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CO_2 refrigeration system design and optimization for LNG driven cruise ships

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ABSTRACT

The cruise industry is in the evolution of eco-friendly technologies due to strict environmental regulations. Liquified natural gas (LNG) is an alternative marine fuel. Compared to conventional diesel fuels, LNG offers a reduced environmental impact and can serve as a transition towards zero emission. LNG is stored in onboard cryogenic tanks at low temperatures. Various techniques can be applied to vaporize the LNG fuel before feeding it to the gas engine. The recovery of this vaporization energy with air conditioning as a heat source can enhance system performance. On a cruise ship, heating, cooling, and ventilation (HVAC) require an average 40 % of the ship's total energy demand. The natural refrigerant CO_2 is an attractive choice due to its compact units, non-toxic nature, and non-flammability, all being primary concerns on a cruise ship. The energy efficiency can be improved by utilizing LNG cold and reducing the need of indirect loops with CO_2 as a refrigerant. This work investigates the LNG cold, waste heat recovery potentials, and CO_2 refrigeration system for cruise ships.

Keywords: Refrigeration, Carbon Dioxide, Cruise Ships, Energy Efficiency, LNG, Cold Recovery

1. INTRODUCTION

The traditional heavy fuel oil (HFO) marine engines are associated with large amounts of sulphur oxides (SO_x), nitrogen oxides (NO_x), and greenhouse (GHG) emissions from exhaust gases (Zhang et al., 2021). The international maritime organization (IMO) leads the UN sustainable development goals of the shipping sector. It aims to reduce CO_2 emissions by 40 % by 2030 and 70 % by 2050 compared to 2008, and also to limit SO_x and NO_x pollution by setting different regulations (IMO, 2021). To comply with the new rules, alternative propulsion systems, fuels and exhaust gas cleaning technologies are under investigation. Among various options, liquified natural gas (LNG) is a favorable marine fuel with 25 % less CO_2, 90 % less NO_x, and about no SO_x as compared to conventional fuel oils (Jafarzadeh et al., 2017). LNG is a reasonable choice to meet the current and upcoming environmental regulations without a significant technical modification.

LNG is stored under cryogenic temperatures between -165 °C and -138 °C, and vaporization/heating is essential before combustion in the gas engine. The recovery of vaporization energy ("cold recovery"), for onboard applications can reduce emissions and overall energy consumption to some extent (Baldasso et al., 2020). Prior studies have extensively investigated LNG cold recovery applications in regasification terminals, including power generation (Gomez et al., 2014), air separation, gas turbine suction air cooling and hydrocarbon liquefaction (Otsuka, 2006), freezing and refrigeration (Dispenza et al., 2009), production of liquid CO_2/dry ice (Hongyu et al., 2010), cryogenic comminution (Lian et al., 2015), seawater desalination (Dhameliya et al., 2015). Recently, the LNG cold recovery researched areas are broadened to transportation sector like LNG-driven refrigerated vehicles (Tan et al., 2010) and fishing vessels (Saeed et al., 2020). Few studies have investigated LNG-driven passenger ships. The common technology evaluated by most authors (Sung et al., 2016, Pasini et al., 2019, Han et al., 2019 and Koo et al., 2019) is the Organic Rankine cycle (ORC) for cold recovery and flue gases heat recovery. Baldasso et al., 2020 analyzed the cold recovery for ferry with 16030 kW engine for HVAC. They reported the primary engines fuel savings of 0.43 % to 0.84 %

with refrigeration system COP of 3 and 5, respectively. However, their work did not discuss the design of HVAC system.

