

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.531 [bar]
Exhaust temp.    T3 =    817.129 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.847
Alpha coeff.              alpha =      1.169 [N*m]
Gas mass flow   m3  =    17.4892 [kg/s]
Epsilon Brown Boveri  =      2.493 [N*m]
Epsilon MAN   =      2.493 [N*m]
Shaft torque          Mt  =   1776.176 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    676.775 [K]
Turbine shaft speed    ntg =  15484.125 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     19.021 [bar]
Exhaust temp.   T4c =    817.129 [K]
Exhaust pressure     p4c =      2.531 [bar]
Exhaust gas mass flow m4c =    17.5858 [kg/s]
Intake air mass flow  m1c =    17.1202 [kg/s]
Mixture coefficient alfa=    33.5431
Rho =     0.4594 [kg/m^3]
Mean effective pressure.         mep =     16.195 [bar]
Mechanical losses fmep=      2.827 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  10202.759 [kW]
Fuel consumption   BSFC =    162.082 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.551 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     85.023  [%]
Total plant efficiency    =     45.236  [%]
```

```
Displayed simulation time =    99.7519 [s]

Blow-off percentual   =     14.350 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    349.258 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.772
Compr. efficiency MAN   EtaC1 =      0.812
Shaft torque          Mc  =   1624.462 [N*m]
Charge air pressure     p2  =      3.179 [bar]
Charge air temp.   T2  =    439.836 [K]
Air flow   Vc =      7.009 [m3/s]
Reduced air flow   Vcrid =      4.664 [m3/s]
Compressor mass flow  m1c =    17.8805 [kg/s]
Shaft speed       ntg =  14908.289 [rev/min]
Reduced shaft speed       nCrid =  12837.452 [rev/min]
Beta compressor MAN           BetaC1 =      3.138
Beta compressor Brown Boveri  BetaC  =      2.158

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.455 [bar]
Exhaust temp.    T3 =    817.354 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.164 [N*m]
Gas mass flow   m3  =    15.7315 [kg/s]
Epsilon Brown Boveri  =      2.419 [N*m]
Epsilon MAN   =      2.419 [N*m]
Shaft torque          Mt  =   1624.475 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    680.374 [K]
Turbine shaft speed    ntg =  14908.289 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     17.391 [bar]
Exhaust temp.   T4c =    817.354 [K]
Exhaust pressure     p4c =      2.455 [bar]
Exhaust gas mass flow m4c =    15.7311 [kg/s]
Intake air mass flow  m1c =    15.3146 [kg/s]
Mixture coefficient alfa=    33.5771
Rho =     0.4105 [kg/m^3]
Mean effective pressure.         mep =     14.659 [bar]
Mechanical losses fmep=      2.732 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   9001.845 [kW]
Fuel consumption   BSFC =    164.164 [g/kWh]
Shaft speed   =    501.000 [rpm]
Torque   Qm  =    171.579 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     75.015  [%]
Total plant efficiency    =     44.663  [%]
```

```
Displayed simulation time =    98.2341 [s]

Blow-off percentual   =     16.800 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    343.211 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.787
Compr. efficiency MAN   EtaC1 =      0.846
Shaft torque          Mc  =   1027.960 [N*m]
Charge air pressure     p2  =      2.405 [bar]
Charge air temp.   T2  =    396.648 [K]
Air flow   Vc =      5.931 [m3/s]
Reduced air flow   Vcrid =      3.311 [m3/s]
Compressor mass flow  m1c =    12.6947 [kg/s]
Shaft speed       ntg =  11633.364 [rev/min]
Reduced shaft speed       nCrid =  10017.430 [rev/min]
Beta compressor MAN           BetaC1 =      2.374
Beta compressor Brown Boveri  BetaC  =      1.632

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.901 [bar]
Exhaust temp.    T3 =    804.289 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.845
Alpha coeff.              alpha =      1.110 [N*m]
Gas mass flow   m3  =    10.8492 [kg/s]
Epsilon Brown Boveri  =      1.874 [N*m]
Epsilon MAN   =      1.874 [N*m]
Shaft torque          Mt  =   1027.969 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    706.055 [K]
Turbine shaft speed    ntg =  11633.364 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     13.034 [bar]
Exhaust temp.   T4c =    804.289 [K]
Exhaust pressure     p4c =      1.901 [bar]
Exhaust gas mass flow m4c =    10.8492 [kg/s]
Intake air mass flow  m1c =    10.5620 [kg/s]
Mixture coefficient alfa=    33.8496
Rho =     0.2808 [kg/m^3]
Mean effective pressure.         mep =     10.589 [bar]
Mechanical losses fmep=      2.445 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   5996.451 [kW]
Fuel consumption   BSFC =    168.595 [g/kWh]
Shaft speed   =    462.000 [rpm]
Torque   Qm  =    123.943 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     49.970  [%]
Total plant efficiency    =     43.489  [%]
```

```
Displayed simulation time =    99.1745 [s]

Blow-off percentual   =     18.900 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    335.157 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.829
Compr. efficiency MAN   EtaC1 =      0.889
Shaft torque          Mc  =    495.308 [N*m]
Charge air pressure     p2  =      1.519 [bar]
Charge air temp.   T2  =    339.117 [K]
Air flow   Vc =      4.518 [m3/s]
Reduced air flow   Vcrid =      1.863 [m3/s]
Compressor mass flow  m1c =     7.1437 [kg/s]
Shaft speed       ntg =   5662.944 [rev/min]
Reduced shaft speed       nCrid =   4876.332 [rev/min]
Beta compressor MAN           BetaC1 =      1.499
Beta compressor Brown Boveri  BetaC  =      1.031

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.329 [bar]
Exhaust temp.    T3 =    788.237 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.819
Alpha coeff.              alpha =      0.996 [N*m]
Gas mass flow   m3  =     5.9506 [kg/s]
Epsilon Brown Boveri  =      1.310 [N*m]
Epsilon MAN   =      1.310 [N*m]
Shaft torque          Mt  =    494.836 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    746.168 [K]
Turbine shaft speed    ntg =   5662.944 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =      8.149 [bar]
Exhaust temp.   T4c =    788.237 [K]
Exhaust pressure     p4c =      1.329 [bar]
Exhaust gas mass flow m4c =     5.9511 [kg/s]
Intake air mass flow  m1c =     5.7936 [kg/s]
Mixture coefficient alfa=    35.1742
Rho =     0.1482 [kg/m^3]
Mean effective pressure.         mep =      6.092 [bar]
Mechanical losses fmep=      2.057 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   3001.894 [kW]
Fuel consumption   BSFC =    177.776 [g/kWh]
Shaft speed   =    402.000 [rpm]
Torque   Qm  =     71.308 [kN*m]
Electric generator power   EGP =      0.000  [kW]
EG percentual power % P_engine      =      0.000  [%]
EG torque  QHyTC =      0.000  [N*m]

Percentual power     =     25.016  [%]
Total plant efficiency    =     41.243  [%]
```

Natural gas - HyTC. Constant speed

```
Displayed simulation time =    99.8687 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.622 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.759
Compr. efficiency MAN   EtaC1 =      0.798
Shaft torque          Mc  =   1803.737 [N*m]
Charge air pressure     p2  =      3.327 [bar]
Charge air temp.   T2  =    449.060 [K]
Air flow   Vc =      7.443 [m3/s]
Reduced air flow   Vcrid =      5.077 [m3/s]
Compressor mass flow  m1c =    19.4622 [kg/s]
Shaft speed       ntg =  15564.683 [rev/min]
Reduced shaft speed       nCrid =  13402.670 [rev/min]
Beta compressor MAN           BetaC1 =      3.283
Beta compressor Brown Boveri  BetaC  =      2.258

