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System Modelling of Organic Rankine Cycle for Waste Energy Recovery System in Marine Applications

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Abstract

Recent regulatory developments in the maritime industry will hasten the shift in usage of conventional marine fuels like HFO and MDO to a cleaner fuel like Liquefied Natural Gas (LNG) to meet its ambitious target of lowering carbon dioxide and other noxious gases emissions. Efficient use of energy and waste heat recovery and onboard a ship will help the industry meet this target in the meantime. This paper presents the findings from the modelling of a dual fuel marine diesel engine and ORC system cooled by vaporising LNG and simulation using a system engineering software, Siemens Simcenter Amesim 16. Engine manufacturer's design data is used as inputs to run the ORC systems running on two working fluids, n-heptane and n-octane to derive the net work and thermal efficiency when installed on a LNG-fuelled Platform Supply Vessel. The ORC system running on n-heptane is found to provide an annual fuel savings of 7% with an estimated payback period of 2.7 years, making it an attractive option for shipowners.

Keywords: Organic Rankine Cycle; ORC; waste energy recovery; system modelling and simulation; LNG-fuelled ship

1. Introduction

Increasing decarbonisation and environmental awareness in the maritime industry is changing how ships will be powered in the future. International Maritime Organisation (IMO) is taking steps as the global regulator to implement measures to control Greenhouse Gases (GHG) and other noxious emissions from the marine industry. In an ambitious

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move, IMO agreed recently on an initial strategy to cut shipping's total greenhouse gases (GHG) by at least 50% from 2008 levels by 2050.

In response to the new regulations, alternative marine fuels like LNG and LPG are touted to replace existing widely-used marine fuel like Heavy Fuel Oil (HFO) and Marine Diesel Oil (MDO). In a report by DNV GL, between 40-70% of the world's ships is predicted to be powered by LNG or LPG by 2050 depending on trade growth scenario [1].

LNG-fuelled ships emit up to 20% less GHGs and negligible sulphur oxides (SOx) and is an ideal solution to reduce carbon dioxide and other noxious gas emissions. Till end of 2017 and excluding gas carriers, there are already 118 ships operating on LNG and a further 123 new LNG-fuelled ships are on order. It is widely expected that LNG will become the marine fuel of choice once most ports are able to provide LNG bunkering when ships call.

One problem facing LNG-fuelled ships is that the liquefied fuel needs to be heated up to its gaseous form and this process requires additional energy that can take up around 5% of the diesel engine rated power. At the same time, abundant heat energy is wasted in the engine exhaust. A possible solution will be to exploit the temperature difference between these heat source and sink in an energy recovery system running on Organic Rankine Cycle (ORC) which is heated by engine exhaust and cooled by vaporising LNG.

Comparison is drawn to the reference ORC system heated by engine exhaust and cooled by seawater. Two probable working fluids, n-Heptane and n-Octane are used for the ORC systems installed onboard a typical LNG Platform Supply Vessel (LNG-PSV).

1.1. Marine ORC with waste heat and LNG cold energy recovery

Research on the application of ORC systems onboard ships had been summarised recently by Mondejar et al. [2] Among the many ideas, is one of an energy recovery system that reuse the engine waste heat and LNG cold energy. However, the concept has not been thoroughly examined and there are only a few instances of related research in the literature.

Sung and Kim proposed an dual-loop ORC onboard a LNG carrier where one loop recovers waste heat from exhaust gas from dual-fuel engines and another loop recovers waste heat from jacket cooling water, LNG fuel and boil-off gas [3]. With n-pentane and R125 as working fluids, they determined the maximum work output to be 5.17% of the engine output.

Zhang et al. developed a multi-objective optimisation model for a combined system that consists of three ORC systems and compared it with separated systems and found that the combined system provides more net output power, lower investment cost and larger heat availability factor [4].

More recently, Tsougranis and Wu suggested a dual utilisation of LNG cryogenic energy and thermal waste energy onboard a LNG-fuelled ferry using a regenerative ORC system running on isobutane with direct expansion of LNG and calculated thermal efficiency of 48% achieved by high vacuum in condenser between -110°C and 330°C [5].

Hence, the application of waste heat and LNG cold energy ORC system has not been investigated for a LNG-PSV and this will form the main focus of this paper.

2. Modelling and simulation

2.1. The case ship – LNG-fuelled PSV

Offshore service vessels (OSVs) refers to a class of ship that serves the offshore oil and gas industry and can include diverse operations like platform supply, anchor handling, construction, fire-fighting, stand-by and rescue, diving support, accommodation etc. Due to different needs, their load patterns differ widely during course of operations.

