COMBINED THERMODYNAMIC AND TURBINE DESIGN ANALYSIS OF TRANSCRITICAL CO_2 CYCLE FOR WASTE HEAT RECOVERY

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ABSTRACT

The utilization of low temperature waste heat has been under great interest in the recent times. One opportunity to utilize low-temperature waste heat sources is to convert the heat into electricity. For this purpose, organic Rankine cycles have been often considered. Another power system capable for converting low temperature heat to electricity is the transcritical CO_2 power cycle. In this study, the utilization of exhaust gas heat at the temperature of 364 ºC and the utilization of high temperature water at the temperature of 90 ºC are investigated with transcritical CO_2 cycles. The investigated waste heat temperature levels and the cycle cooling temperatures were selected based on typical values found in cruise ships using large-scale four stroke gas engines as their main power source.

The transcritical CO_2 cycle analysis and the design of radial inflow turbine for the cycle were combined, in order to investigate the cycle and turbine efficiency under different design power scales and conditions. The turbine geometry was designed based on the cycle operating conditions and mass flow rate, and the turbine efficiency was evaluated for each design case by using existing loss correlations for radial turbines. The investigated turbine losses are including stator loss, passage loss, tip clearance loss, disk friction loss and exit kinetic loss. Based on the results, heat can be utilized by the CO_2 cycle and high efficiency radial turbines can be designed for the investigated transcritical CO_2 cycles recovering waste heat. In addition, the use of CO_2 as the working fluid would allow to have more compact systems, when compared to the use of ORC systems adopting high molecular weight fluids. The main drawbacks of using transcritical CO_2 cycles for waste heat recovery, when compared to the ORC systems, are the high pressure level and high rotational speed of the turbomachinery required in the system.

1 INTRODUCTION

The utilization of waste heat has been under great interest in the recent times as it is one of the most effective ways to increase energy efficiency of different types of industrial processes. There are several technologies that can convert waste heat into electricity, including steam Rankine cycles, organic Rankine cycles and thermoelectric generators. In marine applications the waste heat can be recovered in a form of heat for ship heating purposes, or it can be converted to electricity by using power turbines, advanced turbochargers or by using different types of Rankine cycles(Shu *et al.* 2013). One option in ship applications is also to convert the waste heat to cooling energy by using absorption chillers (Salmi *et al.* 2017). The waste heat recovery potential and opportunities in ships are discussed e.g. in the papers by Shu *et al.* (2013) and Larsen *et al.* (2014). In waste heat to electricity conversion, the use of carbon dioxide as the working fluid in power systems has been under great interest and development. The main benefits of using CO_2 as the working fluid are high efficiency especially in high temperature cycles, compact sized components in respect to the system power scale and high stability of the fluid (Ahn *et al.* 2015). Both supercritical Brayton cycles, operating completely at the supercritical fluid region (Uusitalo *et al,* 2019a) as well as transcritical cycles (Chen *et al.,* 2006) , in where the high pressure side of the process is at the supercritical region and the low pressure side is at subcritical fluid region, have been considered as suitable for waste heat recovery systems using CO_2.

There are already many studies available on utilizing high temperature exhaust heat of large-scale four stroke engines with ORC technology (e.g. Bombarda *et al.* 2010, Uusitalo *et al.* 2014, Song *et al.* 2015). The results of these previous studies have shown that the ORC is capable to increase the power output by about 10 % when connected to large scale reciprocating engine plants. Similar energy saving potential was identified with supercritical Brayton cycle based WHR systems recovering exhaust heat from large engines (Uusitalo *et al.,* 2019a). In addition, the utilization of the lower temperature heat sources from engines have been also investigated by means of ORC technology showing lower power increase when compared to the exhaust heat recovery (Peris *et al.* 2013, Uusitalo *et al.* 2015).

