Energy Use Onboard LNG Steam ships

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Abstract

LNG carriers remain the last stand for steam propelled ships due to their ability to utilise cargo boil off as fuel in their marine boilers. In recent years, the emergence of the dual fuel diesel electric (DFDE) and the slow speed diesel engine with a reliquefaction plant has obviated this advantage. While steam ships still numerically dominate the fleet, accounting for over 70% by number, steam propulsion represents only 8% of future orders. This paper analyses the energy profile of steam propelled LNG so as to understand the energy flows on LNG carriers currently used in the shipping industry. Two approaches are taken: Firstly, an in-depth analysis of onboard energy usage is undertaken so as to determine where the energy goes and how the vessel's operational profile impacts energy distribution, the results obtained being compared against the original design specifications; secondly, to consider future steam propulsion technologies and alternative propulsion technologies such as DFDEs and slow speed diesels in terms of comparing carbon emissions.

Using a case study, the investigations discovered that the LNG carrier's operational efficiency profile lags behind design specifications by between 22% - 29% over the twelve months trading period that was analysed. This difference was traced down to three factors; an increase in propulsion energy requirement due to deterioration of hull/propeller brought about an increase in heat loss to the main condenser; a preceding increase in heat loss by the main boiler; and heat losses due to specific standard operational practices such as steam dumping. Further investigations have shown that traditional steam propelled ships fall behind the DFDEs and slow speed diesel in efficiency and that upgraded re-heat turbines, which offered a 13% increase in efficiency, are also insufficient to close the gap with the other solutions; however, the overall global impact must consider both CO_2 and slip of natural gas. The work reported in this paper has been carried out using academic modelling methods and by using a practical approach supported by extensive observations onboard steam propelled LNG ships.

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1. Introduction

The emergence of the marine diesel engine marked the beginning of the decline of marine steam propulsion plant for most ship types. Diesel engines are more efficient, a lot less complicated and offer improved power density over the most efficient steam propulsion arrangements. As diesel engines developed and improved in output power and reliability they drove steam propulsion to near extinction except in Liquefied Natural Gas (LNG) Carriers.

Liquefied Natural Gas (LNG) is Natural Gas (NG) liquefied for ease of transport. The use of natural gas for energy production has been on a steady upward trend with figures from the International Energy Agency (IEA) suggesting a year on year increase of 2.4% for electricity generation and heating as well as its emergence as a transportation fuel [IEA, 2013]. This is due to the abundance of natural gas worldwide and the increasingly stringent international and national environmental policies that discourage the use of other more polluting hydrocarbons. Liquefying NG reduces its volume by 600 times, thereby improving its energy density for shipping. After liquefying at its source such as in Nigeria and Egypt, LNG is transported in specialized insulated cargo tanks fitted in LNG carriers to areas of demand such as Japan and the UK where it is re-gasified and supplied to homes and power stations.

LNG is transported at -163^{o}C at near atmospheric pressure. Inevitably due to imperfect insulation and sloshing in the tanks the LNG boils off with the rate of boil off typically being 0.13% of the cargo capacity per day. The boil off cannot be contained in the tanks and must either be re-liquefied or burnt in the marine engineering plant for power generation and propulsion. Due to its harmful effects as a greenhouse gas as well as flammability the boil off cannot just be released into the atmosphere.

The need to handle boil off gas onboard the vessel has meant that steam power and propulsion plant has enjoyed a 40 year-long dominance in LNG carriers. Steam boilers can burn natural gas as easily as fuel oil so proved a natural solution for LNG carriers. However, the dominance of steam has more recently been seriously challenged by its old nemesis the diesel engine with the emergence of the Wartsila 50F Dual Fuel Engine in a Diesel Electric (DFDE) configuration and the MAN Slow Speed Diesel with a reliquefaction plant (Andreola & Tirelli, 2007).

The advantage of steam propulsion for LNG Carriers has been undermined by the higher efficiencies offered by diesel engines. The scale of the exodus from steam propulsion is astonishing when the LNG order book is examined. From the 1960s through to the 1990s, steam propulsion plant was fitted in almost all LNG carriers with few exceptions, but by the 2000s this figure was down to 69% and in the current decade has fallen to 21%, and the future order book only stands at 8% [Carksons, 2013]. Despite the paradigm shift, steam propulsion still dominates the LNG carrier fleet with the current LNG fleet standing at 71% steam propulsion, 13% slow speed diesel engine, and 13% DFDE [CIN, 2013] and it appears likely to do so for many years to come unless the existing steam vessels are scrapped or re-engined.

The aim of this paper is to analyse the energy efficiency profile of steam propelled LNG carriers and more generally to understand the energy performance of LNG carriers within the shipping industry. Two major areas of investigation have been undertaken: Firstly, an in-depth analysis of energy use onboard steam propelled LNG carriers to establish where the energy goes and how energy distribution is affected by a vessel's operational profile with the results being compared against the vessel's original design specifications; and secondly, to consider future steam propulsion technologies against alternative propulsion technologies in terms of carbon emissions. The work has been carried out using an academic modelling and a practical approach supported by extensive observations on-board steam propelled LNG ships

1.1 Background and Growth in LNG Shipping

On January 15th 1959 the Methane Pioneer, originally designed as a USA government cargo ship but converted to LNG carrier, left the USA to deliver its first LNG shipment to the UK. The Methane Pioneer was diesel powered with a Conch Tank System and had a modest capacity of about 5,000 m^3. The success of the Methane Pioneer propelled Shell to order two purpose built LNG carriers: the Methane Princess and the Methane Progress both steam powered having a capacity of 27,000 m^3 to enter the Algerian trade in 1964 [6]. In the late 1960's an opportunity arose to transport LNG from Alaska to Japan and two ships with a capacity of 71,500 m^3 were built in Sweden: the SCF Polar and the SCF Arctic both steam powered. In the 1970s the construction of LNG carriers really took off when the USA government encouraged the construction of LNG carriers by providing loan guarantees with a total of 16 steam powered ships being built [Noble 2009, Curt 2004]:

- 10 were built using the moss spherical tank system and were steam driven with an average capacity of 126,300 m^3 with most being delivered in 1978 to engage in trade between Indonesia and Japan.
- 3 were built using the TK MZ 1 tank system (membrane) and were steam driven with a capacity of 125,000 m^3 and all have been in continuous operation until the first was scrapped in 2013.
- 3 were built using a modified Conch system with novel insulation arrangement. These ships turned out to be a disaster as while undergoing trials the tank insulation failed irreparably and the ships were declared a total constructive loss.

The complete dominance of the steam driven propulsion continued all through the 1980s and 1990s with the average capacity of ships delivered in the 1980s being 125,000 m^3 moving to 135,000 m^3 in the 1990s. By 2000, there were some 101 steam powered LNG ships in service.

The early 2000's saw the beginning of a shift away from steam. In 2004, the 1,100 m^3 Pioneer Knusten entered service with first internal combustion engine (ICE) capable of burning NG from the cargo embedded in a Diesel Electric (DE) arrangement [Vuyk, 2006]. This was swiftly followed by further DFDE systems, the 75,000 m^3 GDF Suez Energy and the 155,000 m^3 Provalys which at the time had a larger capacity than any existing steam driven LNG carrier. 2007 saw the arrival of four Q-Flex ships with a capacity of 210,000 m^3 using two-stroke slow speed diesel engines, with a reliquefaction plant to handle the boil off gas. These developments saw the rise in the new DE technology and emergence of two-stroke diesels as a real competitor to the steam turbine for the remainder of the decade.

