A Marine Engineering Plant Model to Evaluate the Efficiency and CO_2 Emissions from Crude Oil Carriers

Joel R. Pérez Osses [a*], Richard W.G. Bucknall [b†]

[a] *University College London, London WC1E 7JE, United Kingdom – Universidad Austral de Chile, Valdivia 5110566, Chile*

[b] *University College London, London WC1E 7JE, United Kingdom*

Abstract

Wet cargo is dominated by crude oil carriers and product carriers which account for around 25% of CO_2 emissions from the shipping sector. Crude carriers predominantly transport their cargos over long distances at constant speed. The CO_2 emissions from these ships are dominated by those from the two-stroke diesel main engines that burn heavy fuel oil (HFO) or intermediate fuel oil (IFO) since these engines also provide waste heat recovery from their exhaust gases that is used for heating (cargo and accommodation) and for generation of electricity for service, cargo and ancillary engine requirements.

This paper focuses on the transportation of crude oil in ocean going ships and examines their CO_2 emissions through the development of a propulsion system model. A model of a two stroke engine has been developed to include combustion, thermodynamic, mechanical and ancillary processes embedded into a marine engineering plant model that includes cargo and accommodation system models. The model is validated using data provided in the public domain as well as actual performance data.

The model is used to examine the relationship between engine efficiency and plant efficiency across different power outputs (ship speed and loading conditions) thereby examining how efficiency and CO_2 emissions change across a typical voyage. Furthermore modifications to the marine engineering plant are also considered, including the use of shaft generators, improved waste heat recovery systems, improved insulation of the cargo tanks, handling and fuel cleaning system and reducing demand from accommodation and general ship services.

The paper concludes by making recommendations into the improvements of the main and auxiliary systems are necessary to reduce CO_2 emissions.

Keywords: Crude oil and product carriers, CO_2 emissions, marine engineering plant model, main and auxiliary systems.

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[*]Corresponding author Tel: +44(0)7793471885

Email address: joel.osses.10@ucl.ac.uk

Joel R. Perez Osses is currently a PhD Researcher in the Marine Engineering Research Group in the Department of Mechanical Engineering at University College London. Prior to this, Joel spent four years in the Merchant Navy, working as a Third Engineering Officer on-board tankers and bulk carriers. Following this Joel lectured at the Universidad Austral de Chile, before moving to UCL, where he obtained an MSc in Marine Engineering. Joel's current research interests are low carbon shipping, focused on propulsion and auxiliary systems.

[†]Richard W. G. Bucknall is Professor of Marine & Power Systems in the Department of Mechanical Engineering at University College London. His research interests are in the fields of electrical systems used in the marine environment and marine propulsion systems and he currently supervises eight PhD students and chairs the Marine Research Group. He is also the Programme Director for the MSc in Marine Engineering and MSc in Power Systems Engineering programmes. Externally he is the Co-Editor of the Journal for Marine Engineering and Technology.

1 Very Large Cargo Carriers (VLCC) revision

1.1 A present view of VLCCs efficiency

Over time, VLCCs have proven to be among the most efficient ships in the whole merchant fleet, yet considerable improvements are considered possible and technologies could be adapted for these vessels to make them even more efficient (IMO, 2009c).

Nowadays the efficiency of these vessels is calculated using the IMO Standards such as the Energy Efficiency Design Index (EEDI), which is mandatory for new vessels, and the Energy Efficiency Operational Indicator (EEOI) which is yet to be made mandatory but is part of a Ship Energy Efficiency Management Plan (SEEMP) which is mandatory for new and existing vessels (IMO, 2009a, IMO, 2009b, IMO, 2011, Bazari and Longva, 2011, IMO, 2013).

Over the past decade VLCC construction has not showed much variation yet the use of Low Speed Diesel Engines (LSDE) as the main propulsion system has been addressed by reducing the specific fuel oil consumption (sfoc) through improving the engine match between the vessel operational speed and shaft speed, a procedure known as de-rating. Significant improvements in the use of shaft generators and more efficient Waste Heat Recovery Systems (WHRS) are also being addressed as the most significant technology options to help reduce the CO_2 footprint from VLCCs.

1.2 A present view of the VLCC fleet

The present fleet of VLCCs account for 766 vessels and their distribution is presented in Figure 1 where it is possible to appreciate the recent increase in new orders. This is a market response to the increase in rising trade on long-haul routes as opposed to the Suez Canal route, which has seen increasing transit prices every year (Clarksons, 2013, Shipping_Intelligence_Network, 2013, Wijnolst and Wergeland, 2009).