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.747 [bar]
Exhaust temp.    T3 =    813.660 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.843
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    19.7999 [kg/s]
Epsilon Brown Boveri  =      2.706 [N*m]
Epsilon MAN   =      2.706 [N*m]
Shaft torque          Mt  =   2130.383 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    663.444 [K]
Turbine shaft speed    ntg =  15564.683 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     21.709 [bar]
Exhaust temp.   T4c =    813.660 [K]
Exhaust pressure     p4c =      2.747 [bar]
Exhaust gas mass flow m4c =    19.9915 [kg/s]
Intake air mass flow  m1c =    19.4622 [kg/s]
Mixture coefficient alfa=    33.5442
Rho =     0.5222 [kg/m^3]
Mean effective pressure.         mep =     18.783 [bar]
Mechanical losses fmep=      2.926 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  11833.089 [kW]
Fuel consumption   BSFC =    158.863 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    219.840 [kN*m]
Electric generator power   EGP =    558.984  [kW]
EG percentual power % P_engine      =      4.724  [%]
EG torque  QHyTC =    361.000  [N*m]

Percentual power     =     98.609  [%]
Total plant efficiency    =     48.333  [%]
```

```
Displayed simulation time =   100.0000 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    348.201 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.758
Compr. efficiency MAN   EtaC1 =      0.797
Shaft torque          Mc  =   1545.933 [N*m]
Charge air pressure     p2  =      2.967 [bar]
Charge air temp.   T2  =    432.291 [K]
Air flow   Vc =      7.088 [m3/s]
Reduced air flow   Vcrid =      4.479 [m3/s]
Compressor mass flow  m1c =    17.1726 [kg/s]
Shaft speed       ntg =  14245.048 [rev/min]
Reduced shaft speed       nCrid =  12266.339 [rev/min]
Beta compressor MAN           BetaC1 =      2.929
Beta compressor Brown Boveri  BetaC  =      2.014

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.536 [bar]
Exhaust temp.    T3 =    814.786 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.846
Alpha coeff.              alpha =      1.170 [N*m]
Gas mass flow   m3  =    17.6406 [kg/s]
Epsilon Brown Boveri  =      2.498 [N*m]
Epsilon MAN   =      2.498 [N*m]
Shaft torque          Mt  =   1951.121 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    674.558 [K]
Turbine shaft speed    ntg =  14245.048 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     19.021 [bar]
Exhaust temp.   T4c =    814.786 [K]
Exhaust pressure     p4c =      2.536 [bar]
Exhaust gas mass flow m4c =    17.6396 [kg/s]
Intake air mass flow  m1c =    17.1726 [kg/s]
Mixture coefficient alfa=    33.5958
Rho =     0.4600 [kg/m^3]
Mean effective pressure.         mep =     16.195 [bar]
Mechanical losses fmep=      2.826 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  10202.843 [kW]
Fuel consumption   BSFC =    162.321 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.552 [kN*m]
Electric generator power   EGP =    573.946  [kW]
EG percentual power % P_engine      =      5.633  [%]
EG torque  QHyTC =    405.000  [N*m]

Percentual power     =     85.024  [%]
Total plant efficiency    =     47.710  [%]
```

```
Displayed simulation time =    99.7589 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    346.318 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.749
Compr. efficiency MAN   EtaC1 =      0.788
Shaft torque          Mc  =   1361.761 [N*m]
Charge air pressure     p2  =      2.675 [bar]
Charge air temp.   T2  =    418.839 [K]
Air flow   Vc =      6.903 [m3/s]
Reduced air flow   Vcrid =      4.059 [m3/s]
Compressor mass flow  m1c =    15.5619 [kg/s]
Shaft speed       ntg =  13186.793 [rev/min]
Reduced shaft speed       nCrid =  11355.080 [rev/min]
Beta compressor MAN           BetaC1 =      2.640
Beta compressor Brown Boveri  BetaC  =      1.815

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.483 [bar]
Exhaust temp.    T3 =    819.304 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.166 [N*m]
Gas mass flow   m3  =    15.9876 [kg/s]
Epsilon Brown Boveri  =      2.446 [N*m]
Epsilon MAN   =      2.446 [N*m]
Shaft torque          Mt  =   1892.354 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    680.482 [K]
Turbine shaft speed    ntg =  13186.793 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     17.033 [bar]
Exhaust temp.   T4c =    819.304 [K]
Exhaust pressure     p4c =      2.483 [bar]
Exhaust gas mass flow m4c =    15.9851 [kg/s]
Intake air mass flow  m1c =    15.5619 [kg/s]
Mixture coefficient alfa=    33.7046
Rho =     0.4155 [kg/m^3]
Mean effective pressure.         mep =     14.277 [bar]
Mechanical losses fmep=      2.757 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   8994.485 [kW]
Fuel consumption   BSFC =    166.319 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    167.103 [kN*m]
Electric generator power   EGP =    695.292  [kW]
EG percentual power % P_engine      =      7.725  [%]
EG torque  QHyTC =    530.000  [N*m]

Percentual power     =     74.954  [%]
Total plant efficiency    =     47.492  [%]
```

```
Displayed simulation time =    99.1063 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    338.653 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.745
Compr. efficiency MAN   EtaC1 =      0.800
Shaft torque          Mc  =    782.608 [N*m]
Charge air pressure     p2  =      1.795 [bar]
Charge air temp.   T2  =    364.111 [K]
Air flow   Vc =      6.137 [m3/s]
Reduced air flow   Vcrid =      2.785 [m3/s]
Compressor mass flow  m1c =    10.6762 [kg/s]
Shaft speed       ntg =   8612.261 [rev/min]
Reduced shaft speed       nCrid =   7415.974 [rev/min]
Beta compressor MAN           BetaC1 =      1.771
Beta compressor Brown Boveri  BetaC  =      1.218

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.942 [bar]
Exhaust temp.    T3 =    822.808 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.846
Alpha coeff.              alpha =      1.118 [N*m]
Gas mass flow   m3  =    10.9689 [kg/s]
Epsilon Brown Boveri  =      1.914 [N*m]
Epsilon MAN   =      1.914 [N*m]
Shaft torque          Mt  =   1483.497 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    719.268 [K]
Turbine shaft speed    ntg =   8612.261 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     11.381 [bar]
Exhaust temp.   T4c =    822.808 [K]
Exhaust pressure     p4c =      1.942 [bar]
Exhaust gas mass flow m4c =    10.9665 [kg/s]
Intake air mass flow  m1c =    10.6762 [kg/s]
Mixture coefficient alfa=    34.1446
Rho =     0.2814 [kg/m^3]
Mean effective pressure.         mep =      8.836 [bar]
Mechanical losses fmep=      2.545 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   5566.904 [kW]
Fuel consumption   BSFC =    181.980 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    103.424 [kN*m]
Electric generator power   EGP =    599.746  [kW]
EG percentual power % P_engine      =     10.767  [%]
EG torque  QHyTC =    700.000  [N*m]

Percentual power     =     46.391  [%]
Total plant efficiency    =     44.631  [%]
```