For passenger ships, current refrigerants in use are R22 and HFCs (globally) and CO_2 for provision cooling and freezing (Hafner et al., 2019). Natural refrigerants are a long-term solution for the shipping industry due to numerous international and national regulations for the use of high GWP (Global warming potential) refrigerants. The usage of these refrigerants is already complicated due to the F-gas regulation and high environmental taxes. Due to such reasons, shipowner companies have a high interest in low GWP refrigerants (Pigani et al., 2016). Zhang et al., (2021) theoretically investigated an integrated trans-critical CO_2 cycle for onboard heating, cooling, engine waste heat recovery. The high-pressure side of CO_2 system was integrated with waste heat, and work was recovered by expansion turbine. Their findings showed operational flexibility and efficient performance, but they suggested more theoretical research and experimental studies. This paper aims to numerically investigate the direct expansion of CO_2 refrigerant in the air-handling units or air conditioning coils for summer mode and the energy-saving potential with LNG cold recovery.

2. METHODS AND DATA

2.1. Reference case

This paper's reference case is a hypothetical cruise ship with a passenger capacity of 330 (excluding crew members) and total engine power of around 8 MW. The operating parameters were adopted from two commercial ships (MS Spitsbergen 2021 and MS Fram 2021) and engine manufacturer (Wärtsilä, 2021) under the following assumptions. The engines were considered dual fuel and equipped with a low-pressure LNG fuel supply system. The fuel consumption of LNG was calculated from its lower heating value (LHV) and a constant engine efficiency. The fuel consumption can also be calculated by specific gas consumption (SGC), but SGC varies with engine load and the data is currently unavailable. So, LHV method was used for approximation. The flow rate of exhaust gases was calculated using a lean air-fuel ratio, and the exhaust temperature was assumed constant. The assumptions were validated by comparing them with the existing available data referred to in introduction of this paper. Table 1 shows the boundary conditions for the reference case.

Table1. Boundary conditions for the reference case

Parameters	Description	Parameters	Description
Engine type	4*dual fuel engines	LNG pressure (bar)	8
Engine rated power (kW)	1980	Lower heating value LNG (MJ/kg)	48.6
Average engine load (%)	85	Engine efficiency (%)	46
LNG supply system	Low pressure	LNG flow (kg/h)	1084
LNG tank	Type C	Exhaust flow (kg/s)	9.15
LNG temperature (℃)	-128.7	Exhaust gas temperature (℃)	365
Inlet temperature to engine (℃)	10		

2.2. LNG cold and exhaust heat recovery

The two standard methods of transferring LNG from the fuel tank to the engine are pressure build-up and cryogenic fuel pump systems. The system described in Fig. 1 is the fuel pump system. For safety reasons, two parallel fuel systems are common for ships with four engines. LNG in the tank (1 bar and -162 ℃) is pressurized to 8 bar by the cryogenic pump and then vaporized in the cold box. The gas valve unit acts as a small storage tank and maintains a smooth flow of gas to the engine. A heat source is required to vaporize the LNG in a cold box. The cold recovery was calculated by the enthalpy difference of methane between -128 ℃ and 10 ℃ under adiabatic conditions. The superheating level of fuel gas was adopted from the engine manufacturer data (Wärtsilä, 2021), which required gas temperature in the range of 0 to 60 °C before the engine. In traditional systems, the engine oil cooling circuit is used as a heat source, but it can be replaced by air conditioning system secondary loops. This can maximize the use of cold in an efficient way.

