Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

Article

An Investigation of the Composition of the Flow in and out of a Two-stroke Diesel engine and Air Consumption Ratio

Mirko Grljušić [1,2,*], Ivan Tolj[1] and Gojmir Radica[1]

[1] Faculty of Electrical Engineering, Mechanical Engineering and Naval Architecture, University of Split, Rudjera Boškovića 32, 21000 Split, Croatia
[2] GM TURBO d.o.o., Vukovarska 58, 21000 Split, Croatia
* Correspondence: mirko.grljusic@fesb.hr; Tel.: +385-98-420-699; Fax: +385-21-305-776.

Abstract: The aim of this research was to investigate the mass, substances and energy flow through two-stroke low speed Diesel engines. For this reason, a zero-dimensional model of the combustion in the engine was developed with a calculated amount and composition of exhaust gases. Due to the large amount of oxygen in the exhaust gases, a ratio of real air consumption and stoichiometric amount of air required for combustion of injected fuel was set. The calculated ratio showed that the engine consumes four times more air than needed for combustion in AFR_{stoich}. In this work it is called the Air Consumption Factor or Ratio and has not been mentioned in scientific literature before. Air Consumption Ratio is defined as a factor of dry or humid air. To be more comprehensive, a modified diagram of composition of the flow in and out of a two-stroke fuel injection engine and the cylinder was drawn.

Keywords: two-stroke engine; uniflow scavenging; exhaust gas composition; two-stroke gas flow performance parameters; air-fuel ratio; air consumption factor/ratio

---

1. Introduction

Low speed, two-stroke, turbocharged diesel engines, are the most common marine propulsion engines used today. These engines are the most efficient among others, exhibiting 50% efficiency. The remaining 50% of the energy released from combustion of fuel is lost to the atmosphere as waste heat.

At the beginning of the development of the internal combustion engine, it was recognized that improvement would be a lengthy and expensive process. Even by the late 1930's, there had been established methods to calculate a large number of physical processes in the engine. In other words, there have been calculations methods developed that enable the analysis of the influential parameters of the working processes in the engine and also provide data for the development of new and improved construction methods.

The theoretical background of engine process calculations rests on the work done by List [1]. Alongside this, the development of methods for the design of the high part of the process, taking into account the increasing number of parameters can be found in [2].

Due to the large impact of changes to the working fluid in the power and efficiency of the engine, developing calculation methods for the low-pressure part of the process are based on the

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

laws of gas dynamics. The simplest one is called the stationary method, which takes into account the processes in the cylinder and distribution devices, ignoring changes of the gas in front of and behind the valve (channel), based on the work of List [3] and Hasselgruber [4].

The models that are described in the papers of McAuly [5] and Woschni [6] approach the real processes in the engine. The processes in the cylinders are described by differential equations derived from the law of conservation of energy and mass, also from the equation of state of gas. Moreover, changes in the properties of the gas due to the compressibility and dissociation were taken into account.

Boy [7] described in detail the process in a propulsion Diesel engine. The model describes the real process in the engine cylinder, the process in the turbocharger, passage channels, and intake distributor by the method of "full – empty".

Some background materials on two-stroke engines can be found in Heywood [8] and Richard Stone [9]. The gas exchange processes are comprehensively treated by Sher [10]. The composition of the inlet and outlet flows of a two-stroke engine and the cylinder are shown by Sher [10] and Van Basshuysen et al [11]. An experimental study of the flow pattern inside a model cylinder of a uniflow scavenged two-stroke engine is presented by Sher et al [12]. The velocity field as well as the turbulent parameters were mapped under steady-flow conditions with the aid of a hot-wire anemometry technique. Ravi and Marathe [13] in their work multidimensional prediction of the scavenging characteristics of a homogeneous charge uniflow scavenged two-stroke cycle engine has been carried out. An engine geometry, having the same dimensions as a General Motors EMD 710 engine has been analyzed. A multidimensional program, CARE, has been developed for this purpose. The fluid flow problem is subdivided into global and local parts and the two parts are solved simultaneously. Combustion is treated as a stoichiometric heat release phenomenon. The computer program is used to study the effects of port/valve sizes and timings on the scavenging characteristics of the engine, provided that the pressures at the inlet and exhaust ports are held constant. It is observed that a larger inlet port area and its early opening-late closing results in a considerable increase in the scavenge ratio, hence resulting in a higher scavenging efficiency and lower trapping efficiency. An increase in the exhaust port area results in an increase in the scavenge ratio but a decrease in the trapping efficiency; this gives rise to an optimum exhaust area for a given inlet area at which the scavenging efficiency is maximal. Carlucci at al. [14] presented an analytical model for the estimation of the trapping efficiency according to the Oswald diagram to the molar concentration of carbon dioxide and oxygen at the tailpipe and then according to the mass flow of air and fuel.

Larsen et al. [15] investigated two-stroke diesel machinery for ships, with five varying configurations to explore the trade-off between increased NOx emissions and the reduction in fuel consumption. By implementing a waste heat recovery system through the use of an organic Rankine cycle and also a hybrid turbocharger, the fuel consumption and NOx emissions were lowered by up to 9% and 6.5% respectively. Andreadis et al. [16] used a large two-stroke marine diesel engine, operating at its full load to explore the pilot injection strategies using simulations of computational fluid dynamics along with an Evolutionary Algorithm. Guan et al. [17] used a modular zero-dimensional engine model that was built in MatLab and Simulink environment to

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

investigate a large two-stoke marine diesel engine's operation. Engine shop trial values were compared with the derived performance parameters of the engine, which was simulated at steady state conditions. The study purpose of Varbanets et al. [18] was to record the methods upon which the ship's diesel process efficiency could be improved. Under the conditions of the fuel equipment and the normal state of the main diesel system, even load distribution between the cylinders was controlled. In their previous researches, the authors investigated the possibility of increasing efficiencies of a low speed two-stroke turbocharged main diesel engine operating with waste heat recovery through combined heat and power production [19, 20]. Spahni et al in the article [21] in particular deals with the new electronically control, two stroke, low speed marine engines. Carlucci et al in the article [22] compared different architectures (single turbocharger, double turbocharger, single turbocharger with an electrically-assisted turbocharger, with intercooler or aftercooler) designed to supercharge an aircraft 2-stroke Diesel engine for general aviation and unmanned aerial vehicles characterized by a very high altitude operation and long fuel distance. A 1D model of the engine purposely designed has been used to compare the performance of different supercharging systems in terms of power, fuel consumption, and their effect on trapping and scavenging efficiency at different altitudes. In the paper [23] Carlucci et al provided several guidelines about the definition of design and operation parameters for a two-stroke two banks uniflow diesel engine, turbocharged with two sequential turbochargers and an aftercooler per bank, with the goal of either increasing the engine brake power at take-off or decreasing the engine fuel consumption in cruise conditions. The engine has been modelled with 0D/1D modelling approach. Wang et al [24] evaluated scavenge port designs for a boosted uniflow scavenge direct injection gasoline engine by 3D CFD simulations. In order to fulfil the potential of the BUSDIG engine, various scavenge ports were designed with different scavenge port number, Axis Inclination Angle and Swirl Orientation Angle, and their effects were evaluated by 3D computational fluid dynamics (CFD) under different intake pressures and engine speeds. The scavenging process was analysed by its delivery ratio, trapping efficiency, scavenging efficiency and charging efficiency.

This investigation was a part of the research how to improve efficiency of Diesel engine plant using waste energy contained in exhaust gases and scavenge air. For this purpose we needed information about quantity of mass, substance and energy flow through the engine and a zero-dimensional model of the combustion and exchange process in the engine was developed. Our 0D program has been directly coupled with NIST-REFPROP Version 9.0 Standard Reference Database [25] so that all physical, chemical and thermodynamic data for all used fluids could be used in our calculations. Calculated amount and composition of the exhaust gases showed good correlation with measured results of MAN Diesel & Turbo. Since our results showed that the amount of an air that passes through the engine is several times greater than required by stoichiometric combustion, it was decided to establish new term: Air Consumption Ratio. At the end of this paper, the parameters of substantive changes in two-stroke engines with fuel injection will be strictly defined. Instead of using the displacement volume as a reference volume what is common in literature, we used V_{max} as a reference volume, which is more appropriate for two-stroke engines. For the purpose of connecting the Air Consumption Ratio with two-stroke gas performance parameters, we proposed an improved version of the figure - Composition of the flows in and out a two-stroke

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

engine and the cylinder. We also found that Air Consumption ratio depends on ratio of Excess of the Air in cylinder and Trapping Efficiency.

2. Engine Model Description

The propulsion engine model depicting the main input and output variables is shown in Figure 1. It is assumed that fluid flow through a propulsion engine is steady. The observer position is stationary with respect to the control volume surrounding the drive motor, in this case, the ship's propulsion engine. In the simplest assessment, the fuel and the air enter into the control volume (CV), while combustion products, work $\dot{W}$ and heat $\dot{Q}_1$ and $\dot{Q}_2$ are taken away at different temperatures T_1 and T_2. For greater accuracy it is assumed that the cylinder oil enters into the control volume, and along with combustion products, emissions exit in low quantities of environmentally-influential substances [26].