The 2010s have so far seen a decline in both steam propulsion and slow speed diesel propulsion in favour of DFDE becoming the propulsion plant of choice. This has been attributed to the dual fuel diesel engines which can burn fuel oil and NG as well as improving efficiency compared to steam turbines. Slow speed diesel propulsion seems to have received little attention due to initial problems with the reliquefaction technology which presents inherent problems when controlling liquid boil off on passage, as well as a gradual shift towards cleaner fuels. Figure 1 shows the global fleet of LNG carriers based on propulsion type and year of delivery [Clarksons, 2013];

Table 1: Current LNG Carrier Propulsion Profile

Propulsion Type	No. 60s	No. 70s	No. 80s	No. 90s	No. 00s	No. 10s	Total
Diesel Electric	0	0	0	0	25	25	50
Motor Ship 2 Stroke	0	0	0	0	41	6	47
Motor Ship 4 Stroke	0	0	1	0	4	6	11
Steam Turbine	2	27	19	46	161	10	265
All LNG Carriers	**2**	**27**	**20**	**46**	**231**	**47**	**373**

1.2 LNG Carrier Fleet Age Profile

The age profile of the global LNG c.arrier fleet is shown in Figure 1. As can be seen the LNG shipping fleet is a relatively young one with about 75% of the fleet being built after the year 2000.

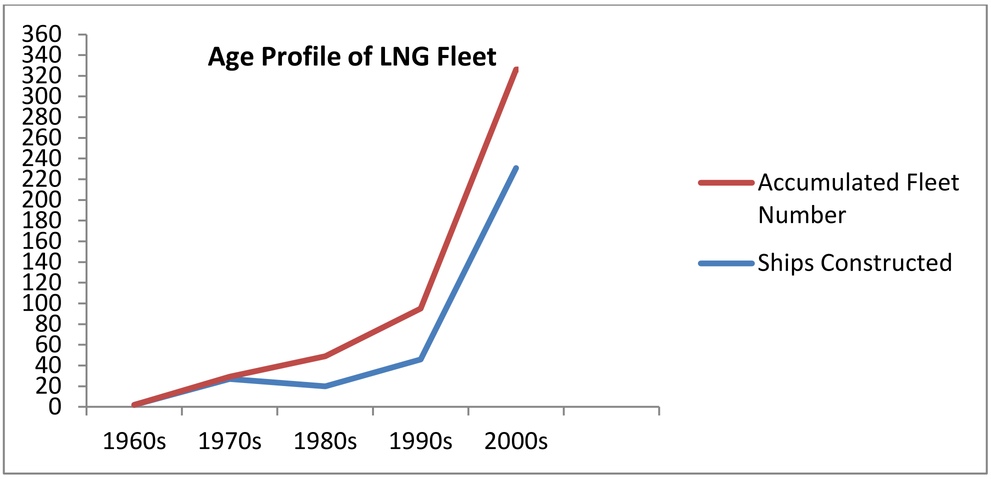


Figure 1: LNGC Age Profile

When cargo capacity instead of number of ships is considered, it is seen that the 75% of the entire fleet built in the 2000s accounts for 79% of the total capacity of the global LNG fleet, indicating a relatively flat increase in the sizes and capacity of LNG carriers. This can be attributed to the fact that the sizes of these ships are limited by the receiving terminals, and despite the advent of the Q-Flex and Q-Max Ships with 215,000 m^3 and 266,000 m^3 respectively, these ships are relatively few and most vessels constructed are still of the 120,000m^3 -160,000 m^3 range which have been used since the 1970s.

1.3 LNG Carrier Order Book

Considering Figure 2, as at end of Q1 2013, there were a total of 88 LNG carriers on order and the figures indicate a decline in steam propulsion for LNG carriers and the increasing dominance of the diesel electric propulsion.

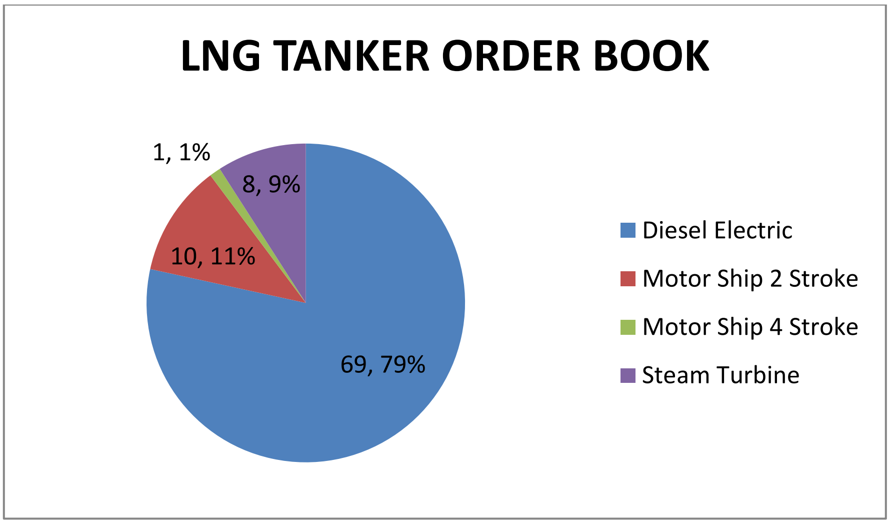


Figure 2: LNGC Order Book

In terms of capacity, the LNG order book seems to indicate a minor rise in average capacities with the absence of Q-Flex or Q-Max ships from the order book means average capacities of 171,133 m^3, 161,826 m^3 and 156,800 m^3 for two-stroke propelled, diesel electric propelled and steam turbine propelled LNG carriers respectively.

Table 2: Average Capacities of LNGCs

Propulsion	Number	Average Capacity
Diesel Electric	69	161826
Motor Ship 2 Stroke	10	171133
Motor Ship 4 Stroke	1	6500
Steam Turbine	8	156800

2.1 Energy Efficiency in Operations- The Case of the Steam Ships

As can be seen from Figure 1, steam ships still numerically dominate the fleet and these ships are likely to still be around for some 20 plus years. In order to analyse the energy efficiency of these ships, a case study was chosen. The choice for the case study was a ship built in the 2000s, a period within which the number of steam ships delivered constituted 43% of the total LNG fleet [Clarksons, 2013]. The choice of this case study also has benefits: firstly, virtually all steam LNG ships regardless of age are similar in their system arrangements and operations as they all have boilers; turbo-alternators; steam feed pumps; main condensers; intermediate heaters and utilise cargo boil off/heavy fuel oil (HFO)/marine gas oil (MGO) in their fuel mixes, with steam dumping being utilised at low loads. Secondly, all steam LNG ships built from 1994 have exactly the same plant configuration as the case study with very minor variations. This translates to 78% of the current steam powered fleet, and 55% of the LNG carrier fleet is powered by an arrangement similar to the case study chosen.