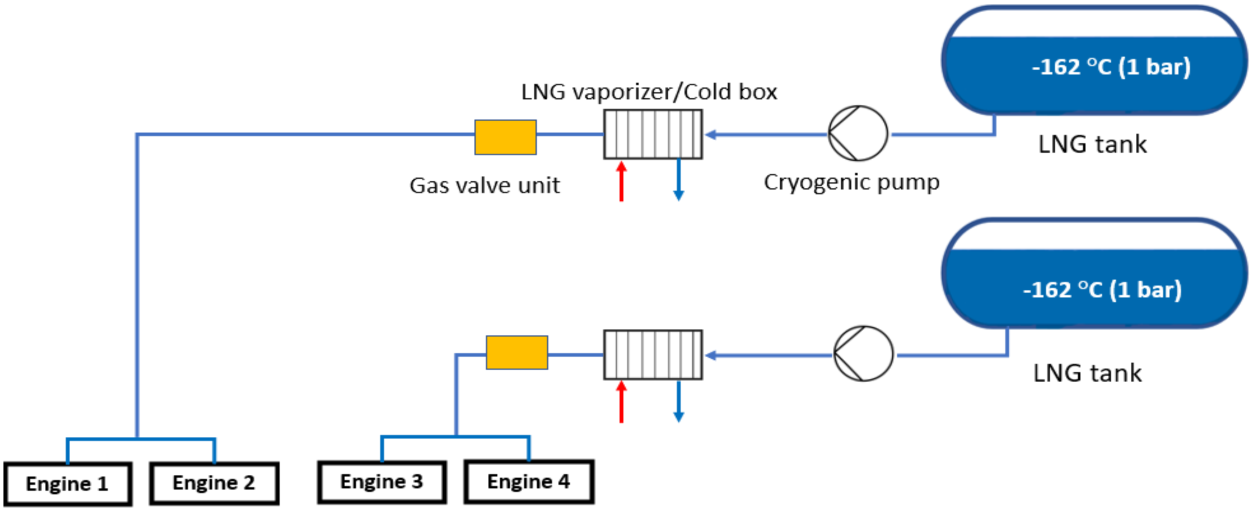

Figure 1: Cryogenic fuel pump system

The engine's exhaust gases contain a considerable amount of heat at high temperatures (at around 350 °C) that can cover heating demands onboard. The values were calculated by using constant specific heat capacity of air and cooling of gases from 365 °C to 120 °C. The LNG/gas has negligible sulphur content, and heat recovery can be made at lower temperatures without acid formation compared to diesel fuel. In warmer climatic conditions, the excess heat can be used to produce cooling by applying absorption cooling technology. Other heat recovery methods are electricity production with an Organic Rankine cycle, however, not included in the scope of this work.

2.3. Refrigeration system design

A CO_2 refrigeration system was investigated for summer conditions by using the dynamic modeling software Dymola with components and libraries from TLK Thermo. In summer, the required temperature in the cabins and the public areas is in the range of 18 °C to 24 °C, and the supply air temperature can be a maximum of 10 °C lower than the indoor temperature. The design cooling and heating load for the reference cruise ship was 1483 kW and 76 kW, respectively, and is distributed in three different zones. The concept evaluated is to circulate CO_2 refrigerant in the air handling unit and cooling fan coils, leading to high evaporation temperature and reducing energy consumption. In the case of other refrigerants, direct circulation in AC coils is not possible due to flammability, toxicity, and other safety issues, and they required low evaporation temperatures for indirect circuits. The reference case was simulated with load variations from 100 % to 50 % for each evaporation temperature of 13 °C to 17 °C with an assumption of only 5K temperature difference from indoor temperature. The high pressure was varied from 85 bar to 80 bar (load dependent), but the temperature before the high-pressure expansion valve was kept at 25 °C. The heat rejection in the condenser/gas cooler was made with in a single heat exchanger operating with a constant inlet seawater temperature of 20 °C.

The refrigeration system layout is presented in Fig. 2. Evaporation temperature was kept constant in the three zones for each simulation. Each zone can have multiple evaporators and coils for air-conditioning, but they were added together for simplicity of simulation effort. The cooling load was controlled by the expansion valves, which take superheat as an input to maintain the required refrigerant flow. The parallel compressor removes gas vapor from the liquid receiver to make sure only liquid flows to the expansion valves before evaporators. The suction gas of both main compressors and the parallel compressor was superheated by an internal heat exchanger for the safety of compressors. The compressor isentropic and volumetric efficiency was set to 0.7. The mechanical work of the gas cooler pump was neglected due to its small value compared to compression work.