As a case study to examine the thermodynamic performance of ORC in marine application, we have selected our case ship to be a Wartsila-designed 4,800dwt LNG- PSV with the principal specifications of the ship and engines shown in Table 1 [6].

The four generators onboard are driven by dual-fuel diesel engines capable of running on diesel and gas modes. During the gas mode, the engines operate on lean-burn Otto process whereby the premixed air-fuel mixture is inducted into the combustion chambers and ignited by a small quantity of diesel pilot fuel. For diesel mode, the normal diesel process applies.

Table 1 Ship and engine specifications

Ship specifications:			Engine specifications:	
Length, overall	:	89.6 m	Diesel Generators	: 2 x 3000kW Wartsila 6L34DF
Breadth	:	21 m	(DG1~4)	: 2 x 960kW Wartsila 6L20DF
Depth	:	9.6m		
Design speed	:	16.4 knots		

LNG is stored in one cryogenic tank of 220m^3 located in the cargo hold section at -163°C and 4 bars. Before being injected into dual fuel engines, it needs to be pumped and vaporised into gaseous state at 6.55 bars and 60°C. Currently, the LNG vaporiser is heated by a glycol/water circuit which is in turn heated by an electric heater. In this usual arrangement, the cold LNG energy is wasted with additional energy being incurred for the glycol/water circulating pump and electric heater. Instead, the vaporising LNG can be used as a heat sink for an ORC that will improve overall thermal efficiencies.

In order to derive the fuel gas consumption and potential energy recovery, the following LNG-PSV operating profile is developed and used as basis for simulation input.

Table 2 PSV operating profile and engine loads

Operation Mode	Units	Steaming 15 knots	Steaming 12 knots	Transit low 7-8 knots	Dynpos Heavy	Dynpos Light	Standby	Harbour
% of operating hours		20%	15%	5%	10%	30%	10%	10%
Operating hours	h	1680	1260	420	840	2520	840	840
Load on DG 1	3000kW	100%	85%	75%	75%	75%	0%	0%
Load on DG 2	3000kW	100%	0%	0%	75%	75%	0%	0%
Load on DG 3	960kW	0%	85%	75%	75%	0%	0%	0%
Load on DG 4	960kW	0%	0%	0%	75%	0%	75%	50%

2.2. ORC-Waste Energy Recovery System

The proposed layout of the ORC system and the reference system is shown in Figure 1. To ease installation in the ship's engine room, a simple ORC system is chosen that consists of an exhaust gas-heated evaporator, turbo-generator, condenser (vaporiser) and a feed pump. An exhaust bypass is provided as the LNG flow rate is not sufficient to cool all the exhaust heat and for safety purpose in case of ORC breakdown.

Exhausts from the engines, typically 300-400°C depending on load, is modelled using exhaust gas data in the software's Pneumatic library. The ORC and LNG lines are modelled using components from the Two-Phase-Flow library due to phase changes in the circuit.

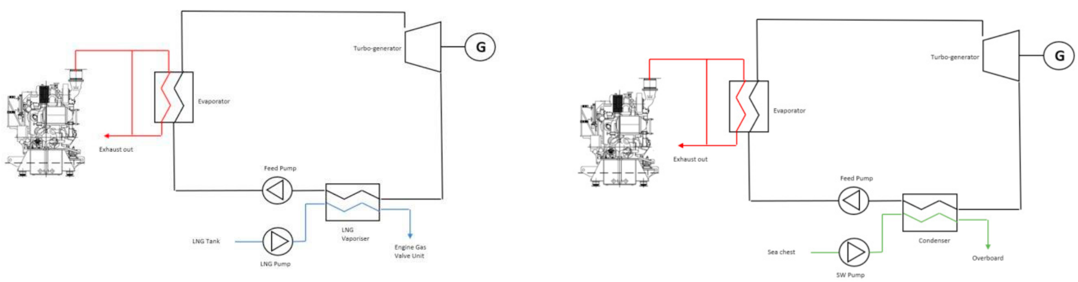

Figure 1 Proposed LNG cooling (left) and reference Seawater cooling (right)

The two organic fluids, n-Heptane and n-Octane are chosen as that they exhibit high thermal stability meaning that it can withstand higher heat sources without thermal decomposition, hence the maximum temperature in the ORC which occurs at evaporator outlet is limited to about 300°C.

