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ORIGINAL PAPER

Optimum operations and performance comparison of CO_2-propane and CO_2-R152a mixture-based transcritical power cycles recovering diesel power plant waste heat

Subha Mondal[1] · Akash Saha[2] · Sudipta De[2]

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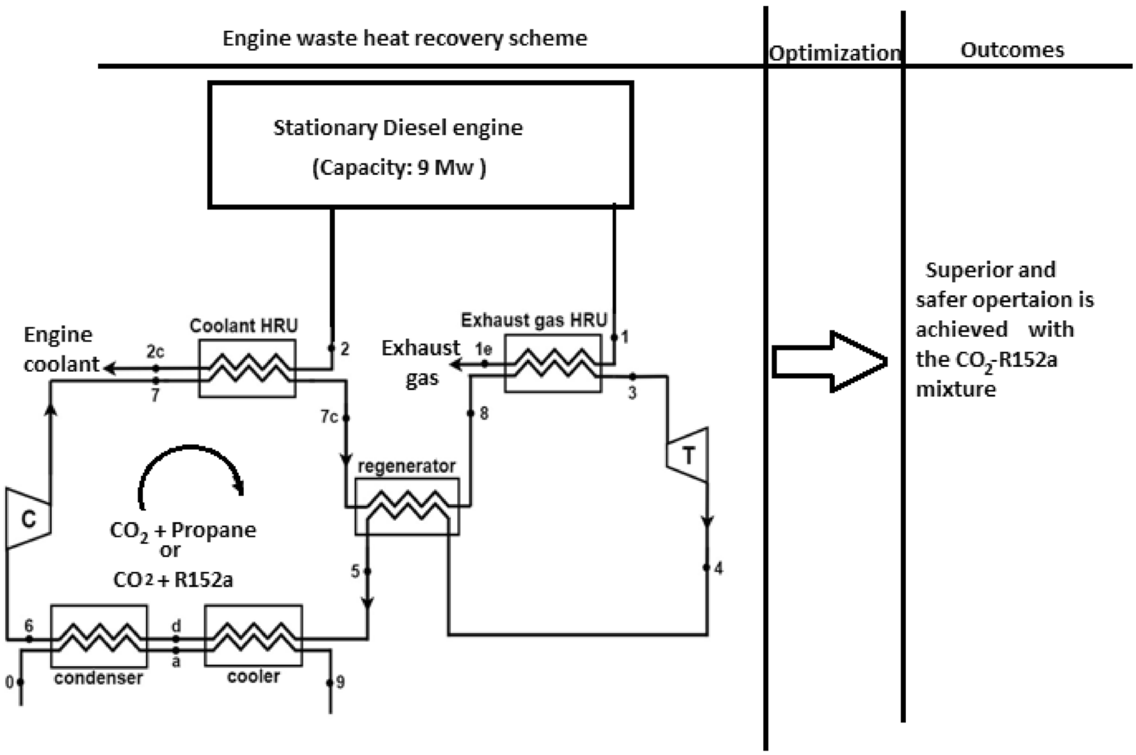
Abstract
Though CO_2 power cycles are preferred for diesel engine waste heat recovery, a very high operating pressure of the CO_2 power cycle is an issue of concern. To address this issue, in the present study, CO_2-propane and CO_2-R152a mixtures with various CO_2 mass fractions are proposed as the working fluid of a regenerative transcritical power cycle recovering waste heat of a diesel power plant. To reduce the possibility of accidental fire hazard; the minimum permissible CO_2 mass fraction is restricted to 0.3. It is observed that reducing CO_2 mass fraction ensures higher output power and lesser levelized electricity cost (LEC), specifically at a lower turbine inlet pressure. Between two considered CO_2-based mixture pairs, the transcritical cycle exhibits a superior performance with CO_2-R152a-based mixtures. The LEC of the presented CO_2-propane-based optimized cycle is about 6.36% lower compared to that of the optimized supercritical CO_2 power cycle. For the CO_2-R152a mixture-based cycle, the corresponding achievable reduction in LEC is about 15.20%. Turbine inlet pressures corresponding to the minimum LECs of the optimized CO_2-propane and CO_2-R152a mixture-based cycles are, respectively, close to 33% and 39% lower than that of the optimized supercritical CO_2 power cycle. As R152a is less flammable than propane, an R152a-based mixture working fluid also ensures a safer operation compared to a propane-based mixture.