```
Displayed simulation time =    99.9971 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    331.063 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.837
Compr. efficiency MAN   EtaC1 =      0.898
Shaft torque          Mc  =    253.943 [N*m]
Charge air pressure     p2  =      1.146 [bar]
Charge air temp.   T2  =    309.888 [K]
Air flow   Vc =      5.343 [m3/s]
Reduced air flow   Vcrid =      1.819 [m3/s]
Compressor mass flow  m1c =     6.9751 [kg/s]
Shaft speed       ntg =   3118.165 [rev/min]
Reduced shaft speed       nCrid =   2685.036 [rev/min]
Beta compressor MAN           BetaC1 =      1.131
Beta compressor Brown Boveri  BetaC  =      0.778

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.469 [bar]
Exhaust temp.    T3 =    827.002 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.828
Alpha coeff.              alpha =      1.030 [N*m]
Gas mass flow   m3  =     7.1655 [kg/s]
Epsilon Brown Boveri  =      1.448 [N*m]
Epsilon MAN   =      1.448 [N*m]
Shaft torque          Mt  =   1553.325 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    766.924 [K]
Turbine shaft speed    ntg =   3118.165 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =      7.157 [bar]
Exhaust temp.   T4c =    827.002 [K]
Exhaust pressure     p4c =      1.469 [bar]
Exhaust gas mass flow m4c =     7.1648 [kg/s]
Intake air mass flow  m1c =     6.9751 [kg/s]
Mixture coefficient alfa=    35.2245
Rho =     0.1782 [kg/m^3]
Mean effective pressure.         mep =      4.772 [bar]
Mechanical losses fmep=      2.385 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   3006.548 [kW]
Fuel consumption   BSFC =    213.395 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =     55.857 [kN*m]
Electric generator power   EGP =    403.269  [kW]
EG percentual power % P_engine      =     13.430  [%]
EG torque  QHyTC =   1300.000  [N*m]

Percentual power     =     25.055  [%]
Total plant efficiency    =     38.967  [%]
```

Natural gas - HyTC. Variable speed

```
Displayed simulation time =    99.8687 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    350.622 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.759
Compr. efficiency MAN   EtaC1 =      0.798
Shaft torque          Mc  =   1803.737 [N*m]
Charge air pressure     p2  =      3.327 [bar]
Charge air temp.   T2  =    449.060 [K]
Air flow   Vc =      7.443 [m3/s]
Reduced air flow   Vcrid =      5.077 [m3/s]
Compressor mass flow  m1c =    19.4622 [kg/s]
Shaft speed       ntg =  15564.683 [rev/min]
Reduced shaft speed       nCrid =  13402.670 [rev/min]
Beta compressor MAN           BetaC1 =      3.283
Beta compressor Brown Boveri  BetaC  =      2.258

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.747 [bar]
Exhaust temp.    T3 =    813.660 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.843
Alpha coeff.              alpha =      1.175 [N*m]
Gas mass flow   m3  =    19.7999 [kg/s]
Epsilon Brown Boveri  =      2.706 [N*m]
Epsilon MAN   =      2.706 [N*m]
Shaft torque          Mt  =   2130.383 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    663.444 [K]
Turbine shaft speed    ntg =  15564.683 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     21.709 [bar]
Exhaust temp.   T4c =    813.660 [K]
Exhaust pressure     p4c =      2.747 [bar]
Exhaust gas mass flow m4c =    19.9915 [kg/s]
Intake air mass flow  m1c =    19.4622 [kg/s]
Mixture coefficient alfa=    33.5442
Rho =     0.5222 [kg/m^3]
Mean effective pressure.         mep =     18.783 [bar]
Mechanical losses fmep=      2.926 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  11833.089 [kW]
Fuel consumption   BSFC =    158.863 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    219.840 [kN*m]
Electric generator power   EGP =    558.984  [kW]
EG percentual power % P_engine      =      4.724  [%]
EG torque  QHyTC =    361.000  [N*m]

Percentual power     =     98.609  [%]
Total plant efficiency    =     48.333  [%]
```

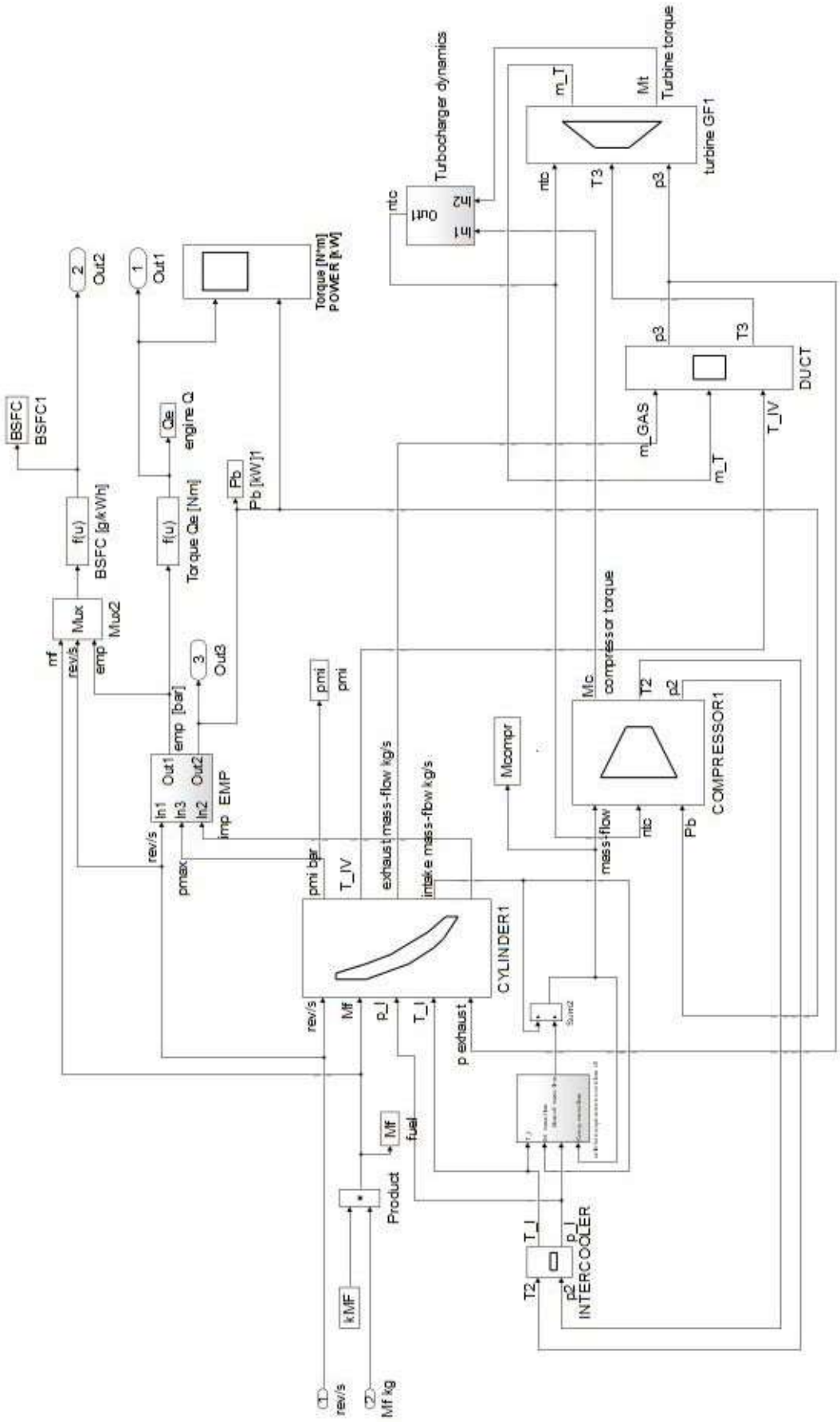
```
Displayed simulation time =   100.0000 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    348.201 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.758
Compr. efficiency MAN   EtaC1 =      0.797
Shaft torque          Mc  =   1545.933 [N*m]
Charge air pressure     p2  =      2.967 [bar]
Charge air temp.   T2  =    432.291 [K]
Air flow   Vc =      7.088 [m3/s]
Reduced air flow   Vcrid =      4.479 [m3/s]
Compressor mass flow  m1c =    17.1726 [kg/s]
Shaft speed       ntg =  14245.048 [rev/min]
Reduced shaft speed       nCrid =  12266.339 [rev/min]
Beta compressor MAN           BetaC1 =      2.929
Beta compressor Brown Boveri  BetaC  =      2.014

