Optimising the operation of a large two-stroke marine engine for slow steaming

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Abstract

This paper studies the effects of extreme de-rating of propulsion machinery following from vessel slow steaming operation as a part of ULYSSES project (EU FP7). An advanced simulation model of a WÄRTSILÄ two-stroke RT-flex50 engine was created to investigate operational optimisation at operating points far below nominal engine rating. The model was validated using experimental test bed data from WÄRTSILÄ and operational data provided by Torm from one of their oil/chemical Handymax tankers, the TORM Lilly, for load conditions from 25% to 100% MCR. Validation shows that the deviation between the simulation result and the real data is within ±5%, hence accurate enough for further analysis using the model. Some retrofit options, e.g. changing start of injection (SOI), compression ratio (CR), swirl ratio (SR) and intake port opening (IPO) are studied and potential performance benefits quantified. The simulation and optimisation results show that the engine performance was improved significantly by changing SOI and CR, while limit effects were found by adjusting SR and IPO. It will be shown that through ultra slow speed steaming (less than 30% MCR) and optimisation of the engine operating parameters that improved fuel economy rates can be achieved, leading to reduced annual CO_2 emissions and fuel costs.

Keywords: Propulsion, Marine, Engine, De-rating, Slow Steaming

1. Introduction

Over the years greater concerns have been placed on the effects of emissions on the environment, especially carbon dioxide (CO_2) emissions and its contributing effect to climate change and global warming. With pressure on the marine industry from governments to cap CO_2 emissions, the International Maritime Organization (IMO) introduced the Energy Efficiency Design Index (EEDI), Energy Efficiency Operation Index (EEOI) and Ship Energy Efficiency Management Plan (SEEMP) to MARPOL Annex VI for the monitoring and reduction greenhouse gas (GHG) emissions from shipping.

The formation of CO_2 is closely related to the amount of fuel burnt during the combustion process, with 3.0-3.2 kg of CO_2 being produced per kilogram of marine fuel combusted, depending on the specific fuel being used (Cooper, 2002; Fridell, 2006; Johnsen, 2000; IMO, 2000). The specific emission rates are determined by the fractional weight of carbon contained in the fuel, normally ranging between 86.5-87% (Cooper, 2002). Fuel consumption and, thus, engine efficiency are the main drivers in reducing CO_2 emissions and shipping contribution to GHG emissions.

Slow steaming is increasingly used by ship-owners in times of high fuel prices, low shipping demand and high shipping supply to reduce operational costs. Since the required propulsion power scales over-proportionally with the vessel speed, significant fuel cost reductions can be achieved with only moderate reductions in service speed. The vessel service speed, therefore, provides a useful control variable for ship-owners to maximise income as a response to fuel prices and freight rates.

The propulsion power required in a ship scales approximately with the third order of the vessel speed. As an example, Faber et al. (2010) reported that a speed reduction of 10% results in a reduction of 27% in the required shaft power. Taking into account the increase in voyage time when operating at reduced speed, a rule of thumb is that fuel consumption for a voyage scales approximately quadratic with vessel speed (Faber et al., 2010). Hence, substantial fuel savings can be realised by reducing the operating speed, leading to reduced overall fuel costs and CO_2 emissions for a vessel.

Figure 1 illustrates the required propulsion power for a vessel with a large two-stroke main engine and a fixed-pitch propeller. It is clear that for large reductions in operating speed, major power, and thereby, energy savings are possible. Maersk and MAN Diesel have reported that by reducing the speed of the 8000 TEU container ship Maersk Salalah from 24 to 22 knots and optimizing the MAN B&W main engine for this new operating point, the power demand dropped by approximately 27%. Fuel savings for a voyage would be in the order of 25% (Greenship, 2011).

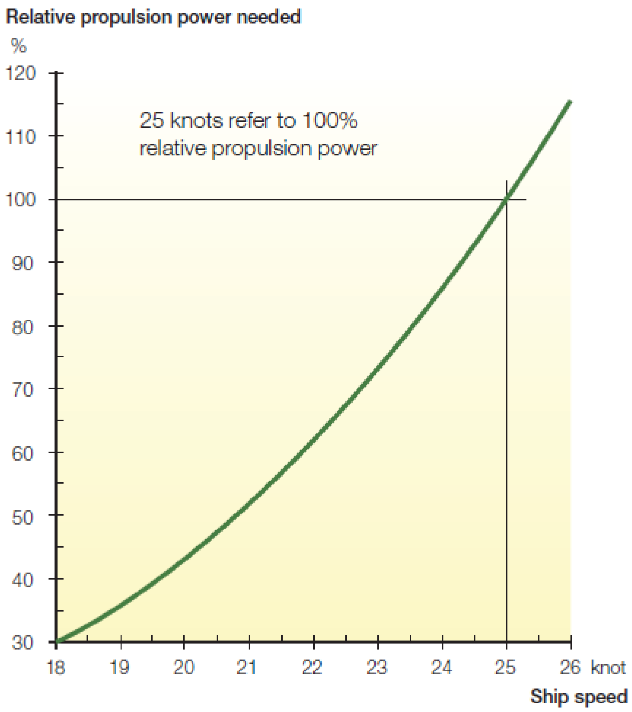


Figure 1: Required propulsion power for a vessel as a function of speed
Source: MAN (2011a)

Figure 2 and Figure 3 show the operating costs for service between Europe and Asia over different sailing speeds. As seen below, reducing the speed gives significant reductions in fuel cost, however with a penalty in voyage time. To maintain the same regularity in the service, more ships are then required, increasing the capital expenditure and (to a lesser degree) operating costs.