The process was to analyse the design characteristics as well as the actual operational conditions and undertake a comparison between the two to indicate how efficiently the LNG carrier is being operated. A ship with the following characteristics was chosen:

Table 3: Case Study Ship Principal Characteristics

1. Ship Principal Characteristics		
Characteristics	**Value**	**Comments**
Ship Type	LNG Carrier	
Date of Delivery	2006	
Summer Draught	12.32 m	
Draught, Ballast	9.78 m (Normal & Heavy Weather)	
Cargo Tank Capacity	141,052 m^3	At 100%
Deadweight, Summer Draught	79,541 t	
Displacement, Summer Draught	113,567 t	
Service Speed	19.25 knots	Design Draught 11.25 m
2. Propulsion Engine		
Descriptive Notes: Steam Turbine Driven Shaft Via Gearbox		
Make and Model	Mitsubishi MS 36-2 Steam Turbine	
Rating (Turbine)	23,500 kW	HP Turbine: 5,685 rpm LP Turbine: 3,351 rpm Propeller: 81 rpm
Specific Fuel Consumption at rated power		
Propeller	5 Bladed 8.6 m diameter	Fixed Pitch
3. Generators and Boilers		
Turbo Generators: Two Steam Turbo Generators		
Make and Model (Turbine)	HHI RG92-2	8145rpm
Generator	4062.5 kVA at 1800 rpm	
Specific Fuel Consumption at rated Power	13.65 t/h	Steam
Diesel Generators: Two 6-Cylinder Direct injection Diesel Engines		
Engine Make and Model	Hyundai MAN- B&W 8L28/32H	2 x 1,600 kW at 720 rpm
Generator	2,000 kVA at 720 rpm	
Fuel	LSDO	
Boilers: Two Top Fired Water Tube		
Make and Model	HHI 2 X MB-3E	
Rating	47 t/h 515^{O}C at 60 Bar	Maximum 55 t/h
Rated Fuel Consumption	4001 kg/h	Maximum Burner Capacity
Fuel	HFO/Methane/MGO	

In this study, assumptions were kept at an absolute minimum, with typical values, obtained onboard or calculated based on real life operating conditions for all modes of operations (ballast, loaded, loading, discharging, manoeuvring, Off Port Limit operations) over a 12 month period.

2.2 Design/Actual Energy Flow Analysis

The first step involved an energy flow analysis showing how all the energy is used. To do this the design heat balance and flow diagrams [Kim, Kwak & Park, 2006] were utilized to develop energy flow diagrams (EFD), which essentially details how much fuel the plant should consume at different modes of operation- this was used for the 'design specification'. For actual scenarios recorded, the Kyma Steam Plant Analyser [Kyma, 2009] was used which basically monitored the engine and plant performance in actual conditions, as well as back up information such as obtained from plant logs, electronic archive sheets and noon reports were utilised over a 12 month period. The different modes of operation analysed includes 100% Maximum Continuous Rating (MCR) Dual Burning, 90% MCR Dual Burning, 70% MCR Dual/HFO Burning, 50% MCR Dual/HFO Burning, 30% MCR Dual/HFO Burning, in Port Loading, in Port Discharging, and hotel load (anchor) conditions. Based on these conditions, the EFDs and Sankey diagrams were utilized to better visualise the results, the samples of which are shown in Figure 3 and Table 4 respectively.

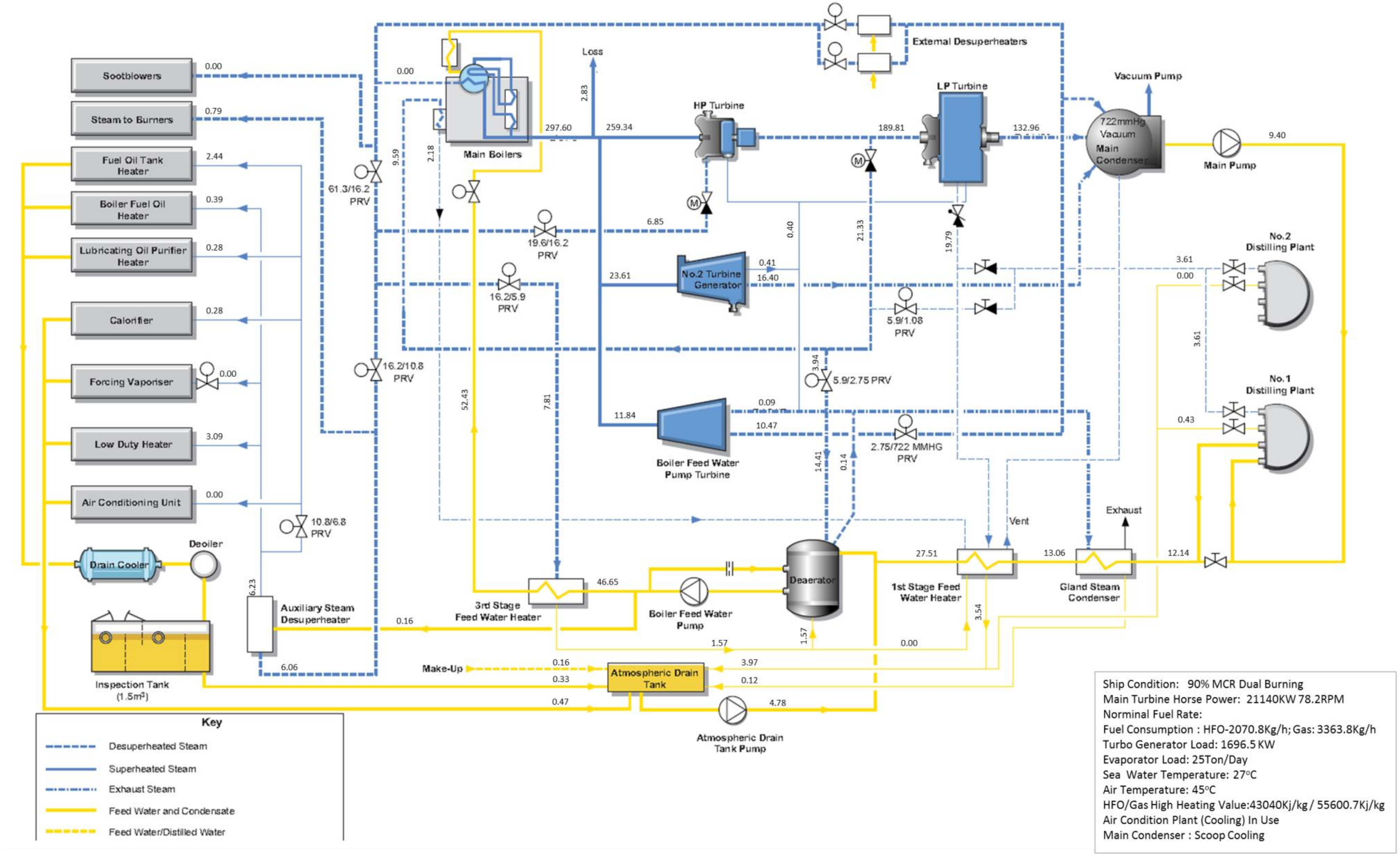
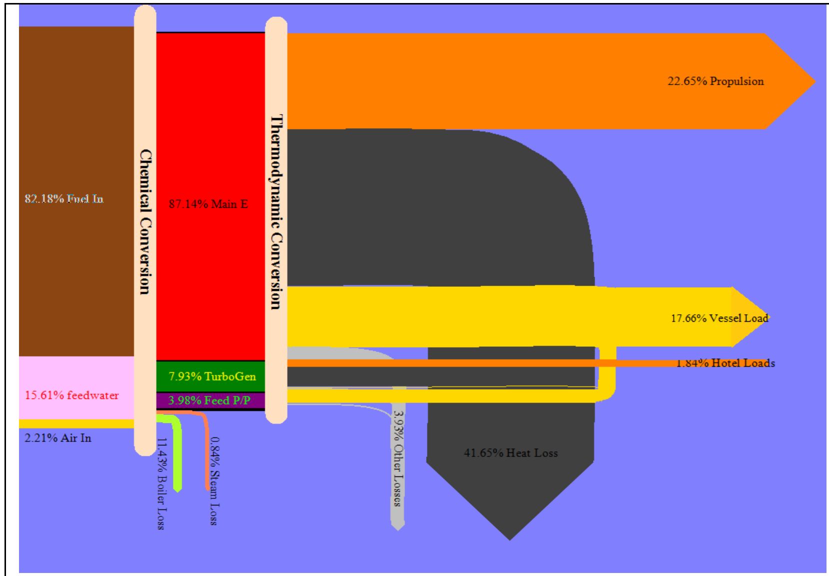
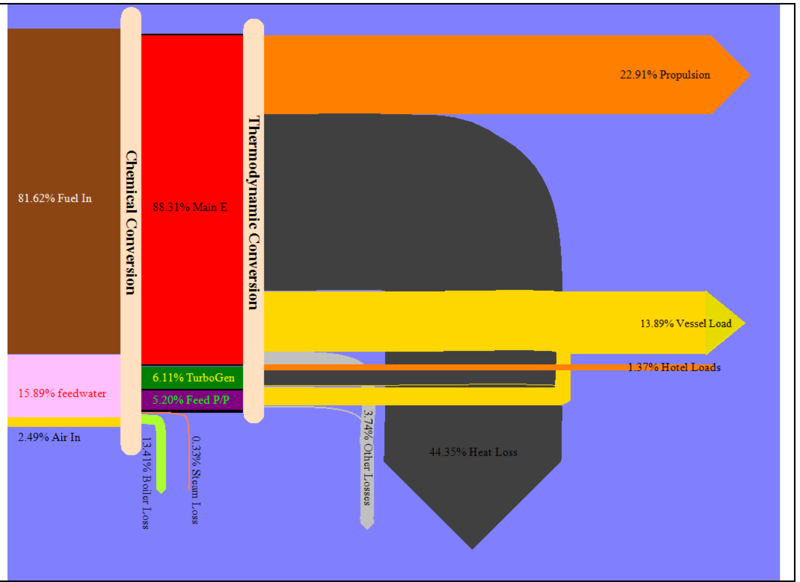