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.536 [bar]
Exhaust temp.    T3 =    814.786 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.846
Alpha coeff.              alpha =      1.170 [N*m]
Gas mass flow   m3  =    17.6406 [kg/s]
Epsilon Brown Boveri  =      2.498 [N*m]
Epsilon MAN   =      2.498 [N*m]
Shaft torque          Mt  =   1951.121 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    674.558 [K]
Turbine shaft speed    ntg =  14245.048 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     19.021 [bar]
Exhaust temp.   T4c =    814.786 [K]
Exhaust pressure     p4c =      2.536 [bar]
Exhaust gas mass flow m4c =    17.6396 [kg/s]
Intake air mass flow  m1c =    17.1726 [kg/s]
Mixture coefficient alfa=    33.5958
Rho =     0.4600 [kg/m^3]
Mean effective pressure.         mep =     16.195 [bar]
Mechanical losses fmep=      2.826 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =  10202.843 [kW]
Fuel consumption   BSFC =    162.321 [g/kWh]
Shaft speed   =    514.000 [rpm]
Torque   Qm  =    189.552 [kN*m]
Electric generator power   EGP =    573.946  [kW]
EG percentual power % P_engine      =      5.633  [%]
EG torque  QHyTC =    405.000  [N*m]

Percentual power     =     85.024  [%]
Total plant efficiency    =     47.710  [%]
```

```
Displayed simulation time =    99.6735 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    346.306 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.761
Compr. efficiency MAN   EtaC1 =      0.801
Shaft torque          Mc  =   1342.559 [N*m]
Charge air pressure     p2  =      2.710 [bar]
Charge air temp.   T2  =    418.756 [K]
Air flow   Vc =      6.729 [m3/s]
Reduced air flow   Vcrid =      4.009 [m3/s]
Compressor mass flow  m1c =    15.3705 [kg/s]
Shaft speed       ntg =  13201.796 [rev/min]
Reduced shaft speed       nCrid =  11367.999 [rev/min]
Beta compressor MAN           BetaC1 =      2.675
Beta compressor Brown Boveri  BetaC  =      1.839

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      2.460 [bar]
Exhaust temp.    T3 =    814.000 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.851
Alpha coeff.              alpha =      1.164 [N*m]
Gas mass flow   m3  =    15.7898 [kg/s]
Epsilon Brown Boveri  =      2.424 [N*m]
Epsilon MAN   =      2.424 [N*m]
Shaft torque          Mt  =   1837.832 [N*m]
Turbine outflow pressure p4  =      1.015 [bar]
Turbine outflow temp.   T4  =    677.227 [K]
Turbine shaft speed    ntg =  13201.796 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     17.376 [bar]
Exhaust temp.   T4c =    814.000 [K]
Exhaust pressure     p4c =      2.460 [bar]
Exhaust gas mass flow m4c =    15.7885 [kg/s]
Intake air mass flow  m1c =    15.3705 [kg/s]
Mixture coefficient alfa=    33.7144
Rho =     0.4103 [kg/m^3]
Mean effective pressure.         mep =     14.644 [bar]
Mechanical losses fmep=      2.732 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   8992.477 [kW]
Fuel consumption   BSFC =    164.263 [g/kWh]
Shaft speed   =    501.000 [rpm]
Torque   Qm  =    171.401 [kN*m]
Electric generator power   EGP =    650.115  [kW]
EG percentual power % P_engine      =      7.249  [%]
EG torque  QHyTC =    495.000  [N*m]

Percentual power     =     74.937  [%]
Total plant efficiency    =     47.863  [%]
```

```
Displayed simulation time =    99.6139 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    339.750 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.774
Compr. efficiency MAN   EtaC1 =      0.831
Shaft torque          Mc  =    797.885 [N*m]
Charge air pressure     p2  =      1.953 [bar]
Charge air temp.   T2  =    371.925 [K]
Air flow   Vc =      5.618 [m3/s]
Reduced air flow   Vcrid =      2.716 [m3/s]
Compressor mass flow  m1c =    10.4108 [kg/s]
Shaft speed       ntg =   9210.976 [rev/min]
Reduced shaft speed       nCrid =   7931.525 [rev/min]
Beta compressor MAN           BetaC1 =      1.927
Beta compressor Brown Boveri  BetaC  =      1.325

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.882 [bar]
Exhaust temp.    T3 =    799.511 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.845
Alpha coeff.              alpha =      1.107 [N*m]
Gas mass flow   m3  =    10.6925 [kg/s]
Epsilon Brown Boveri  =      1.855 [N*m]
Epsilon MAN   =      1.855 [N*m]
Shaft torque          Mt  =   1252.519 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    703.292 [K]
Turbine shaft speed    ntg =   9210.976 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =     12.734 [bar]
Exhaust temp.   T4c =    799.511 [K]
Exhaust pressure     p4c =      1.882 [bar]
Exhaust gas mass flow m4c =    10.6939 [kg/s]
Intake air mass flow  m1c =    10.4108 [kg/s]
Mixture coefficient alfa=    34.2209
Rho =     0.2738 [kg/m^3]
Mean effective pressure.         mep =     10.301 [bar]
Mechanical losses fmep=      2.434 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   5832.869 [kW]
Fuel consumption   BSFC =    168.987 [g/kWh]
Shaft speed   =    462.000 [rpm]
Torque   Qm  =    120.562 [kN*m]
Electric generator power   EGP =    416.936  [kW]
EG percentual power % P_engine      =      7.141  [%]
EG torque  QHyTC =    455.000  [N*m]

Percentual power     =     48.607  [%]
Total plant efficiency    =     46.489  [%]
```

```
Displayed simulation time =    99.7245 [s]

Blow-off percentual   =      0.000 [%]

INTERCOOLER DATA:
Outflow temp.   T1c =    332.120 [K]

COMPRESSOR DATA:
Compr. efficiency Brown Boveri   EtaC =      0.845
Compr. efficiency MAN   EtaC1 =      0.906
Shaft torque          Mc  =    310.323 [N*m]
Charge air pressure     p2  =      1.239 [bar]
Charge air temp.   T2  =    317.479 [K]
Air flow   Vc =      4.266 [m3/s]
Reduced air flow   Vcrid =      1.533 [m3/s]
Compressor mass flow  m1c =     5.8785 [kg/s]
Shaft speed       ntg =   3523.633 [rev/min]
Reduced shaft speed       nCrid =   3034.183 [rev/min]
Beta compressor MAN           BetaC1 =      1.223
Beta compressor Brown Boveri  BetaC  =      0.841

EXHAUST MANIFOLD DATA:
Exhaust pressure      p3 =      1.333 [bar]
Exhaust temp.    T3 =    778.882 [K]