The use of transcritical cycles in waste heat recovery has been also investigated in the literature. Chen et al. (2006) compared a transcritical CO_2 cycle and low temperature ORC for recovering low temperature waste heat. Their results showed the transcritical CO_2 cycle reached slightly higher power output when compared to the ORC. They also concluded that the use of CO_2 resulted into more compact cycle. Shu *et al.* (2016) investigated the use of transcritical CO_2 cycle for recovering engine exhaust heat and cooling water heat in a single cycle. Their results showed that the system could produce additional 9 kW power when connected to an engine with a power output of 43.8 kW. Manjunath *et al.*(2018) studied a combination of supercritical CO_2 cycle for power production and transcritical CO_2 cycle for producing cooling energy from waste heat in a gas turbine powered ship. They estimated that the ship overall efficiency could be increased by more than 11% with the proposed system.

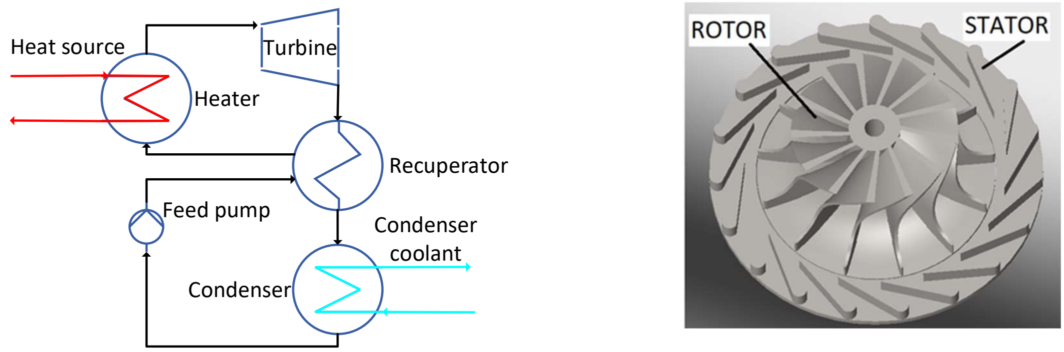
In this study, the utilization of waste heat from large scale four-stroke gas engines by means of transcritical CO_2 cycle is investigated. The investigation concentrates on the cycle thermodynamic analysis as well as on the turbine design and turbine losses with different cycle parameters. The heat source and cycle cooling temperatures were selected based on typical values found in cruise ships, using gas-fired reciprocating engines as their main power source. In modern cruise ships part of the engine waste heat is utilized for heating purposes in the ship, but in many operating conditions the amount of waste heat is significantly larger than can be utilized in the ship heating operations. In addition, as the ships are equipped with several main engines and auxiliary engines, the waste heat of all the engines in operation cannot be utilized for heating (Uusitalo *et al.* 2019b). Thus, the conversion of waste heat to electricity of a single engine is investigated in this paper.

2 METHODS

The engine waste heat data is collected from a data sheet by engine manufacturer for 14.4 MW sized 12-cylinder gas engine (Wärtsilä, 2021). Exhaust gas temperature of 364 °C and mass flow of 26.2 kg/s were used in the simulations. High temperature (HT) cooling water loop temperature of 95 °C before the WHR system and 74 °C after the WHR were used. The heat power available in the engine cooling loop is 4884 kW per engine. A condensing temperature of 25 °C was used in the simulations as this temperature was estimated to be achievable in ships operating in cold climate and using sea water as the condenser coolant, such as in the Baltic sea. The corresponding condensing pressure of CO_2 is 64.3 bar. The utilization of the exhaust heat (Case 1) and engine cooling water heat (Case 2) were studied in separate transcritical cycles. The process diagram of recuperated transcritical CO_2 cycle is presented in Figure 1a. The exhaust gas heat recovery cycle was studied with turbine inlet temperatures of 200 °C, 240 °C, 280 °C and 320 °C. The engine HT water heat utilization was simulated with turbine inlet temperature of 80 °C. Both the cycles were investigated with different turbine inlet pressures.