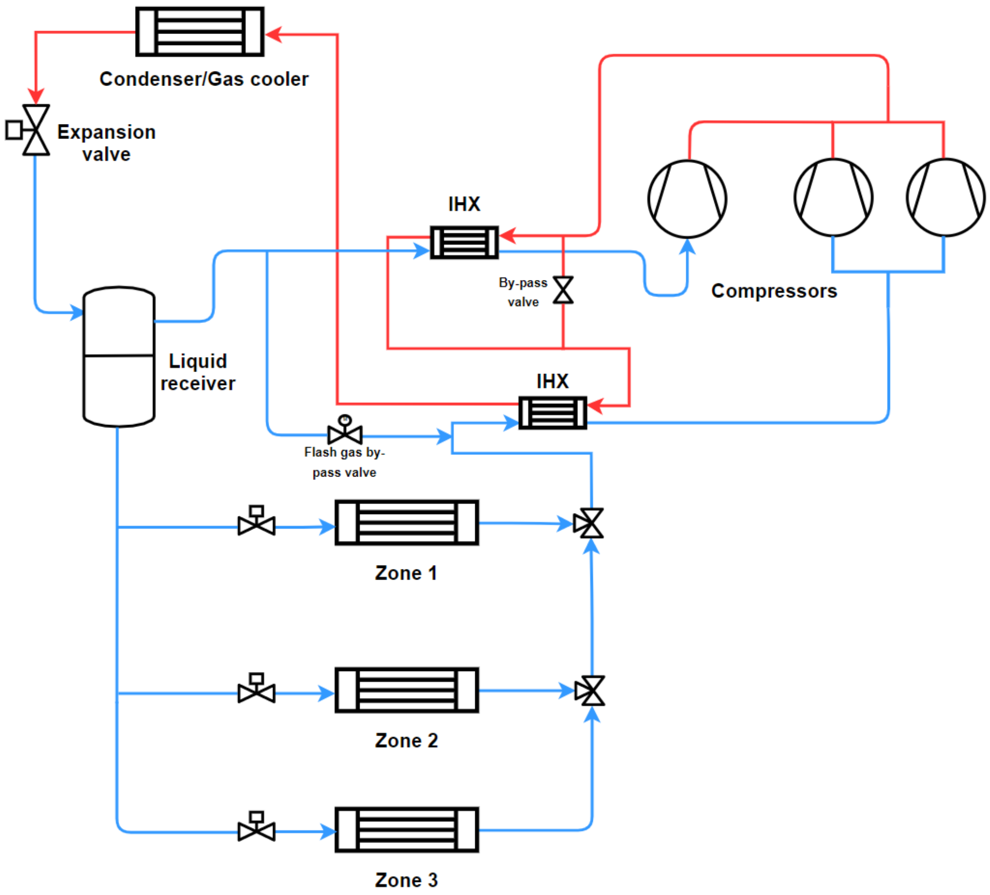

Figure 2: Refrigeration system layout

The cooling load profile of the reference ship for 24 hours is shown in Fig. 3. The load is minimum during the night, gradually increasing from 7 am, being at its maximum during midday when the sun is at its peak. The load is then decreasing in a polynomial way and reaches the lowest value at 24 hours.