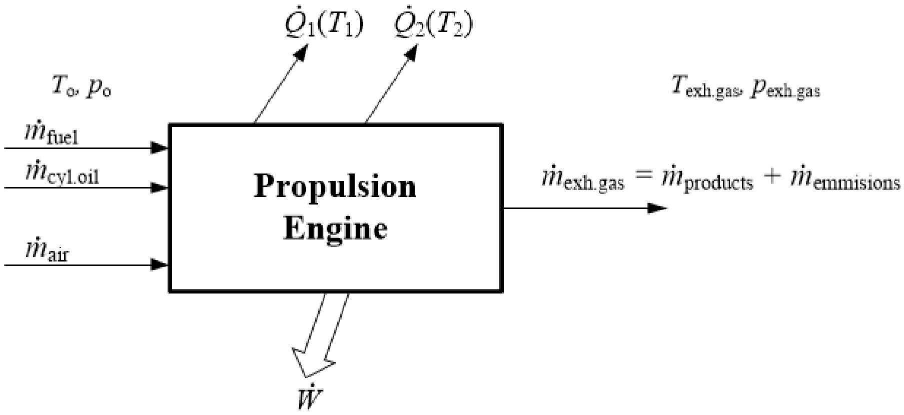


Fig. 1. The propulsion engine model depicting main input and output substances and energy flows.

The engine operates in the environment temperature T_0, in which the unused heat can be rejected. The environment in this case is the atmosphere, but the environment is also the cooling water from the sea or river. The environment can be considered as large enough when T_0 does not change as a result of heat transfer. For proper operation of the engine, it is necessary to lubricate all the parts that rub against each other, and this task is performed by lubricating oil. This oil takes part of the friction work and heat; heat carrier from the cylinder liner, cylinder cover, lubricating oil, scavenged air and the cooling water. In this paper, the reference state of the environment is taken at STP, where the temperature of the intake air and cooling water is 25.0 ° C at 1.00 bar. At steady state conditions the continuity equation is $\dot{m}_{in} = \dot{m}_{out}$.

The first law of thermodynamics can be expressed as conservation of energy in the control volume. Disregarding changes in kinetic energy of the all substances flow, conservation of energy for steady state flow can be written as:

$$\dot{Q} - \dot{W} = (\sum_i \dot{m}_i \cdot h_i)_{out} - (\sum_i \dot{m}_i \cdot h_i)_{in} \tag{1}$$

where $\dot{Q}$ is the sum of flows heat output, $\dot{W}$ flow of taken (obtained) shaft work, $\dot{m}_i$ and h_i are the mass flow rates and enthalpies of the individual substance flow. In Equation (1) as the form of the first law of thermodynamics, the work is technical. This means that it is done by moving the

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

fluid with respect to the machine control volume, and not due to changes in volume of fluid that moves in relation to the control volume. Accordingly, below is $\dot{W} = \dot{W}_{\text{tech}}$. Now:

$$\dot{W} = \dot{m} \cdot (\sum h_{\text{in}} - \sum h_{\text{out}}) + \dot{Q} \tag{2}$$

As a result of heat transfer to the environment $\dot{Q}_0 = -\dot{Q}$, its entropy increases for $\Delta\dot{S}_0$. Therefore $\Delta\dot{S}_0 \equiv -\dot{Q}/T_0$ and $-\dot{Q} = T_0 \cdot \Delta\dot{S}_0$.

Now, the first law of flow processes is:

$$\dot{W}_{\text{tehn}} = \dot{m} \cdot (\sum h_{\text{in}} - \sum h_{\text{out}}) - T_0 \cdot \Delta\dot{S}_0 \tag{3}$$

If the working fluids, control volume and environment are considered united as a single isolated system, then from the Clausius inequality (second law for all processes) the total amount of the change in entropy is:

$$\dot{m}(s_{\text{out}} - s_{\text{in}}) + \Delta\dot{S}_0 \geq 0$$

$$T_0 \cdot \Delta\dot{S}_0 \geq T_0 \cdot \dot{m} \cdot (s_{\text{in}} - s_{\text{out}})$$

$$\dot{W}_{\text{tehn}} \leq \dot{m} \cdot \left[(\sum h_{\text{in}} - \sum h_{\text{out}}) - T_0 \cdot (s_{\text{in}} - s_{\text{out}})\right]$$

If all processes are reversible, the maximum output power is:

$$\dot{W}_{\text{tehn, max}} = \dot{m} \cdot \left[(\sum h_{\text{in}} - \sum h_{\text{out}}) - T_0 \cdot (s_{\text{in}} - s_{\text{out}})\right] \tag{4}$$

In the equation (4) a change enthalpy shows how greater work would be obtained, when there are no heat exchanges with the environment. Rearranging follows:

$$\dot{W}_{\text{tehn, max}} = \dot{m} \cdot \left[(h - T_0 s)_{\text{in}} - (h - T_0 s)_{\text{out}}\right]$$

or per unit mass flow:

$$\frac{\dot{W}_{\text{tehn, max}}}{\dot{m}} = w_{\text{tehn, max}} = (h - T_0 s)_{\text{on}} - (h - T_0 s)_{\text{out}} = e_{\text{x}} \tag{5}$$

Expression $h - T_0 s$ known as exergy function ζ, and e_x exergy per 1 kg input or output substance.

In the case of the internal combustion engine or a fuel cell, as shown in Fig. 1, if the air and fuel enter at atmospheric temperature and pressure, maximum work from the fuel is obtained if the process is in equilibrium and the combustion products leave the system at atmospheric temperature and pressure. If so, defined state of equilibrium combustion products with the environment, from the equation (5) per 1 kg of reactants follow:

$$w_{\text{tehn.max}} = (h_{\text{reactants}} - h_{\text{products}})_{T_0, p_0} - T_0 \cdot (s_{\text{reactants}} - s_{\text{products}})_{T_0, p_0} \tag{6}$$

Per 1 kg of fuel is:

$$w^*_{\text{tehn.max}} = \Delta H'_{T_0, p_0} - T_0 \cdot (s^*_{\text{reactants}} - s^*_{\text{products}})_{T_0, p_0} \tag{7}$$

Where :

$$w^*_{\text{tehn.max}} = w_{\text{tehn.max}} \cdot (1 + \lambda m_{\text{a stoich}}) \tag{8}$$

$\Delta H'_{T_0, p_0}$ is the enthalpy of reaction of combustion or fuel caloric value. The suffix * means a quantity corresponding to 1 kg of fuel, and without suffix * amount per kg of products. In that $\Delta H'_{T_0, p_0}$ would be a lower caloric value if H_2O in the combustion products was in the vapour state and the higher caloric value if the H_2O in the combustion products was in the liquid state.

When the liquid hydrocarbon fuel burns with air and forms carbon dioxide and water, under their full balance with the atmosphere, it is considered their states with the ambient temperature T_0

and the pressures equal to their partial pressures into the atmosphere. In that case $s^*_{\text{products}} \approx s^*_{\text{reactants}}$ with $\left[\Delta H'\right]_{T_0, p_0}$ is equal to fuel higher caloric value. It follows for these fuels $w^*_{\text{tehn,max}} \approx \left[\Delta H'\right]_{T_0, p_0}$. For pure carbon lower and higher caloric values are the same, for hydrogen $w^*_{\text{tehn,max}} = 0.823\left[\Delta H'\right]_{T_0, p_0}$. For gaseous hydrocarbons $w^*_{\text{tehn.max}}$ is a few % lower than $\left[\Delta H'\right]_{T_0, p_0}$. But since in practice, the full amount of $w^*_{\text{tehn.max}}$ is very difficult to achieve, the exergy efficiency calculated as $\eta_{ex} = w^*_{\text{tehn}}/H_d$ will be used for the purpose of comparison.

2.1. Data MAN B&W CEAS_ERD (Engine Room Dimensioning)

For Newbuilding 447 „Donat“ Tankerska plovidba d.d. Zadar, in Shipyard and Diesel engine Factory SPLIT, a propulsion engine 6S70MC-C7 was chosen with the following characteristics:

bore = 700 mm	stroke = 2 800 mm	s/d = 4,0	c_m = 8,5 m/s
SMCR L_1:	n = 91 min^{-1}	p_{me} = 19,0 bar	*NMCR* = 18 660 kW

Using MAN B&W CEAS_ERD (Engine Room Dimensioning) [27] with selection of the specified operating point i.e. *SMCR* (Specified Maximum Continuous Rating) is 100% *NMCR* (Nominal Maximum Continuous Rating) at 91 min^{-1} and marked by point L_1 on Figure 2.

Optimising point is 85% *SMCR* and marked with light blue point on the propeller curve, since *NCR* (Normal Continuous Rating) which is also Service Rating *SCR*, 80% *SMCR*.