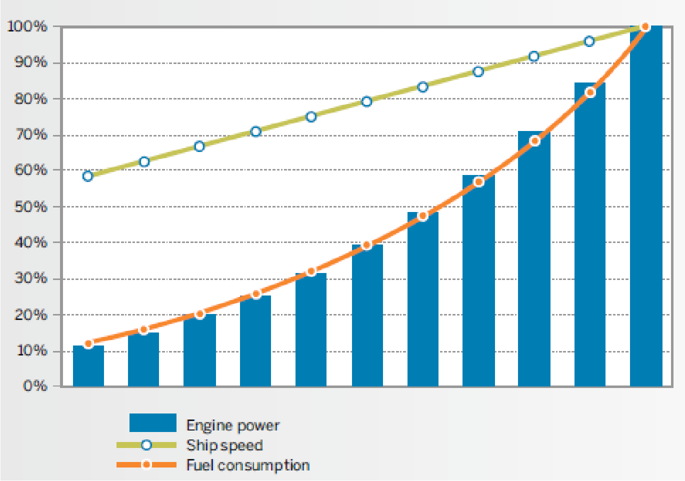


Figure 2: Correlation between ship speed, fuel consumption and engine power
Source: Wiesmann (2010)

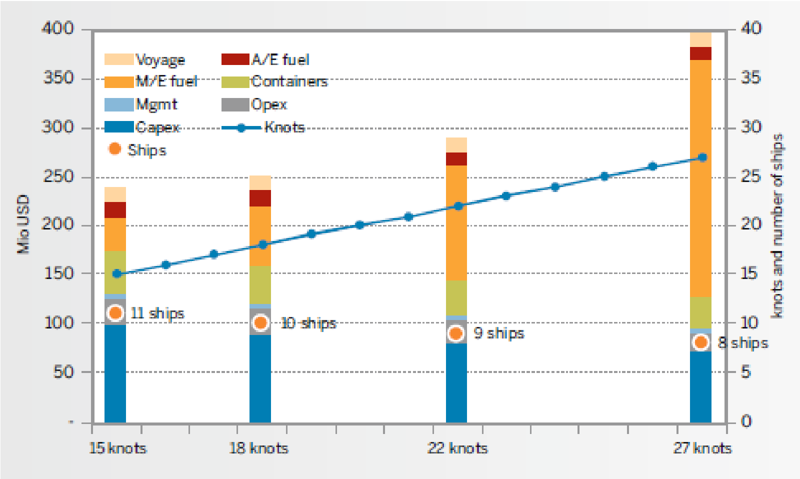


Figure 3: Operating costs for Europe-Asia service for different ship speeds
Source: Wiesmann (2010)

Low-load operation and de-rating of propulsion engines is unproblematic within certain limits around the engine's design operating point. In general, continuous operation down to around 50% MCR is usually possible without significant penalties in fuel consumption or operational risks. For loads below 50% MCR down to around 30% MCR, there is little technical risk but modifications are usually desired in order to avoid heavy fuel efficiency penalties. For loads below 30% MCR, special care is required in order to avoid excessive wear or damage to the engine or auxiliaries.

Radically reducing the service speed of an existing vessel will influence a number of operational variables in the machinery system due to the significant reduction in the required propulsive power. Such major changes will inevitably lead to the propulsion machinery operating outside the design conditions, with potential adverse effects on fuel efficiency, exhaust gas emissions formation, maintenance requirements, etc.

In this work, effects of very low power operation on WÄRTSILÄ engine will be analysed and potential performance-enhancing modifications explored. The theoretical study will be carried out based on a model validated by manufacturer specifications and experimental data. The engine performance under ultra-slow steaming will be simulated and some retrofit options will be studied and potential performance benefits quantified. The effectiveness of the slow speed steaming and the optimisation of the engine to reduce overall fuel consumption will be shown, with respect to the emissions, propulsion and costs.

2. Simulation setup and investigation strategy

The software using for model setup and simulation is DIESEL-RK, a research tool for engine thermodynamic simulation and optimisation. It mainly characterised by a RK-model for calculating multi-zone diesel fuel spray mixture formation and combustion; a multi-parametric and multi-dimensional optimisation tool for simultaneously optimising engine parameters; and a fuel spray evolution visualisation tool which allows researchers to choose the best fuel injection and combustion system parameters (DIESEL-RK, 2012).

The engine chosen for this study is a Wärtsilä 7RT-flex50. It is a two-stroke, seven-cylinder slow speed engine with a cylinder bore of Ø500mm, running on marine diesel oils (MDO) or heavy or residual fuel oils (HFO/RFO), depending on operational conditions and sailing area. Table 1 shows the main specifications of this engine (Wärtsilä, 2010).

Table 1: Main specifications of Wärtsilä 7RT-flex50

Cylinder bore	500 mm
Engine speed	99-124 rpm
Piston stroke	2050 mm
BMEP	19.5 bar
Mean piston speed	8.5 m/s

Test bed data provided by WÄRTSILÄ and operational data provided by Torm from one of their oil/chemical Handymax tankers, the TORM Lilly, such as engine load, engine revolutions, fuel oil consumption, etc., are used as simulation input or for validation. The engine performance is simulated under five different ultra-slow speed-steaming loads, namely, 10% 15%, 20%, 25% and 30% for the extreme de-rating situations. Finally, optimisation is performed regarding some factors that affect the engine performance, such as start of injection (SOI), compression ratio (CR), swirl ratio (SR) and intake port opening (IPO). The test bed data used for modelling was taken under controlled conditions with the engine running on MDO

3. Results and discussions

3.1. Model Validation

Figure 3 and Figure 4 show the comparisons of engine test and the model simulation results. At full load, the simulated engine power and SFC are 10958kW and 180.6g/kWh, respectively, and for 25% load, the simulated engine power and the SFC were found to be 2842kW and 185.5g/kWh, respectively.