✉ Sudipta De
de_sudipta@rediffmail.com;
sudipta.de@jadavpuruniversity.in

[1] Department of Mechanical Engineering, Aliah University, Kolkata 700160, India

[2] Department of Mechanical Engineering, Jadavpur University, Kolkata 700032, India

Graphical abstract



Keywords Diesel power plant waste heat · CO_2-based mixture · Transcritical power cycle · Output power · Levelized electricity cost

List of symbols

Symbol

A	Area of heat exchange (m^2)
AOH	Annualized operating hours (h)
C_p	Specific heat (kJ/kg.K)
C_p^0	Purchase cost ($)
C_{OM}	Cost of operations and maintenance ($)
CRF	Capital recovery factor (Dimensionless)
e	Specific exergy (kJ/kg)
$\dot{E}_i$, $\dot{E}_D$, $\dot{E}_L$	Exergy in flow, destruction and loss (kW)
F	Correction factor (Dimension less)
h	Specific enthalpy (kJ/kg)
LEC	Levelized energy cost ($/kWh).*LT:* Lifetime (years)
LMTD	Log mean temperature difference (K)
$\dot{m}$	Mass flow rate (kg /s)
mf	Mass fraction (Dimensionless)
Nu	Nusselt number
P	Pressure (MPa)
Pr	Prandtl number
Q	Heat transfer (kW)
Re	Reynolds number
s	Specific entropy (kJ/kg.K)
S_T, S_L	Transverse and longitudinal pitch (m)
t	Temperature (°C)
T	Temperature (K)
u	Fluid velocity (m /s)
U	Overall heat transfer coefficient (W/m^2.K)
$\dot{W}$	Power (kW)

Greek symbol

α	Convective heat transfer coefficient (W/m^2.K)
η	Efficiency (Dimensionless)

Subscript

CO_2	Carbon dioxide
c	Compressor
cond	Condenser
cw	Coolant water
EG	Exhaust gas
i	Inlet
Jcwav	Available flow rate of the engine coolant
Jcw	Engine coolant entering the coolant HRU
t	Turbine
WF	Working fluid
w	Water

II	2nd law
I	1St law
1-1e	Flue gas states
2-2c	Jacket coolant water states
3–7,7c,8	Working fluid cycle states
0	Ambient state

Abbreviations

CW	Coolant water
CEPCI	Chemical engineering plant cost index
EG	Exhaust gas
GWP	Global warming potential
HRU	Heat recovery unit
LEC	Levelized energy cost
LMTD	Log mean temperature difference
ODP	Ozone depletion potential
ORC	Organic Rankine cycle
TIP	Turbine inlet pressure
TIT	Turbine inlet temperature

Introduction

A considerable part of energy input to a diesel engine is usually rejected to its immediate surroundings as waste heat. Thus, through efficient waste heat recovery the energy utilization efficiency of a diesel engine could be significantly improved. Turbocompounding, CO_2-based power cycle and organic Rankine cycle (ORC) are some of the technology options reported in the literature for the diesel engine waste heat recovery (Mondal and De 2022).

Turbocompounding is a preferred technology for moving vehicles operating at a higher load over a longer period. It was observed that the efficiency of a marine diesel engine with turbocompounding would be even above 50% (Hireth and Prenninger, 2007). However, produced back pressure due to the turbocompounding affects the purging process of the burnt gas from the combustion chamber (Aghaali and Ångström 2015).

Recently, several researchers had proposed to use ORCs for recovering the diesel engine waste heat. Yu et al. (2013) proposed to use R245fa as the working fluid of an ORC operating with the waste heat of a diesel engine coupled with a generator. Shu et al. (2014) reported that a diesel engine waste heat-driven ORC using cyclo-hexane as the working fluid would improve brake specific fuel consumption by 10%. Yang (2016) proposed an ORC layout recovering waste heat from the exhaust gas, jacket cooling water, scavenging air cooling water and lubricating oil of a large marine engine simultaneously. Boodaghi et al. (2021) reported that a dual loop ORC coupled with a heavy-duty diesel engine would deliver 310 kW power at an engine speed of 1800 RPM. R245fa and R134a had been working fluids for the high-temperature and low-temperature loops, respectively. Lion et al. (2019) revealed that the annual fuel cost of the marine diesel engine with an ORC-based waste heat recovery system was close to 5% lower compared to that of the marine diesel engine without waste heat recovery.

Recently in many of the waste heat recovery schemes, CO_2 was considered as the working fluid for power cycles due to its environment-friendly and non-flammable characteristics (Mondal and De 2015a, 2015b; Zhao et al. 2020). CO_2 is also preferred for engine waste heat recovery due to its stability at a higher temperature and compact design of the CO_2 turbomachinery (Hosseinpour et al. 2022). Song et al. (2018) proposed a diesel engine waste heat-driven supercritical CO_2 power cycle with two pre-heaters. Incorporation of the 2nd pre-heater resulted in about 6.7% enhancement of output power. Pan et al. (2020) showed that diesel engine waste heat-driven recompression CO_2 power cycle with dual turbine would reduce auxiliary fuel consumption substantially. Mondal et al. (2020) proposed a CO_2-organic cascade cycle to maximize the heat recovery from a large marine diesel engine. Sakalis (2021) showed that the achievable range of efficiency improvement of a standalone marine diesel engine due to the waste heat recovery through supercritical CO_2 power cycles would be 6.6–7.25%. Zhang et al. (2020) revealed that the maximum achievable thermal efficiency of an engine waste heat-driven optimized recompression CO_2 power cycle would be about 35.86%.