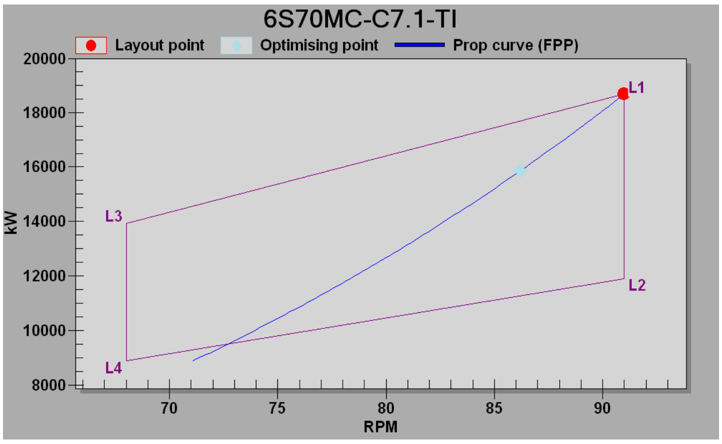


Fig. 2. The quad of mean effective pressure and engine speed with chosen *NCR* [27]

Lines $L_1 - L_3$ and $L_2 - L_4$ are isobars and refer to the respective mean effective pressure p_{me}, and lines $L_1 - L_2$ & $L_3 - L_4$ are isotachs and relate to the constant speed n of engine. The blue curve that passes through the point L1 is the heavy propeller curve, and it is expected that the voyage of the ship will mainly reflect the actual conditions of navigation as hull fouling (covered by shells and algae), and environmental conditions (waves, wind, ocean currents, etc.).

Optimized power is 15.861 kW at 86.2 min^{-1}, and Service Power or *NCR* is taken to be 80% *SMCR* which is 14.928 kW at 84.5 min^{-1}.

2.2. The flow of substances through the engine

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

Data for ISO ambient condition according to [27] are shown in tables 1 and 2 for engine loads from 50% to 100%. Specific cylinder oil consumption $b_{e, c.o.}$ is 0.60 g/kW h at all engine loads. From Table 2 is obvious that scavenges air cooler heat has great potential especially at higher engine loads.

Table 1. *SFOC* and exhaust gases data, ISO conditions [27].

Load % *SMCR*	Power kW	Speed min^{-1}	*SFOC* g/kWh	Exhaust gas amount kg/h	Exhaust gas temp. °C
100	18,660	91.0	170.9	172,800	240.5
95	17,727	89.5	168.8	166,200	235.6
90	16,794	87.9	167.1	159,600	232.0
85	15,861	86.2	165.6	152,900	229.7
80	14,928	84.5	164.4	146,000	228.8
75	13,995	82.7	163.5	138,900	229.1
70	13,062	80.8	162.9	131,500	230.8
65	12,129	78.8	162.9	123,800	233.7
60	11,196	76.8	163.3	115,800	238.0
55	10,263	74.6	163.9	107,500	243.6
50	9,330	72.2	164.8	98,800	250.5

Ambient Air Suction Temperature: 25.0 °C Cooling Water Temperature: 25.0 °C

Table 2. Heat exchanger power at various engine loads, ISO conditions [27]

1 Engine load (% SMCR) **4** Scavenge air amount +/- 5% (kg/h) **9** Jacket water cooler heat -15/+0% (kW)
2 Engine power (kW) **5** Scavenge air pressure (bar abs) **10** Main lubrication oil heat (kW)
3 Engine speed (r/min) **6** Scavenge air temperature before air cooler (°C)
7 Scavenge air temperature after air cooler (°C)
8 Scavenge air cooler heat (kW)

ISO Ambient Conditions Air suction temperature:25.0°C Cooling water temperature:25.0°C

1 %	2 kW	3 r/min	4 kg/h	5 bar(abs)	6 °C	7 °C	8 kW	9 kW	10 kW
100	18,660	91.0	169,600	3.79	193.0	37.0	7,420	2,390	1,260
95	17,727	89.5	163,200	3.62	186.0	36.0	6,890	2,290	1,240
90	16,794	87.9	156,800	3.45	179.0	34.0	6,360	2,200	1,220
85	15,861	86.2	150,300	3.28	172.0	33.0	5,840	2,100	1,200
80	14,928	84.5	143,500	3.11	164.0	32.0	5,310	2,010	1,180
75	13,995	82.7	136,600	2.94	156.0	31.0	4,790	1,920	1,160
70	13,062	80.8	129,400	2.77	148.0	30.0	4,270	1,830	1,130
65	12,129	78.8	121,800	2.60	139.0	29.0	3,750	1,730	1,100
60	11,196	76.8	114,000	2.43	130.0	29.0	3,250	1,640	1,070
55	10,263	74.6	105,800	2.26	121.0	28.0	2,750	1,550	1,040
50	9,330	72.2	97,300	2.09	111.0	27.0	2,270	1,460	1,000

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

Scavenge air amount, pressure and temperature are very important for modelling of mass, substance and energy flow during breathing phase of two stroke marine Diesel engine. Scavenge air cooler heat, Jacket water cooler heat and data from Table 1 are necessary for calculation of waste energy which could be used in Waste Heat Recovery (WHR) processes.

Basic data with mass, substances and energy flow through the engine has been shown on Figure 3 showing mass, energy, shaft work and heat flow through the control surface. In this open circuit heat engine, reactants (fuel and air) cross the control surface at inlet and products of combustion (exhaust gases) leave it at exit, while only work, but no heat crosses the control surface. Note, that the exhaust gases, though hot, convey energy but no heat. $\dot{Q}_1$ and $\dot{Q}_2$ represent heat losses to the environment through cooling water. Cylinder oil $\dot{m}_{c.o.}$ was added at engine inlet since most of this oil burns in cylinder. Air pressure after turbocharger is 250 mm of water column higher than in the air distributor.

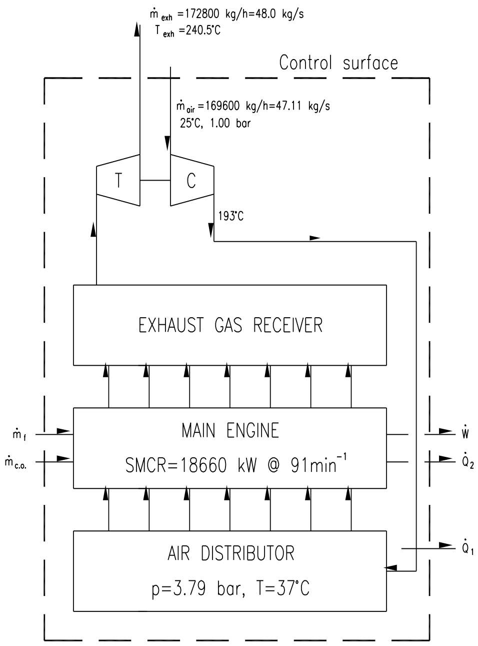


Fig. 3. Schematic diagram of the conventional main engine 6S70MC.C7 for ISO conditions and 100% *SMCR*

In engine enters $\dot{m}_{a.h.} = 169\ 600\ \mathrm{kg/h} = 47.1111\ \mathrm{kg/s}$ at 100% *SMCR* and ISO standard conditions (25.°C air temperature at the turbocharger inlet and fresh water cooling temperature, ambient air pressure 1 bar and relative air humidity $\Phi = 30\%$).

$$x = 0.622 \frac{\Phi \cdot p_{ws}}{p - \Phi \cdot p_{ws}} = 0.622 \frac{p_p}{p - p_p} \tag{9}$$

Where x = vapor quality, p_{ws} = water saturation pressure [Pa], p = humid air pressure [Pa], and p_P = partial vapor pressure [Pa].

$$\dot{m}_{H_2O} = x \cdot \dot{m}_{a.h.} \tag{10}$$

$$\dot{m}_{a.d.} = \dot{m}_{a.h.} - \dot{m}_{H2O} \tag{11}$$

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

2.3. Calculation of combustion based on fuel, cylinder oil and air consumption

Although the scavenged air consumption was taken from [27] at ISO standard conditions, the composition of the air was calculated by NIST [25] using our initial values. The fuel composition for MDO and cylinder oil composition were taken into account, as well. The heat input from the fuel is expressed in terms of its Lower Calorific Value while the fuel and air mass flow rates must meet the following relation:

$$\dot{W} = \dot{m}_{\mathrm{f}} \cdot \Delta H'_{T_0, p_0} - (\dot{m}_{\mathrm{a}} + \dot{m}_{\mathrm{f}}) \cdot c_{\mathrm{p,\,exh\,gas}} (T_{\mathrm{exh\,gas}} - T_0) + \dot{Q} \tag{12}$$

Specific mass flow through the engine:

$$smf_{\mathrm{react}} = smf_{\mathrm{exh.gas}} = smf_{\mathrm{a}} + smf_{\mathrm{f}} + smf_{\mathrm{c.o.}} = z_{\mathrm{b}} + b_{\mathrm{f}} + b_{\mathrm{c.o.}} \tag{13}$$

Where subscript a denotes air, f fuel and c.o. cylinder oil.

AIR

Oxygen content at the inlet of the cylinder:

$$N_{\mathrm{O2}} = \frac{smf_{\mathrm{a.d.}}}{M_{\mathrm{a.d.}}} \cdot r_{\mathrm{O2}} \quad \frac{\mathrm{kmol\ O_2}}{\mathrm{kW\ h}}$$

Where N denotes Specific Molar Flow Rate [kmol/kWh], M Molar Mass [kg/kmol] and r Molar and Volume fraction.

FUEL

Fuel energy is: $\dot{E}_{\mathrm{f}} = \dot{m}_{\mathrm{f}} \cdot \Delta H'_{T_0, p_0}$ kJ/s .