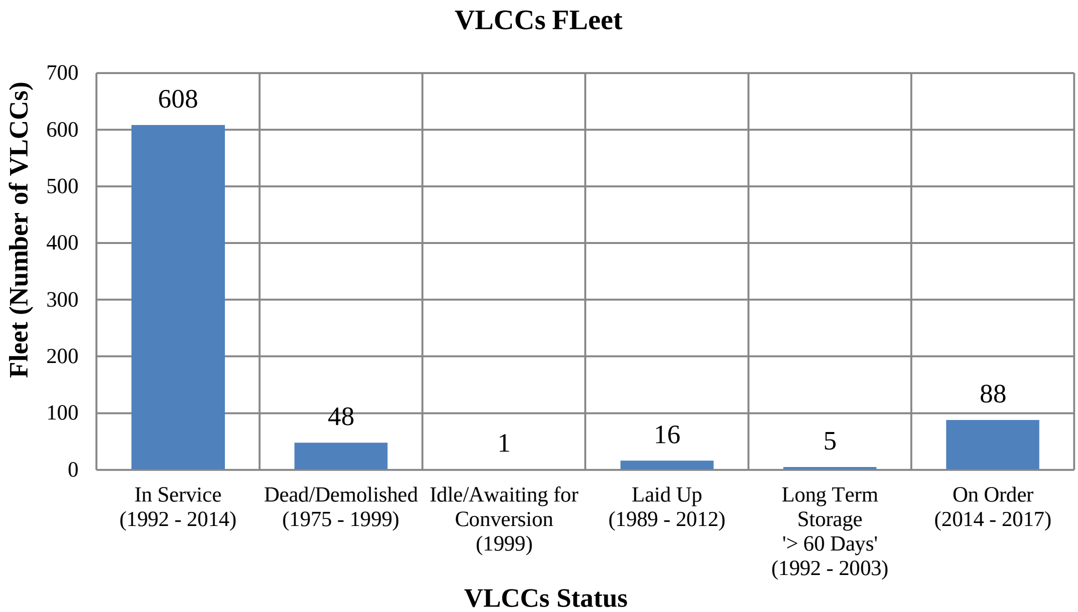

Figure 1 VLCCs Fleet Status

From Figure 1, VLCCs ‘In Service’ are vessels constructed between 1992 and 2014. All use two stroke LSDEs burning HFO and IFO with Fixed Pitch Propellers (FPP) as the main propulsion system, navigating at speeds between 12 knots and 21.5 knots.

VLCCs ‘Dead/Demolished’ were constructed between 1975 and 1999. All of these vessels being scrapped between May 2010 and April 2014. The two stroke LSDEs in these VLCCs had a maximum cylinder pressure set up to 16 bar compared with typically 21 bar today which is an important consideration when considering engine efficiency.

VLCCs ‘Laid up’ and ‘Long term storage (up to 60 days)’ were constructed between 1989 and 2012 and their service has been considered detrimental to the surrounding environment because of the need for continuous operation of the auxiliary systems burning fuel to keep the vessels operational and ready to navigate.

VLCCs ‘On Order’ have been contracted to be built between May 2014 and 2017. Of this group, 22 VLCCs are scheduled to be delivered by the end of 2014 and some are being fitted with the new engine from MAN, the G-Series (MAN, 2013b, MAN, 2013a), having maximum cylinder pressures of up to 20 bar. This engine has been de-rated and operates at low speed of up to 68 rpm. This new engine allows the use of larger propellers, which are up to 10 m in diameter.

New engines and larger diameter propellers considerably increase the efficiency of the main propulsion plant, and the ship performance overall, decreasing sfoc and hence emissions (MAN, 2013a). For the ‘In Service’ fleet 5 VLLCs are already fitted with this new engine (Clarksons, 2013).

In 2015, 15 new VLCCs are expected to be delivered. From this group it has been found that some vessels are going to be fitted with the slightly newer engine from Wärtsilä, the X-Series, which is an engine working up to 21 bar as maximum cylinder pressure, has been de-rated and operates at low speeds up to 84 rpm (Wärtsilä, 2013)

In 2016, 45 new VLCCs are expected to be delivered and 15 are already contracted to be fitted with the G-Series from MAN Engines (MAN, 2013b) and 4 with the X-Series from Wärtsilä Engines (Wärtsilä, 2013). A further 6 VLCCs are expected to be delivered in 2017 (Clarksons, 2013).

2 Marine engineering plant in VLCCs

2.1 Marine Engineering Plant Requirements:

The purpose of the marine engineering plant (MEP) is to provide the vessel with propulsion and auxiliary power. Propulsion power (P_E) from equation 1 is the mechanical shaft power to turn the propeller which is determined by the ship’s speed (v_S) and its ship resistances (R) which are a function of ship’s speed, loading condition, environmental conditions, hull form fouling, etc.

$$P_E = R * v_S \quad \text{Equation 1}$$

Auxiliary power requirement is governed by ship activity state which may be classed as in port (loading, unloading), at anchor, at sea (full speed, slow steaming, etc.), manoeuvring, etc. The auxiliary power requirement is normally electrical power to drive pumps, fans, etc. and heat to drive turbines, heat cargo and accommodation. In modern ships other forms of power e.g. hydraulics and pneumatics are driven primarily by electrical motors.

The MEP therefore provides mechanical power, electrical power and heat which will vary for each activity state and is influenced by external factors.

2.2 Propulsion and auxiliary plant

The greatest power demand is propulsion in the region of 24 MW. The two stroke LSDE as the prime mover is used for propulsion because it offers a number of advantages over other prime movers such as four-stroke diesel engines and gas turbines. Whilst two-stroke engines as can be seen in Figure 2 are large and heavy they offer good power to weight ratio and good efficiency, usually in excess of 50% due mainly to low speed typically between 58 rpm and 120 rpm. Cheaper low cost fuels, mostly residuals like IFO and HFO, are typically used. The engine drives the propeller directly to reduce transmission losses and is reversible.

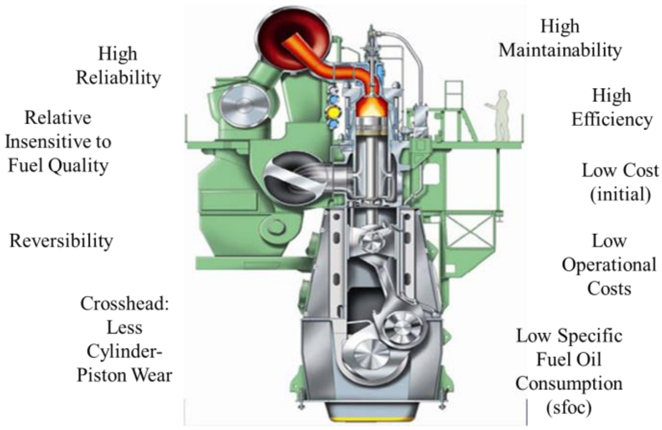

Figure 2 Sectional view and advantages of two stroke LSDE (MAN, 2013b)

The auxiliary plant has to provide electricity and heat when the main engine is operating at varying loads (underway) and when it is stopped (port/anchor) to meet a variety of demands that may be classified as engine room, hotel and cargo.

A common propulsion system model and auxiliary systems of a typical VLCC MEP is shown in Figure 3 from which it is possible to appreciate the main components and their interactive distribution.