Though substantial studies were conducted on engine waste heat-driven CO_2 power cycles, a very high turbine inlet pressure is a serious limitation of CO_2 power cycles. To reduce this limitation, many researchers had recommended using a mixture of CO_2 and organic fluid as the working fluid of power cycles (Shu et al. 2020). Liu et al. (2019) showed experimentally that while recovering the engine waste heat, a power cycle with a (0.6/0.4) CO_2/R134a mixture by volume exhibited better performance compared to the power cycle using pure CO_2. Shu et al. (2018) reported that using a (0.3/0.7) CO_2/R32 mixture by volume as the working fluid, the optimum operating pressure of an engine waste heat-driven power cycle could be decreased by even 1.4 MPa compared to that of the corresponding power cycle operating with pure CO_2. Yang (2017) also showed that the operating pressure and energy cost of a transcritical power cycle could be reduced by using different CO_2-based working fluid mixtures. Liu et al. (2021) reported that while using different CO_2-hydrocarbon pairs as the working fluid, an engine exhaust gas-driven split transcritical cycle exhibited the best performance with a CO_2-propane-based mixture.

It appears that engine waste heat recovery for additional power and other utility services is an emerging subject in the field of energy study. In the present study, operations of a regenerative transcritical power cycle with CO_2-propane mixture and CO_2-R152a mixture are explored to recover waste heat from a stationary diesel power plant. Quality and quantity of waste heat carried away with the exhaust gas differs substantially from that of the waste heat rejected by the engine coolant. Recoverable waste heat from these two sources can be optimally utilized to deliver the maximum output power at any specified turbine inlet pressure. Thus, an optimization algorithm is employed to find out the turbine inlet temperature corresponding to this operating condition. The optimization algorithm is also useful to determine the operating parameters that minimize the levelized electricity cost (LEC). The use of CO_2-based mixture not only ensures an environment-friendly operation but also provides some safeguards against any accidental flame propagation. In other words, the intention of the present study is to recommend a suitable CO_2-based working fluid mixture for the regenerative transcritical power cycle recovering low-grade and medium-grade waste heat from the jacket cooling water and the exhaust gas of a diesel power plant, respectively. Cycle power output, turbine inlet pressure and LEC are the considered performance parameter of the study.

Selection of working fluid

The success of any waste heat recovery scheme greatly depends on the selection of a suitable working fluid. The selected working fluid should exhibit reasonably good operating performance along with good economy and a minimal adverse effect on the environment. While selecting a working fluid, issues of the flammability and commercial availability should also be taken into consideration. As discussed in the introduction, many of the researchers are proposing CO_2 as one of the possible working fluids for engine waste heat recovery due to its commercial availability and also stability at a higher temperature. CO_2 is a non-flammable working fluid with zero ozone depletion potential (ODP). Global warming potential (GWP) of CO_2 is unity and this impact is not significant as it only re-circulates with possible small leakages only. The temperature profile of supercritical CO_2 also fits well with temperature profiles of the waste gases from diesel engines. In spite of so many advantages, a very high operating pressure and associated difficulties of the CO_2 power cycle is an issue of great concern (Xu et al. 2019). Thus, in the present study, to reduce the operating pressure of the supercritical CO_2 power cycle, different compositions of CO_2-propane and CO_2-R152a mixtures are used as working fluids of the waste heat-driven cycle.

It is necessary to mention that GWP and ODP of propane are 3 and 0, respectively. On the other hand, the GWP and ODP of R152a are 140 and 0, respectively (Sendil Kumar and Elansezhian. 2014). Thus, all considered compositions of the mixture working fluid ensure an acceptable environment-friendly operation. It should be noted that an inert working fluid can be mixed with a hydrocarbon to reduce the possibility of accidental flame propagation due to the burning of the hydrocarbon (Mondal and De 2019; Garg et al. 2013a, 2013b). In the considered working fluid mixture, CO_2 acts primarily as the diluents suppressing the higher flammability of Propane. The mass fraction of CO_2 is not allowed to fall below 0.3 to ensure the safety against the high flammability of propane (Garg et al. 2013a). It is necessary to note that R152a is less flammable than propane. Thus, the mixture of CO_2-R152a is also less flammable compared to the mixture of CO_2 propane. Mixing CO_2 with R152a also reduces the GWP value of R152a. It is necessary to note that the chance of accidental flame propagation is reduced with an increase in value of the lower flammability limit (LFL). Thus, variations of LFL of two considered mixture pairs are estimated by the suggested correlation of Schroeder (2016) and plotted against the CO_2 mass fraction in Fig. 1a. It is observed that the LFL of each working fluid mixture pair increases with an increase in CO_2 mass fraction. It is also evident from Fig. 1b that the GWP of the CO_2-R152a mixture pair decreases with an increase in CO_2 mass fraction in the mixture. It is evident from Figs. 1a, b that if the CO_2 mass fraction is increased beyond 0.3, the LFL of the CO2-propane mixture goes above 3% and the GWP of the CO2-R152a mixture comes below 100. As higher LFL and lower global warming potential (GWP) are two desirable properties of an ideal working fluid, the CO_2 mass fraction is not reduced below 0.3 in each working fluid mixture.

For any mixture of working fluids, very high glide of temperature is not desirable as it would result in a composition shift (Dai et al. 2014). It appears from Fig. 2a that temperature glides of the considered CO_2-based mixtures for all considered mass ratios remain well below 50 °C.