The amount of carbon, hydrogen and sulphur from fuel at the inlet of the cylinder:

$$\dot{m}_{\mathrm{C,f}} = g_{\mathrm{C,f}} \cdot \dot{m}_{\mathrm{f}} \quad \mathrm{kg/s}$$

$$smf_{\mathrm{C,f}} = b_{\mathrm{e}} \cdot g_{\mathrm{C,f}} \quad \mathrm{kg\ C/kWh}$$

$$N_{\mathrm{C,f}} = \frac{b_{\mathrm{e}} \cdot g_{\mathrm{C,f}}}{M_{\mathrm{C}}} = \frac{smf_{\mathrm{C,f}}}{M_{\mathrm{C}}} \quad \frac{\mathrm{kmol\ C}}{\mathrm{kW\ h}}$$

$$N_{\mathrm{H,f}} = \frac{b_{\mathrm{e}} \cdot g_{\mathrm{H,f}}}{M_{\mathrm{H}}} = \frac{smf_{\mathrm{H,f}}}{M_{\mathrm{H}}} \quad \frac{\mathrm{kmol\ H}}{\mathrm{kW\ h}}$$

$$N_{\mathrm{S,f}} = \frac{b_{\mathrm{e}} \cdot g_{\mathrm{S,f}}}{M_{\mathrm{S}}} = \frac{smf_{\mathrm{S,f}}}{M_{\mathrm{S}}} \quad \frac{\mathrm{kmol\ S}}{\mathrm{kW\ h}}$$

Where g is Mass fraction.

CYLINDER OIL

The amount of carbon and hydrogen from cyl oil at the inlet of the cylinder:

$$N_{\mathrm{C\ c.o.}} = \frac{smf_{\mathrm{HC\ c.o.}} \cdot g_{\mathrm{C\ c.o.}}}{M_{\mathrm{C}}} = \frac{smf_{\mathrm{C\ c.o.}}}{M_{\mathrm{C}}} \quad \frac{\mathrm{kmol\ C}}{\mathrm{kW\ h}}$$

$$N_{\mathrm{H\ c.o.}} = \frac{smf_{\mathrm{HC\ c.o.}} \cdot g_{\mathrm{H\ c.o.}}}{M_{\mathrm{H}}} = \frac{smf_{\mathrm{H\ c.o.}}}{M_{\mathrm{H}}} \quad \frac{\mathrm{kmol\ H}}{\mathrm{kW\ h}}$$

Sulphur content at the inlet is ignored as the size of a lower order.

Calculation of the amount of unburnt fuel and cylinder oil in exhaust gases

Mass of hydrocarbons entered the cylinder:

$$smf_{\mathrm{HC}} = smf_{\mathrm{HC\ f}} + smf_{\mathrm{HC\ c.o.}} \tag{14}$$

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

HC share in the total mixture of the reactants is $smf_{\mathrm{HC}} / smf_{\mathrm{react}}$

Emissions and unburnt fuel in the exhaust gases

According to MAN data for MC/ME low speed Diesel engines measured emissions are:

1 500 vppm NOx (90 ÷ 95 % NO i 5 ÷ 10 % NO_2)

70 ppm SOx

60 ppm CO

180 ppm HC

$\dot{E}_{\mathrm{f\ unburnt}} = \dot{m}_{\mathrm{f\ unburnt}} \cdot \Delta H'_{T_0, p_0}$ kJ/s

$$\%\ \text{fuel unburnt} = \frac{smf_{\mathrm{f\ unburnt}}}{b_{\mathrm{e}}} \cdot 100 = 0.962482658\ \% \tag{16}$$

$$\%\ \text{unburnt fuel and cyl oil} = \frac{smf_{\mathrm{HC\ t\ unburnt}}}{b_{\mathrm{e}} + b_{\mathrm{c.o.}}} \cdot 100 = 1.298610269\ \% \tag{17}$$

$N + O \rightarrow NO$

Stoichiometric amount of oxygen required for combustion:

$C + O_2 \rightarrow CO_2$

$H_2 + 1/2\ O_2 \rightarrow H_2O$

$S + O_2 \rightarrow SO_2$

2.4. Air Consumption Ratio/Factor

Now, the Air Consumption Ratio/Factor is:

$$\Lambda_{\mathrm{AC}} = \frac{N_{\mathrm{O2,in}}}{\Sigma N_{\mathrm{O2,f}}} \tag{18}$$

a ratio of oxygen entered the engine to oxygen necessary for stoichiometric combustion of fuel.

Stoichiometric (dry air) Air-Fuel Ratio is:

$$AFR_{\mathrm{a.d.\ stoich}} = \frac{smf_{\mathrm{a.d.\ stoich}}}{smf_{\mathrm{f}}} = \frac{N_{\mathrm{a.d.\ stoich}}}{N_{\mathrm{f}}} \tag{19}$$

Mole fraction of the fuel is difficult to calculate because the exact composition of the fuel is unknown. Therefore, the calculation is done by mass fraction.

Since the stoichiometric combustion 0.1709 kg fuel / kWh required 0.01808288 kmol O2 / kWh will be set ratio with the amount of O_2 in dry air.

$$x = \frac{r_{\mathrm{O2\ ISO}}}{\Sigma N_{\mathrm{O2\ stoich,f}}} = \frac{0.2096\ \text{kmol O}_2 / \text{kmol air dry}}{0.01808288\ \text{kmol O}_2/\text{kWh}} = 11.59107393\ \text{kWh/kmol air dry}$$

Now one can calculate the amount of O_2 and other components in air (N_2 and Ar) in kg/kWh necessary for ideal stoichiometric combustion of 0.1709 kg/kWh fuel.

$$smf_{\mathrm{O2,stoich}} = N_{\mathrm{O2,stoich}} \cdot M_{\mathrm{O2}} \quad \text{kg O}_2/\text{kWh}$$

$$smf_{\mathrm{N2,stoich}} = \frac{r_{\mathrm{N2}}}{x} \cdot M_{\mathrm{O2}} = N_{\mathrm{O2,stoich}} \cdot M_{\mathrm{O2}} \quad \text{kg N}_2/\text{kWh}$$

$$smf_{\mathrm{Ar,stoich}} = \frac{r_{\mathrm{Ar}}}{x} \cdot M_{\mathrm{Ar}} = N_{\mathrm{Ar,stoich}} \cdot M_{\mathrm{Ar}} \quad \text{kg Ar/kWh}$$

$$smf_{\mathrm{a.d.stoich}} = smf_{\mathrm{O2,stoich}} + smf_{\mathrm{N2,stoich}} + smf_{\mathrm{Ar,stoich}}$$

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

$$AFR_{\text{stoich}} = (\dot{m}_{\text{a}} / \dot{m}_{\text{f}})_{\text{stoich}} \text{ stoichiometric air fuel ratio} \tag{20}$$

AFR_{stoich} for fuel combustion (without cyl oil):

$$AFR_{\text{stoich}} = \frac{smf_{\text{a.d. stoich}}}{smf_{\text{f}}} \quad \text{kg}_{\text{a.d.}}/\text{kg}_{\text{f}}$$

$$AFR_{\text{actual}} = \frac{smf_{\text{a.h. in}}}{smf_{\text{f}}} \quad \text{kg}_{\text{a.h.in}}/\text{kg}_{\text{f,in}}$$

Minimum amount of air required for combustion of fuel:

$$\dot{m}_{\text{a,stoich}} = \dot{m}_{\text{f}} \cdot AFR_{\text{stoich}} \quad \text{kg air/s} \tag{21}$$

And Air Consumption Ratio can be defined as:

$$\Lambda_{\text{AC}} = \frac{(\dot{m}_{\text{a}} / \dot{m}_{\text{f}})_{\text{actual}}}{(\dot{m}_{\text{a}} / \dot{m}_{\text{f}})_{\text{stoich}}} \tag{22a}$$

$$\Lambda_{\text{AC}} = \frac{\dot{m}_{\text{a.d. actual}}}{\dot{m}_{\text{a,stoich}}} \tag{22b}$$

2.5. Calculation of exhaust gas composition using NIST refprop09

The pressure drop through the exhaust system (spark arrestor, exhaust gas silencer, exhaust gas boiler and piping was given by Abbo [28].

Total quantity kmol/kWh in the exhaust gases

$$\Sigma N_{\text{t,exh.gas}} = (N_{\text{O2,in}} - N_{\text{O2,burn}}) + N_{\text{H2O,in}} + N_{\text{N2,in}} + N_{\text{Ar,on}} + N_{\text{HC,unburn}} + (N_{\text{CO2act}} + N_{\text{H2O,burned}} + N_{\text{SO2act}}) \tag{23}$$

The share of components in the exhaust gases:

$$r_{\text{i}} = \frac{N_{\text{i}}}{\sum_{\text{i}} N_{\text{i}}} \tag{24}$$

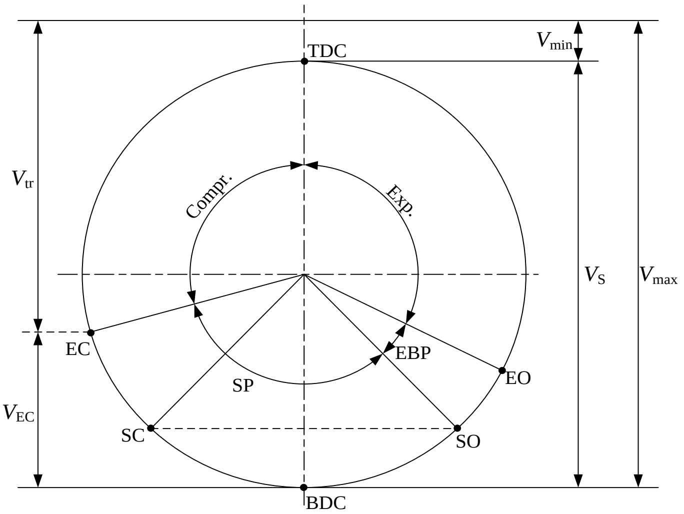
Where "i" is the index of chemical ingredient in the mixture of exhaust gases.