The calculation principles of the cycle are based on solving the energy balance and fluid state at inlet and outlet of each process component. A commercial thermodynamic library Refprop (Lemmon *et al.* 2018) was used for determining the CO_2 thermodynamic properties at different process nodes. In the exhaust gas heat recovery, recuperator was used as it significantly increases the cycle efficiency. In the low temperature loop no recuperator was included in the analysis due to the limited benefits of using the recuperator in the low temperature cycle. The exhaust gas temperature at the heater outlet of 120 ºC was used and the recuperator effectiveness was varied in such a way that the minimum temperature difference between the exhaust gas and CO_2 at the heater cold end is 20 ºC. The recuperator effectiveness was limited to a maximum value of 0.8 in the analysis. Pump efficiency of 70 % was used in the simulations. The pump motor efficiency as well as the turbogenerator electric and mechanical efficiency were not considered.

The design and analysis of a radial-inflow turbine was coupled with the cycle analysis tool to be able to estimate the turbine geometry, rotational speed and efficiency at different cycle design operating conditions. The design principles for defining the turbine geometry were based mainly on following the turbine design guidelines of Rohlik (1972) and Balje (1981). An in-house code developed at LUT was used and the step by step radial turbine design procedure is presented in Uusitalo et al. (2021) for supercritical CO_2 turbines. An example of radial turbine geometry is presented in Figure 1b.



(a) (b)

Figure 1. Simplified process diagram of the investigated transcritical CO2 cycle. (a) shows the cycle layout with recuperator and (b) shows and example of radial turbine geometry including stator vanes and rotor.

The turbine for each case was designed to reach the specific speed of 0.6, that is in general, close to the design specific speed allowing to reach peak efficiencies for radial inflow turbines (Balje, 1981, Rohlik, 1972). In a recent study, it has been shown that the turbine efficiency vs. specific speed curve for supercritical CO_2 driven radial turbines follows relatively well the generalized curves for radial turbines suggesting maximum isentropic efficiencies at specific speeds between 0.5 to 0.6 (Uusitalo et al., 2021). Thus, it can be expected that the selection of the $N_s = 0.6$ allows to reach high isentropic efficiencies also for the investigated transcritical cycles, while it should be noted that it is not necessarily the N_s value that maximizes the turbine efficiency in all the investigated cases. By selecting a lower N_s the turbine rotational speed would be decreased and the rotor diameter increased. With higher N_s, the turbines would require higher rotational speed but would result to more compact turbine wheel. Overall, the turbine design N_s was remained as constant for all the investigated cases to simplify the analysis and to limit the number of the simulation cases. The rotational speed of the turbine was calculated to reach the targeted N_s value from the definition of the specific speed defined as,

$$N_s = \frac{\omega {q_{v2}}^{0.5}}{{\Delta h_s}^{0.75}} \qquad (1)$$

The turbine geometry and velocity triangles were solved based on the turbine inlet conditions and outlet pressure gained from the cycle analysis. The flow angle of $\alpha_1 \approx 70°$ was used at the rotor inlet, the rotor discharge was assumed to be axial ($c_{u2} = 0$) and the outlet blade angle at the tip and hub was defined based on the angle of the relative velocities. The rotor diameter ratio of $D_2/D_1 = 0.7$ and the hub to tip diameter ratio of 0.3 were used for all the designs. Diameter ratio for the stator vane inlet of $1.3D_1$ and stator vane outlet of $1.05D_1$ were used. The blade height at the rotor inlet was defined by using the continuity equation and the shape of the velocity triangle at the rotor inlet. The inlet and outlet blade thicknesses were defined as a fraction of the rotor inlet and outlet radii. Tip clearance of 0.3 mm was used and the velocity ratio C_{u1}/U_1 was set to 0.95 by adjusting the pressure between the stator and rotor in all the turbine designs. The loss correlations used for estimating the isentropic efficiency of the turbine includes enthalpy-based loss correlations for stator loss, rotor passage loss, tip clearance loss, disk friction loss, incidence loss and exit kinetic loss. The different loss correlations and their literature references are summarized in Table 1. The validation of the turbine design and loss analysis method in respect to the CO_2 driven radial turbine designs available in the literature, is presented in detail in Uusitalo et al. (2021). The turbine isentropic efficiency was defined for each design case as,