Finally, it also appears from Fig. 2b that mixing propane as well as R152a with CO_2 results in higher critical temperatures compared to that of pure CO_2. This eliminates the requirement of a low-temperature heat sink (at less than 30 °C) and permits operation of the exhaust gas heat-driven cycle in transcritical mode even for an ambient temperature of 25 °C. Operation in transcritical mode appreciably reduces compressor power consumption compared to the power consumed by the compressor of a supercritical cycle.

System description

The layout of the proposed power cycle recovering waste heat from a diesel power plant is presented in Fig. 3a. Figure 3b is the corresponding temperature entropy diagram. It appears from Table 1 that the exhaust gas and the engine coolant are the two principal carriers of the rejected waste heat from the considered diesel power plant. Thus, the presented power cycle consists of two heat recovery units

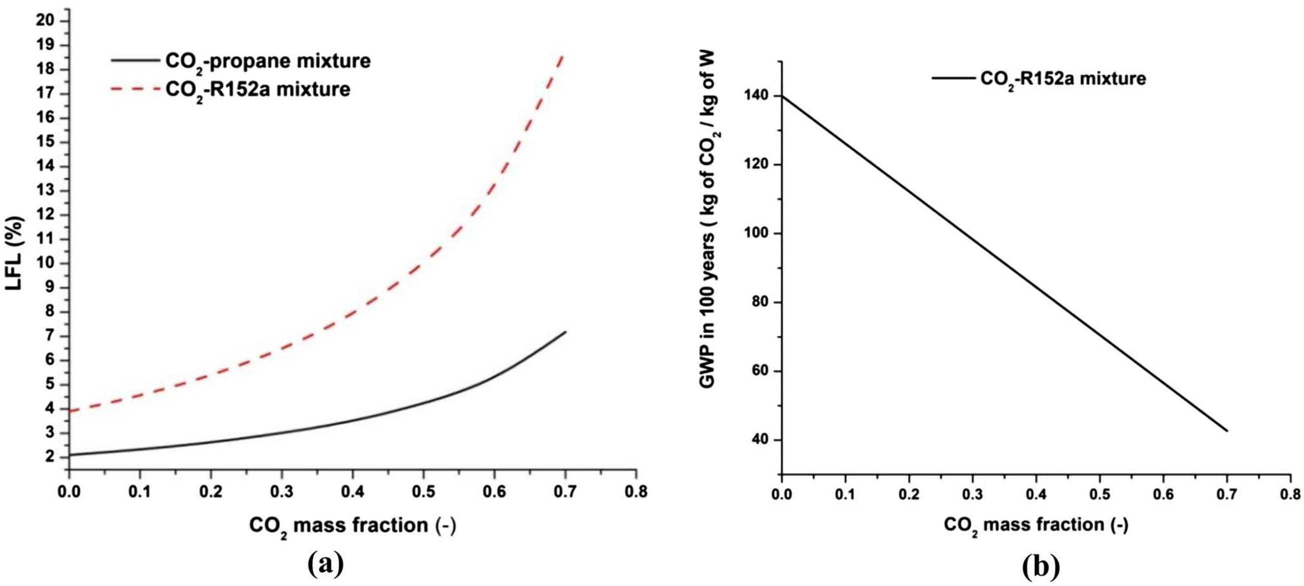


Fig. 1 **a** Lower flammability (LFL) of the mixture working fluid versus CO_2 mass fraction, **b** GWP of the CO_2-R152a mixture vs. CO_2 mass fraction