2.6. The composition of the flow in and out of a two-stroke engine and the cylinder

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

The absence of the separate induction and exhaust strokes in the two-stroke engines is fundamentally different from four-stroke engines. In two-stroke engine after a short blowdown exhaust at the end of expansion, the gas exchange process has the induction and exhaust processes occurring simultaneously when the piston is near BDC. The most efficient scavenges process will be the one whereby the products of combustion are completely replaced by a fresh charge, at the charge pressure and charge temperature. In fact, the fresh charge does not replace the products of combustion by a perfect displacement mechanism, and charging efficiency will be always smaller than unity. Another very important feature of the two-stroke exchange gas process is unavoidable supercharging or turbocharging, which is not necessary in four-stroke engines. In figure 4, is shown the timing diagram for a two-stroke engine.



EC = Exhaust valve Closed
EO = Exhaust valve Opening
SC = Scavenging port Closed
SO = Scavenging port Opening
SP = Scavenging Period
EBP = Exhaust Blowdown Period

Fig. 4. The timing diagram for a two-stroke Diesel engine

When defining parameters of substance changes in the two-stroke engine with fuel injection, the following symbols will be used:

p_{sc} = scavenge air pressure at the entrance of scavenging ports

T_{sc} = scavenge air temperature at the entrance of scavenging ports

T_s = gas temperature in cylinder at the end of scavenge process

$\dot{m}_{sc}$ = mass of air delivered to engine at pressure p_{sc} and temp. T_{sc} per sec

$m_{sc,h}$ = mass of humid air delivered to the cylinder/cycle at pressure p_{sc} and temp. T_{sc}

$m_{sc,d}$ = mass of dry air delivered to the cylinder/cycle at pressure p_{sc} and temp. T_{sc}

$m_{a,H2O}$ = mass of water delivered with a humid air to the cylinder/cycle at pressure p_{sc} and temp. T_{sc}

m_{ar} = mass of fresh air retained in the cylinder / cycle

$m_{ar,t}$ = total air mass in the cylinder / cycle after the SPC and EVC

m_r = residual mass of gases in cylinder / cycle

m_{rp} = residual (recirculated) mass of products in the cylinder / cycle

m_{tr} = m_{ar}+m_r = mass of trapped charge (fresh air and residual gas in the cylinder/cycle) after SC and EC

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

m' = mass of air that could be caught in the cylinder / cycle at a pressure p_{sc} and a temperature T_{sc}

$m_{a.d.\ stoich,}$ = mass of dry air required for stoichiometric combustion in the cylinder / cycle

$m_{a.h.\ stoich}$ = mass of humid air required for stoichiometric combustion in the cylinder / cycle

$m_{a,excess}$ = excess air in the cylinder/cycle

$m_{a.d.,s\text{-}c}$ = mass of dry air, directly blowout cylinder / cycle

m_f = the mass of fuel injected into the cylinder / cycle

m_{co} = the mass of cylinder oil injected into the cylinder / cycle

$m_{a.d.\ out}$ = the total mass of the dry air at the outlet of the cylinder / cycle

m_p = mass products of combustion at the exit from the cylinder / cycle (including moisture from the air and emissions)

m_{exh} = the total mass of the exhaust gas at the outlet of the cylinder / cycle

$$m_{sc} = \frac{\dot{m}_{sc}}{(n/60)\cdot z}$$

$$m' = V_{max}\cdot\rho_{sc} = V_1\cdot\rho_{sc}$$

$m_p = m_{react} = m_{a,.d.\ stoich} + m_{a,H2O} + m_f + m_{co}$

$m_{exh} = m_{sc,h} + m_f + m_{co} = m_p + m_{a.d.out}$

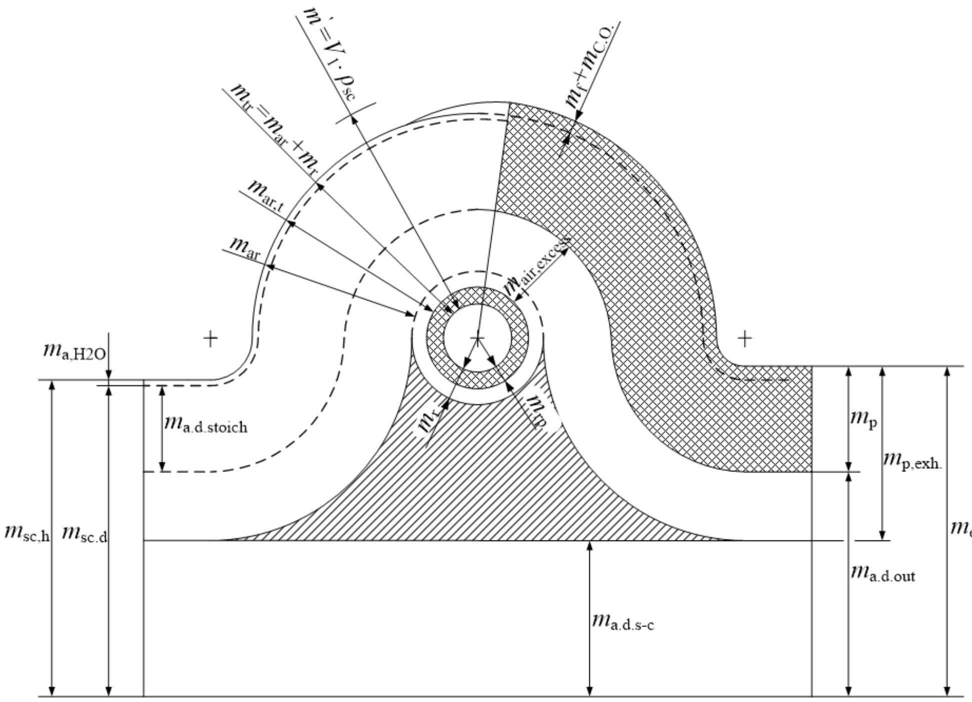


Fig. 5. The composition of the flow in and out of a two-stroke fuel injection engine and the cylinder after [10 and 11]

Figure 5 is very important and it enables to properly define the parameters of working fluid exchange in the two stroke engines. Such figure was suggested by Sher and gradually was modified and improved. Figure 5 is significantly improved compared to the same such figures presented in literature [8, 9, 10 and 11]. On this figure fuel was introduced as one of the important constituents which pass through the cylinder. Also, intake humid air which takes part in combustion is shown as dry air and water.

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

Parameters of working fluids exchange can be now defined according to Fig. 5:

(a) Delivery ratio (Scavenge ratio) Λ_{d}

$$\Lambda_{\mathrm{d}} = \frac{\text{mass of delivered air}}{\text{reference mass}} = \frac{m_{\mathrm{sc}}}{m'} = \frac{m_{\mathrm{sc}}}{V_1 \cdot \rho_{\mathrm{sc}}} \tag{25}$$

It compares the actual mass of air for scavenging the engine to the one required by the ideal turbocharging process. The reference mass is defined as the product of cylinder maximum volume and density of the ambient air or with high supercharging engine as the product of the cylinder maximum volume and density of air in the air distributor.

(b) Charging efficiency η_{ch}

$$\eta_{\mathrm{ch}} = \frac{\text{mass of delivered air retained}}{\text{reference mass}} = \frac{m_{\mathrm{ar}}}{m'} = \frac{m_{\mathrm{ar}}}{V_1 \cdot \rho_{\mathrm{sc}}} \tag{26}$$

It shows how successfully the cylinder volume filled with fresh air.

(c) Scavenging efficiency η_{sc}

$$\eta_{\mathrm{sc}} = \frac{\text{mass of delivered air retained}}{\text{mass of trraped cylinder charge}} = \frac{m_{\mathrm{ar}}}{m_{\mathrm{ar}} + m_{\mathrm{r}}} = \frac{m_{\mathrm{ar}}}{m_{\mathrm{ar,t}} + m_{\mathrm{rp}}} \tag{27}$$

It indicates to what extent the residual (recirculation) gases in the cylinder m_r are replaced by a fresh charge m_{ar}.

The total mass of charge (trapped cylinder charge) is the sum of the mass of fresh air $m_{ar,t}$, the mass of burnt gas and the mass of unburnt fuel from previous cycle m_{rp}.

(d) Retaining Efficiency η_{rt}

$$\eta_{\mathrm{rt}} = \frac{\text{mass of delivered air retained}}{\text{mass of delivered air / cycle}} = \frac{m_{\mathrm{ar}}}{m_{\mathrm{sc}}} \tag{28}$$

It shows how much air came directly to the exhaust.