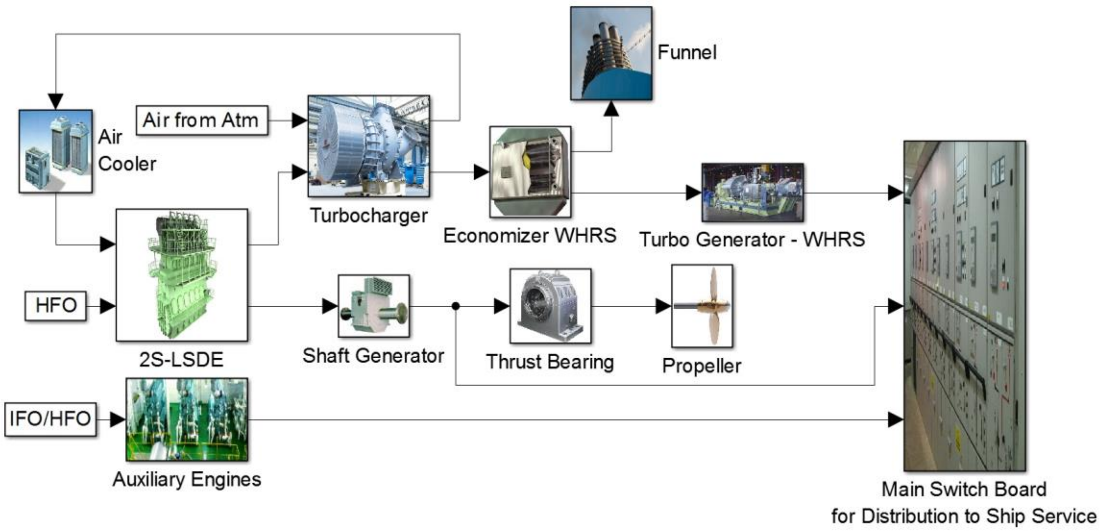

Figure 3 Common MEP configuration of existing VLCCs

2.3 Vessel Operations

2.3.1 Underway

When underway the main engine is operating and it can be exploited for generating electricity and heating for auxiliary needs.

Electricity can be generated in a number of different ways: In a WHRS system, a turbo-generator can be driven by the steam generated in an economizer located in the exhaust stream of the main engine (Cowley, 1992). Using a turbine alternator driven by the exhaust gases from the main engine (Woud, 2002). By connecting a shaft generator to the main engine shaft (power take off - PTO) and using renewable sources such as photovoltaic panels (minimal contribution).

Under certain circumstances excess electricity may be generated, in which case electricity can be inputted onto the main engine shaft through a shaft mounted motor (power take in – PTI). Combined PTO and PTI can also be facilitated.

The heating demand includes fresh water generation, accommodation heating (including galley) and, if on a loaded passage, cargo heating.

2.3.2 Manoeuvring

When manoeuvring the engine speed is uncertain and the quantity of waste heat from the main engine is uncertain and cannot be relied upon. For electricity the main source is the use of auxiliary engines. For heating the main sources are steam from boilers.

2.3.3 In Port and at anchor:

When in port or at anchor the main engine is not operating and waste heat from the main engine is not available. Heat and electricity therefore need to be supplied from alternative sources such as boilers and auxiliary engines.

3 Modelling approach

Having explained the details regarding the MEP requirements, the main and auxiliary plant systems and the different vessel operations it is possible to describe the modelling approach.

The approach is to develop a model to analyse and to improve the understanding of the performance of the MEP of VLCCs. Analysis of the VLCC MEP includes three major aspects, firstly to analyse the performance and efficiency of the plant under different load conditions, secondly performing a heat balance analysis and thirdly identify new waste heat recovery sources to improve the performance of the MEP.

The efficiency of the plant is considered based in the IMO Regulations namely EEDI because using this introduces a means of evaluation of the CO_2 emissions of the MEP.

The approach has been to consider performance under steady-state conditions and not transient conditions for analysis at this time. Performance under steady-state condition as a first approach can provide clues and paths to perform deeper analysis of the MEP, allowing more detailed modelling of the complexity of the machinery systems into a more reasonable and comprehensive form that can later be treated under transient conditions.

The modelling approach segregates the complexity of the machinery on board VLCCs into four main groups as can be seen in Figure 4; Supply, which can be any source of energy on board i.e. fuel. Conversion, which can be any equipment designed to release the energy from the fuel source i.e. engines and boilers. Residual, which is the process of energy transfer between supply and demand. Demand, which is the energy requirement of the vessel i.e. propulsion, electricity and heat and the components which absorb the energy. There are also feedback paths between each of these blocks.

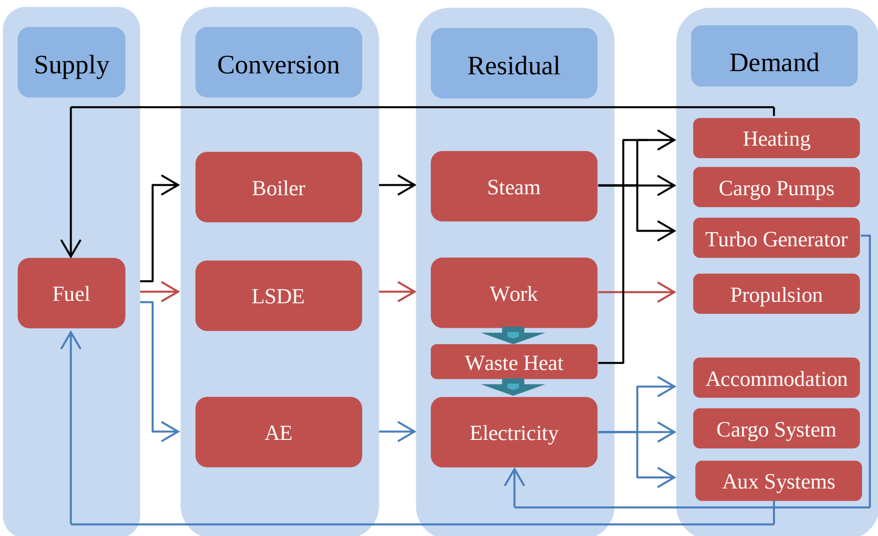

Figure 4 MEP’s algorithm of operational condition and conversion systems

The connection between the main groups presented in Figure 4 describes the aforementioned VLCC's operation and includes the main propulsion and auxiliary equipment.