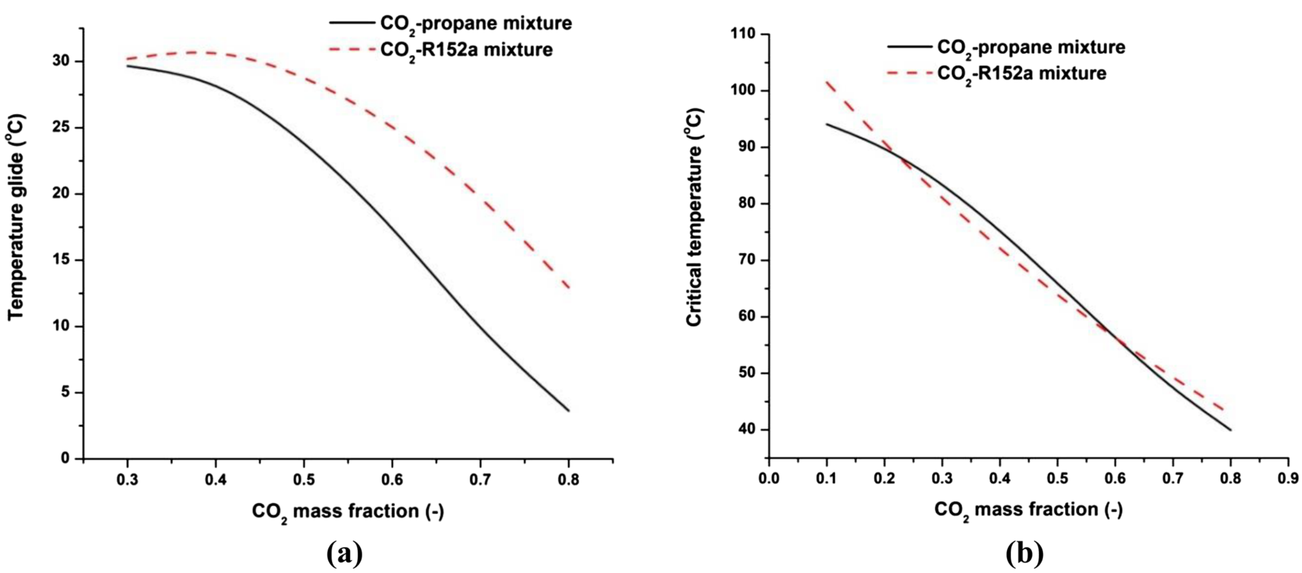


Fig. 2 **a** Effects of varying CO_2 mass fraction in each considered mixture working fluid on the temperature glide during condensation, **b** Effects of varying CO_2 mass fraction in each considered mixture working fluid on the critical temperature

(HRUs) recovering waste heat from the exhaust gas and also the same from the engine coolant. In the engine coolant HRU, the working fluid is heated from the compressor exit state (i.e., 7) to the regenerator inlet state (i.e., 7c). The hot working fluid after undergoing a heating process in the regenerator (process 7c-8) enters into the exhaust gas HRU. In the exhaust gas HRU, the working fluid is heated up to the turbine inlet state. The mixture working fluid exiting the exhaust gas HRU is expanded through the radial flow turbine (process 3–4) to produce the output power. A fraction of the turbine power is expensed to run the compressor.

Modeling and methodology

The objective of this study is to explore an optimum waste heat recovery scheme for a diesel power plant using CO_2-propane and CO_2-R152a mixture-based transcritical

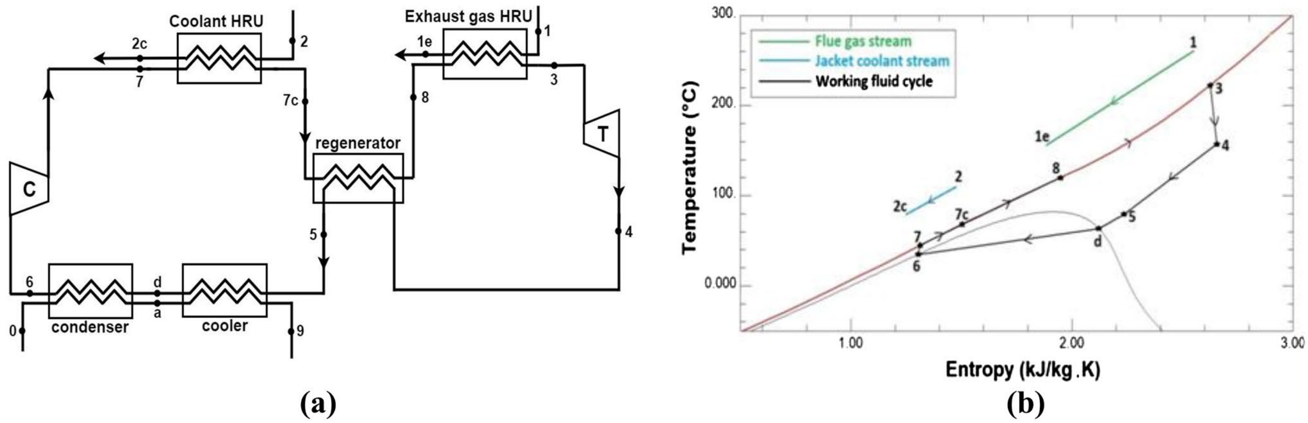


Fig. 3 **a** Layout of the waste heat-driven cycle, **b** T-s diagram of the waste heat-driven cycle

Table 1 Diesel power plant waste heat data (Morawski et al. 2021)

Input parameters	Value
Engine shaft power (MW)	9
Electric power (MW)	8.73
Specific fuel consumption (g /kWh)	183
Exhaust gas temperature (°C)	345
Mass flow rate of exhaust gas (kg/s)	16.7
Temperature of jacket cooling water	78.4
Cooling water available (kg/s)	27.02

power cycles. A mathematical model is developed for this work based on the following simplified assumptions:

i. The system operates at steady flow condition
ii. Isentropic efficiencies of the compressor and the turbine are both assumed to be 85%.
iii. The ambient condition is specified to be 100 kPa and 25 °C.
iv. The chemical exergy has been ignored as the composition of the working fluid remains constant throughout the cycle.
v. All the heat exchangers (HE) are assumed to be of shell and tube type.
vi. In both the HRUs, the working fluid is assumed to flow through tubes of the HE.
vii. Working fluid mass flux is assumed to be constant at 350 kg/m^{2}s
viii. Flue gas velocity over the tube bank is not allowed to go above 10 m/s.
ix. Maximum velocity of engine coolant in the HRU is assumed to be 0.75 m/s
x. In the regenerator, high-pressure working fluid flows through the tube, while the low pressure working fluid flows through the shell side.
xi. The cooling water is available at 25 °C and the minimum cycle temperature is assumed to be 35 °C.
xii. Flue gas contains SO_2 and the acid dew point temperature of the flue gas is assumed to be 120 °C (so the flue gas can be cooled up to a minimum of 130 °C for additional safety).
xiii. The whole heat exchanger system is designed with a pinch point temperature difference of 10 °C.