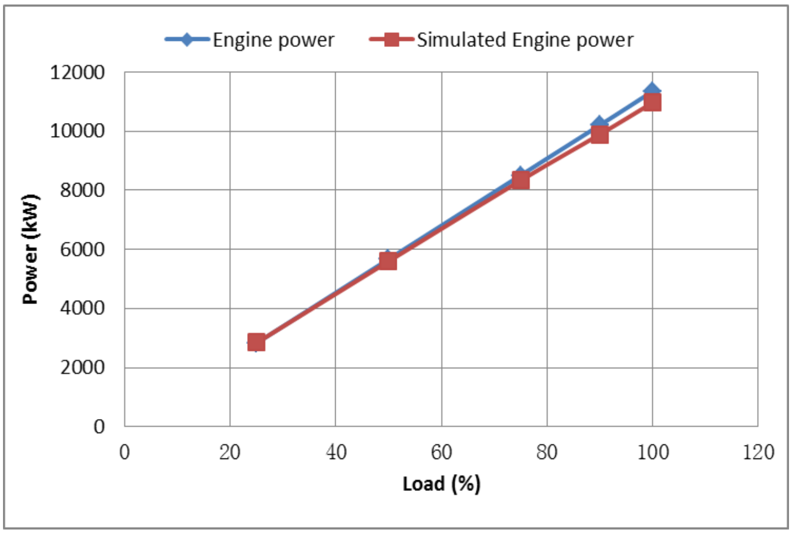


Figure 3: Comparison of tested and simulated engine power

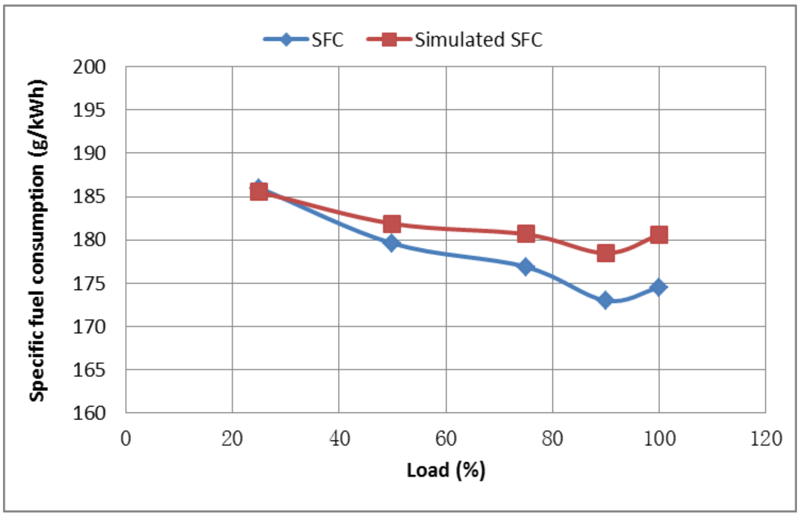


Figure 4: Comparison of tested and simulated SFC

The standard deviation for the difference between the 7RT-flex50 engine test data on-board the Torm Lilly and the model simulation results was calculated. The standard deviation was used to determine accuracy if the model created and the maximum error that could be expected. An error of 5% or less was deemed an acceptable for the engine modelling based on the data available.

Figure 5 shows a plot of the standard deviation between the engine's power as given by the original test bed data and the model simulation results. Exploring the standard deviation, it is clear that the difference between the engine power from the test bed data and the model simulation results is less than 5% over the full operating range, which is well within acceptable error levels. The deviations in engine power ranged between 0.2% at the 25% engine load and -3.4% at full load.

Similarly, the standard deviation between the engine test bed data and the model simulation results for the SFC is shown in Figure 6. The standard deviation for the SFC varied between -0.2% and 3.4%. Again, this is more than acceptable for the model as the variation for each load was less than the desired 5% error. The developed model, therefore, appears more than suitable for exploring engine operational characteristics.

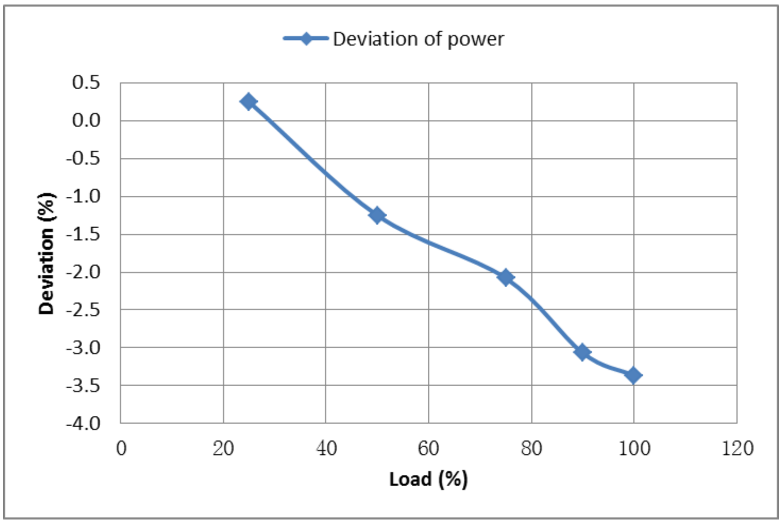


Figure 5: Deviation of improved simulated engine power

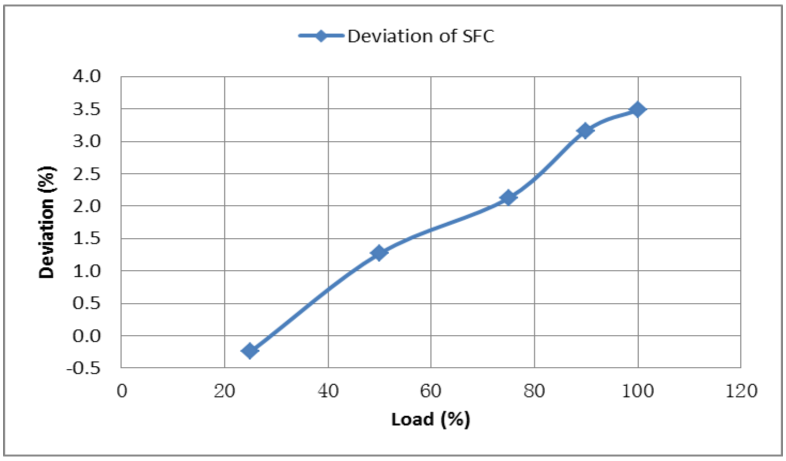


Figure 6: Deviation of improved simulated SFC

3.2. Simulation of ultra-slow speed steaming

For the purpose of slow steaming, the engine operation condition must be set before the simulation is performed. The follow assumptions were made:

- Engine loads. The loads were set between 10% and 30%, with steps of 5 percentage .
- Engine speed. Based on the given experimental data, the relationship between engine load and engine revolution n was determined based on a regression formula:
  n=0.00002×(load×100) ^3-0.0076×(load ×100)^2+1.2891×load×100 +50.301 rpm

- Fuel consumption (FC). It is the required input in DIESEL-RK to determine engine power and for calculating the SFC by dividing FC by the engine power. It is also the dependent of engine load which a theoretical figure is given by another regression formula except 25% load:

  FC= 19.213×load×100+52.243 kg/h

The assumption of engine revolution and engine power related to the engine loads are shown in Table, except those of the 25% load, which are given by the test bed data for the engine.

Table 3: The engine revolution and engine power for slow speed steaming

load	%	10	15	20	25	30
Engine revolution	rpm	62.5	68.0	73.2	78.1	82.7
Engine power	kW	1134	1701	2268	2835	3402
Fuel consumption	kg/h	244.4	340.4	436.5	527.31	628.6

The simulated engine power is shown in Figure 7. It shows the changes in engine power with respect to the engine load for both theoretical and simulated .The engine thermal efficiency decreases, which is reflected by the increasing SFC, as shown in Figure8. The results of engine power are relatively high compared to theoretical ones; with the maximum deviation at 10% load of 4.1%.

At 10% load, the SFC is 12% higher in comparison to a 30% load, being 207.1g/kWh as compared to 184.8g/kWh, respectively. At 25% load, the SFC was compared to the test bed data for the engine of 186g/kWh, matching the expected theoretical SFC. It was found that at this load the simulated SFC was only marginally less than both the theoretical and actual SFC values, with a negligible standard deviation of less than 0.3%. Although the theoretical and simulated SFC standard deviation increases as load decreases, with a standard deviation of 7.4g/kWh (3.7%) at 10% engine load it is still well within acceptable modelling errors.

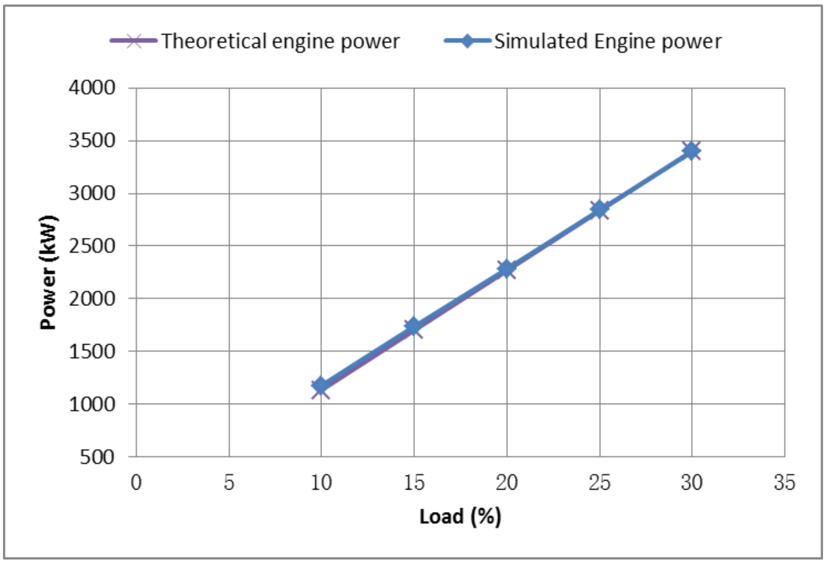


Figure 7: Simulated engine power of slow speed steaming

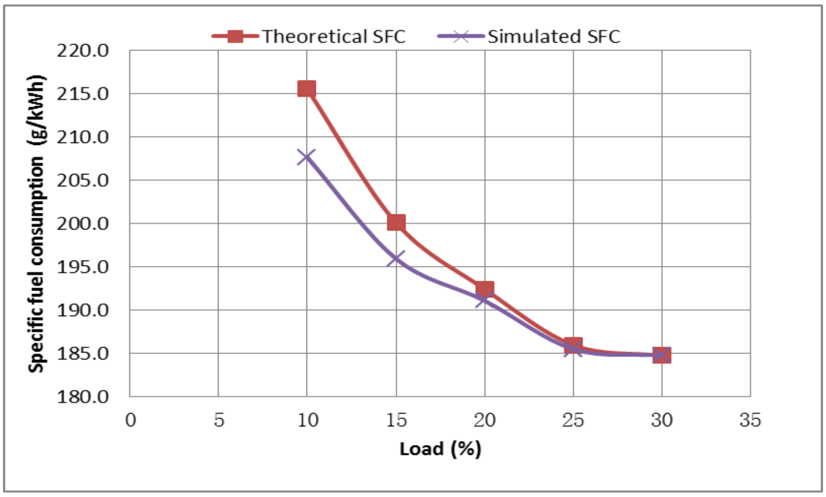


Figure 8: Simulated engine SFC of ultra-slow speed steaming

3.3. Optimisation for slow speed steaming

Usually, the lowest speed at which an engine would run short term on a ship is about 30%. Too low operation speed over longer periods of time will inevitably lead to mechanical problems, reliability deterioration, and lifetime & maintenance issues (MAN, 2011b). In this study, the engine load chosen for the optimisation study is set to 25% load with engine speed of 78.1 rpm.