TURBINE DATA:
Turbine efficiency          etaT =      0.820
Alpha coeff.              alpha =      0.997 [N*m]
Gas mass flow   m3  =     6.0363 [kg/s]
Epsilon Brown Boveri  =      1.315 [N*m]
Epsilon MAN   =      1.315 [N*m]
Shaft torque          Mt  =    807.420 [N*m]
Turbine outflow pressure p4  =      1.014 [bar]
Turbine outflow temp.   T4  =    736.728 [K]
Turbine shaft speed    ntg =   3523.633 [rev/min]

ENGINE DATA:
Indicated mean pressure         imp =      8.151 [bar]
Exhaust temp.   T4c =    778.882 [K]
Exhaust pressure     p4c =      1.333 [bar]
Exhaust gas mass flow m4c =     6.0384 [kg/s]
Intake air mass flow  m1c =     5.8785 [kg/s]
Mixture coefficient alfa=    35.5578
Rho =     0.1488 [kg/m^3]
Mean effective pressure.         mep =      6.092 [bar]
Mechanical losses fmep=      2.058 [bar]
Teta injection    TETAIN =     -6.000 [°]
Power              Pb  =   3001.941 [kW]
Fuel consumption   BSFC =    178.434 [g/kWh]
Shaft speed   =    402.000 [rpm]
Torque   Qm  =     71.310 [kN*m]
Electric generator power   EGP =    175.272  [kW]
EG percentual power % P_engine      =      5.839  [%]
EG torque  QHyTC =    500.000  [N*m]

Percentual power     =     25.016  [%]
Total plant efficiency    =     43.490  [%]
```

ANNEX II: Simulink® model and MatLab® code


Turbocharger dynamics
Turbine torque
turbine GF1
DUCT
COMPRESSOR1
CYLINDER1
INTERCOOLER
Torque [N*m]
POWER [kW]
BSFC
BSFC1
BSFC [g/kWh]
Torque Qe [Nm]
engine Q
Pb [kW]1
compressor torque
mass-flow
exhaust mass-flow kg/s
intake mass-flow kg/s
pmi bar
p exhaust
fuel
Product
kMF
Mux
Mux2
EMP
emp [bar]
Out1
Out2
Out3
rev/s
Mf kg

```
clear

clc

MANengineData;

%% FUEL
MC1per=53.2;                    % Fuel percentage (100% MCR)
tavMf=[0 MC1per; tts MC1per];   % Fuel vector
%% SPEED
G=100;                          % Insert speed desired percentage from MCR
N=514*G/100;                    % Revolutions per minute
revs=N/60;                      % Revolutions per second

%% COMPRESSOR CONSTANTS
maproj=23.7;                    % air mass to enter the cylinder
T3i=740;
p3i=3.57;

%TURBO CONSTANTS
T4i=596;
m3i=24.294;

% CONSTANTS FOR SPEED CALCULATION
ntc=22000/60;


%% SIMULATOR
EngineTest;                     % simulink file
```

```
% clear
% clc

%% VALUES FOR THE SIMULATION
ts= 0.03;                                % simulation step
tts = 10 ;                               % simulation time length (orig. 10 [s])
decim = 10;                              % decimals for displayed data periods
ndmax = 10000;                           % maximum displayed data number
xx = .85;                                % mean results exclusion

%% ENGINE
lambdaV=0.896;                           % cylinder filling coefficient. Check
excel file
T_I=273+43+40;                           % Intake manifold temp. in project
conditions [K]
p_I=4.29000;                             % Intake manifold pressure in project
conditions [Pa]
pa=1.01325;                              % Atmospheric pressure [Pa]
D = 0.51;                                % Cylinder bore [m]
C = 0.60;                                % Stroke [m]
R = C/2;                                 % Crank length [m]
L_r = R/0.3;                             % Connecting rod length [m]
Ncyl=12;                                 % Number of cylinders
Vcyl=pi*(D^2)/4*C;                       % Cylinder displacement
Vivc=0.9143;                             % Ratio for the intake valve closing
volume and displacement
CR=15;                                   % Compression ratio
Vc = Vcyl/(CR-1);                        % Combustion chamber volume [m^3]
etacomb=1;                               % Combustion efficiency
Ti= 404;                                 % Cylinder walls temp. [k]
Hi= 42700*10^3;                          % Diesel heating value [J/Kg]
Rgas = 287;                              % Air universal constant [J/kg K]
NC=52;                                   % Cetane number
etaV=0.88;                               % Air renewal efficiency
p3=p_I/1.2;                              % Exhaust pressure

%% ANGlES
tetaIL=30;                               % Injection length angle
IVC=-40;                                 % Intake valve closing advance from BDC
exhAdv=0;                                % Exhaust advance
tetaCL=50;                               % Combustion length angle

%% COMPRESSOR
nC=22000/60;                             % Compressor speed introduction value.
Ta=273+25;                               % Atmospheric temperature
Tref = 288;                              % Refference temp. [K]
pref = 1.01325;                          % Refference pressure [bar]
Ka = 1.4;                                % cp/cv air
cpa = 1000;                              % Air constant pressure heat capacity
[J/kg/K]
load etac.dat;                           % Load compressor efficiency table
kceta = 1;                               % Compressor efficiency correction load
betac.dat;                               % Beta table (compression ratio)
compressor
kmassa=1/(23.7/7.45);                    % Compressor mass-flow correction.
kmassaBlowOff= 1.0;                      % Blow-off valve mass-flow correction
(OK=1,066)
kEtaC = 0.81/0.77*1.025;                 % Compressor efficiency correction for
enhanced model. (OK=1.025)
```

```
kbetacor = 4.29/(2.95);                % Beta correction
knC= 15500/(nC*60)*15500/15238;        % Compressor speed correction.

% Reduced speed vector (per minute/1000):
ncr = [8 9 10 11 12 13 14 15 15.5 16 17 18];
ncr = ncr*1000/60;                     % Turbocompressor speed per second

% Reduced volume flow vector v1r [m^3/s]:
v1r = [1.5 2 2.5 3 3.5 4 4.5 5 5.5 6 6.2 6.5 7 7.5 8 8.5]';

% Correction for the compressor under 60% MCR
Power=        [25,  35.08, 46.1, 56.5, 67.5, 70, 80, 90, 100]*12000/100;   %
Input vector
deltaeta=[1.02, 1.02, 1.04, 1.02, 1, 1, 1, 1, 1,];      % Efficiency increase
Pbdelay=12000;

%% TURBINE CONSTANTS: (Brown Boveri)
J = 0.72/5;             % Moment of inertia [kg*m^2]
Kj = 1/(2*p_I*J);
kAt = 1;
Stgeo = .01527*kAt;    % Turbine geometrical section [m^2] % original value =
0.01527
nimprps = 258;         % Induced speed rps --> 15480 rpm
kEtaT=1.025;           % Efficiency correction (OK=1.025)
kepsilon=1;            % Expansion ratio correction (from A420 engine)
kgasmass=24.3/8.4;     % Gas mass-flow correction.
epst = [1 1.5 2 2.5 3 3.5 4]; % epsilon vector turbine table
alfat = [.92 1.043 1.133 1.17 1.182 1.19 1.195]; % vector alfat
etat=[0.8, 0.825, 0.835, 0.845, 0.84, 0.825, 0.815];

%% BLOW OFF VALVE:
Va = 0.3;                    % Intake manifold volume [m^3]
Cd = 0.7;                    % Discharge coefficient
df = 0.026;                  % Blow-off valve throat diameter [m]
omegaWG = p_I*df^2;          % Blow-off valve throat area

%% TC HYBRID
QHyTC = 0;                   % Electric generator torque [Nm]