As shown in Table 3 below, the selected working fluids although are non-toxic, special precaution will need to be in place to handle them as their flashpoints are below the typical engine room temperature. However, the level of safety precautions will be similar for usage of LNG as fuel already stipulated in marine classification rules.

Table 3 Comparison of properties of organic fluids

	Formula	$T_{freezing}$ (°C)	$T_{boiling}$ (°C)	T_{crit} (°C)	P_{crit} (bar)	Flashpoint (°C)	Autoignition temperature (°C)
Methane	CH_4	-182.46	-161.49	-82.59	44.08	-187.2	536.9
n-Heptane	C_7H_{16}	-90.58	98.43	266.98	27.36	-4.1	203.9
n-Octane	C_8H_{18}	-56.77	125.68	296.17	24.97	12.9	205.9

The evaporation pressure is set at 9bara taking into account that the evaporating temperature should not exceed the autoignition temperature for safety reasons. The condensing pressure are 0.01bara and 0.1bara for the proposed and reference cycle respectively due to the temperatures of the cooling medium.

For the LNG stream, a cryogenic pump take suction from LNG tanks at -163°C and 4bara and supply to the condenser at 6.5bar. After vaporisation, the gaseous LNG at 60°C is led to the Gas Valve Unit (GVU) which regulates the flow pressure before being combusted in the diesel engines.

For the seawater cooling stream, the inlet temperature is taken to be 32°C and outlet temperature is 45°C.

The heat exchangers i.e. evaporator and condenser are modelled using the efficiency-NTU method which is geometry independent and hence is useful for early design phase when equipment details are not known. By this method, the heat transfer rate between the cold fluid and the hot fluid cannot exceed a maximum value, Q_{max}:

$$Q_{max} = C_{min}.(T_{hot,in} - T_{cold,in})$$

where C_{min} is the minimum of the heat flow rate capacity, $T_{hot,in}$ and $T_{cold,in}$ are inlet temperatures at the heat exchanger hot and cold side respectively.

$$C_{min} = min(|\dot{m}_{hot}|.C_{p,hot}, |\dot{m}_{cold}|.C_{p,hot})$$

where $\dot{m}$ is the mass flow rate and C_p is the specific heat capacity.

Hence, efficiency-NTU method states that heat flux can be defined as below:

$$Q = \epsilon.Q_{max}$$

where ϵ is the efficiency or thermal effectiveness of the heat exchanger.

The pumps and expander are modelled with efficiency characteristics of the fixed positive displacement machines i.e. volumetric, isentropic, mechanical efficiencies are set at 80%, while electrical efficiency is assumed at 95%. The correlations for volumetric and isentropic efficiencies are stated below:

Table 4 Relations for volumetric and isentropic efficiencies

	Pump	Turbine
Volumetric efficiency, η_{vol}	$\eta_{vol} = \dot{m} / \rho_s.N.\Delta$	$\eta_{vol} = \rho_s.N.\Delta / \dot{m}$
Isentropic efficiency, η_{is}	$\eta_{is} = (h_{4s} - h_{3s}) / (h_4 - h_3)$	$\eta_{is} = (h_2 - h_1) / (h_{2s} - h_{1s})$

where η_{vol}=volumetric efficiency, $\dot{m}$=mass flow rate, ρ_s=suction density, N=rotational speed, Δ=displacement

η_{is}=isentropic efficiency, (h_{4s}-h_{3s})=isentropic enthalpy increase across pump, (h_4-h_3)=actual enthalpy increase across pump, (h_2-h_1)=actual enthalpy drop across turbine, (h_{2s}-h_{1s})=isentropic enthalpy drop across turbine

Standard components in the Simcenter Amesim are built using correlations found in the literature and validation of models are done via these researches. Furthermore, at the system level, Siemens collaborated closely with industry partners like Renault to ensure models are also validated at the system level.

3. Results and discussions

3.1. Analysis of Cycle Efficiency and Net Work Output

When comparing the performance of different working fluids in ORC, thermal efficiency, η_{th} and net work output, W_{net} are two important measures and they are derived as follows:

$$\eta_{th} = \frac{W_{turbine} - W_{pumps}}{Q_{in}}$$

$$W_{net} = W_{turbine} - W_{pumps}$$

where $W_{turbine}$=work output from turbine, W_{pump}=work input for pump, Q_{in}=heat input

Firstly, the condensing pressure is varied between 0.01bara to 10bara while keeping the evaporating pressure at 9bara as mentioned before (see Figure 2). It is found that n-Heptane has higher thermal efficiency and net work output than n-Octane. Also, when condensing pressure is reduced, the thermal efficiency and net work output is increased. For condensing pressure of 0.01bara, the thermal efficiency is 23.46% and net work output is 176.18kW/kg for ORC running on n-Heptane.