Thermodynamic performance estimation

Cycle power output and the 2nd law efficiency are the two parameters used to assess the thermodynamic performance of the waste heat recovery scheme. For any specified composition of the working fluid, the working fluid mass flow rate is estimated from the energy balance of the exhaust gas HRU as presented below:

$$\dot{m}_{\mathrm{WF}} = \frac{\dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}} \left(T_1 - T_{1\mathrm{e}}\right)}{h_3 - h_8} \tag{1}$$

The mass flow rate of the engine coolant (i.e., Jacket cooling water) through the engine coolant HRU is estimated as

$$\dot{m}_{\mathrm{Jcw}} = \frac{\dot{m}_{\mathrm{WF}} \left(h_{7\mathrm{c}} - h_7\right)}{C_{\mathrm{pJCw}} \left(T_2 - T_{2\mathrm{c}}\right)} \tag{2}$$

Turbine power output, compressor power output, net cycle power output and 1st law efficiency are estimated by Eqs. (3)–(6).

Table 2 component exergy destruction equations

Component	Exergy destruction (kW)
Turbine	$\dot{m}_{\mathrm{WF}} T_{\mathrm{o}}(s_4 - s_3)$
Compressor	$\dot{m}_{\mathrm{WF}} T_{\mathrm{o}}(s_7 - s_6)$
Condenser	$\dot{m}_{\mathrm{WF}} T_{\mathrm{o}}(s_6 - s_d) + \dot{m}_{\mathrm{cond}} T_{\mathrm{o}}(s_a - s_{\mathrm{o}})$
Cooler	$\dot{m}_{\mathrm{WF}} T_{\mathrm{o}}(s_d - s_5) + \dot{m}_{\mathrm{cond}} T_{\mathrm{o}}(s_9 - s_a)$
Regenerator	$\dot{m}_{\mathrm{WF}} T_o\{(s_8 - s_{7c}) + (s_5 - s_4)\}$
Exhaust gas HRU (EGHRU)	$\dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}} T_o \ln\left(\frac{T_{1e}}{T_1}\right) + \dot{m}_{\mathrm{WF}} T_o(s_3 - s_8)$
Jacket cooling water HRU (JCWHRU)	$\dot{m}_{\mathrm{Jcw}} C_{\mathrm{pJcw}} T_{\mathrm{o}} \ln\left(\frac{T_{2c}}{T_2}\right) + \dot{m}_{\mathrm{WF}} T_{\mathrm{o}}(s_{7c} - s_7)$

$$\dot{W}_{\mathrm{t}} = \dot{m}_{\mathrm{WF}}(h_3 - h_4) \tag{3}$$

$$\dot{W}_{\mathrm{c}} = \dot{m}_{\mathrm{WF}}(h_7 - h_6) \tag{4}$$

$$\dot{W}_{\mathrm{net}} = \dot{W}_{\mathrm{t}} - \dot{W}_{\mathrm{c}} \tag{5}$$

$$\eta_{\mathrm{I}} = \frac{\dot{W}_{\mathrm{net}}}{\dot{m}_{\mathrm{Jcw}} C_{\mathrm{pw}}(T_2 - T_{2c}) + \dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}}(T_1 - T_{1e})} \tag{6}$$

Equations for exergy destructions (irreversibilities) of different components are summarized in Table 2:

Now, 2nd law efficiency of the overall system is expressed as

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{\mathrm{net}}}{\dot{E}_{\mathrm{EGi}} + \dot{E}_{\mathrm{Jcwi}}} = \frac{\dot{E}_{\mathrm{Egi}} + \dot{E}_{\mathrm{Jcwi}} - \Sigma\dot{E}_{\mathrm{D}} - \dot{E}_{\mathrm{L\,Eg}} - \dot{E}_{\mathrm{L\,Jcw}} - \dot{E}_{\mathrm{L\,coolant}}}{\dot{E}_{\mathrm{EGi}} + \dot{E}_{\mathrm{Jcwi}}} \tag{7}$$

Exergy inputs with engine exhaust gas and engine coolant are estimated by Eq. (8) and (9), respectively:

$$\dot{E}_{\mathrm{EGi}} = \dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}}(T_1 - T_o) - \dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}} T_o \ln\left(\frac{T_1}{T_o}\right) \tag{8}$$

$$\dot{E}_{\mathrm{Jcwi}} = \dot{m}_{\mathrm{Jcwav}} C_{\mathrm{pw}}(T_2 - T_o) - \dot{m}_{\mathrm{Jcwav}} C_{pJCw} T_o \ln\left(\frac{T_2}{T_o}\right) \tag{9}$$