%% INTERCOOLER CONSTANTS:
dTvalve = 40;                % Intake temperature increase MCI (original 40)
Kpint = 0.98;                % Loses  (200mm C.A.)
TiH2O = 273+15;              % Water temp. [K]
Vint = 0.1;                  % Intercooler volume [m^3]
Ra = 287;                    % Intercooler air constant [J/kg K]
E =0.86;                     % Intercooler efficiency (original 0.88)

%% EXHAUST MANIFOLD CONSTANTS:
Vcs = p_I*D^2/4*C*3*Ncyl;    % Exhaust manifold volume [m^3] (G. Ferrari pag. 191
iVcs = 1/Vcs;

%% GAS PROPERTIES CONSTANTS (Langen)
aa = 992.1;             % coeff. cp air (orig. = 992.1)(cpa = aa + ba T)
ba = 0.136;             % coeff. cp air (orig. = 0.136)(cpa = aa + ba T)
aap = 703.2;            % Coeff. cv air (orig. = 703.2) (cva = aap + ba T)
Ka = 1.4;               % cp/cv air

%% DIFFERENTIAL EQUATIONS INITIAL DATA
p2i=4.29;               % [bar] Intake manifold pressure
```

```
T2i=493;                % [K] Intercooler inlet temp.

%% FUEL
MFprojtot=0.0123*12;    % Fuel mass per cylinder
kMF=MFprojtot/100;


%% CAMPORA METHOD
fmeptab=[1.25, 1.934, 2.292, 2.507, 3.03];     % Mechanical loses table for max.
speed
MFtab=[25, 50, 75, 85, 100]*kMF;               % Fuel table

%% INJECTION ANGLE
tetaInj= [-13,-14,-15,-16,-18,-19,-19,-20];        % Injection start angle
depending on engine speed
speed=   [0.3,0.4,0.5,0.6,0.7,0.8,0.9,1]*514/60;   % Speed
```

ENGINE/Cylinder

```
function [pmi,T4c,p4c,gasflow,airMassFlow,alpha,ro,pmax,p, V] =
fcn(revs,MF,p_I,T_I,
D,C,CR,Hi,Ti,tetaIS,lambdaV,L_r,Ncyl,ps,IVC,Vr,etaV,NC,tetaIL,exhAdv,tetaCL)
%%
teta=[1:722];
p=[1:722];
x=[1:722];
V=[1:722];
mass=[1:722];
Tg=[1:722];
u=[1:722];
hi=[1:722];
dQrOverTeta=[1:722];
dVOverTeta=[1:722];
k=[1:722];
dpOverTeta=[1:722];
ptr=[1:722];
dQbOverTeta=[1:722];
dW=[1:722];
mres=1;
me=[1:722];
up1=[1:722];
uist=[1:722];
M=[1:722];
p0=[1:722];
T0g=[1:722];
dmeOverTeta=[1:722];
Tgext=[1:722];

%% OTHER COTANTS
N=revs;
p_I=p_I*100000;                        % [Pa]
ps=ps*100000;                          % [Pa]
tetaIS=round(tetaIS);
pa=101325;                             % Atmospheric pressure ISO condition [Pa]
Vcyl=pi*D^2/4*C;                       % Cylinder displacement [m^3]
R=C/2;                                 % Crank length  [m]
up = 2*N*C;                            % Piston mean velocity; [m/s]
Rgas = 287;                            % Air universal constant [J/kg K]
Vc = Vcyl/(CR-1);                      % Combustion chamber volume [m^3]
Hincomb= -2000000;                     % [J/Kg] Fuel injection enthalpy at 900 BAR
and 90°C

%% EXHAUST FLOW COEFFICIENT
d=D/3.5;                               % Intake port diameter [m]
dv=1.2*d;                              % Valve diameter
Nvalv=2;                               % Number of exhaust valves
G=[0.1   0.2  0.3];                    % Exhaust valve displacement/diameter ratio
vector pag. 229 INTERNAL COMBUSTION ENGINE FUNDAMENTALS
H=[0.676 0.73 0.573];                  % Exhaust flow coefficient vector
lvsudvExh=0.15;
lvExh=lvsudvExh*dv;                    % Exhaust valve displacement [m]
ArExh=pi*dv*lvExh;                     % Exhaust port opening area [m^2]
Cdsc=interp1(G,H,lvsudvExh);

%% INTAKE VALVE CLOSING ADVANCE ANGLE (MILLER CYCLE)
IVCr=(IVC+180)*pi/180;                  % Radiants conversion
```

```
xICV=C/2*(1+1/(R/L_r)-cos(IVCr)-(1/(R/L_r))*sqrt(1-(R/L_r)^2*(sin(IVCr))^2)); %
Piston position from TDC [m]
VACA=pi*D^2/4*xICV;                     % Volume calculation [m^3]

%% WIEBE COSTANTS
a = 6.9;                                % Wiebe law constant
mw = 1.2;                                % Wiebe law constant

%% IGNITION DELAY
T_II=T_I*CR^(1.35-1);
p_II=p_I/1000000*CR^1.35;
Z=(61.884/(NC+25)*(1200/T_II-0.582)+6.85/(10*p_II-12.4)^0.63);  %Hardenberg and
Hase
tetaID=round((0.36+0.22*up)*exp(Z));  % Ignition delay angle [°]

%% COMBUSTION ANGLES
tetaCS = tetaIS + tetaID;               % Combustion start angle [°]
tetaE = tetaCS + tetaCL ;               % Combustion end angle [°]

%% AIR AND EXHAUST GAS
ro = p_I/(Rgas*T_I);                    % Air density [kg/m^3]
mfProjCycle=MF/Ncyl;                    % Fuel mass per cylinder from
initialization file motoreprovaGO

maProjCycle= lambdaV*ro*Vcyl*etaV*Vr; % Air mas per cycle per cylinder [kg]
mExhGasCycle=maProjCycle+mfProjCycle; % Exhaust gas mass  [kg]
mfCycle=mfProjCycle;                    % Fuel mass [kg]


%% CYCLE START
tetaCS =(tetaCS+360)*pi/180;            % [rad]
tetaE =(tetaE+360)*pi/180;              % [rad]
ignAdv = tetaIS;                        % Ignition advance [°]


%Counter
Tstent=900;
kd=0;
errTs=100;
while errTs>0.1
    kd=kd+1;
    if kd>10
        break
    end
   Ts=Tstent;
for id=1:721

  % INTAKE
  if id < 181+IVC+1
    teta(1)=0;
    p(1)=p_I;
    mres=ps*Vc/(Rgas*Ts);               % Gas from previous cycle
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));     % Piston position [m]
    V(id) = pi*D^2/4*x(id)+Vc;          % Cylinder instant volume [m^3]
    mass(id)=(V(id)-Vc)/VACA*(lambdaV*ro*Vcyl*etaV)*Vr; % Mass entering cylinder
[kg]
    Tg(id)=p(id)*V(id)/((mres+mass(id))*Rgas);
    u(id)=6.18*up;                      % pag. 614 Ferrari
```