Figure 3: 90% MCR Dual Burning Vessel EFD

Table 4: 90% MCR Design/Actual Summary Vessel Condition





Ship Condition (Design)	**90% MCR Dual Burning**		**Ship Condition (Actual)**	**90% MCR Dual Burning**	
Main Turbine Horsepower	21,140 kW 78.2 rpm		Main Turbine Horsepower	21,215 kW 77.6rpm	
Nominal Fuel Rate (F/G)	303.9 g/kWh	234.9 g/kWh	Nominal Fuel Rate (F/G)	372.2 g/kWh	293.9 g/KWh
Fuel Consumption	2,070.8 kg/h	3,363.8 kg/h	Fuel Consumption	851 kg/h	5,575 kg/h
Fuel Energy in	276.16 GJ/h		Fuel Energy in	341.15 Gj/h	
Turbo Generator Load	1,696.5 kW		Turbo Generator Load	1595 kW	
Evaporator Load	25 t/day		Evaporator Load	24 t/day	
Sea Water Temperature	27^{o}C		Sea Water Temperature	22^{o}C	
Air Temperature	45^{o}C		Air Temperature	35^{o}C	
HFO/Gas Heating Value	43,040 kJ/kg	55,600.7 kJ/kg	42391kj/kg	42,391 kJ/kg	43,040 kJ/kg
Air Condition Plant	Cooling (In Use)		Air Condition Plant	Cooling (In Use)	
Main Condenser	Scoop Cooling		Main Condenser	Scoop Cooling	

Over the 12 month period the vessel did not achieve its design MCR with the maximum achieved being 91%. Yet with at lower %MCR, the actual values indicated a 10.8% greater amount of fuel energy being needed. From the energy flow diagrams, the 10.8% increase in fuel requirement is represented as additional heat lost with majority of it (98%) being lost in the main condenser. The T/A, feed pump and other consumers all fall within acceptable range with their performance remaining close to design specifications.

At 90% MCR comparability between both design and actual conditions is possible as the vessel regularly operates at this %MCR. From the Energy Flow Diagram (EFD) it is seen that the fuel consumption is 23.5% higher than design conditions. There is a 47% additional energy lost in the main boiler, with 71% of this extra energy being lost in the main condenser. Other main engine losses (conversion losses etc.) account for the remainder. The T/A, feed pump and other consumers all fall within acceptable range.

At 70% MCR, the vessel utilises 26.5% more fuel energy than design specifications. Initially, there is a 67.7% increase in heat loss from the boiler, and a 39% increase in heat lost to main condenser. The T/A Feed Pump and other consumers fall within range.

At 50% MCR the vessel utilises 41% more fuel energy than at design conditions. In the boiler there is a 92% increment on the heat lost, while there is a 59% increase in the heat lost in the main condenser. The T/A, feed pump and other consumers all exhibit minor increment in losses but generally fall within range.

At 30% MCR, the vessel uses 45% more fuel than the design specifications. There is a 65% increase in the heat lost in the boiler, while a 90.5% increase of the energy lost to the main condenser. The performance of the T/A, feed pump, as well as other consumers all fall within range.

At loading conditions, the vessel utilises 83% more fuel than its design specifications. Initially there is a 112% increase in heat lost in the boilers (during loading conditions the Main Engine is shutdown so there is no Main Engine heat loss to the main condenser) however a substantial amount of heat is lost as ‘dump steam’- this is normally used when the vessel is at very low loads to maintain the boiler load above a certain threshold to ensure plant stability.

At discharging conditions, the vessel utilises 31% more fuel than design specifications. In the boiler there is a 27% increase in heat loss, while although the dump steam is reduced compared to during loading conditions due to the increased amount of load on the T/As, this dump still accounts for 80% of the extra energy put into the boilers.

For hotel load conditions (at anchor), the vessel utilises 273% more fuel than design specifications. Initially there is a 128% increase in heat energy lost from the boiler. 74% of this extra fuel requirement is lost as dump steam to the main condenser, with minimal losses in the T/A, feed pump and other consumers.