Exergy losses with different outgoing streams are presented as follows:

$$\dot{E}_{\mathrm{LEG}} = \dot{m}_{\mathrm{EG}} C_{\mathrm{pEG}}(T_{1e} - T_o) - \dot{m}_{\mathrm{EG}} C_{\mathrm{pg}} T_o \ln\left(\frac{T_{1e}}{T_o}\right) \tag{10}$$

$$\dot{E}_{\mathrm{L\,Jcw}} = \dot{m}_{\mathrm{Jcw}} C_{\mathrm{pw}}(T_{2c} - T_o) - \dot{m}_{\mathrm{Jcw}} C_{\mathrm{pJcw}} T_o ln\left(\frac{T_{2c}}{T_o}\right) + (\dot{m}_{\mathrm{Jcwav}} - \dot{m}_{\mathrm{Jcw}}) e_{\mathrm{Jcwi}} \tag{11}$$

$$\dot{E}_{\mathrm{Lcoolant}} = \dot{m}_{\mathrm{cond}}\{(h_9 - h_0) - T_o(s_9 - s_0)\} \tag{12}$$

Economic assessment

The objective of the economic assessment simultaneously with the thermodynamic performance study is to estimate the levelized electricity (LEC) cost for the proposed waste heat recovery scheme. The economic assessment begins with the bare module cost estimation of equipment constituting the waste heat recovery system.

The bare module cost of any heat exchanger is a function of its heat transfer surface area and associated operating pressure. As shown in Fig. 4, the heat exchanger has been discretized into N subsections assuming an equal enthalpy drop is occurring across each subsection. Finally, the temperatures of the tube side fluid are evaluated accordingly at each subsection.

The LMTD at each (i.e., ith) section has been calculated as

$$\mathrm{LMTD_i} = \frac{(T_{\mathrm{i}} - T(h_{\mathrm{i}})) - (T_{\mathrm{i+1}} - T(h_{\mathrm{i+1}}))}{ln\left(\frac{T_{\mathrm{i}} - T(h_{\mathrm{i}})}{T_{\mathrm{i+1}} - T(h_{\mathrm{i+1}})}\right)} \tag{13}$$

The heat recovery at each and every subsection is estimated as

$$\dot{q} = \frac{\dot{Q}_{\mathrm{FGHRU}}}{N} \tag{14}$$

The area required at each and every subsection has been calculated as:

$$A_{\mathrm{i}} = \frac{\dot{q}}{\mathrm{FU_i LMTD_i}} \tag{15}$$

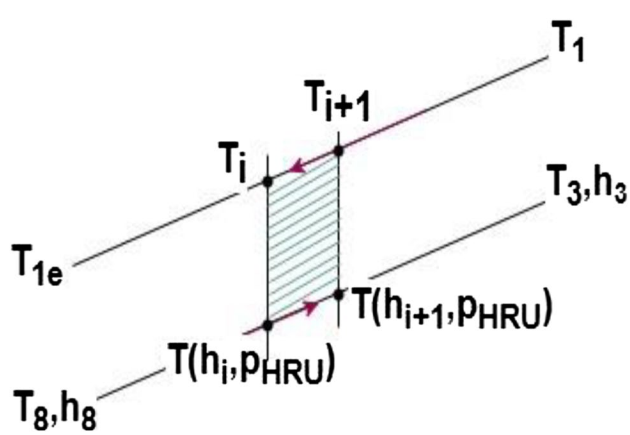


Fig. 4 Discretization of the flue gas HRU

In above equation, F is the correction factor and U_i is the overall heat transfer coefficient. The overall heat transfer coefficient is expressed as

$$\frac{1}{U_\mathrm{i}} = \frac{1}{\alpha_\mathrm{si}} + \frac{1}{\alpha_\mathrm{ti}} \tag{16}$$

α_{si} and α_{ti} are shell side convective heat transfer coefficient and tube side convective heat transfer coefficient for element i of the HE.

Shell side heat transfer coefficient is estimated by using following correlation of fluid flowing over the tube bank (Cengel and Ghajar 2011):

$$\mathrm{Nu}_\mathrm{si} = 0.35\left(\frac{S_\mathrm{T}}{S_\mathrm{L}}\right)^{0.2}\mathrm{Re}_\mathrm{si}^{0.6}\mathrm{Pr}_\mathrm{si}^{0.36} \tag{17}$$

For supercritical CO_2, Protopopov co.relation (without wall correction factor) is applied to estimate the associated Nusselt number (Pioro et al. 2004). Heat transfer area of a heat exchanger is estimated by adding areas of all discretized elements.