$$\eta_s = \frac{\Delta h_s - \Sigma \Delta h_{loss}}{\Delta h_s} \tag{2}$$

In where Δh_s is the isentropic enthalpy change over the turbine and $\Sigma\Delta h_{loss}$ is the sum of the different loss sources. The cycle and turbine design calculation was iterated until there were no significant changes in the cycle or turbine geometry and efficiency values between the iteration rounds.

Table 1. Investigated turbine losses and the literature references for the loss models.

Loss	Literature reference
Stator loss	Whitfield and Baines (1990)
Rotor passage loss	Nasa CETI model (Moutapha et al. 2003)
Incidence loss	Whitfield and Wallace (1973)
Tip clearance loss	Jansen (1970)
Disk friction loss	Daily and Nece (1960)
Exit kinetic loss	Rahbar et al. (2015)

3 RESULTS

The main results of the study are summarized in the following. First, the cycle analysis results are presented for the exhaust gas heat recovery (Case 1) with turbine inlet temperatures of 200 ºC, 240 ºC, 280 ºC and 320 ºC. Second, the results of the HT water utilization are presented. Finally, the results of the turbine design analysis and the result summary are presented. The results for the recuperated cycle utilizing exhaust heat are presented in the following. In Figure 2a turbine power, 2b cycle net power output, 2c pump power and 2d CO_2 mass flow are presented with different turbine inlet temperatures and pressures.

As the turbine inlet pressure is increased both the turbine and feed pump power were increased. With the highest turbine inlet pressures the cycle net power output starts to decrease as the pump power consumption is increasing more than the turbine power output with the respective change in pressure. Out of the studied conditions, the highest cycle net power output is reached with the turbine inlet temperature of 280 ºC and 320 ºC reaching about 1.5 MW net power output, that corresponds to about 10.7 % of the gas-engine power output, with the turbine inlet pressures between 200 bar to 300 bar. Thus, the waste heat recovery potential from the exhaust gases seems to be in the same order of magnitude for the transcritical CO_2 cycle, when compared to the previous studies regarding the use of high temperature ORCs and Kalina cycle (Bombarda et al. 2010, Uusitalo et al. 2014). The lower turbine inlet temperatures of 200 ºC and 240 ºC results to lower net cycle power outputs when compared to the higher turbine inlet temperatures. The lower turbine inlet temperature cases can have the maximum degree of recuperation as the temperature difference between the CO_2 and exhaust gas remains sufficient at the heater cold end. In these cases, the exhaust gas temperature level at the heater cold end could have been further lowered from the selected minimum value of 120 ºC that would have allowed to extract more heat from the exhaust gases. With the highest turbine inlet temperatures, the degree of recuperation must be lowered from its maximum value, to maintain the minimum of 20 ºC temperature difference at the heater cold end. The mass flow rate of the CO_2 is varying between 20 kg/s to 30 kg/s in the investigated cases. The cycle mass flow rate slightly decreases as the turbine inlet temperature or the turbine inlet pressure are increased. The results on the turbine geometry and efficiency for the Case 1 are presented later in this section.