A summary of the extra energy in GJ/h requirement of actual conditions compared to design conditions is shown in Table 5.

Table 5: Difference between Actual and Design energy Usage.

Actual-Design	90%MCR	70%MCR	50%MCR	30%MCR	Loading	Discharging	OPL
Boiler Heat Loss	17.64	16.84	18.09	10.11	8.82	3.03	9.28
Propulsion	0.27	-1.5552	0.81	-0.5004	0	0	0
M/E Heat Loss	46.25	39.75	47.13	49.41	0.00	0.00	0.00
M/E Other Losses	2.51	6.56	2.69	2.07	0.00	0.00	0.00
T/A Hotel Load	-0.4374	0.08064	0.26784	-0.3132	-5.40	-1.86	0.56
T/A Heat Loss	-0.84	0.02	0.67	0.34	-12.47	-4.39	2.70
T/A Other Losses	-0.20	0.08	0.03	0.60	4.38	4.36	4.61
Feed P/P Heat Loss	0	0	0	0	-0.56	0.00	-4.54
Feed P/p Other Loss	0.14	0.93	1.02	1.10	1.02	1.02	0.93
Steam loss	-1.45	-0.21	0.10	0.45	0.00	-0.16	0.25
Dump	0	0	0	0	49.25	19.42	57.19
Vessel Load	-0.47	-5.58	-0.59	-6.29	5.41	5.32	6.37
Total Extra Fuel	**63.41**	**56.90**	**70.21**	**56.98**	**50.46**	**26.75**	**77.34**

The next step involved using the Marine Environment Protection Committee (MEPC) mandated Energy Efficiency Operational Index (EEOI) to analyse the vessel's efficiency over the same period using the data from the case study. In this case study, the EEOI is used as a representative value of the ship's efficiency over a period and represents the trading pattern of the vessel. The trading period covered here is between 19/06/2012 and 26/05/2013 during which the vessel delivered six cargoes, all originating from Bonny, Nigeria. Two cargoes each delivered to Japan and Korea, while one each to Spain and India. The results of the EEOI for these voyages are shown in Table 6; all input values for the calculation were obtained from onboard vessel's records.

Table 6: Actual EEOI over 6 Voyages

Voyage	Distance (nm)	LNG Delivered(t)	LSDO Consumed(t)	LNG Consumed(t)	HFO Consumed(t)	EEOI (CO_2/t.nm)
1- NG-JPN	22,559.3	129,683	12	4,694	2,676	32.37
2- NG-IND	16,242.8	128,887	33	4,649	637	31.74
3- NG-SPA	7,179.5	134,445	39.2	2,039	247	29.65
4- NG-JPN	22,406.1	128,848	4	4,861	2,505	32.65
5- NG-SKR	22,289.7	128,596	14	5,261	2,438	34.35
6- NG-SKR	22,249.6	128,480	7	5,304	2,568	35.26

A comparison was also done between the actual EEOI of these voyages and the design (i.e. best case) EEOI. The EEOI was recalculated based on the design specifications of the plant to ascertain how more efficiently these cargoes should have been delivered. To achieve this, all log input for the plant during the 6 voyages were analysed and based on the fuel consumption design specifications at those conditions, an equivalent fuel consumption for each specific voyage was obtained, with the LNG/LSDO consumption constant, this was used in calculating the EEOI. The results are displayed in Table 7:

Table 7: Calculated Design EEOI of Last 6 Voyages

Voyage	Distance(Nm)	LNG Delivered(t)	LSDO Consumed(t)	LNG Consumed(t)	HFO Consumed(t)	EEOI (CO_2/tonNm)
1- NG-JPN	22,559.3	12,968	12	4,694	1042	24.63
2- NG-IND	16,242.8	128,887	33	4,649	-539	23.91
3- NG-SPA	7,179.5	134,445	39.2	2,039	-325	21.53
4- NG-JPN	22,406.1	128,848	4	4,861	949	25.18
5- NG-SKR	22,289.7	128,596	14	5,261	341	24.20
6- NG-SKR	22,249.6	128,480	7	5,304	394	24.70

As can be seen from the design EEOI obtained above, the drop in operational efficiency of the vessel when benchmarked against designer specifications is between 22.9% and - 29.9%.

2.3 Improving Energy Efficiency of Steam Propelled LNG Ships

Ideally this will be twofold; the first will be from an operational perspective and the other from a design stand point. For operational measures, a further analysis of the case study carried out by utilising the energy usage analysis in Table 5 to ascertain which individual components have efficiency losses and introduce options for mitigating the losses.

From the energy usage analysis, three major sources of the inefficiencies are easily identified:

The first is in the steam consumption of the MT as design specifications indicate that at to achieve 100% MCR while dual firing, each boiler would be producing 47.5t/h steam, whereas in actual conditions, both boilers are in fact producing 52.5 t/h steam, but only achieving 91% MCR. A closer look at the EFDs- design and actual, indicate that the extra 10t of steam is going to the main propulsion turbine and the feed pump steam consumption is also increased so as to supply the extra feed water required to produce the steam. This trend is quite common throughout the different modes of operation. At 90% MCR the MT requires 18t/h more steam than design conditions; at 70% MCR 17t/h more; at 50% MCR 21t/h; and at 30% MCR 18t/h more steam is required. The higher steam consumption invariably leads to a higher amount of exhaust steam whose heat energy is lost in the Main Condenser. The higher steam consumption was traced back to a significant drop in hull and propeller performance: The performance trail done immediately after dry dock when extensive cleaning work had been carried out on the propeller and hull, the vessel achieved 98.6% MCR with the boilers producing 47.5t/h each and the MT steam consumptions at the design specifications . At 90%MCR while the vessel was fully loaded, the steam consumption was 3t/h more than design conditions compared to the 18t/h it is during the period analysed.

The second source is due to the heat lost in the main boilers. Invariably, due to the higher fuel consumption due to the higher amount of steam required by the MT, the heat losses in the boiler increases, and this coupled with a slight drop in boiler efficiency from post dry dock conditions accounted for 17-26% of extra losses across the different modes of operation.

The third is due to actual operational standard procedures during periods when the MT is not in use or at very low loads. During these conditions, a minimum steam flow is set on the boilers and this minimum is higher than the steam demanded by the plant. In this case study, the minimum was set at 40t/h from the boilers for all conditions where the MT was stopped. This figure was compared to 20t/h for loading, 29t/h for discharging, and 11 t/h for OPL conditions being the actual required steam flow at these conditions, with the excess steam being dumped in the main condenser as wasted energy. The justification for this is that there is a limitation on how low a boiler can go in terms of steam produced, based on a minimum flow of HFO/Gas fuel to the boilers as well as the superheated steam exit conditions at very low boiler loads being poor. By shutting down a boiler, at the above conditions, then the above limitations' are eliminated, as for one boiler the load is large enough to maintain the minimum gas/fuel requirement, as well as keep the quality of superheated steam at the desired quality. The firing rate of the boilers can also be manually reduced, regardless of the minimum HFO/Gas requirement, as far as reasonably practicable. Also the HFO/Gas relationship should not be ignored, as NGs' higher calorific value and lower carbon factor, will mean a lot less CO_2 is emitted when gas burning, in the boilers than when HFO burning. But for the period considered during this case study, gas burning was not carried out in discharge ports, with either HFO or LSDO being used.