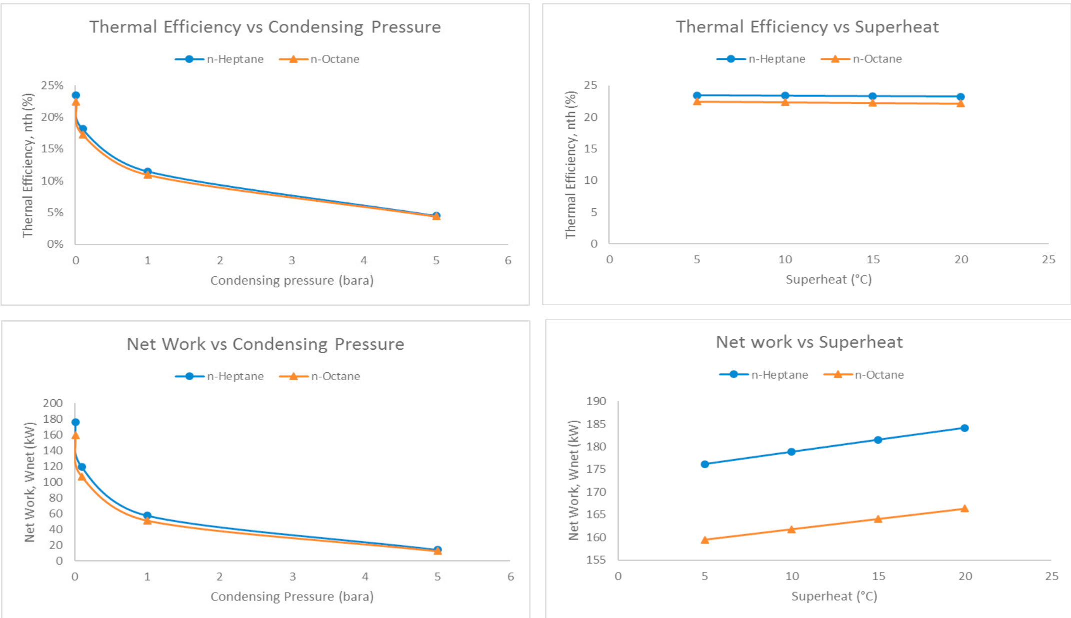

Figure 2 Cycle efficiency and net power output varying condensing pressure (left) and superheat (right)

A second analysis is performed to see how the extent of superheat at the evaporator outlet affects the two measures for each working fluid for a condensing pressure of 0.01bara (see Figure 2). It can be seen that n-Heptane offers higher thermal efficiency and net work output. With increasing superheat, the decrease in thermal efficiency is little (~1%) while net work output increases by about 5%.

From these results, it can be seen that n-Heptane offers higher efficiency and net work output than n-Octane. A lower condensing pressure also promotes higher efficiency and net work output. Increasing superheat has little effect on thermal efficiency but increases the net work output, but this is limited by the autoignition temperature of the fluid.

3.2. Deriving the fuel savings and payback time when ORC is installed on LNG-PSV

In this part of the study, n-Heptane is used as the working fluid in the two ORC configurations shown in Figure 1 due to its superior thermodynamic performance. For the LNG cooling case, the condensing pressure is fixed at 0.01bara while for reference seawater cooling case, it is set at 0.1bara to allow heat exchange to take place considering second law of thermodynamics.

For each of the engine load of 50%, 75%, 85% and 100%, the mass flow rate of working fluid is obtained by the following relations. The reason for the two equations is that mass flow rate of working fluid is governed by the heat transfer in the LNG vaporiser for LNG cooling case and in the exhaust gas evaporator for the seawater cooling case.

With the mass flow rate obtained, all the cycle information like heat input, heat output and net work can be calculated and the overall Power Savings can also be derived.