```
    hi(id)=3.26*D^(-0.2)*(p(id)/1000)^0.8*Tg(id)^(-0.53)*u(id)^0.8;
    dQrOverTeta(id) = ((pi*(D^2)/2)+(pi*D*(x(id)+C/(CR-1))))/
(2*pi*N)*hi(id)*(Tg(id)-Ti);   % Heat transferred through cylinder walls
    dVOverTeta(id)= Vcyl/2*(sin(teta(id))+(((R/L_r)*sin(2*teta(id)))/(2*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2)))); %dV/dteta
    k(id)=1.4-(Tg(id)-300)/12000;
    cp=1005;
    dpOverTeta(id)=-(k(id)-1)/V(id)*dQrOverTeta(id)-
k(id)/V(id)*p(id)*dVOverTeta(id)+(k(id)-
1)/V(id)*((lambdaV*ro*Vcyl*etaV)*Vr/VACA*dVOverTeta(id))*cp*T_I;
    p(id+1) = p(id)+dpOverTeta(id)*pi/180;
    teta(id+1)= teta(id)+pi/180;
  end

  % COMPRESSION
  if id >= 181+IVC+1 && id < 361+ignAdv
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));
    V(id) = pi*D^2/4*x(id)+ Vc;
    Tg(id)=p(id)*V(id)/((mres+maProjCycle)*Rgas);
    u(id)=2.28*up;
    hi(id)=3.26*D^(-0.2)*(p(id)/1000)^0.8*Tg(id)^(-0.53)*u(id)^0.8;
    dQrOverTeta(id) = ((pi*(D^2)/2)+(pi*D*(x(id)+C/(CR-1))))/
(2*pi*N)*hi(id)*(Tg(id)-Ti);
    dVOverTeta(id)= Vcyl/2*(sin(teta(id))+(((R/L_r)*sin(2*teta(id)))/(2*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2))));
    k(id)=1.4-(Tg(id)-300)/12000;     % Heat capacity ratio cp/cv
    dpOverTeta(id) = -dQrOverTeta(id)*(k(id)-1)/V(id)-
p(id)*dVOverTeta(id)*k(id)/V(id);
    p(id+1) = p(id)+dpOverTeta(id)*pi/180;
    teta(id+1)= teta(id)+pi/180;
  end

  % INJECTION (without combustion)
  if id >= 361+ignAdv && id <= 361+ignAdv+tetaID
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));
    V(id) = pi*D^2/4*x(id)+ Vc;
    mass(id)=(id-(361+ignAdv))/tetaIL*mfCycle;
    Tg(id)=p(id)*V(id)/((mres+maProjCycle+mass(id))*Rgas);
    u(id)=2.28*up;                    % pag. 614 Ferrari
    hi(id)=3.26*D^(-0.2)*(p(id)/1000)^0.8*Tg(id)^(-0.53)*u(id)^0.8;
    dQrOverTeta(id) = ((pi*(D^2)/2)+(pi*D*(x(id)+C/(CR-1))))/
(2*pi*N)*hi(id)*(Tg(id)-Ti);
    dVOverTeta(id)= Vcyl/2*(sin(teta(id))+(((R/L_r)*sin(2*teta(id)))/(2*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2))));
    k(id)=1.4-(Tg(id)-300)/12000;
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));
    cp=1005;
    dpOverTeta(id)=-(k(id)-1)/V(id)*dQrOverTeta(id)-
k(id)/V(id)*p(id)*dVOverTeta(id)+(k(id)-1)/V(id)*mfCycle/
(tetaIL*pi/180)*Hincomb;
    p(id+1) = p(id)+dpOverTeta(id)*pi/180;
    teta(id+1)= teta(id)+pi/180;
  end

  % COMBUSTION
  if id > 361+ignAdv+tetaID && id<541-exhAdv
```

```
    dQbOverTeta(id)= a/(tetaCL*pi/180)*(mw+1)*((teta(id)-tetaCS)/
(tetaCL*pi/180))^mw *exp(-a*((teta(id)-tetaCS)/
(tetaCL*pi/180))^(mw+1))*mfCycle*Hi;
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));
    V(id) = pi*D^2/4*x(id)+ Vc;
    mass(id)=(id-(361+ignAdv+tetaID))/tetaIL*mfCycle;

    if id <=361+ignAdv+tetaIL
       Tg(id)=p(id)*V(id)/((maProjCycle+mres+mass(361+ignAdv+tetaID)
+mass(id))*Rgas);
       k(id)=1.4-(Tg(id)-300)/12000; % Heat capacity ratio cp/cv

       % Woschini - Annand
       C2=0.7;                        % Empiric coefficient
       sigma0=56.7*10^-9;             % Stefan-Boltzmann constant (black body)
[W/(m^2 K^4)]
       ptr(id)=p(181+IVC)*(V(181+IVC)/V(id))^1.32; % giancarlo ferrari
       u(id)= 2.28*up+3.24*10^(-3)*V(id)*Tg(181+IVC)/
(p(181+IVC)*V(181+IVC))*(p(id)-ptr(id));
       hi(id)= 3.26*D^(-0.2)*(p(id)/1000)^0.8*Tg(id)^(-0.53)*u(id)^0.8;
       dQrOverTeta(id) = ((pi*(D^2)/2)+(pi*D*(x(id)+C/(CR-1))))/
(2*pi*N)*(hi(id)*(Tg(id)-Ti)+(C2*sigma0*(Tg(id)^4-Ti^4)));
       dVOverTeta(id)= Vcyl/2*(sin(teta(id))+(((R/L_r)*sin(2*teta(id)))/
(2*sqrt(1-(R/L_r)^2*(sin(teta(id)))^2))));
       dpOverTeta(id) =(dQbOverTeta(id)-dQrOverTeta(id)-k(id)/(k(id)-
1)*p(id)*dVOverTeta(id))*(k(id)-1)/V(id)+(k(id)-1)/V(id)*mfCycle/
(tetaIL*pi/180)*Hincomb;
       p(id+1) = p(id)+dpOverTeta(id)*pi/180;
       teta(id+1)= teta(id)+pi/180;
    else
       Tg(id)=p(id)*V(id)/((mExhGasCycle+mres)*Rgas);

       % Woschini - Annand
       C2=0.6;                        % Empiric coefficient
       sigma0=56.7*10^-9;             % Stefan-Boltzmann constant (black body)
[W/(m^2 K^4)]
       ptr(id)=p(181+IVC+1)*(V(181+IVC+1)/V(id))^1.32;
       u(id)=2.28*up+3.24*10^(-3)*V(id)*Tg(181+IVC+1)/
(p(181+IVC+1)*Vcyl)*(p(id)-ptr(id));
       hi(id)=3.26*D^(-0.2)*(p(id)/1000)^0.8*Tg(id)^(-0.53)*u(id)^0.8;
       dQrOverTeta(id) = ((pi*(D^2)/2)+(pi*D*(x(id)+C/(CR-1))))/
(2*pi*N)*(hi(id)*(Tg(id)-Ti)+(C2*sigma0*(Tg(id)^4-Ti^4)));
       dVOverTeta(id)= Vcyl/2*(sin(teta(id))+(((R/L_r)*sin(2*teta(id)))/
(2*sqrt(1-(R/L_r)^2*(sin(teta(id)))^2))));
       k(id)=1.4-(Tg(id)-300)/12000; % Heat capacity ratio cp/cv
       dpOverTeta(id) =(dQbOverTeta(id)-dQrOverTeta(id)-k(id)/(k(id)-
1)*p(id)*dVOverTeta(id))*(k(id)-1)/V(id);
       p(id+1) = p(id)+dpOverTeta(id)*pi/180;
       teta(id+1)= teta(id)+pi/180;
    end

  end

  % EXHAUST
  if id >=541-exhAdv
    x(id) = C/2*(1+1/(R/L_r)-cos(teta(id))-(1/(R/L_r))*sqrt(1-
(R/L_r)^2*(sin(teta(id)))^2));
    V(id) = pi*D^2/4*x(id)+ Vc;
```