3.3.1 Fuel injection timing

Injection timing is a major factor influencing the combustion process. The optimisation range is 4° crank angle (CA) after top dead centre (ATDC) to 8° CA before top dead centre (BTDC).

Simulations show that the engine power increases when injection timing increases from 4° CA BTDC, as shown in Figure 9. A maximum being reached between 4° CA BTDC and 5° CA BTDC, after which it drops. As all other operational variables are fixed, the trend in specific fuel consumption mirrors that of the power output, as shown in Figure 10. Delayed injection usually results in the decreasing of maximum cylinder pressure, as combustion takes place at the time piston accelerating down the cylinder, whereas injection too early leads to increased combustion pressure before piston reached top dead centre. The brake mean effective pressure will drop and is reflected in lower engine power output. It should be noted that in optimising fuel injection timing, other factors will also be important, such as in-cylinder mechanical and thermal loading, and emissions formation. As can be seen from Figures 9 and 10, with injection timing of 4° CA BTDC, the power is 2876kW and SFC is 183.3g/kWh, respectively, both improving by 1.2% compared to that of injection timing of 0° CA.

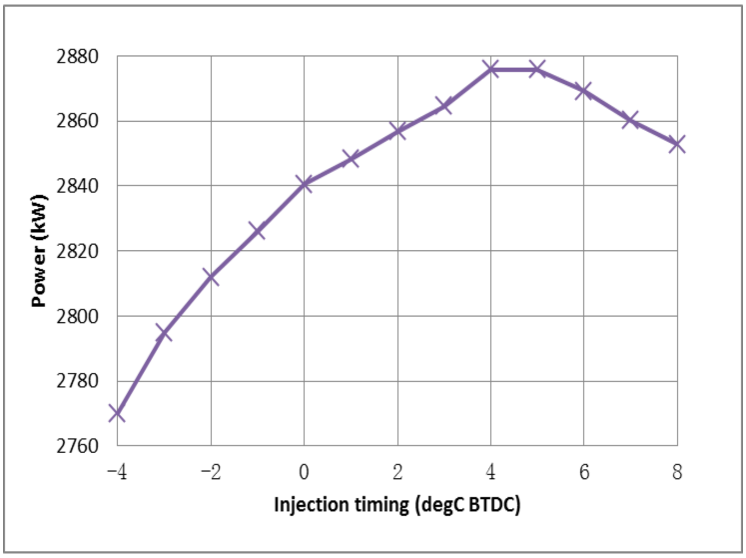


Figure 9: Injection timing dependence in engine power optimisation

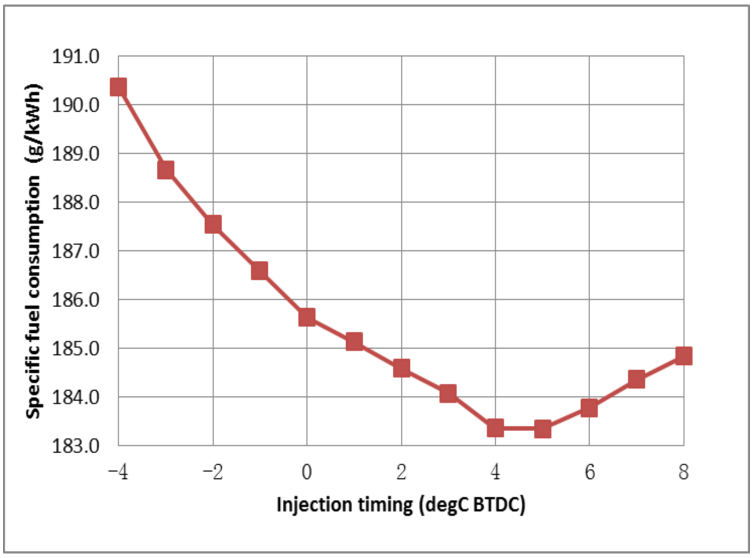


Figure10: Injection timing dependence in SFC optimisation

3.3.2 Compression ratio (CR)

Compression ratio is defined as the ratio of the combustion chamber volume before and after compression. High CR gives a higher theoretical cycle efficiency, but leads to higher combustion pressures and combustion temperature. This increases the mechanical and the thermal loads within the engine, and influences heat losses from the combustion chamber and emissions formation. The optimisation range set for this work is 16-20. The range was chosen to enable reasonable and realistic operation of the engine simulation.

The simulation results, as shown in Figure 11, indicates that as the CR increases from 16 to 20, the engine power increases linearly from 2782kW to 2872kW, and the SFC decreases from 189.6g/kWh to 183.6g/kWh (see Figure). Compared to the default value, the engine performance improved by 1.1% at CR=20. In the RT-flex50 engine, it is possible to increase CR by reducing the thickness of compression shims or increase the height of piston or the length of connecting rod. The former is easy to realise, but has limited effect on the CR. The latter two solutions rely on the redesign of components of the engine, but can increase CR to a higher level.