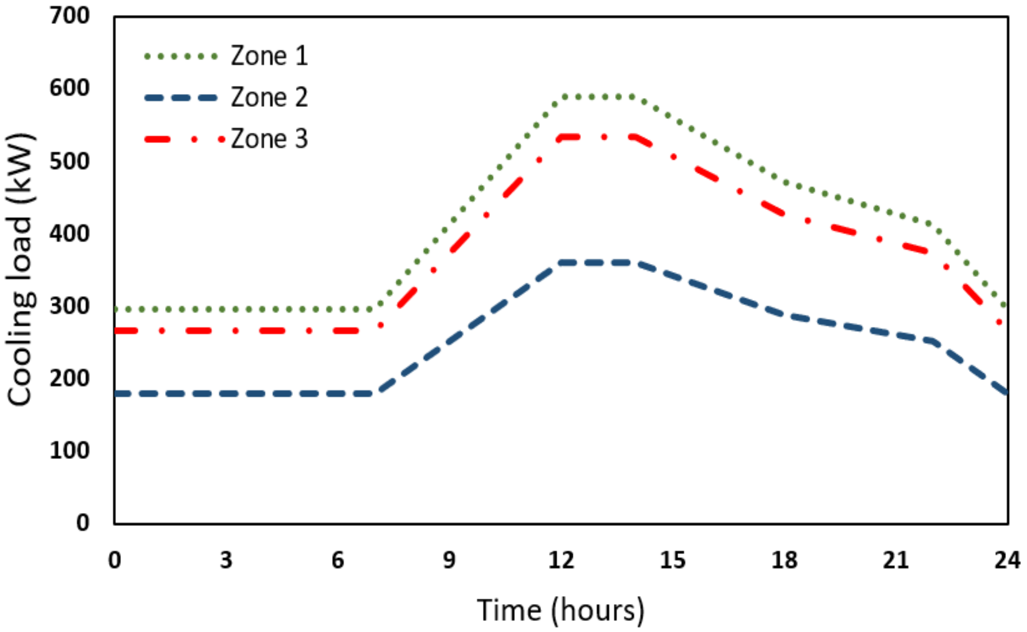

Figure 3: Cooling load of three different zones

3. RESULTS AND DISCUSSIONS

The cold recovery potential from the LNG fuel is shown in Fig. 4. As seen, the recovered cold energy is 265 kW at 100 % engine load and 106 kW at 40 % engine load. Possible heat sinks or applications for this cold were discussed by some researchers, as mentioned in the introduction, but utilizing it in an efficient way is the main challenge. Integration of the cold recovery with HVAC system possesses a more significant potential for energy efficiency compared to engine oil cooling. The LNG cold recovery at the average engine load of 85 % is 225 kW, and the maximum cooling load for the reference ship is 1483 kW. By utilizing cold recovery, the AC cooling system size can be reduced to 1258 kW. The engine load can be less than 85 % at peak cooling demand, but the integration of thermal energy storage can compensate for the reduced AC system size. If the COP of the refrigeration system is assumed to 5, the potential fuel savings will be 0.67 % at average engine load and 0.79 % at full load.

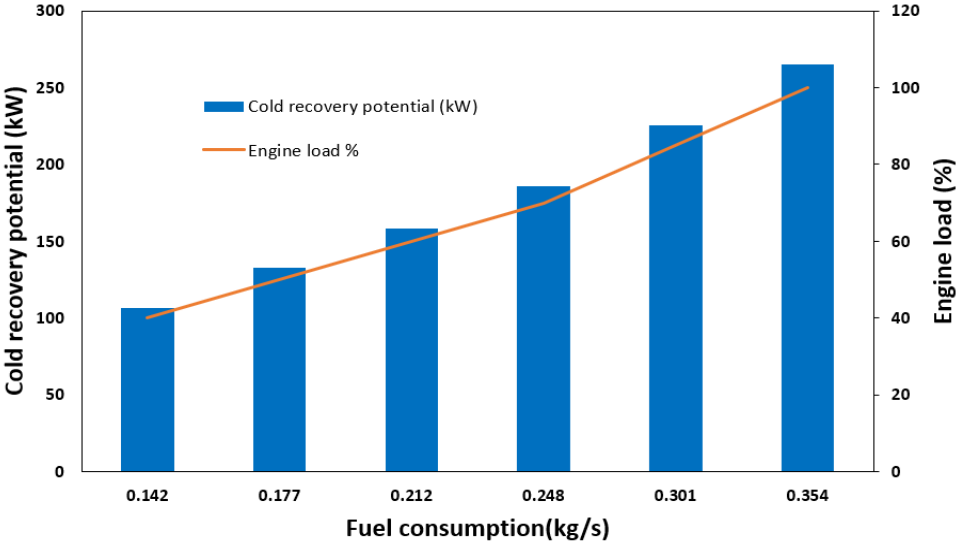

Figure 4: LNG cold recovery versus fuel consumption

The heat recovery from the exhaust gases of the engines is presented in Fig. 5. The heat recovery at 100 % engine load is 2715 kW and 1090 kW at 40 % engine load. This high-temperature heat recovery can be used for onboard hot water production, electricity production using ORC, heat to power production by thermoelectric effect, or the combination of different applications. The different applications will be analyzed in further work.