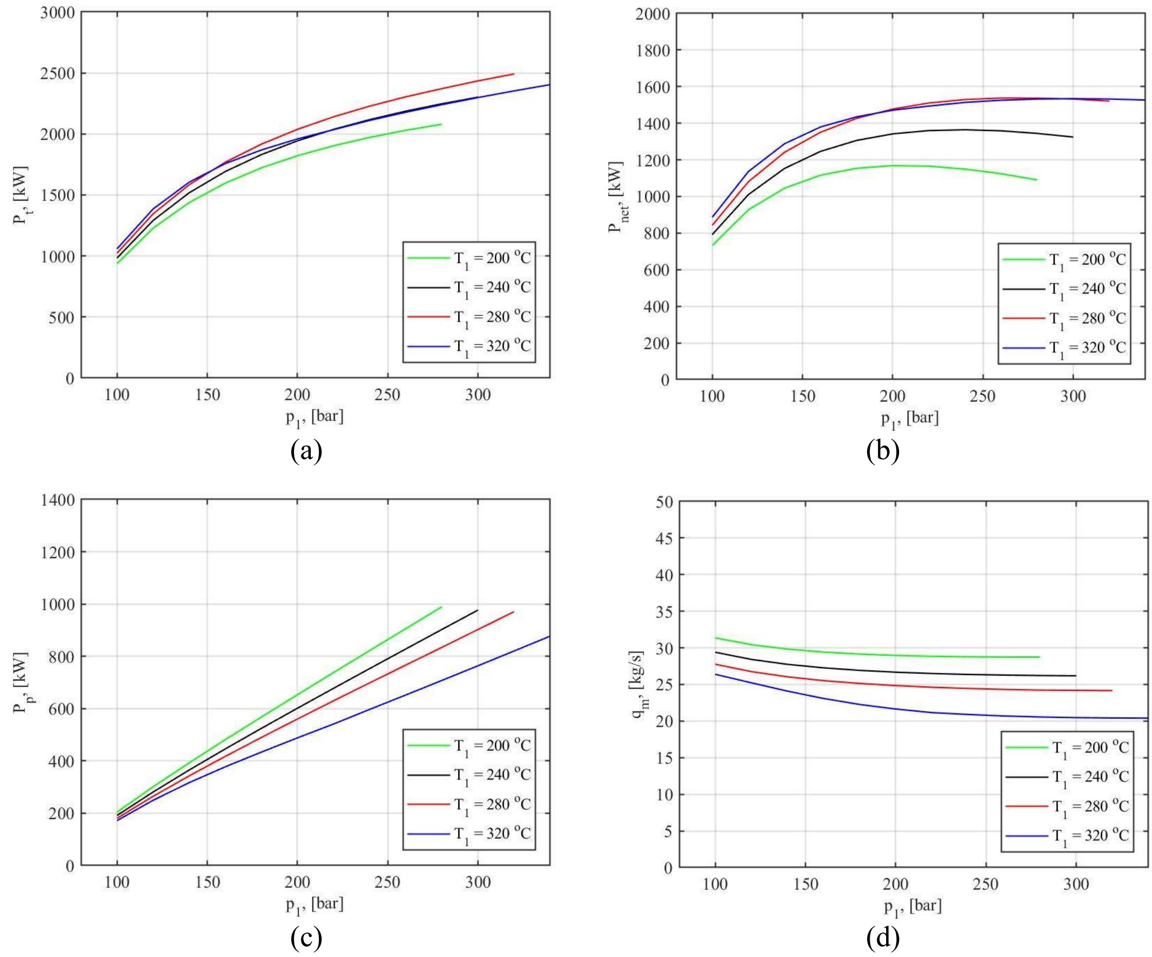
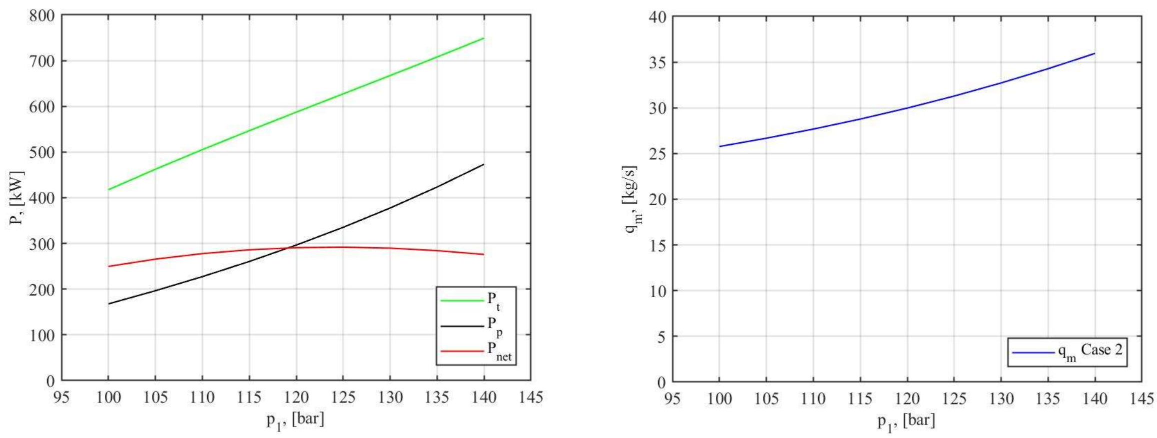