3.1 Modelling approach of the MEP

Discretizing the complexity of the machinery is feasible. Considering fuel as the main source of energy on board, the modelling approach presented and described in this study leads to a dependency of the main conversion units on the main supply. Therefore the MEP energy model is a closed loop that considers the heating demand as a requirement, and need to have a suitable supply to perform a correct analysis of the MEP.

The processes are described in the modelling approach presented in Figure 5, where the main relation between supply and conversion groups is presented, and the residuals from them are connected to two of the main demands of the MEP, propulsion and electricity.

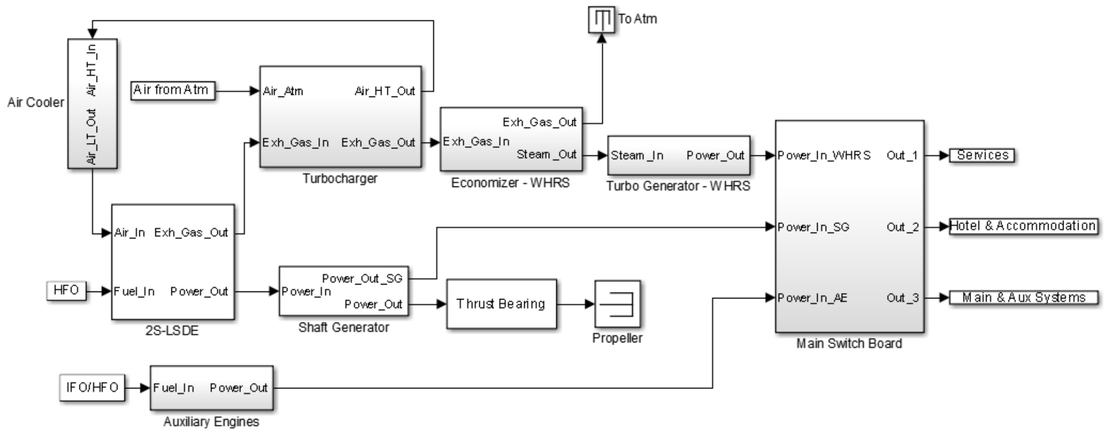

Figure 5 First approach of the MEP energy model developed using MATLAB/Simulink

Each block presented in Figure 5 is a discretization of the main supplies and mathematical formulation has been used to develop the MEP energy model. The two stroke LSDE block is the key block which contains subsystems including the combustion and thermodynamic models.

Every other block is considered as auxiliaries and their development and configuration follows the same pattern as the two stroke LSDE block, i.e. using a set of subsystems.

4 Model development

Having an appropriate discretization and a modelling approach of the MEP of VLCCs it is possible to describe the MEP energy model development. This includes the process of controlling the performance of the engine and the operational characteristics to evaluate the MEP performance.

4.1 Propulsion system model

A database of VLCCs, which includes information of main characteristics and propulsive systems, has been used to obtain values regarding power required for propulsion. Values obtained for a set of VLCCs, as presented in section 1, have been evaluated and an average value of 24 MW has been considered as the normal continuous rating (NCR) power output which is 75% of the maximum continuous rating required to propel a VLCC fitted with a FPP (for the purposes of this paper). Main characteristics of a VLCC are presented in Table 1 as reference.

Length (m)	Length Pp (m)	Draught (m)	Beam (m)	Ship Size (dwt)	GT (Tons)	Built	Speed (kn)	Cargo Capacity (m^3)	Fuel	Output Power (MW)	EEOI	EEDI Attained	EEDI Required
330	316	21.5	60	297,412	156,975	2012	16	324,660	HFO	24	2.67	2.24	2.60

Table 1 VLCC main characteristics

4.2 Two stroke engine model

This model is a representation of the power speed curve developed by the diesel engine to understand its performance at different loads and its overall performance and efficiency within the context of the MEP.

The two stroke engine model includes combustion, thermodynamic and heat balance models covering the most important aspects of its operation and capabilities to perform under different loads.

The main equations and boundary conditions ruling this model are those required to represent the working principle and the geometry of the engine and those necessary to represent the energy conversion approach to develop the remainder of the models.

Equations included for the modelling are those to determine values for: displaced volume, clearance volume, compression ratio, crank radius to stroke ratio, bore to stroke ratio, ratio between trapped cylinder volume at any crank angle and the clearance volume, mean piston speed, instantaneous piston velocity, engine brake power, brake mean effective pressure, indicated mean effective pressure, torque, crankshaft rotational speed, mechanical efficiency, trapped efficiency, fuel efficiency, air to fuel ratios, delivery ratio (being the most important) (Woodward, 1981, Woodyard, 2009, Woud, 2002).

This set of equations and boundary conditions have been set up in a script using MATLAB and in tables using Excel to be read and properly evaluated later in the model developed in Simulink.

4.2.1 Combustion model

The combustion model is a representation of the necessary details required to evaluate the performance of the engine. A simplistic approach considering the amounts of fuel and air has been used to get details regarding composition, mixing and exhaust gas generation (Heywood and Sher, 1999). The mixture composition of air, fuel and burned gas is important because they determine the development of the combustion process and exhaust gas generation, which is a valuable response of the model to be considered later in the development of more appropriate and efficient systems.

Chemical energy is converted into thermal energy by means of a combustion reaction of the fuel with air at ambient conditions which is boosted by a turbocharger. Equations ruling the combustion model are those presented in section *4.2* plus boundary conditions at different points of the operating principle of the diesel engine (Woud, 2002).