From a design stand point, it would appear that the viable measures would be to place the main boilers and the main turbine as the core target of improvement. To this end the two remaining designers and suppliers - Kawasaki and Mitsubishi Heavy Industries have developed the reheat cycle plant where steam pressure and temperature are raised, and it is claimed to have improved efficiency by more than 13% [JIME, 2008]. The idea is that the plant operates in a regenerative cycle with steam tapped from the hp turbine and fed to a reheater in the boiler before supply to the ip turbine. The efficiency gain derives from a higher steam pressure condition (6MPa increased to 10 MPa) and a higher steam temperature condition (510^{o}C to 560^{o}C) and improved turbine operating efficiency [Marine Prop, 2008].

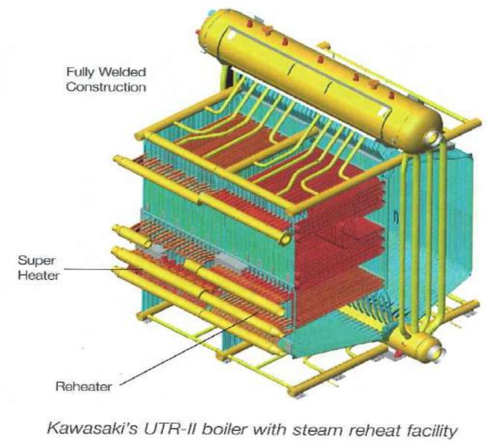


Figure 4: Steam Reheat Boiler

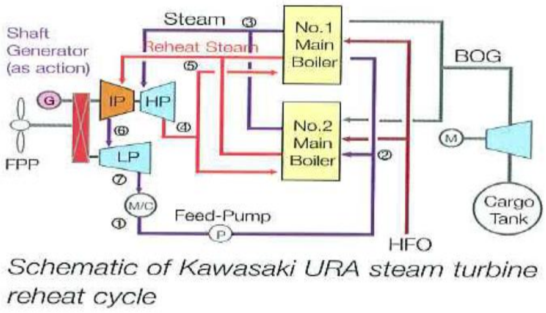


Figure 5: Steam Turbine Reheat Cycle

3.1 Energy Efficiency in Design- Propulsion Comparisons

The EEDI is basically a statistical analysis of the existing ships in order to develop a reference curve against which future ships can be designed. This reference curve is then set as a start line against which future developments will be benchmarked. While the EEDI is not required for existing ships, a statistical analysis of the existing fleet against a common reference curve is necessary so as to assess and compare the design efficiencies of the three different propulsion systems within the fleet. At the MEPC 65/4/13, Society of International Gas Tankers and Terminal Operators (SIGTTO), Denmark, Japan and Liberia proposed a single reference line for all LNG carriers installing different kinds of propulsion systems by evaluating the greenhouse gas emissions from the different propulsion systems taking in account the treatment of boil off gases, which is a specific issue to this ship class [IMO, 2013]. The equation is shown below:

$$EEDI = 2465.7dwt^{-0.482} \text{..................... (1)}$$

For the purpose of this study, this is taken as the EEDI baseline.

In calculating the index value of existing ship designs the following equations obtained from resolution MEPC 231(65) are represented in Figure 6 [MEPC, 2013],

	Direct Drive Diesel	Dual Fuel Diesel – Electronic (DFDE)	Steam Turbine
Margins	$Engine: 10\%$ $Sea: 20\%$	$Engine: -$ $Sea: 20\%$	$Engine: -$ $Sea: 20\%$
Design Margin	$Margin = \frac{0.9}{1.2}$ $Margin = 75\%$	$Margin = \frac{1}{1.2}$ $Margin = 83\%$	$Margin = \frac{1}{1.2}$ $Margin = 83\%$
P_{ME} Formula[1]	$P_{ME(i)} = 0.75 \cdot (MCR_{ME(i)} - P_{PTO(i)})$	$P_{ME(i)} = 0.83 \cdot \frac{MPP_{(i)}}{\eta_{Electrical(i)}}$	$P_{ME(i)} = 0.83 \cdot (MCR_{ME(i)} - P_{PTO(i)})$
SFC_{ME} in g/kWh (Fuel)	$190\ (HFO)$	$175\ (FBO)$	$285\ (FBO)$
P_{AE} Formula[2]	$P_{AE} = 0.025 \cdot \sum_{i=1}^{nME} MCR_{ME(i)} + 250 + Capacity \cdot BOR \cdot 15$	$P_{AE} = (0.025 + 0.02) \cdot \sum_{i=1}^{nME} P_{ME(i)} + 250$	$P_{AE} = 0$
Index Formulae	$3.1144 \cdot \frac{190 \cdot \sum_{i=1}^{nME} P_{ME(i)} + 215 \cdot P_{AE}}{Capacity \cdot V_{ref}}$	$2.75 \cdot \frac{175 \cdot \sum_{i=1}^{nME} P_{ME(i)} + 175 \cdot P_{AE}}{Capacity \cdot V_{ref}}$	$2.75 \cdot \frac{285 \cdot \sum_{i=1}^{nME} P_{ME(i)}}{Capacity \cdot V_{ref}}$

Figure 6- EEDI Formulas

Then results are displayed in the Figure 7, Figure 8, & Figure 9. For the purpose of this study the following parameters and assumptions were made:

- The Boil Off Ratio (BOR) for the direct drive diesel was assumed to be 0.15%/day
- Capacity is taken as the deadweight at summer load line draft
- Vref from the data of the ships service speed from Clarkson's World Fleet Register.
- PME, PAE, SFC, CFME, CFAE are all obtained from Figure 6.
- Data on each ships main Installed power is used as MCR_{ME}
- Some ships data entries are blank and do not contain any data; these were nevertheless added to the calculations and are represented on the graph with no values.