Figure 2. (a) Turbine power, (b) cycle net power, (c) pump power consumption and (d) CO_2 mass flow rate with different turbine inlet temperatures as a function of turbine inlet pressure.

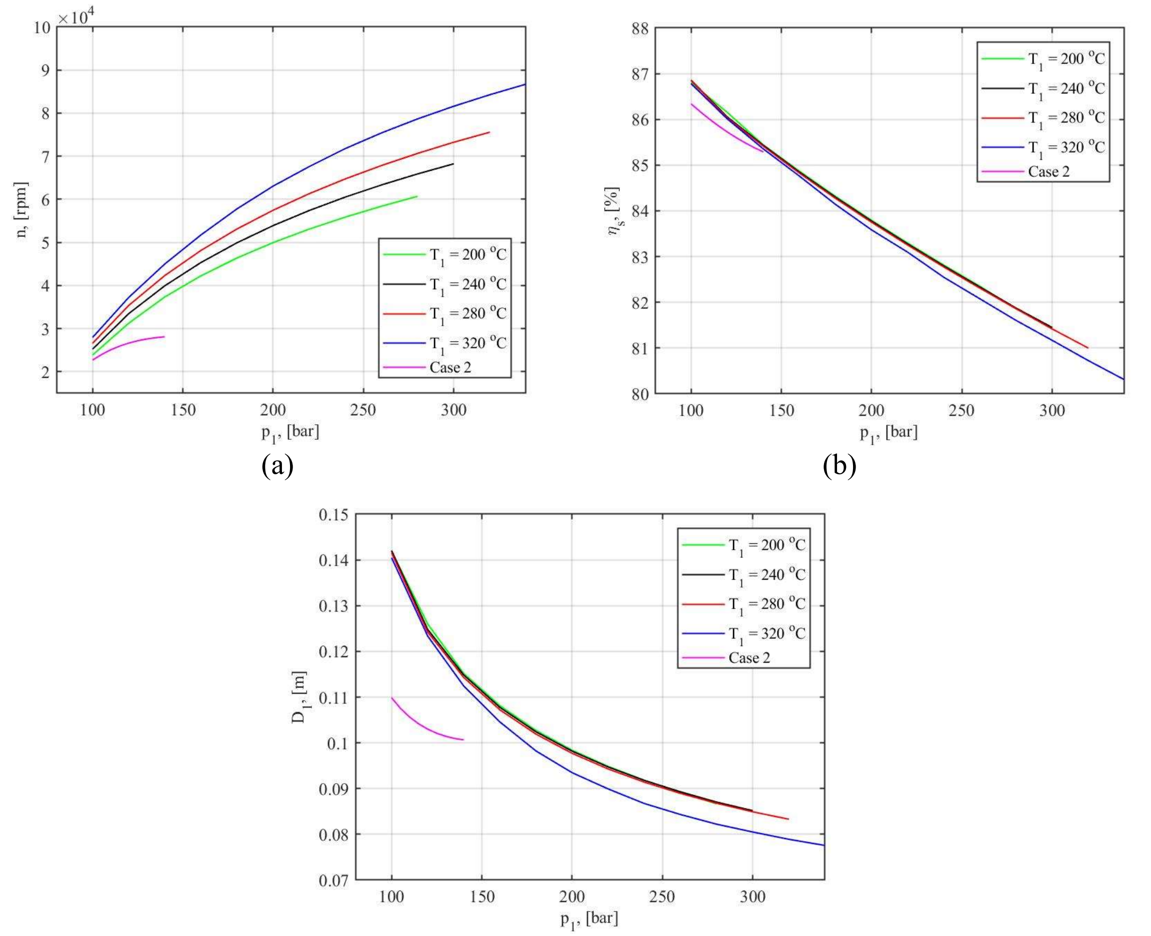
The results for the engine HT cooling loop (Case 2) are presented in Figure 3a and 3b. Figure 3a presents the turbine, pump and net power output of the cycle with different inlet pressures and Figure 3b shows the CO_2 mass flow rate in the low temperature WHR cycle. The low temperature transcritical CO_2 cycle could produce about 300 kW net power from the HT cooling water loop of the engine. This corresponds to about 2 % of the gas engine power output. The highest cycle net power output was simulated with turbine inlet pressure of 125 bar. The flow rate of CO_2 slightly increases as the turbine inlet pressure is increased. This can be explained by the fact that the heater inlet temperature is slightly higher with high pump outlet pressures, resulting to lower enthalpy change and higher mass flow in the heater as the heat power with all the studied conditions is remained as 4884 kW. The net power output of the cycle starts to be reduced with high turbine inlet pressures of above 125 bar as the pump power consumption increases more than the turbine power output is increasing with the respective increase in pressure. It should be noted that the simulations for the HT water heat recovery were carried out by using only a single turbine inlet temperature of 80 °C, that was selected in order to reach the turbine inlet temperature relatively close to the temperature of the HT water. If a smaller temperature difference between the CO_2 and HT water was used, leading to higher turbine inlet temperature, the efficiency of the HT loop system could be further improved.