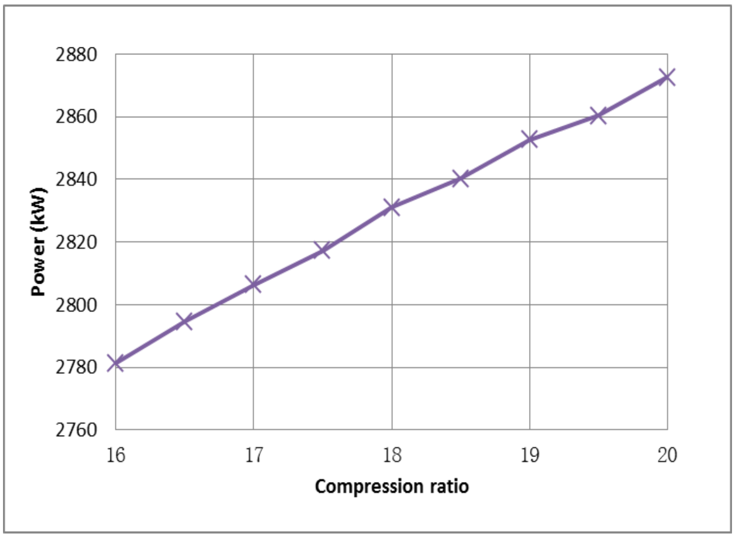


Figure 11: Compression ratio dependence in engine power optimisation

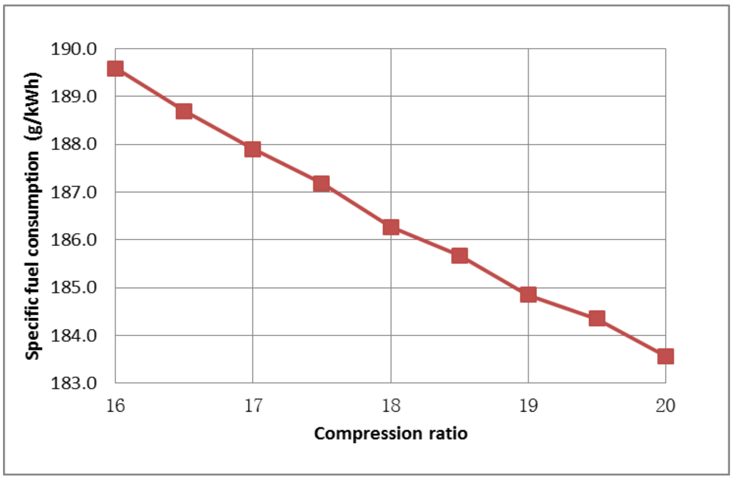


Figure 12: Compression ratio dependence in SFC optimisation

3.3.3 Swirl ratio (SR)

Swirl ratio is the in-cylinder vortex speed relative to engine revolution. It is mainly affected by the shape of air passage/scavenge ports and the piston bowl. This study set the SR to 4 for the original shallow bowl piston, and defined the optimisation range as 2-8.

Usually, a strong swirl ratio improves fuel atomisation and combustion rates in deep bowl combustion chambers. However, too strong a swirl may lead to the overlap of fuel spray, hence, may deteriorate the combustion process. The simulation results show that a weak swirl are suitable for low load operations of the engine, but poses negligible improvement to the shallow bowl piston, as shown in Figure and Figure . The engine performance decreases from SR=2 and reaches to the bottom at SR=6, while the variations in power and SFC are just 6 kW and 0.4 g/kWh, respectively. It can be concluded the optimum SR is 2, but this variable is of less influential on the engine's optimisation.

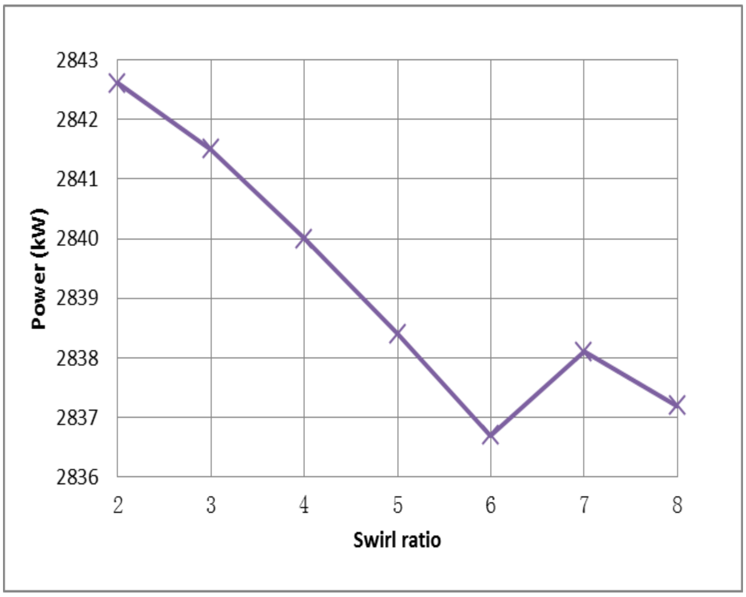


Figure 13: Swirl ratio dependence in engine power optimisation

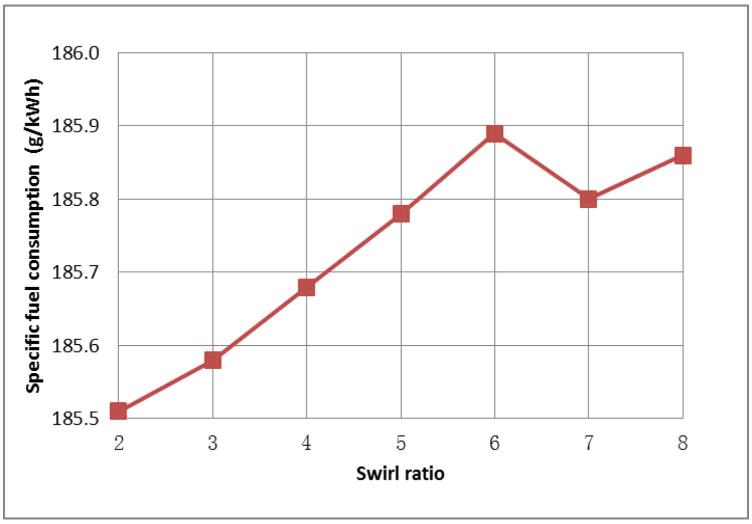


Figure 14: Swirl ratio dependence in SFC optimisation

3.3.4 Intake port opening (IPO)

Intake port opening timing will affect the charge air pressure. The optimisation range used is 26-38° CA BBDC.