4.2.2 Thermodynamic model

The thermodynamic model is a representation of the Seiliger cycle, commonly known as dual fuel cycle. This model is used to describe the pressure volume indicator of the working principle of the two stroke diesel engine. This air standard cycle is a combination of the Diesel and Otto cycles (Woud, 2002, Turesson, 2009).

The equations ruling this modelling are those presented before in section *4.2* with particular presence of the geometric compression ratio, the effective compression ratio and adding variables to define the increase in the pressure and the isobaric expansion during combustion (Woud, 2002, Morsy El Gohary and Abdou, 2011).

As was mentioned before this modelling has been considered for steady-state rather than transient so the evaluation of the thermodynamic model obeys analytical rules based in the stages and ratios defined and followed for the working principle concerning the volumes, pressures and temperatures of the Seiliger process (Woud, 2002, Hui et al., 2013, Skogtjarn, 2002).

4.3 Mechanical and ancillary model

In this section three auxiliary services are considered for modelling and based in the fuel and fresh water as the supplies needed for them to operate. WHRS, fuel system and fresh water generator (FWG) are considered as the systems to be modelled because of their direct influence on the main engine performance at different loads of the MEP and are considered as main contributors of the heat balance analysis of the MEP.

All the models in this section are based in the heat rejection principle based in the differential working temperature (*ΔT*), mass flow rate ($\dot{m}$) and specific heat (*Cp*) of the working fluids, (see Equation 2).

$$Q = \dot{m} * Cp * \Delta T \qquad \text{Equation 2}$$

4.4 Heat balance model

The heat balance modelling is based on the heat losses from the engine and the gain from these losses by the auxiliary systems, such as WHRS and FWG. As in the previous mechanical and ancillary model, this heat balance model follows the heat rejection and gain principle stated and describe in Equation 2.

The engine thermal losses considered are; scavenging air losses, jacket cooling water losses, exhaust gas losses, frictional losses, lubrication oil losses and heat radiated.

4.5 MEP energy model

Based on the model development and description of the subsystems in the previous sections, the MEP energy model is developed and presented in Figure 6, which differs from the model presented in Figure 5, mainly in the modelling of the auxiliary systems and in the form of inclusion of the subsystems described in previous section. The blocks of this model are segregated to show most of the fluids and conditions which have been set to perform the analysis of the plant.

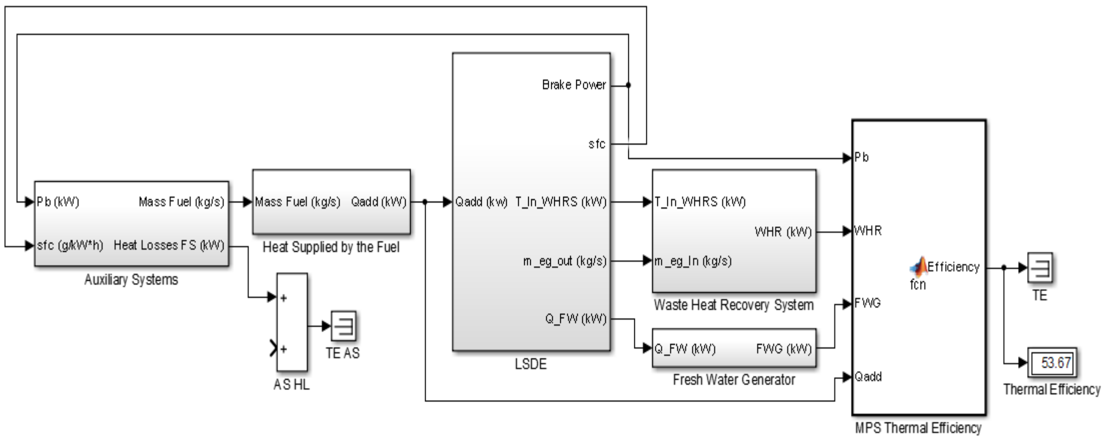

Figure 6 MEP energy model

The main block named LSDE includes the subsystems for combustion, thermodynamic and heat balance models, mentioned in the previous section as part of the evaluation of the engine which needs the heat added by the fuel to perform its evaluation. From this block a loop is created for a better visualization of the fuel system and its heat losses as part of the auxiliary systems.

From the LSDE block it is possible to appreciate the output signals to evaluate the performance of the WHRS and FWG which are considered as gains from the losses of the LSDE that affect the performance of the plant.

The thermal efficiency of the MEP is evaluated considering the LSDE only as a first approach and after the results from the auxiliary systems, WHRS and FWG are added to determine how much gain in efficiency these systems generate.

Taking the LSDE block from Figure 6, its discretization is presented in the form of a subsystem model in Figure 7 from which it is possible to appreciate the consideration of every major system influencing the performance of the engine. Some details are omitted for sake of space and diagramming but the overall performance of the engine is captured in this block where the inputs and outputs are based on the model developed to characterise the two stroke LSDE in the most accurate form.

The block BP stands for brake power developed by the engine and the modelling described in previous section is embedded here. The blocks friction losses (Q_FL), lubrication oil losses (Q_LO) and heat radiated losses (Q_HR) have been described as percentages losses from the heat added from the fuel (Qadd) as a simplistic representation of the models.