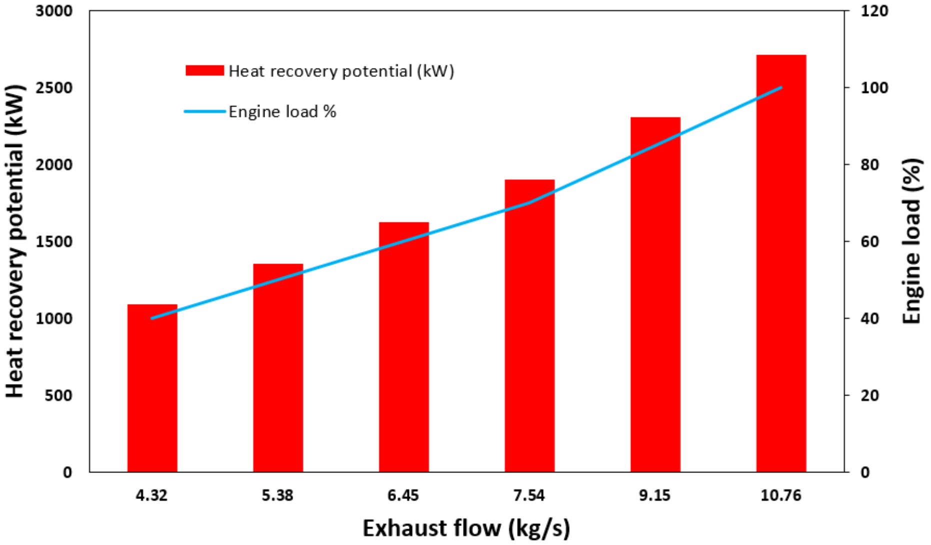

Figure 5: Heat recovery versus exhaust flow gases

The simulation model of the refrigeration system was run for the 24 h load profile shown in Fig. 3. Five different simulations were performed with evaporation temperatures of 13 °C to 17 °C. The power consumption presented in Fig. 6 is the sum of the main compressor and the parallel compressor. The maximum power consumption was in the 13 °C evaporation case due to the higher pressure ratio as compared to other cases, and the minimum was observed in 17 °C case. The maximum power consumption in 13 °C, 14 °C, 15 °C, 16 °C, and 17 °C cases was 256.3 kW, 242.7 kW, 229.3 kW, 216.1 kW, and 203.1 kW, respectively. It can be concluded that at the maximum cooling capacity, each °C increase in evaporation temperature will save an average 13 kW (and 65 kW for 5 °C increase).