Figure 15 and Figure 16 reveal that with the decreasing of IPO timing, the engine performance improves, until it reaches a top at 29° CA BBDC, with the engine power and SFC of 2857kW and 184.6 g/kWh, respectively, both improved by 0.5% compared to the nominal value. This revealed that too early Intake port opened would not help to air charging, as exhaust gas will flow back to the scavenge air cylinder and block the air intake. While the large delay in IPO will result in insufficient air charge, hence will deteriorate combustion.

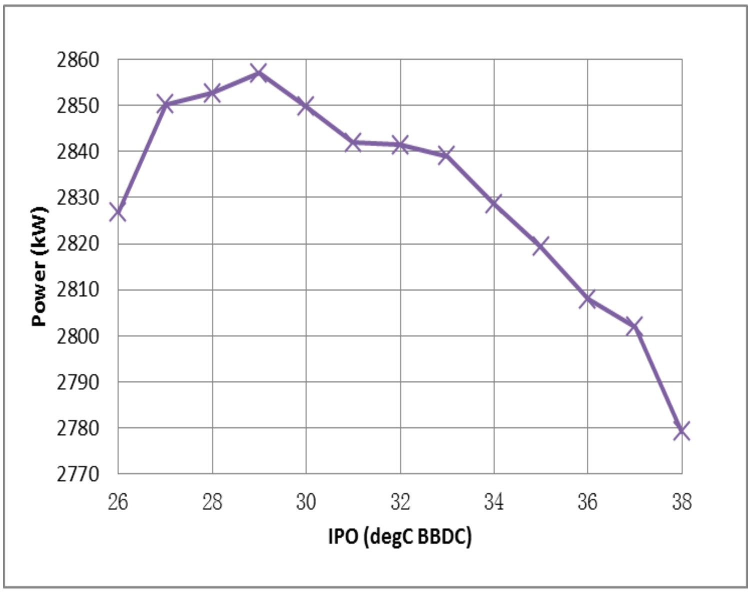


Figure 15: Intake port opening dependence in engine power optimisation

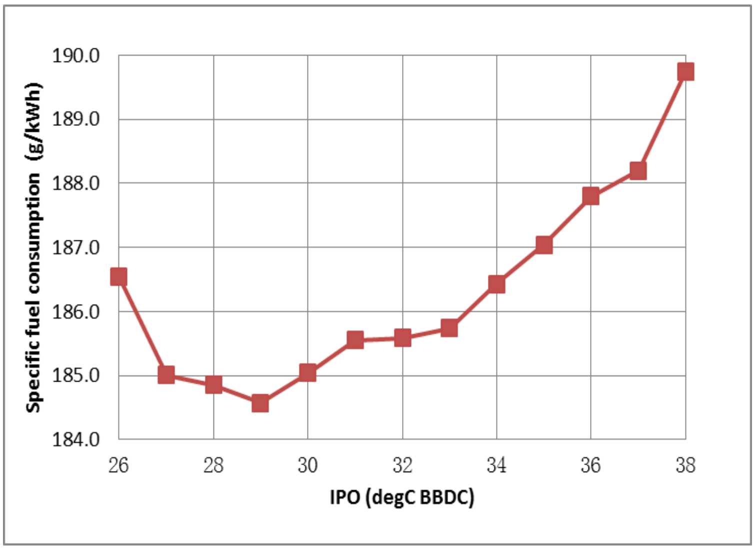
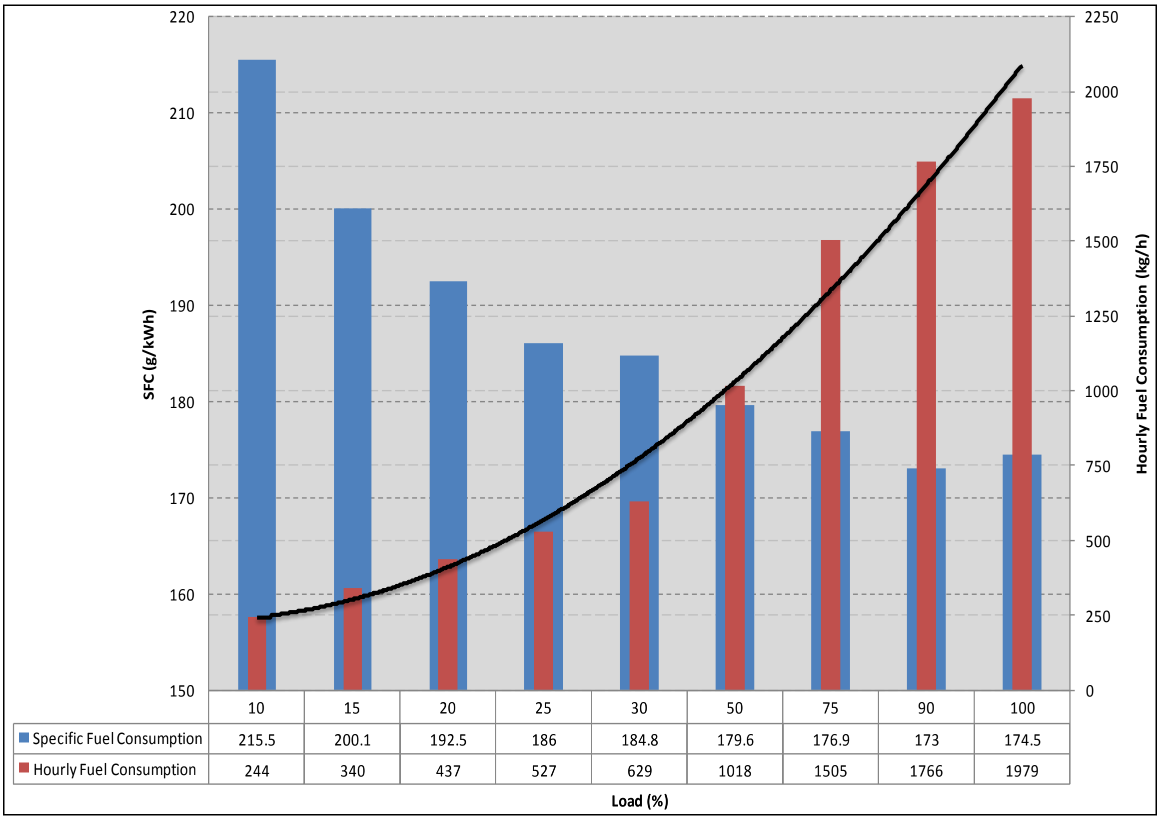