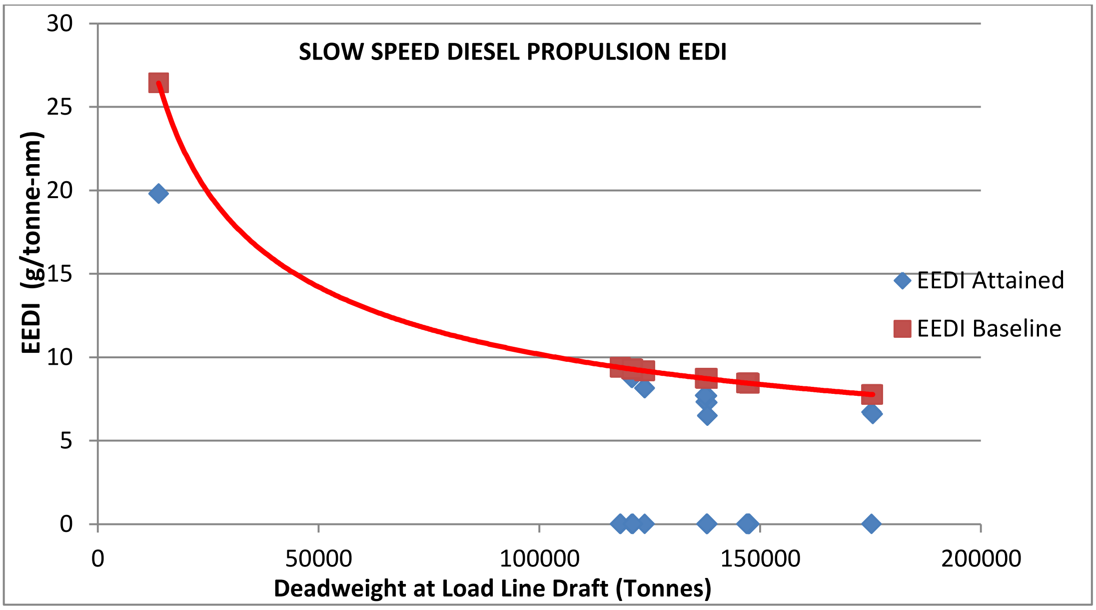


Figure: 7 EEDI of Slow Speed Diesel Propulsion

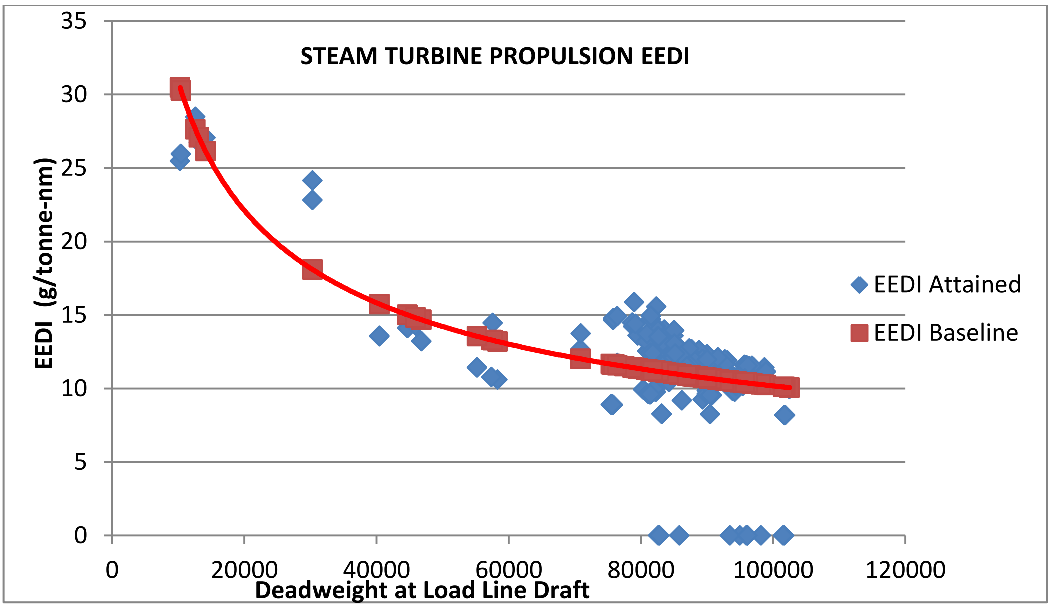


Figure 8: EEDI of Steam Turbine Propulsion

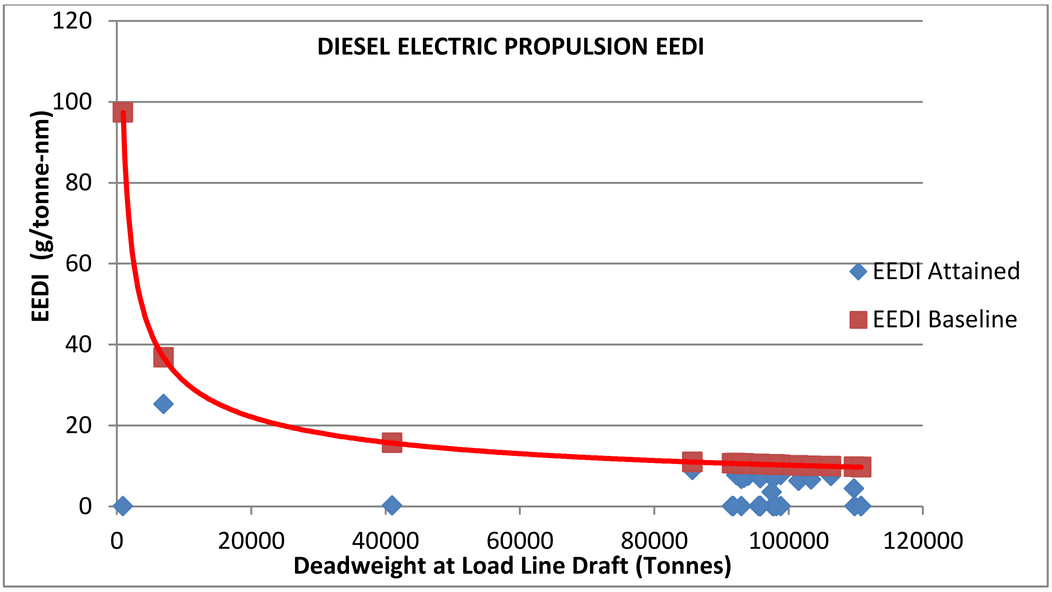


Figure 9: EEDI of Diesel Electric Propulsion

From the results, it is seen that the DFDEs offers the highest efficiency in EEDI terms as all ships analysed fell below the baseline curve. The slow speed diesel ships however exhibited the same property, but that was only when its reliquefaction technology was operational. In the absence of this, these vessels become highly inefficient, due to the addition of the CO_2 values emitted from the combustion of the BOG in the Gas Combustion Unit (GCU). The steam propelled carriers displayed the lowest efficiencies in EEDI terms as expected with majority of the vessels going above the baseline value. Even when the 13% improvement in efficiency offered by the reheat turbine is theoretically applied to the existing steam fleet, thereby reducing the specific fuel consumption from 285g/kWh to 250g/KWh, just over 50% of the steam fleet still fall below the baseline curve as seen in the Figure 10. This also shows that despite a 13% increase in efficiency, future steam carriers would still be less efficient than the present DFDE or slow speed diesel alternatives in EEDI terms.