(a) (b)

Figure 3. The effect of turbine inlet pressure on (a) turbine, pump and net power, and (b) on CO_2 mass flow rate

The results for the turbine designs and predicted efficiencies are presented Figure 4a-c. The figures 4a-c contain both the results for Case 1 with different turbine inlet temperatures and results for Case 2.



(a) (b) (c)

Figure 4. The effect of turbine inlet pressure and temperature on (a) turbine rotational speed, (b) turbine efficiency and (c) turbine diameter.

In general, the designed radial turbines have relatively high rotational speed requirement ranging from about 22 krpm to over 80 krpm. The simulated turbine efficiencies are ranging from 80 % to 87 % and the turbine diameters are ranging from 80 mm to over 140 mm. As the design turbine inlet pressure is increased the turbine efficiency and diameter decreases, whereas the optimal rotational speed is increased. The reduction in the turbine efficiency with the increasing turbine inlet pressures is mainly originating from the increased pressure ratio over the turbine that increases the flow velocities resulting to increased turbine losses. The turbine of the low temperature system has requirement for lower rotational speed and it reaches slightly lower efficiency and larger turbine wheel when compared to the transcritical CO_2 cycle utilizing the hot exhaust gas heat. In general, the turbines are significantly compact in respect to the investigated power scale, especially when compared to typical ORC turbines using high molecular weight fluids (e.g. vanBuijtenen 2003). The main loss sources for the turbines were observed to be the stator loss, passage loss, and tip clearance loss, whereas the disk friction loss and incidence loss had only a rather small contribution on the total losses. In addition, the kinetic energy at the rotor discharge was assumed to be lost energy that further reduces the turbine efficiency.