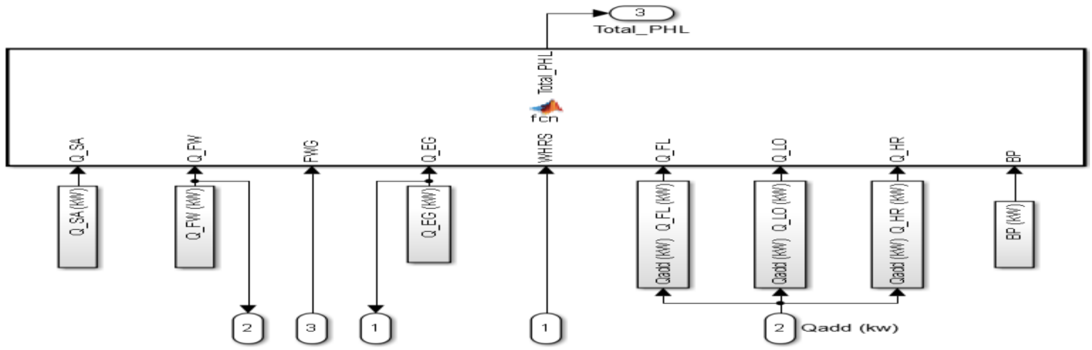

Figure 7 Heat balance of two stroke LSDE as part of MEP energy model

The heat losses through the scavenging air (Q_SA), FW cooling system (Q_FW) and exhaust gases (Q_EG) are dominated by equation 2 when considering values for the respective working fluids.

The total power heat losses (Total_PHL) is the value used for the MEP model presented in Figure 6 to evaluate the thermal efficiency of the propulsion system which is later tested in the overall performance of the plant.

5 Validation

Validation of the MEP energy model developed by the author is necessary and to do so data provided in the public domain and actual performance data from engine manufacturer MAN B&W (MAN, 2013b) has been used. Specific information regarding the new G-Series of engines has been obtained through the website to get the necessary information to perform a validation process of the performance of the MEP energy model.

The information from the manufacturer includes selection of the engine at specific operational condition and technical data, some of it is presented in Table 2, which also includes the number of new VLCCs that are been contracted to be fitted with this engine.

Information regarding the engine has been used from their technical digital library to validate results presented in this paper (B&W, 2014, B&W, 2012d, B&W, 2012a, B&W, 2012b). This information is open access and most of it can be easily put through validation using equation 2 presented in *4.3* and in some cases, such as in the jacket cooling water system, where tables are available with the corresponding values at the specific engine load.

Number of VLCCs	Engine Manufacturer	Engine Model	Maximum Power Output (kW)	Sfoc (g/kW*h)	Speed (rpm)
14	MAN B&W	7G80ME-C9.2	32,970	166	65 to 68

Table 2 7G80ME-C9.2 engine data (B&W, 2014, B&W, 2012b, B&W, 2012c)

The selection of this engine for validation of the MEP energy model has been considered because this engine represents the latest technology in two stroke LSDE and because it has been considered to fitting on board new VLCCs.

Results from the MEP energy model are contrasted against the evaluation of the MEP using data from the engine manufacturer as can be seen in Figure 8 from which it is possible to describe that the total heat losses from the engine manufacturer follows a linear increase as engine load increases compared to the MEP energy model which is non-linear at lower loads and exhibits maximum differential at higher loads.

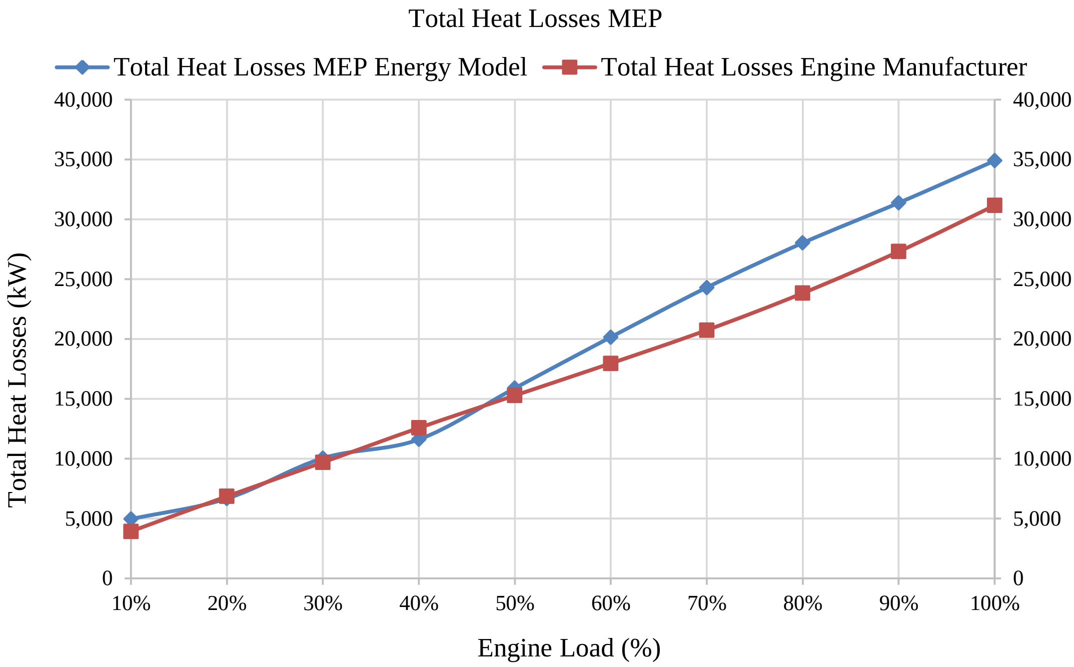

Figure 8 Total heat losses MEP

At higher loads the performance of the MEP energy model shows increased inaccuracy reaching a maximum of 17% higher heat losses at 80% engine load than engine manufacturer. At lower loads the MEP energy model shows accuracy having a maximum of -7% less heat losses difference. For high and low loads the differences are coming through the jacket cooling water system and the lubrication oil system.

6 Results

Results are based on the response of the model at different engine loads to simulate different vessel operation states. Three vessel operations were described in section *2.3* and in this section vessel underway and vessel at manoeuvring operations are going to be discussed. The actual state of development of the MEP energy model is not yet able to represent all the auxiliary systems mentioned in section 2.3 to cover all operations described.

The propulsion system of a VLCC is set-up to operate at an NCR which, in most cases, is about 75% of the MCR. This percentage allows to the engine to have enough sea and engine margin to operate. In addition the NCR is established by the IMO in its regulations to perform the evaluation of the energy efficiency design index (EEDI).