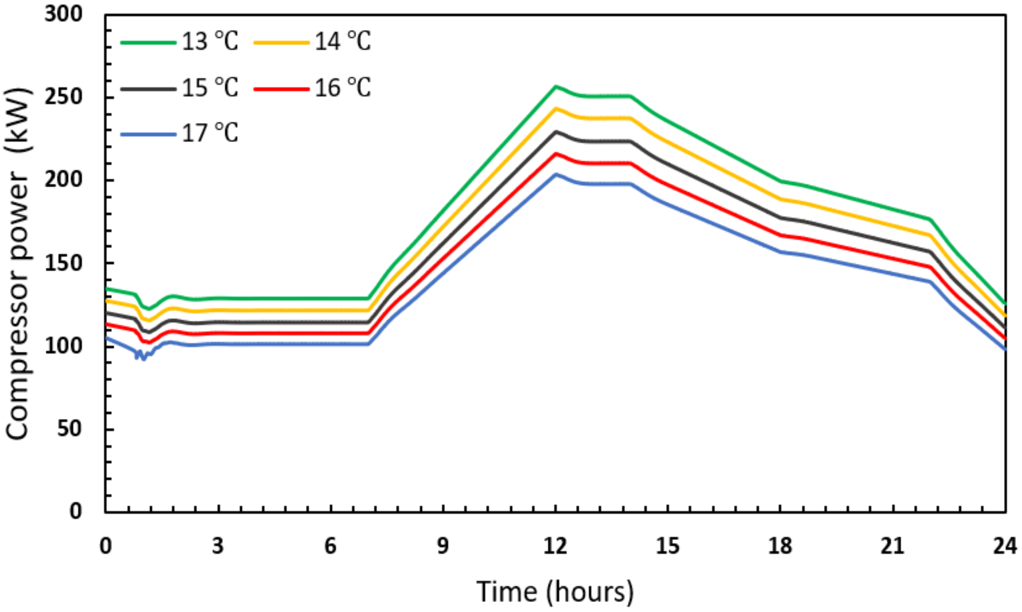

Figure 6: Cooling unit power consumption versus time

A general trend was observed for COP in all cases. High COP at low loads and low COP at high loads. At high loads, the pressure ratio slightly increased as the system had to dissipate more heat, and it reduced at lower loads. The COP at maximum load in 13 °C, 14 °C, 15 °C, 16 °C, and 17 °C cases was 5.5, 5.8, 6.2, 6.5 and 7.1, respectively. The average COP difference at maximum load for each °C rise in evaporation temperature is 0.4.