The charging efficiency can be defined in terms of the delivery ratio and retaining efficiency,

$$\eta_{\mathrm{ch}} = \Lambda_{\mathrm{d}} \cdot \eta_{\mathrm{rt}} \tag{29}$$

At the closing of the scavenging ports and exhaust valve the retaining efficiency becomes the trapping efficiency.

(e) Trapping efficiency η_{tr}

$$\eta_{\mathrm{tr}} = \frac{\text{mass of trapped air charge}}{\text{mass of delivered air / cycle}} = \frac{m_{\mathrm{ar,t}}}{m_{\mathrm{sc}}} \tag{30}$$

(f) Relative charge (Volumetric efficiency) Λ_{tr}

$$\Lambda_{\mathrm{tr}} = \frac{\text{mass of trapped cylinder charge}}{\text{reference mass}} = \frac{m_{\mathrm{tr}}}{m'} = \frac{m_{\mathrm{tr}}}{V_1 \cdot \rho_{\mathrm{sc}}} \tag{31}$$

The factor of relative charge is an indication of the degree of charging, and it is the ratio of charging efficiency and the scavenging efficiency.

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at *Energies* **2017**, *10*, , 805; doi:10.3390/en10060805

$$\Lambda_{\mathrm{tr}} = \eta_{\mathrm{ch}} / \eta_{\mathrm{sc}} \tag{32}$$

When the reference mass in the definition of delivery ratio is air mass and gas trapped in the cylinder m_{tr} (or a close approximation of it), then:

$$\eta_{\mathrm{sc}} = \Lambda_{\mathrm{d}} \cdot \eta_{\mathrm{tr}} \tag{33}$$

(g) Air-Fuel Ratio (*AFR*) for stoichiometric fuel combustion:

$$AFR_{\mathrm{stoich}} = \frac{m_{\mathrm{a\ stoich}}}{m_{\mathrm{f}}} = \frac{\dot{m}_{\mathrm{a\ stoich}}}{\dot{m}_{\mathrm{f}}} \tag{20}$$

Minimal quantity of air necessary for stoichiometric burning of injected fuel:
$\dot{m}_{\mathrm{a.d.\ stoich}} = \dot{m}_{\mathrm{f}} \cdot AFR_{\mathrm{stoich}}$ refers to the dry air. Now,

$$AFR_{\mathrm{actual}} = \frac{m_{\mathrm{a\ actual}}}{m_{\mathrm{f}}} = \frac{\dot{m}_{\mathrm{a\ actual}}}{\dot{m}_{\mathrm{f}}} \tag{34}$$

(h) Excess of the air in cylinder, relative ratio $\Lambda_{\mathrm{excess}}$ is:

$$\Lambda_{\mathrm{excess}} = \frac{(m_{\mathrm{a}} / m_{\mathrm{f}})_{\mathrm{actual}}}{(m_{\mathrm{a}} / m_{\mathrm{f}})_{\mathrm{stioch}}} = \frac{m_{\mathrm{ar,t}}}{m_{\mathrm{a.h.\ stoich}}} = \frac{m_{\mathrm{a.h.\ stoich}} + m_{\mathrm{a,excess}}}{m_{\mathrm{a.h.\ stoich}}} \tag{35}$$

$\Lambda_{\mathrm{excess}} = 1/\Phi$ where Φ is Equivalence Ratio.

(i) Air Consumption Factor/Ratio Λ_{AC} :

$$\Lambda_{\mathrm{AC}} = \frac{\text{mass of air delivered/cycle}}{\text{mass of air necessary for stoichometric fuel combustion/cycle}} = \frac{m_{\mathrm{sc}}}{m_{\mathrm{a\ stoich}}} \tag{22b}$$

$$\Lambda_{\mathrm{AC}} = \frac{m_{\mathrm{sc}}}{m_{\mathrm{a\ stoich}}} = \frac{m_{\mathrm{sc}} \cdot m_{\mathrm{ar,t}}}{m_{\mathrm{ar,t}} \cdot m_{\mathrm{a\ stoich}}} = \frac{\Lambda_{\mathrm{excess}}}{\eta_{\mathrm{tr}}} \tag{36}$$

In the steady state flow, $\Lambda_{\mathrm{AC,dry}} = \dot{m}_{\mathrm{sc,d}} / \dot{m}_{\mathrm{a\ stoich}}$ or $\Lambda_{\mathrm{AC,h}} = \dot{m}_{\mathrm{sc,h}} / \dot{m}_{\mathrm{a\ stoich}}$

3. Results and discussion

Table 3. Fuel and cylinder oil consumption and mass composition at 100% *SMCR*, ISO conditions

Substance	$\dot{m}$ kg/s	b_e kg/kWh	g_c %	g_h %	g_s %	g_{CA} %
Fuel	0.8858317	0.1709	85.76	13.82	0.42	-
Cyl. oil	0.00311	0.0006	83.60	13.40	0.50	2.5
Cyl. Oil HC	0.0030167	0.000582	86.18	13.82	-	-

Table 4. Air consumption and mass composition at 100% *SMCR*, ISO conditions

Substance	$\dot{m}$ kg/s	*smf* kg/kWh	gN_2 %	gO_2 %	gAr %	gH_2O %
Humid air	47.1111	9.08896	75.1187	23.02169	1.2615	0.597175
Dry air	46.829775	9.03468332	75.57	23.16	1.2691	-
H_2O in air	0.281336	0.054277	-	-	-	-

% unburnt fuel = 0.9625 %

$$\%\ \text{unburnt fuel and cyl oil} = \frac{smf_{\mathrm{HC\ t\ unburnt}}}{b_{\mathrm{e}} + b_{\mathrm{c.o.}}} \cdot 100 \tag{17}$$

% unburnt fuel and cylinder oil = 1.2986 %

Thus, the total amount of unburnt hydrocarbons from fuel and cylinder oil is 1.3% over the adjusted values.

From the point of view of balance of energy, the emissions do not play a significant role. Depending on the design of the engine, the amount of unburnt fuel can reach more than 3%. However, due to environmental pollution, today much attention is paid to reducing emissions and reducing the CO_2 content in the exhaust gases. As of 14.03.2011. MAN Diesel & Turbo data, modern 6SME-C diesel engines have emissions of only 0.3 g / kWh CO and 0.4 g / kWh HC. In the reporting engine, total unburnt hydrocarbons from fuel and cylinder oil in *SMCR* is 2.227 g / kWh (HC, CO, soot), of which the fuel is only 1.645 g / kWh.

Table 5. Fuel composition, stoichiometric oxygen consumption for fuel and cylinder oil combustion at 100 % *SMCR*, ISO standard conditions

	C	H_2	S	Σ
Proportion of components in the fuel, N kmol/kWh	0.012202468	0.011716047	0.0000223886	0.023940904
The oxygen consumption for the fuel combustion, N kmol O_2/kWh	0.012202468	0.005858024	0.0000223886	0.01808288
The oxygen consumption for the complete combustion of fuel and cyl. oil, N kmol O_2/kWh	0.012244227	0.005877973	0.0000223886	0.018144589
The oxygen consumption for 98.7% fuel and cyl. oil combustion, N kmol O_2/kWh	0.012078002	0.005820837		
Products after complete combustion, N kmol/kWh	0.012244227	0.011755946	0.0000223886	0.024022562
Products after 98.7% combustion, N kmol/kWh	0.012078002	0.011641673		

Minimum amount of air required for combustion of fuel:

$$\dot{m}_{\mathrm{a,stoich}} = \dot{m}_{\mathrm{f}} \cdot AFR_{\mathrm{stoich}} = 12.94965028 \text{ kg air/s} \quad \text{at } 100\%\ SMCR$$

Table 6. Stoichiometric Air Consumption and Composition.

Substance		O_2	N_2	Ar	Σ
Dry Air stoich	kg/kWh	0.5786341	1.88798344	0.0317073	2.49832481
g	%	23.16	75.57	1.2691	100
r	%	20.96	78.12	0.92	100

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

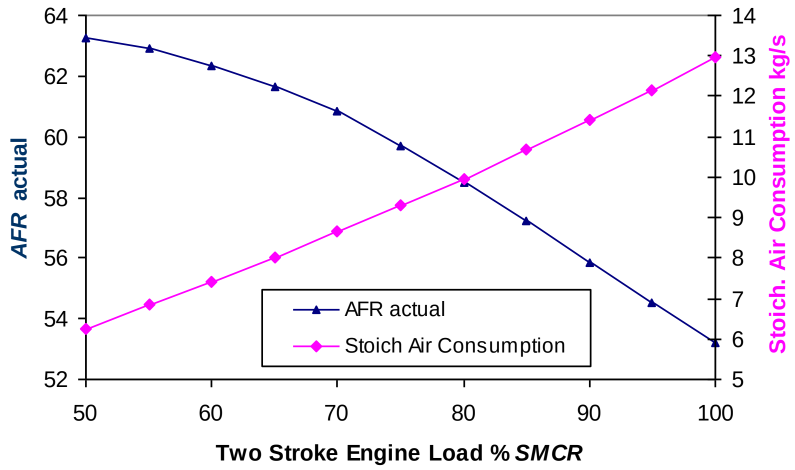


Fig. 6. AFR_{actual} and Stoichiometric Air Consumption in function of main engine load

Figure 6 shows AFR_{actual} and Stoichiometric Air Consumption change with main engine loads. Actual (total) *AFR* is very close to the results presented in the work of Guan et al [29] for the main engine load from 50 to 100% *SMCR*.