To reach the desired operational NCR the engine must past through different loads and depending on conditions such as sea state and area of navigation, the engine might spend significant time operating at lower or higher loads than the NCR.

The following results show the response of the plant in terms of thermal efficiency and heat balance analysis.

6.1 Thermal efficiency

The thermal efficiency of the plant is the result of the evaluation of total heat added by the fuel and how much of this heat is used directly for propulsion. Figure 9 shows results of the MEP energy model evaluating the thermal efficiency of the 7G80ME-9.2 MAN B&W Engine at different loads for the engine only and with the inclusion of the WHRS and FWG systems.

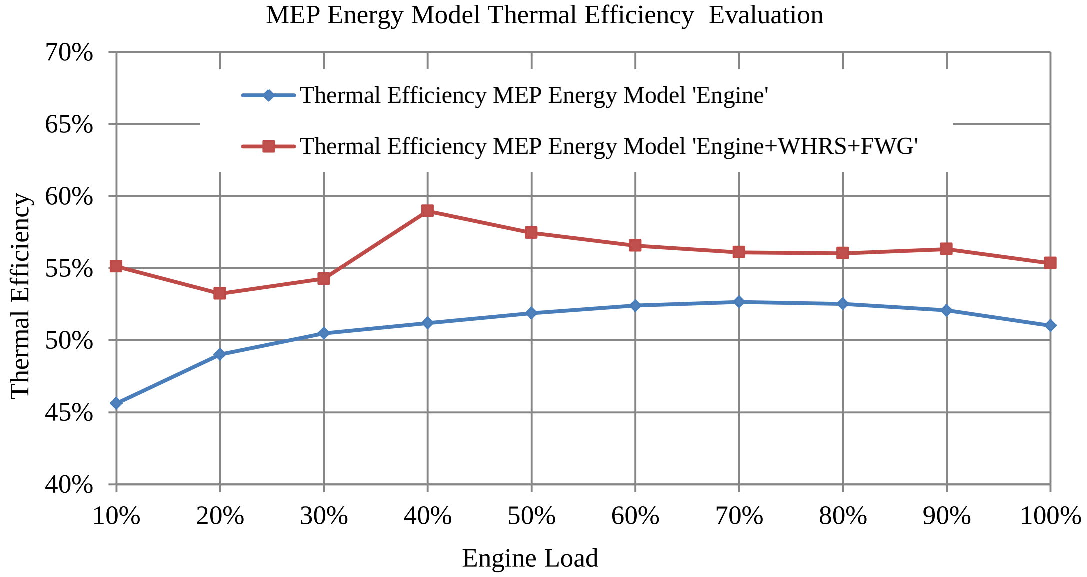

Figure 9 MEP energy model thermal efficiency evaluation

From Figure 9, the blue curve represents the thermal performance of the engine. Greatest efficiency is obtained between 60% and 80% engine loads achieving a peak value of almost 53%.

The red curve represents the plant performance when the WHRS and FWG system are added to recover waste heat. At high engine load, between 60% and 80%, the increase in thermal efficiency is 3.5%. At higher engine loads the difference reaches a maximum of 4.5%. At lower loads the thermal efficiency increases significantly especially at 40% load reaching 8% increase in efficiency.

6.2 Heat balance analysis

A heat balance analysis is the result of the evaluation of the thermal efficiency of the engine (BP) when comparing the thermal losses through the scavenging air (Q_SA), exhaust gases (Q_EG), friction (Q_FL), jacket cooling system (Q_FW), lubrication oil and heat radiated (Q_HR) as the main systems to the heat added by the fuel when it is burned and its energy is released by the engine (Q_Add) as is described in Equation 3.

$$Q\_Add = BP + Q\_SA + Q\_EG + Q\_FL + Q\_FW + Q\_LO + Q\_HR \qquad \text{Equation 3}$$

After evaluating Equation 3 at different engine loads, results of the heat balance analysis are presented graphically in Figure 10 where the major influences on the heat balance of the engine are shown and it is clearly shows the error margin of the heat balance of the engine, especially at higher loads.