The results summary of the simulation cases reaching the highest net power output for the cycle are presented in Table 2, including some details on the turbine geometry and main losses.

Table 2. Result summary of simulation cases with the highest net power output

	Case 1	**Case 2**
Net power	1537 kW	292 kW
Turbine power	2371 kW	627 kW
Pump power	834 kW	335 kW
CO_2 mass flow	24.2 kg/s	31.3 kg/s
Heater power	7050 kW	4884 kW
Turbine inlet temperature	280 °C	80 °C
Turbine inlet pressure	280 bar	125 bar
Turbine isentropic efficiency	81.9 %	85.6 %
Turbine rotational speed	70600 rpm	27180 rpm
Turbine inlet diameter	87 mm	102 mm
Turbine tip outlet diameter	61 mm	71 mm
Inlet blade height	4.5 mm	7.5 mm
Outlet blade height	21.3 mm	25.0 mm
Relative flow angle at outlet (meanline)	54°	56°
Isentropic enthalpy change	119.6 kJ/kg	23.4 kJ/kg
Stator loss	5.9 kJ/kg	1.0 kJ/kg
Passage loss	3.9 kJ/kg	0.6 kJ/kg
Tip clearance loss	5.3 kJ/kg	0.6 kJ/kg

4 CONCLUSIONS

Waste heat recovery from large scale gas-fired engine was investigated with transcritical CO_2 cycles. From the exhaust gas heat, it was estimated that about 1.5 MW additional electric power could be generated with the transcritical CO_2 cycle. This corresponds to about the 10.7 % of the engine power output. A maximum of 0.3 MW was modelled to be produced from the high temperature engine coolant loop which corresponds to about 2.0 % of the engine power output. Thus, significant efficiency improvement potential in cruise ship energy systems could be achieved by using transcritical CO_2 cycles and the magnitude of the energy savings is in line with the previous studies using ORCs for recovering high and low temperature heat from large-scale engines. The analysis also showed that the turbine of the system can be designed to have high efficiency and extremely compact size with respect to the power scale. These features can be considered as a significant advantage for WHR system to be used in a marine application as the installation space for the WHR system is often limited. The main drawback of the investigated cycle is that high pressure levels are needed for maximizing the cycle net power output. In addition, the required turbine rotational speeds were observed to be rather high.

It should be highlighted that the aim of the study was to include a preliminary evaluation of the radial turbine design for the investigated WHR cycles to be able to evaluate the size, rotational speed and obtainable turbine efficiency. A more detailed analysis taking into account the effect of different design N_s, rotor inlet flow angle, and diameter ratios on the turbine efficiency and loss distribution is recommended to be carried out in the future as these parameters were remained as constant for all the investigated cases. The turbine wheel manufacturability and material strength analyses are recommended to be carried out as these aspects affect e.g. the minimum thicknesses of rotor blades and stator vanes which also have a direct influence on the turbine losses. In addition, further analysis and considerations on the turbogenerator design, including the electric machine, rotor dynamics, sealing technology and bearings design should be carried out. In this study, a constant pump efficiency of 70 % was used for all the investigated cases. Further analysis could be carried out also for the feed pump of this kind of system as the cycle net power output was observed to be sensitive on the power consumption of the pump. In addition, detailed heat exchanger design for the investigated cycles could be studied in detail in the future, to be able to better evaluate the overall size of the process equipment as well as the effect of heat exchanger pressure losses on the cycle performance.

NOMENCLATURE

P	power	(kW)	n	rotational speed (rpm)	
h	enthalpy	(kJ/kg)	ω	angular speed	(rad/s)
T	temperature	(K, ºC)	**Subscript**		
p	pressure	(bar)	s	isentropic	
q_m	mass flow rate	(kg/s)	loss	loss	
q_v	volumetric flow	(m^3/s)	t	turbine	
D	diameter	(mm)	1	turbine inlet	
b	blade height	(mm)	2	turbine outlet	

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