```
    k(id)=1.4;
    p(id)=ps;
    Tg(id)=(Tg(541-exhAdv-1)*(1-((k(id)-1)/k(id))*(1-ps/p(541-exhAdv-1))))+25;
   %Tg(id)=Tg(541)*(ps/p(541))^((k(id)-1)/k(id));
    teta(id+1)= teta(id)+pi/180;
  end
end

errTs= abs((Ts-Tg(721))/Tg(721))*100;
Tstent=Tg(721);
end


%% GENERAL CALCULATIONS
for id=1:720
    dW(id)=(p(id)+p(id+1))/2*(V(id+1)-V(id));
end

   [pressmax, Pressmax]=max(p);
   pmax=pressmax/100000;              % Maximum pressure [bar]
   pmi= sum(dW(:))/Vcyl/100000;       % Indicated mean pressure [bar]
   fmep= 0.97+0.15*(N*60/1000)+0.05*(N*60/1000)^2; % friction mean effective
pressure [bar]
   bmep = pmi-fmep;                   % Effective mean pressure [bar]
   Rend=bmep*100000*Vcyl/(mfCycle*Hi)*100;
   BHP = bmep*100*Vcyl*N/2;           % Effective power [kW]
   BSFC= mfCycle*1000*3600*N/2/BHP;   % Brake specific fuel consumption
   Me= BHP/(2*pi*N)*1000;             % Engine torque [N*m]

T4c=Tg(721);
p4c=p(721);
mExhGasCycle=lambdaV*ro*Vcyl+mfCycle; % Exhaust gas per cylinder per cycle
gasflow=mExhGasCycle*N/2*Ncyl;        % Exhaust mass-flow [kg/s] total
airMassFlow=lambdaV*Vcyl*ro*N/2*Ncyl; % Air mass-flow. Renewal not considered
alpha=maProjCycle/mfProjCycle;
```

ANNEX III: Symbols and acronyms

a	*Wiebe* law efficiency factor
a_a	*Langen* coefficient
a_{ap}	*Langen* coefficient
A_{Eff}	Effective turbine area
b_a	*Langen* coefficient
BDC	Bottom dead centre
BHP	Brake horse power
bmep	Brake mean effective pressure
BSFC	Brake specific fuel consumption
C	Piston stroke
c	Local speed of sound
C_1	*Woschni – Annand* coefficient 1
C_2	*Woschni – Annand* coefficient 2
c_p	Constant pressure heating capacity
CR	Compression ratio
c_V	constant volume heat capacity
D	Cylinder bore
E	Electromotive force
EGP	Electric generator power
f	*Wiebe* law fuel burnt fraction
f_0	Resonance frequency
f_m	Disturbance frequency
fmep	Friction mean effective pressure
g	Gravitational acceleration
h	Enthalpy
HFO	Heavy fuel oil
h_i	Woschni-Annand heat transfer coeff.

H_i	Lower calorific value
$H_{injected\ fuel}$	Injected fuel enthalpy
HT	High temperature
imep	Indicated mean effective pressure
IVC	Intake valve closing advance
J	Moment of inertia
k	Electric machine constant
k_{mass}	Compressor scale factor
k_n	Compressor Speed factor
L	Angular momentum
L_m	Intake manifold length
L_r	Connecting rod length
LT	Low temperature
m	Mass
$\dot{m}$	Mass-flow
$\dot{m}_A$	Air mass-flow consumed by the engine
$\dot{m}_A$	Intake air mass-flow rate
$m_{A\ proj\ cycle}$	Air mass per cycle and cylinder
$\dot{m}_C$	Compressor mass-flow
MCR	Maximum continuous rate
m_F	Mass of fuel used per full engine cycle
$m_{F\ ciycle}$	Mass of fuel used per cycle and cylinder
$m_{F\ per}$	Selected percentage of fuel to use
$m_{F\ proj}$	Maximum fuel per full engine cycle
$\dot{m}_{GAS}$	Exhaust gas mass-flow rate
$m_{GAScycle}$	Combustion gas mass per cycle

MGO	Marine gas oil
m_{rem}	Mass of the remaining combustion gasses in the cylinder
$\dot{m}_T$	Turbine gas mass-flow
$\dot{m}_{T\,cor}$	Corrected turbine gas mass-flow
m_w	*Wiebe* law form factor
n	Engine speed in rev/s
NC	Cetane number
n_{opt}	Optimum engine speed for best λ_V
n_{TC}	Turbocharger shaft speed
p	Pressure
P	Power
p_0	Initial exhaust pressure
p_{atm}	Atmospheric pressure
p_B	Chen-Flynn, pressure to overcome
p_{C1}	Compressor inlet pressure
p_{C2}	Compressor outlet pressure
P_E	Engine brake effective power
P_{EG}	Electric generator power
p_I	Pressure at intake stroke start
p_{II}	Pressure at compression stroke start
p_{III}	Pressure at compression stroke end
p_{IV}	Pressure at exhaust stroke start and end
p_{max}	Maximum pressure recorded in the cycle
p_{T3}	Turbine inlet pressure
p_{T4}	Turbine outlet pressure
p_{tr}	Instantaneous pressure in the cylinder in case no combustion is given

$\dot{q}$	Newton's law heat flow
Q	Transferred heat
Q_b	Combustion released heat
Q_r	Cylinder transferred heat
R	Crankshaft radius
r	Radius
R_{gas}	Gas constant
R_L	Electrical load
S_m	Intake manifold section area
T	Temperature
T_{C1}	Compressor inlet temperature
T_{C2}	Compressor outlet temperature
TDC	Top dead centre
T_g	Combustion gas instant temperature
T_i	Cylinder wall or liner temperature
T_{T3}	Turbine inlet temperature
T_{T4}	Turbine outlet temperature
u	In-cylinder gas velocity
u_p	Mean piston speed
V	Instant volume
v	Velocity
$\dot{V}_C$	Compressor volumetric air-flow
V_{CC}	Combustion chamber volume
V_{cyl}	Cylinder displacement
VIT	variable injection timing
V_{IVC}	Cylinder volume at *IVC* point
VIVT	variable intake valve timing

V_m	Exhaust manifold volume
V_r	Displacement and V_{IVC} ratio
W	Work
X_{IVC}	Piston position at *IVC*
Z	*Hardenberg and Hase* coefficient
z	Height increase from a certain datum
α	Air-fuel ratio
α_F	Combined *imp* calculation method coeff.
α_T	Turbine flow coefficient
β	Compression ratio
γ	Adiabatic coefficient
δ_η	Compressor efficiency factor
ε	Emissivity coefficient
η_c	Compressor isentropic efficiency
η_{EG}	Electric generator efficiency
η_{IC}	Intercooler efficiency
η_T	Turbine isentropic efficiency
η_{TPP}	Total power-plant efficiency
η_V	Cylinder air renewal efficiency
θ	Crankshaft rotation angle
θ_{CD}	Angle of combustion delay
θ_{CE}	Combustion end angle
θ_{CL}	Combustion length angle
θ_{CS}	Combustion start angle
θ_{EVC}	Exhaust valve opening
θ_{IL}	Injection length angle

θ_{IS}	Injection start angle
θ_{IVC}	Intake valve closing advance angle
λ	Air-fuel equivalence ratio
λ_V	Cylinder filling coefficient
π	Pi number
π_T	Turbine expansion ratio
ρ	Density
σ_0	*Stefan–Boltzmann* constant
τ	Torque
ϕ	Fuel-air equivalence ratio
τ_C	Compressor torque
ω	Angular velocity