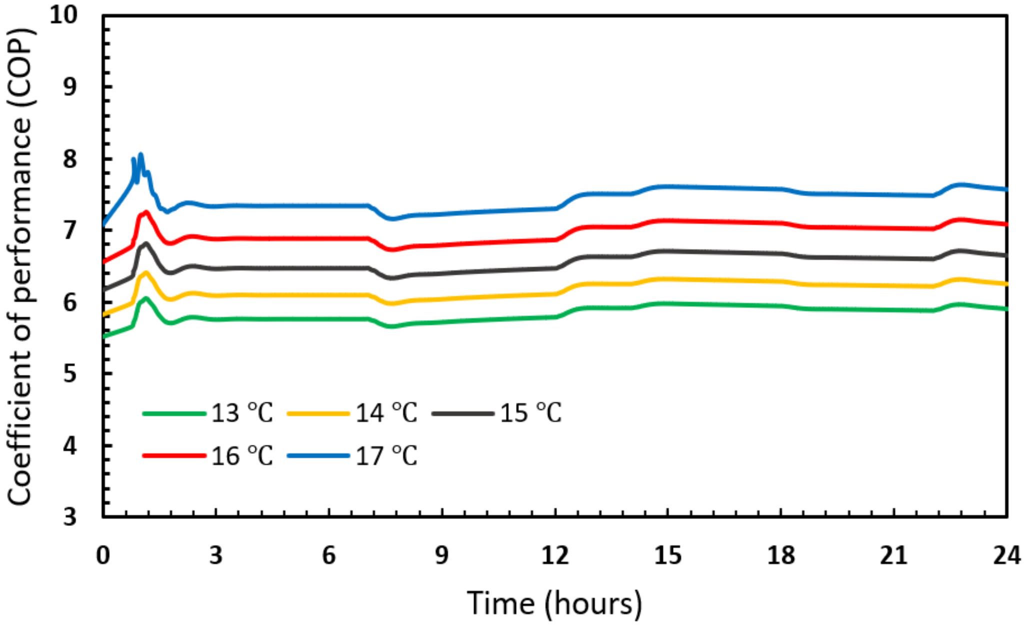

Figure 7: Coefficient of performance versus time

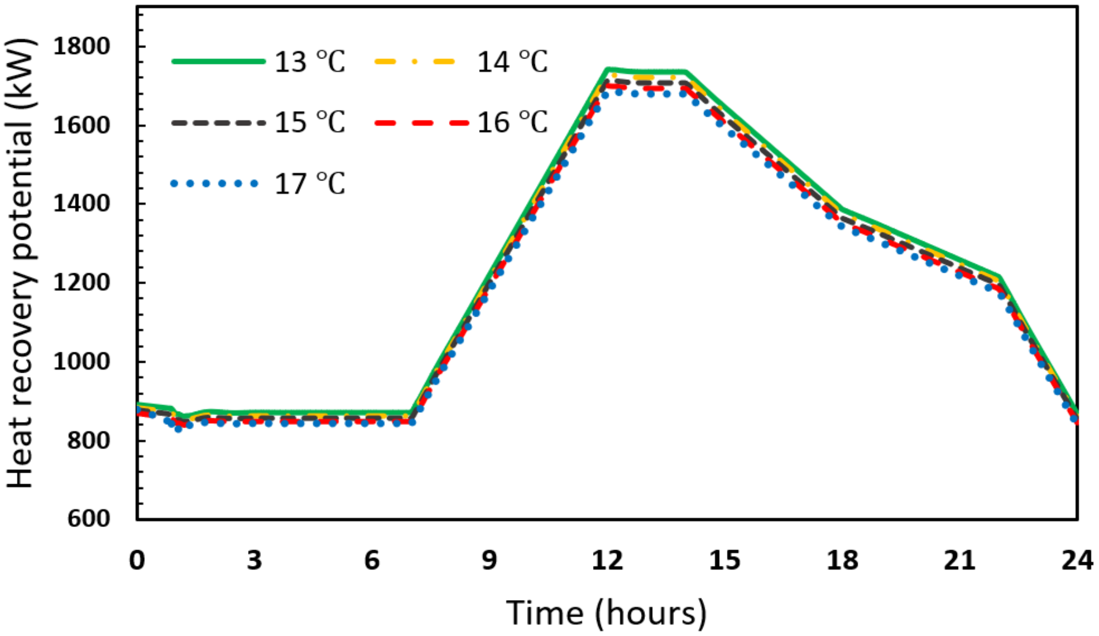

Figure 7: Refrigeration system heat recovery versus time

The temperature before the gas cooler varied from 58 °C to 55 °C. The heat recovery from the gas cooler is low temperature as compared to recovery from engine exhaust gases. The heat recovery at maximum load in 13 °C, 14 °C, 15 °C, 16 °C, and 17 °C cases was 1740 kW, 1726 kW, 1713 kW, 1700 kW, and 1687 kW, respectively. The average heat recovery difference for each evaporation case is 13 kW. Depending on the applications and temperature levels, heat recovery can be made in two heat exchangers.

4. CONCLUSION

In this paper, thermal system analysis of a reference cruise ship driven by LNG fuel was performed by calculations and simulations in Dymola. A cruise ship was investigated with a passenger capacity of 330, engine power around 8 MW, designed cooling load 1483 kW, and heating load of 76 kW, in summer mode. Results showed that 225 kW LNG cold could be recovered at the average engine load of 85 %. By utilizing this cold for air conditioning application, 0.67 % of fuel can be saved at maximum cooling capacity. The heat recovery from the exhaust gases of the engine at 85 % load was estimated at 2.3 MW. Evaluations of CO_2 refrigerant direct expansion in air handling units and coils showed impressive results for energy saving compared to indirect systems. Each °C increase in evaporation will save an average of 13 kW of the compressor power. Besides, the CO_2 system also provides flexibility for integrated cooling and heating. The work of this paper will extend to further development of CO_2 refrigeration system for cruise ships and integration with thermal energy storage. Other thermal systems will also evaluate further for the best possible optimization

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