The relations used to find working fluid mass flow rate, m_{orc} and power savings, P_{sav} is shown below:

Table 5 Relations for working fluid mass flow rate and power savings

	LNG cooling case	Seawater cooling case
Working fluid mass flow rate, m_{orc}	$m_{orc} = \frac{m_{LNG}\Delta h_{vap}}{\Delta h_{cond}}$	$m_{orc} = \frac{m_{exh}\Delta h_{exh}}{\Delta h_{evap}}$
Power savings, P_{sav}	$P_{sav} = P_{wnet} + P_{vap}$	$P_{sav} = P_{wnet} - P_{SW\ Pump}$

where m_{LNG}=mass flow rate of LNG, Δh_{vap}=enthalpy increase from LNG vaporisation, Δh_{cond}=enthalpy drop across ORC condenser, m_{exh}=mass flow rate of exhaust, Δh_{exh}=enthalpy drop from exhaust gas, Δh_{evap}=enthalpy increase across ORC evaporator, P_{wnet}=Net work output from ORC, P_{vap}=Power input to heat up vaporiser, $P_{SW\ Pump}$=Power input to seawater pump

A summary of the power savings for each engine load and engine model is presented in Table 6.

Table 6 Summary of power savings at different engine operating loads for each engine model

	LNG cooling case (p_2=0.01bara)		SW cooling case (p_2=0.1bara)	
Engine model	6L34DF	6L20DF	6L34DF	6L20DF
Engine Load	P_{sav} (kW)	P_{sav} (kW)	P_{sav} (kW)	P_{sav} (kW)
100%	200.91	70.04	119.44	32.28
85%	174.01	60.94	114.07	34.08
75%	157.94	54.35	109.57	34.43
50%	114.70	38.56	89.83	31.56

In order to further obtain the total energy savings, E_{sav} in a year, the P_{sav} is multiplied by the number of hours the LNG-PSV spends at the load based on the operational profile shown in Table 2 and summed up for each configuration. The specific fuel gas consumption of 7500kJ/kWh is multiplied to E_{sav} to find out the total fuel gas quantity (in MJ) that could have been saved per year. This is converted to British Thermal Units (btu) and based on recent fuel gas price of USD$9.74/mmBTU [7], the fuel savings per year is $158,630 for LNG cooling and $103,727 for seawater cooling and with a typical annual fuel cost of about $2.3 million per year, the fuel savings take up 7.1% and 4.6% respectively.

Table 7 Fuel savings from ORC

	Energy savings, E_{sav} (kWh)	Gas quantity (MJ)	mmBTU	Fuel Savings/year	% Fuel Cost/year
LNG Cooling	2,290,956	17,182,173	16,286	$158,630	7.1%
SW Cooling	1,498,035	11,235,266	10,649	$103,727	4.6%

Song [8] managed to compile a list of ORC module costs against Net Power Output collected from non-exhaustive set of ORC manufacturers and from published scientific publications. From the graph, for an ORC with a Net Power Output of 100kW, the unit cost is expected to be about USD$3500/kW. This will lead to CAPEX of about USD $420K for LNG cooling and USD$1.26 million for seawater cooling configuration. Hence, a simple payback time of 2.7 and 12.2 years is expected for the installation of ORC using LNG cooling and seawater cooling respectively (see Table 8). Hence, the ORC cooled by LNG could be an economically feasible option for shipowners.

Table 8 Estimated capital cost and payback time

	Installed Power (kW)	Capital cost	Payback (years)
LNG Cooling N-Heptane	120	$420,000	2.7
SW Cooling N-Heptane	360	$1,260,000	12.2

4. Conclusion

This paper discusses the possibility of installing an Organic Rankine Cycle (ORC) system onboard a LNG-fuelled Platform Supply Vessel (LNG-PSV) with waste energy recovery from engine exhaust and cold energy from the vaporising LNG fuel.

Two organic fluids, n-Heptane and n-Octane are used as working fluids for the ORC and is modelled and simulated in a commercial-off-the-shelf 1D system simulation software known as Simcenter Amesim which saves time building model from scratch and is able to perform transient simulations.

Comparing with the reference case of ORC cooled by seawater, the proposed configuration cooled by LNG running on n-Heptane as working fluid showed superior thermodynamic performance and greater fuel savings per year of about 7%.

When considering the capital costs of installing ORC system onboard, payback time of about 2.7 years is derived and becomes an attractive option for shipowners with LNG-fuelled ships as waste heat from engine exhaust can now be used to vaporise the LNG and provide additional electrical power.

In conclusion, ORC systems can be a viable means of achieving higher energy efficiency onboard ships to meet tightening environmental regulations and also reduce carbon emissions.

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