Figure 16: Intake port opening dependence in SFC optimisation

4. Benefits of Slow Speed Steaming

Slow speed steaming has been shown to benefit overall fuel consumption and emission rates over time. From the modelling, the theoretical specific fuel consumption for the engine on-board the reference ship will increase as the load reduces. At low loads the SFC increases as the load decreases. At very low-loads, 10% MCR, the theoretical SFC is 215.5g/kWh, a significant increase of 48.5g/kWh compared to that of the real normal engine-operating load of 80% MCR, increased by 29%. Although, the SFC at very low-loads increases the hourly fuel consumption for the engine will decrease with decreasing load, as shown in Figure 17.



	10	15	20	25	30	50	75	90	100
Specific Fuel Consumption	215.5	200.1	192.5	186	184.8	179.6	176.9	173	174.5
Hourly Fuel Consumption	244	340	437	527	629	1018	1505	1766	1979

Figure 17: The theoretical SFC and hourly fuel consumption plotted with respect to engine load

Form the plot it is clear that the hourly fuel consumption rate of the engine is far less at 10% engine load then that for the nominal operation of the engine at around 85% MCR and, therefore, hourly CO_2 emissions. However, through extreme de-rating, to optimise the engines operation at very low load, this can be further improved as shown in Table 4 and Table 5.

Table 4: A table of SFC and fuel consumption over time for optimised engine settings at 25%MCR

	SFC	Hourly	Daily	Yearly	Annual CO_2 Emissions		Annual Cost ($US)	
	g/kWh	kg/h	Tons	Tons	Low (Tons)	High (Tons)	150 $US/Ton	700 $US/Ton
85%MCR (180 days)	166.6	1606	39	6937	20812	22199	$1,040,595.60	$4,856,112.78
25%MCR (300 days)	186.0	527	13	3797	11390	12149	$569,494.80	$2,657,642.40
FIT (25%MCR = 4^o CA BTDC)	183.3	520	12	3742	11225	11973	$561,227.94	$2,619,063.72
CR (@25%MCR = x20)	183.6	521	12	3748	11243	11992	$562,146.48	$2,623,350.24
SR (@25%MCR = x2	185.5	526	13	3786	11359	12117	$567,963.90	$2,650,498.20
IPO (@25%MCR = 29^o CA BBDC)	184.6	523	13	3768	11304	12058	$565,208.28	$2,637,638.64

For the reference ship, it was estimated that running the engine at 25% MCR as apposed to nominal operation of the engine at 85% MCR, the speed of the ship through the water would be around 9knots compared to 15 knots respectively. This represents a 60% reduction in speed and therefore will increase the sailing times for similar trips from 180 days to 300 days. Even with this increase in trip time, the calculation shows that a significant saving in annual fuel consumption is possible through the operation of the engine at 25% load, of nearly 3200 tons for the same number sailings. Optimisation the fuel injection timing for the engine to 4 ° CA BTDC would further improve annual fuel consumption savings by 50 more tons.

This leads to reduced CO_2 emission and fuel costs. Using a low of 3kg/kg of fuel burnt and a high of 3.2kg/kg of fuel burnt the CO_2 emissions were estimated. For the estimated annual fuel consumption for the reference ship, CO_2 emissions were shown to reduce by nearly 9500 tons for an engine load of 25% MCR. By optimising the engine settings for very low engine loads, the CO_2 emission were shown in the modelling to further reduce by up to 165 tons.

Similarly, savings in the annual fuel costs for the reference are shown to be significant when operating the engine at very low loads. At 25% MCR, the savings are nearly US$500,000 for a low of US$150 per ton of fuel (IFO380, 2005-2011 price low) and US$2.2million for a high of US$700 per ton of fuel (IFO380, 2005-2011 price high). However, the ship would spend more time at sea per trip and, therefore, the number of cargo carrying trips would be significantly reduced. The modelling showed optimisation of the fuel injection timing for the engine the fuel cost saving would be further increased by up to US$50,000.

5. Conclusions

In this paper, a model was established based on a WÄRTSILÄ RT-flex50 for studying the engine performance at slow steaming operation. The model was validated under load conditions from 25% to 100%, using test bed data of the engine. Validation showed that the deviation between the simulation results and the engine data was within ±5%, accurate enough for the further work. The engine power output and the specific fuel consumption, from 10% to 30% loads, were simulated and optimised for a 25% load. The conclusions from the subsequent modelling results are:

- Start of injection (SOI) and compression ratio (CR) gave a significant improvement in engine performance. By adjusting SOI to 4 or 5° CA BTDC, the engine's performance improved by 1.2%, and for a CR = 20 an improvement of 1.1% on default engine setting could be achieved in the model simulations.

- Limited effects were found for changes in swirl ratio (SR) and inlet port opening (IPO). A weak swirl ratio gave negligible improvement to the shallow bowl piston and the engine performance increased slightly (by 0.5%) when IPO was set to 29° CA BBDC.
- Through the modelling and optimisation of the engine at very low load, it can be shown that improved annual fuel consumption savings can be made by optimising the fuel inject timing and compression ratio. This in turn leads to reduced CO_2 emissions and fuel costs.

6. Acknowledgements

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