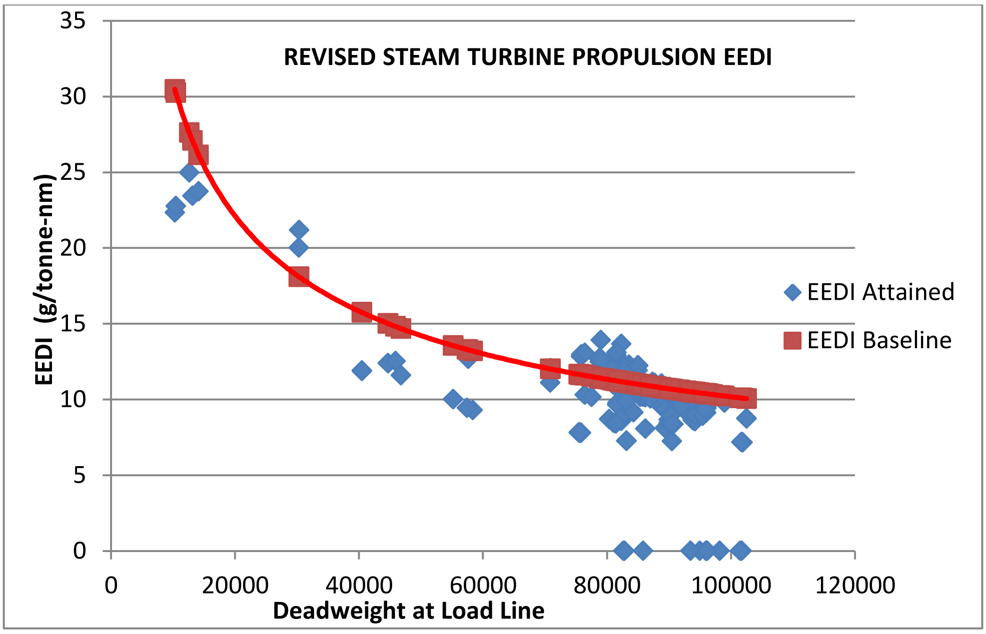


Figure 10: Revised EEDI of Steam Turbine Propulsion

3.2 Discussion

The only perceivable benefit of steam propulsion compared to DFDEs would be the lower exhaust gas emissions. The issue of methane slip is particularly an issue in the DFDEs as methane is estimated to have a global warming potential (GWP) of 25 times that of CO_2 [IMO, 2009]. With the methane slip of the DFDEs estimated at 4g/KWh to 8 g/kWh, essentially 2.2% to 2.4%, and that from marine boilers estimated to be less than 0.1% [US EPA, 2004] would mean that the equivalent total GWP of the DFDEs would rise by about 50% when methane emissions are added thereby making the future steam reheat cycle propulsion system at par and comparative to it. It is however important that while considering future steam propulsion systems within the LNG shipping industry, the future of other propulsion systems should be considered as well. The newer generation DFDEs offer a reduction in methane slip to around 0.8% [Heraldson, 2011] while even the future slow speed diesel propulsions even offer better promise as despite having the highest thermal efficiency of all three propulsion types, the slow speed diesel type is less dominant in the LNG shipping industry due to its capabilities in handling boil off gas. Although all the slow speed diesel engines have reliquefaction plants that handle this boil off, initial reliability issues with these reliquefaction plants as well as new emission requirements has made it seem out of place to have an LNG carrier not utilising the cleaner boil of gas as fuel and instead using large amounts of power to reliquefy it and send back to the cargo tanks [MAN, 2012].

The new ME-GI propulsion systems which are two-stroke engines capable of burning gas have been introduced by MAN B&W as a proffered solution to this oddity. The concept of the ME-GI system is based on a high pressure gas injection principle with pilot fuel ignition, ensuring that the same high thermal efficiency of the diesel combustion process for heavy fuel oil burning can be achieved. MAN B&W claims that this would have an advantage over the carburetted premixed Otto cycle gas process currently being used by the DFDEs due to the fact that the gas does not take part in the compression stroke; thereby eliminating the risk of knocking and thus high compression/expansion ratios can be utilized offering higher energy efficiency and low gas emissions [Juliussen, Kryger, & Andreasen, 2011].

4. Conclusions and Further Work

The aim of this study was to analyse the energy profile of steam propelled LNG carriers, and in order to do that two objectives were set. As these steam-propelled carriers are the least efficient and numerically dominant LNG Carrier propulsion system, the first objective involved performing an in-depth study of its operational energy use characteristics, identifying which areas are performing inefficiently and where improvements might be possible. The results obtained from a case study where the vessel performance was benchmarked against the designer's specifications, taken over different modes of operations, indicated three major sources of inefficiencies:

- The first being an increase in heat lost to the main condenser due to a reduction in hull/propeller performance leading to a higher amount of main turbine steam consumption to achieve its required output and invariably a higher amount of energy lost to the condenser.
- The second is that the higher amount of steam required by the system, coupled with a slight drop in boiler efficiency means more energy is lost during the energy conversion of fuel energy to steam energy in the boiler when compared to design specifications.
- The third source is due to an established practice of steam dumping, where a minimum load is kept on the boilers at periods where vessel is in port, anchored or slow steaming, with the unused extra energy dumped in the main condenser as heat loss. This extra energy lost during this practice range from 31% in discharge ports to 237% in OPL/Anchorage conditions.

It was also found out that the remedy to the sources of inefficiencies are quite easily achievable, and have been practiced quite successfully during the period being considered, such as hull and propeller cleaning was identified to reduce the ME heat losses by 68%, while also reducing the boiler efficiency losses by 49%, totalling a reduction in extra CO_2 emitted of 64%. Also by burning BOG, a savings of 25% of CO_2 emissions is realised when compared to burning HFO at full load conditions due to the higher calorific value and the lower carbon factor of the gas. Also heat lost due to steam dumping can be eliminated by shutting down one boiler when the loads have reduced to levels that require the steam to be dumped; also by reducing the firing rate manually would also reduce the amount of heat currently being lost to steam dumping. All these solutions have been carried out at different times during the period considered for this case study but are not standard practice due to both internal and external factors. For example, hull and propeller cleaning are limited by dry-dock periods and owner/charterers requirements, while the choice of fuel either HFO or BOG is determined by the cargo owners/charterer's requirements, while the shutting down of a boiler at low loads which usually is in port is dependent on whether the ports or terminals would allow it.

The second objective involved analysing the current and future design of steam LNG ships and benchmarking them against alternative propelled types. On this it was seen that in terms of the EEDI, the DFDE offers the best option, while steam propulsion offers the least, with slow speed diesel becoming the least efficient in instances where the BOG reliquefaction system is not operational. Even the current reheat turbine, which offers a 13% improvement in efficiency over the conventional steam ships, did not improve steam propulsion efficiency significantly when compared to the DFDEs and the slow speed diesel systems. The slow speed diesel, however, has promise in terms of the future, as the new ME-GI propulsion systems, a slow speed diesel that can burn gas utilises its higher efficiency over the four-stroke DFDEs to possibly become the most efficient system for future LNG Carriers. The problem however with gas burning diesel engines, is the issue of methane slip as current DFDEs have an estimated methane slip of 2.2% to 2.4%, compared to the <0.1% from steam ships. But with the newer ME-GI propulsion systems promising to offer a reduction in methane slip by 20 to 40 times compared to current DFDEs, it would seem the future is diesel and not steam, if low carbon shipping is the goal.

As seen from the study, that despite steam propulsion being the least efficient of the three propulsion types, its operational profile is also below design specifications due to specific propulsion type standard operational practices. Therefore for future work, in-depth studies of energy use on-board the other two LNG propulsion systems: the DFDEs and the slow speed diesel are required. This is because as seen in the energy study for steam propulsion, that while the design fuel consumption can be one value, the actual fuel consumption might be a lot higher depending on the propulsion type standard specific operational practices. Hence, an in-depth study would highlight how these standard practices are impacting on the efficiency of the propulsion type and introduce options that can attempt to mitigate them, while also painting a holistic picture of the energy efficiency performance profile of LNG shipping.

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