In this work, was defined and introduced the new term, Air Consumption Ratio/factor. This factor can be applied for two-stroke engines, especially for two-stroke fuel injection spark ignition engines and for two-stroke Diesel engines. Since the engine runs humid air, $\Lambda_{\mathrm{AC,humid}}$ is the definition which should be used, although $\Lambda_{\mathrm{AC,dry}}$ is a more useful definition. Figure 7 shows changes of Air Consumption Ratio in function of main engine load for humid air and calculated dry air. At Service Power which is usually 80% of *SMCR*, air consumption is four times bigger than it is necessary for stoichiometric combustion.

$$\Lambda_{\mathrm{AC,dry}} = \frac{\dot{m}_{\mathrm{a.d.\ actual}}}{\dot{m}_{\mathrm{a\ stoich}}} = 3.616296516$$

$$\Lambda_{\mathrm{AC,humid}} = \frac{\dot{m}_{\mathrm{a.h.\ actual}}}{\dot{m}_{\mathrm{a\ stoich}}} = 3.638021884$$

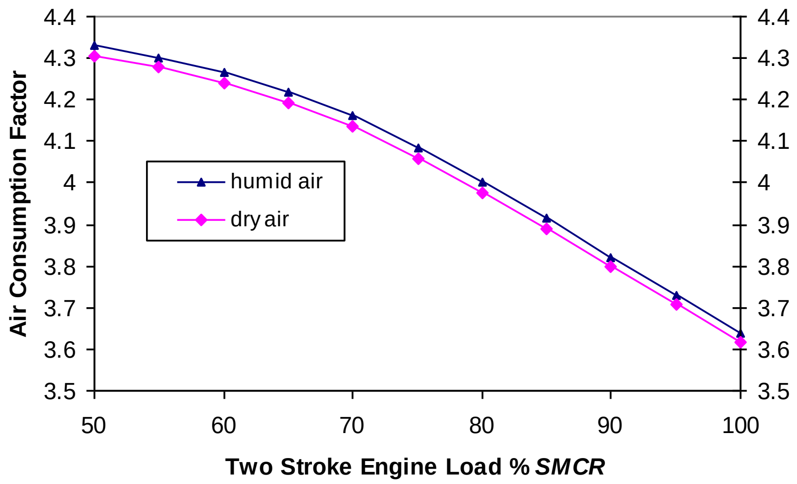


Fig. 7. Air Consumption Ratio in function of main engine load

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 26 May 2017 doi:10.20944/preprints201705.0192.v1

Peer-reviewed version available at Energies 2017, 10, , 805; doi:10.3390/en10060805

Table 7. The share of the components involved in the combustion

	C unburnt	H unburnt	C in CO_2	H in H_2O	S in SO_2	O_2 burnt
Fraction in exh. gas kmol/kWh	0.000 166	0.000 114	0.012 080	0.011 640	0.000 022	0.017 920

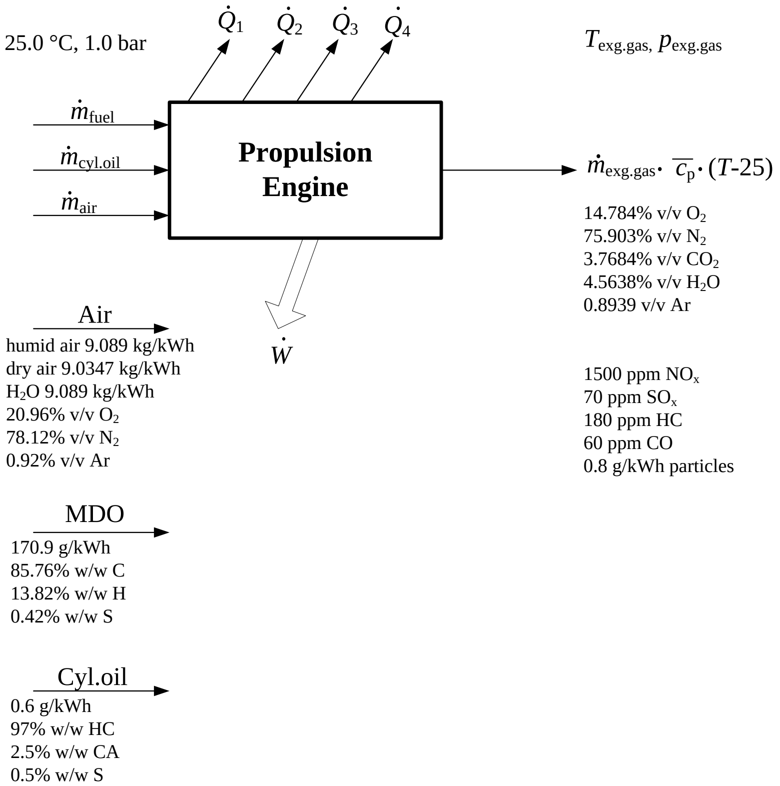
Table 8. The exhaust gas composition at 100% *SMCR*, ISO conditions

	O_2	CO_2 (+ SO_2)	H_2O	Ar	N_2	H_6C_{14}
r	0.147839424	0.037683715	0.045638029	0.008938867	0.759026424	0.000873541
$r \cdot M$	4.730713744	1.658460303	0.822169089	0.357089869	21.26260721	0.07527737

$$M_{\text{exh.gas}} = \Sigma r_{\text{i}} \cdot M_{\text{i}} = 28.90631758 \text{ kg/kmol} \quad (37)$$

$$R_{\text{exh.gas}} = \frac{\Re}{M_{\text{exh.gas}}} = 287.6305492 \text{ J/kg K} \quad (38)$$

In figure 8, cylinder oil and emissions data are from MAN source [27]. Inlet amount of air is from MAN data, but composition is calculated using NIST program. MDO data are data we got from Diesel factory – Shipyard SPLIT. Exhaust gas composition is calculated by our 0D program in combination with NIST program. Although emission data is orientated, it can be assumed that the figures relating to the emissions of NOx, CO, HC and particles when using HFO are very close to the actual values of the MDO fuel, for which our calculation was made. This assumes that all the cylinder lubricating oil enters and burns in the combustion chamber, and the small percentage that remains scraped by the piston rings and remains below the piston is negligible.



Peer-reviewed version available at *Energies* **2017**, *10*, , 805; doi:10.3390/en10060805

Fig. 8. The calculated substance flow through the two-stroke low speed engine at 100 % *SMCR.*

As the size of the lower order NOx and SOx in the exhaust gases will be ignored, while unburnt HC and soot (C) in the calculation is taken over hexane whose calorific value approximately corresponds to the calorific value of MDO. In order to simplify the exhaust gas calculation SO_2 is included in CO_2. Results of our calculation for volume/molar composition are shown in figure 9 for 100% *SMCR* engine load. Table 9 shows the exhaust gas composition over 50 to 100% of main engine load.

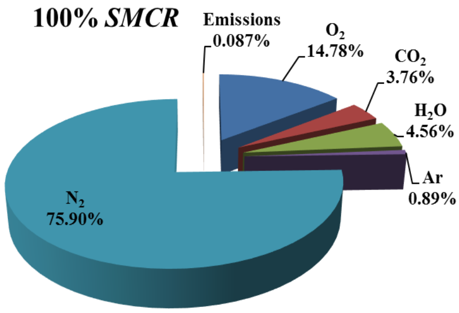


Fig. 9. Exhaust gas composition of 6S70MC MAN B&W low speed two stroke engine

Table 9. Exhaust gas composition at various main engine loads.

% *SMCR*	50	60	70	80	90	100
r_{O2} %	15.72927012	15.65486	15.52668	15.31897	15.06865	14.78394245
r_{CO2} %	3.170466448	3.217451	3.298467	3.42984	3.588208	3.768371514
r_{H2O} %	3.991802162	4.036903	4.11451	4.240189	4.391613	4.563802879
r_{Ar} %	0.896522031	0.896313	0.895954	0.895375	0.894678	0.893886725
r_{N2} %	76.12641422	76.10865	76.0782	76.02904	75.96987	75.90264236
$r_{emissions}$ %	0.085525019	0.085829	0.086186	0.086585	0.086981	0.08735407

Table 9 shows that there is no drastic change in the composition of the exhaust gas in the main engine load ranging from 50 to 100 *SMCR.* The biggest change is in the composition of CO_2 and H_2O as expected. The oxygen proportion in the exhaust gas is high and ranges from 15.73% (50% *SMCR*) to 14.78% (100% *SMCR*). The main reason is a short circuiting of air during the gas exchange process. The high consumption of air helps to clean exhaust gases in the cylinders, but also lowers the temperature of the exhaust gases whose energy could be used in the WHR system. It also requires a higher capacity turbocharger. At the service power (80% *SMCR*) the engine consumes four times more air than is required for stoichiometric fuel combustion.