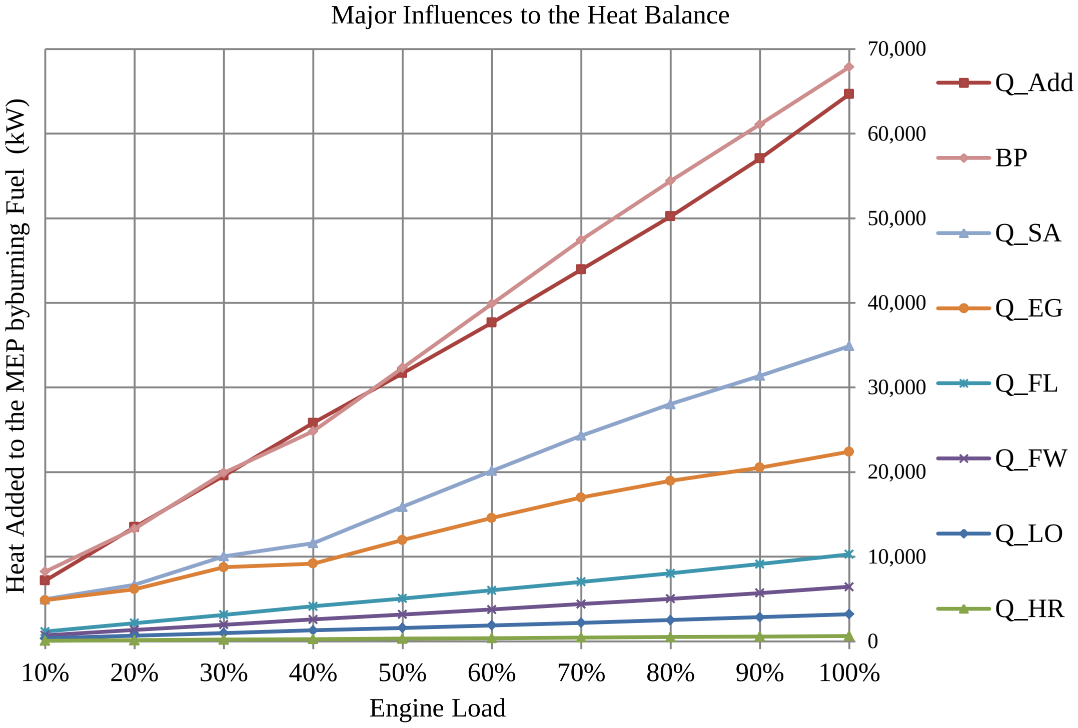

Figure 10 Heat balance of the engine of the MEP energy model

Figure 10 shows the near linear characteristic of the heat balance factors with some variation at lower loads, especially for the heat rejected by the scavenging air system and the exhaust gases between 30% and 40% engine load.

Overall, it has been found that losses through the exhaust gases are the highest, having maximum effect at lower loads, contributing almost 50% of the heat balance of the engine at 10% engine load.

Scavenging air heat losses show variation mainly due to the amount of air needed and passing through the engine at different engine loads, having significant effect at higher loads when more air is needed to allow the engine to burn the fuel needed to operate. These losses reach a 20% of the heat balance of the engine at 100% engine load.

7 Discussion

Results of the thermal efficiency and heat balance analysis were presented and discussed in the previous section separately but to understand the overall performance of the MEP energy model's ability to evaluate the MEP a combined discussion of the results is required.

Taking into account the thermal efficiency of the plant at 10% engine load is at its minimum, the heat balance analysis shows that the exhaust gases losses are maximum and when these losses are provided to the WHRS, the higher values of regain heat are found to increase the efficiency of the plant by up to 10%. Reasons behind this behaviour of the plant can be explained by the fact that WHRS operates under a relatively restricted temperature differential.

The temperature differential accounts for the exhaust gas temperature after the turbocharger and the temperature of the gases after the WHRS which is limited to 180°C because of the sulphur content in the exhaust gases and the need to avoid condensation that will lead to the formation of sulphuric acid (Woodward, 1981).

With this limit in the performance of the WHRS, it is easy to understand the lower efficiency gain of the plant when the engine itself is performing more efficiently (lower sfoc and lower exhaust gas temperature). The implication of this leads to discussion of alternatives that compensate for the limitations of the WHRS e.g. operate the plant at lower loads where more heat can be extracted from the exhaust gases without forgetting the fact that at lower loads less mass flow rate of exhaust gases are going to be available for work within the WHRS and that the engine will be less efficient.

Considering combined results at 40% engine load the behaviour of the plant follows a tendency of reduction of the heat rejected through the exhaust gases and an increment of the influence of the rejected heat through the scavenging air. The thermal efficiency of the plant shows an increment following the same behaviour described previously for 10% engine load, higher values of the exhaust gases temperature differential.

Considering that sfoc follows a tendency of decrease when the engine load increases to a minimum value at NCR it is possible to argue that working at lower loads could be an operational option to increase the efficiency of the plant. Operation at lower loads also leads to the vessel operating at lower speed, an operational condition known as slow steaming.

Summarizing, the higher temperature differential found at lower engine loads leads to increase in plant performance, but more details of the fuel consumed compared to other loads is necessary to properly evaluate the feasibility to operate VLCCs at slow steaming.

Future work

Based on the model development, validation, results and discussion sections a set of future work recommendations is presented:

- Modifications to the MEP energy model are necessary to reduce the error margin and focussing on the evaluation of the heat rejected by the factors affecting the heat balance of the engine. This will lead to a more accurate evaluation of the thermal efficiency of the plant. This will require more detailed data and modelling to get a closer match to the real performance of the plant, especially at lower loads where evaluation by the model is more sensitive to the factors that affect the practical feasibility to operate VLCCs at these loads.

- A more restricted set of boundary conditions at different loads are necessary to reduce the inaccuracy of response of the model based on more details of the components of the model.

- For the WHRS, and considering the development of the technology associated with the system, a more accurate model is necessary to evaluate its performance at lower loads where results are showing higher temperatures of the exhaust gases.

- Inclusion of a mechanical efficiency model to account for the complete propulsion plant of the vessel.

- Inclusion of power generation system models to get a more complete visualization of the performance and efficiency at different vessel operation modes for the plant.

- Inclusion of new technologies associated with the propulsion and auxiliary plants to increase the efficiency of the MEP i.e. upgrade the two stroke engine model to de-rated performance.

Conclusions

This paper has presented the results of a study aimed at evaluating the energy modelling of a VLCC's MEP to assess its efficiency and possible improvements to achieve higher efficiencies. Results showed the feasibility of the modelling approach to improve the efficiency of the VLCC's MEP configuration using MATLAB/Simulink.

Results also showed the feasibility of using the exhaust gases and jacket cooling water energy from the two stroke LSDE to waste heat absorbed by the WHRS and FWG systems to improve the thermal efficiency of the MEP.

Analysing the plant at different engine loads allowed performing an analysis of the MEP energy model where the thermal efficiency of the plant is affected mainly by the performance of the engine at lower and higher loads than its NCR operational point. Results showed that the thermal efficiency of the MEP increases at lower loads reaching a 8% at 40% engine load and a maximum of 10% of increase of the thermal efficiency when is operating at 10% engine load.

Operating at lower loads suggests that slow steaming operation for VLCCs is an option but more details of the performance of the plant at lower loads is necessary to get a valid conclusion including an optimization of the modelling and the boundary conditions at which analysis of the performance of the MEP is carried out.

Looking further into the future, the author aims to improve the modelling of the MEP to reduce the error percentage found when validation is checked against engine manufacturer’s data. The author also aims to develop a deeper understanding of the overall performance of the plant, including other energy conversion systems models, a mechanical efficiency model and a power generation system model.

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