Thanks to Dr Fredrik H. Andersen from MAN we got results of Performance measurements, exhaust measurements and fuel specification and consumption conducted by MDT for an undisclosed 6S70MC-C7 engine. The data we got from Dr Andersen are for validating the model, not for publication since the data are confidential. Even though used MDO is not the same as fuel used in our calculation, and tested engine probably has different piston geometry, atomizer and engine lay-out, there are very good correlation between our results and performed factory test measurement.

4. Conclusions

This work developed a 0D model of combustion in a Diesel engine. It was applied on a 6S70MC-C7 low speed engine and by using NIST program obtained thermodynamic parameters of air and fuel at the inlet of the engine and parameters of the exhaust gases. Our results are in very good correlation with MAN Performance measurement and Exhaust measurement conducted on same engine type.

The obtained data show that the composition of exhaust gases doesn't change significantly in the range of 50 to 100 % *SMCR*. Also, it is observed that the quantity of oxygen in exhaust gases in the complete observed range is high at value ca. 15% v/v. Dividing the total quantity of air which enters the engine with the air quantity required for the stoichiometric combustion, the Air Consumption ratio/factor is obtained and shown in figure 7, which has not been previously used in scientific literature. In the main engine service rating, this factor value is at about four, which means that at this engine load it consumes four times more air than is required by stoichiometric combustion. The reason for this is the great quantity of air flows directly from air distributor to exhaust during the exchange of working fluid. This achieves better blowout of cylinders and less residual exhaust gas in the cylinders but decreases the exhaust gas temperature, therefore it lower efficiency of the WHR process. Further, for such a high quantity of fresh air a turbo charger at higher capacity is required, which additionally decreases the energy of exhaust gases available for WHR process. Better swirling flow within the cylinder and bigger diameter of exhaust valve could probably decrease the Air Consumption Ratio.

New term Λ_{AC} is defined as $\Lambda_{\mathrm{AC,\ humid}}$ and $\Lambda_{\mathrm{AC,\ dry}}$. Since, humid air passes through the engine, $\Lambda_{\mathrm{AC,\ humid}}$ should be used, although $\Lambda_{\mathrm{AC,\ dry}}$ would be more useful. Although displacement volume V_s is used as a reference volume in literature, we used V_{max} for two stroke engines what is much more appropriate.

Analysis of scavenge parameters of two stroke engines shows: $\Lambda_{\mathrm{AC}} = \Lambda_{\mathrm{excess}} / \eta_{\mathrm{tr}}$. This shows that Air Consumption Ratio depends on Relative Ratio and Trapping Efficiency. It is necessary to keep high Relative ratio about 2 what means controlling/lower peak temperature which is main parameter governing NOx formation. Better Trapping Efficiency can be achieved by decreasing mass of air directly blowout cylinder. By measuring and calculation Air Consumption Ratio and Relative Ratio, Trapping efficiency can be calculated.

Figure 5 shows the flow of working substances in a cylinder and from the cylinder according to the lit. [10] and [11]. Figure 5 is modified and significantly improved version of figure shown in the literature [11] particularly in relation to literature [8, 9 and 10]. Two-stroke gas flow performance parameters can be easy defined using this improved figure 5.

Acknowledgments

The authors would like to thanks Dr Fredrik H. Andersen, Development Engineer of MAN Diesel & Turbo, Performance and Emission (LDF2), Marine Low Speed Research & Development, Copenhagen SV - Denmark for his valuable and helpful technical information.

This work was supported by the Ministry of Science, Education and Sports of the Republic of Croatia.

Reference

1. List, H., Thermodynamik der Verbrennungskraftmaschine, Serija List –Verbrennungskraft maschine, Band 2, Springer Verlag, Wien, 1939.
2. Vibe, I. I., Novoe o raboĉem cikle dvigatelei, Mašgiz, Moskva-Sverdlovsk,1962.
3. List, H., Der Ladungswechsel der Verbrennungskraftmaschine, Der Zweitakt, serija: List-Verbrennungskraftmaschine, Band 4, Teil 2, Springer Verlag, Wien, 1950.
4. Hasselgruber, H., Verallgemeinerte Berechnung des stationären Ladungswechsel von Verbrennungskraftmaschinen, Forschung Ing. Wes. Nr. 5 1961.
5. McAulay, K. J. et al., Development and Evaluation of the Simulation of the Compression - Ignition Engine. - SAE Paper 650451, 1965.
6. Woschni, G., Elektronische Berechnung von Verbrennungsmotor - Kreisprozessen, MTZ 26, 1965.
7. Boy, P., Beitrag zur Berechnung des instationären Betriebsverhaltens von mittelschnellaufenden Schiffsdieselmotoren: Dissertation, Hannover: TH, 1980.
8. Heywood, J. B., Internal Combustion Engine Fundamentals, McGraw-Hill Book Company, USA 1988.
9. Stone, R., Introduction to Internal Combustion Engines, Palgrave, NY, Third edition 1999.
10. Sher, E.: Scavenging the two-stroke engine, Prog. Energy Combust. Sci, **1990**, Vol 16, pp. 95-124, Pergamon Press plc
11. Van Basshuysen, R., Schäfer, F., Handbuch Verbennungsmotor, Grundlagen, Komponenten, Systeme, Perspektiven, 5. Auflage, MTZ, IAW GmbH
12. Sher, C., Hossian, I., Zhang, Q., Winterbone, D. E., Calculation and Measurement in the Cylinder of a Two-Stroke Uniflow-Scavenged Engine Under Steady Flow Conditions, Experimental Thermal and Fluid Science, **1991**, 4, 418-431
13. Ravi, M. And Marathe, A., Effect of Port Sizes and Timings on the Scavenging Characteristics of a Uniflow Scavenged Engine, SAE Technical Paper 920782, **1992**, doi:10.4271/920782.
14. Carlucci, A. P., Ficarella, A., Laforgia, D., Longo, M., An easy and inexpensive way to estimate the trapping efficiency of a two-stroke engine, ATI 2015 – 70th Conference of the ATI Engineering Association, Energy Procedia 82 (2015) 17-22.
15. Larsen, U., Pierobona, L., Baldib, F., Haglinda, F., Ivarsson, A., Development of a Model for the Prediction of the Fuel Consumption and Nitrogen Oxides Emission Trade-Off for Large Ships, Energy, 80,**2015**, 545–555 doi:10.1016/j.energy.2014.12.009.
16. Andreadis, P., Chryssakis, C., Kaiktsis, L., Optimization of Injection Characteristics in a Large Marine Diesel Engine Using Evolutionary Algorithms, SAE Technical Paper 2009-01-1448
17. Guan, C., Theotokatos, G., Chen, H., Analysis of Two Stroke Marine Diesel Engine Operation Including Turbocharger Cut-Out by Using a Zero-Dimensional Model, Energies 2015, 8, 5738-5764.
18. Varbanets, R.A., Karianskiy, S.A., Analyse of Marine Diesel Engine Performance, Journal of Polish CIMAC 2012.

Peer-reviewed version available at *Energies* **2017**, *10*, , 805; doi:10.3390/en10060805

19. Grljušić, M., Medica, V., Račić, N., Thermodynamic Analysis of a Ship Power Plant Operating with Waste Heat Recovery through Combined Heat and Power Production", Energies, 2014, 7, 7368-7394
20. Grljušić, M., Medica, V., Radica, G., Calculation of Efficiencies of a Ship Power Plant Operating with Waste Heat Recovery through Combined Heat and Power Production, Energies, 2015, 8, 4273-4299.
21. Spahni, M., Schumacher, B., Karamitsos, A., Concepts, Testing and First Service Experience with the Generation X-Engines, WinGD Low-speed Engines Licensees Conference 2015.
22. Carlucci, A. P., Ficarella, A., Laforgia, D., Renna, A., Supercharging system behavior for high altitude operation of an aircraft 2-stroke Diesel engine, Energy Conv. And Manag. 101, **2015**. 470-480, dx.doi.org/10.1016/
23. Carlucci, A. P., Ficarella, Trullo,GL., Performance optimization of a Two-Stroke supercharged diesel engine for aircraft propulsion, Energy Conv. And Manag. 122, **2016**. 279-289, dx.doi.org/10.1016/
24. Wang, X., Ma, J., and Zhai, H., Evaluations of Scavenge Port Designs for a Boosted Uniflow Scavenged Direct Injection Gasoline (BUSDIG) Engine by 3D CFD Simulations, SAE Technical Paper 2016-01-1049, **2016**, doi: 10.4271/2016-01-1049.
25. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.0; National Institute of Standards and Technology: Gaithersburg, MD, USA, **2010**.
26. Grljušić, M. Thermodynamic Analysis and Optimisation of Marine Low Speed Propulsion Diesel Engine Operating with Waste Heat Recovery. Ph.D. Thesis, University of Rijeka, Rijeka, Croatia, 10 September 2012.
27. MAN Diesel and Turbo, Marine/Low Speed/CEAS-ERD Engine Room Dimensioning, Main Engine Room Data, 6S70MC-C7.1-TI. Available online: http://apps.mandieselturbo.com/ceas/client/erd.aspx (accessed on 9 November **2014**).
28. Aabo, K., Large Marine Diesels in Challenging Times, CIMAC Circle, Marimtec 2009.
29. Guan, C.; Theotokatos, G.; Zhou, P., Chen, H. Computational investigation of a large containership propulsion engine operating at slow steaming conditions. *Appl. Energy* **2014,** *130,* 370-383.