Once the areas of all heat exchangers, turbine and compressor capacities are estimated, the purchase cost of any equipment is estimated by using the following equation (Turton et al. 2013)

$$\log_{10} C_\mathrm{p}^0 = K_1 + K_2 \log_{10} Z + K_3\left[\log_{10} Z\right]^2 \tag{18}$$

Hence, the bare module cost of individual equipment is expressed as

$$C_\mathrm{bm} = C_\mathrm{p}^0\left(B_1 + B_2 F_\mathrm{m} F_\mathrm{p}\right) = C_\mathrm{p}^0 F_\mathrm{BM} \tag{19}$$

Table 3 Coefficient for total cost estimation (Turton et al. 2013)

Equipments	Performance parameters (Z)	K_1	K_2	K_3	B_1	B_2	F_M	C_1	C_2	C_3
FGHRU	A_{FGHRU} (m^2)	4.8306	−0.8509	0.3187	1.63	1.66	1.3178	−0.00164	−0.00627	0.0123
Jacket CWHRU	A_{CWHRU} (m^2)	4.3247	−0.3030	0.1634	1.63	1.66	1.3178	−0.00164	−0.00627	0.0123
Regenerator	A_{reg} (m^2)	4.8306	−0.8509	0.3187	1.63	1.66	1.3178	−0.03881	−0.11272	0.08183
Condenser	A_{cond} (m^2)	4.3247	−0.3030	0.1634	1.63	1.66	1.3178	−0.00164	−0.00627	0.0123
Turbine	W_t (kW)	2.2476	1.4965	−0.1618	–	–	3.4825	–	–	–
Compressor	W_c (kW)	2.2897	1.3604	−0.1027	–	–	2.433	–	–	–

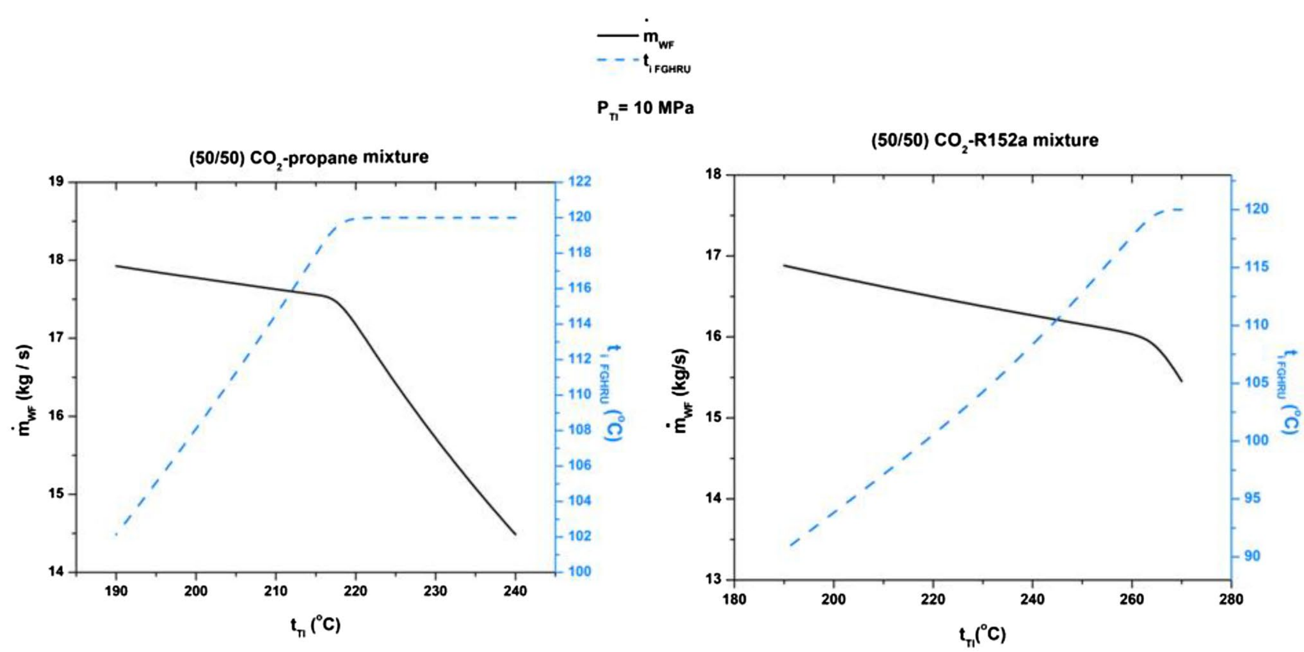


Fig. 5 Effects of turbine inlet temperature on working fluid mass flow rate and working fluid inlet temperature to FGHRU

F_{BM} is the bare module factor. F_m&F_p are material correction and pressure correction factors, respectively. The pressure correction factor for heat exchangers is estimated by the following equation:

$$\log_{10} F_p = C_1 + C_2 \log_{10}(P) + C_3\{\log_{10}(P)\}^2 \quad (20)$$

In Eq. (20), P is the operating pressure in bar gauge. The total equipment cost is converted in to the cost of current date by applying Eq. (21).

$$C_{Tot} = \left(\sum C_{BM,eq}\right) \times \left(\frac{\mathrm{CEPCI}_{\text{current year}}}{\mathrm{CEPCI}_{2001}}\right) \quad (21)$$

The values of different coefficients of bare module cost estimation are summarized in Table 3.

Finally, the levelized electricity cost is estimated as follows

$$\mathrm{LEC} = \frac{\mathrm{CRF} \times C_{Tot} + C_{OM}}{\dot{W}_{net} \times \mathrm{AOH}} \quad (22)$$

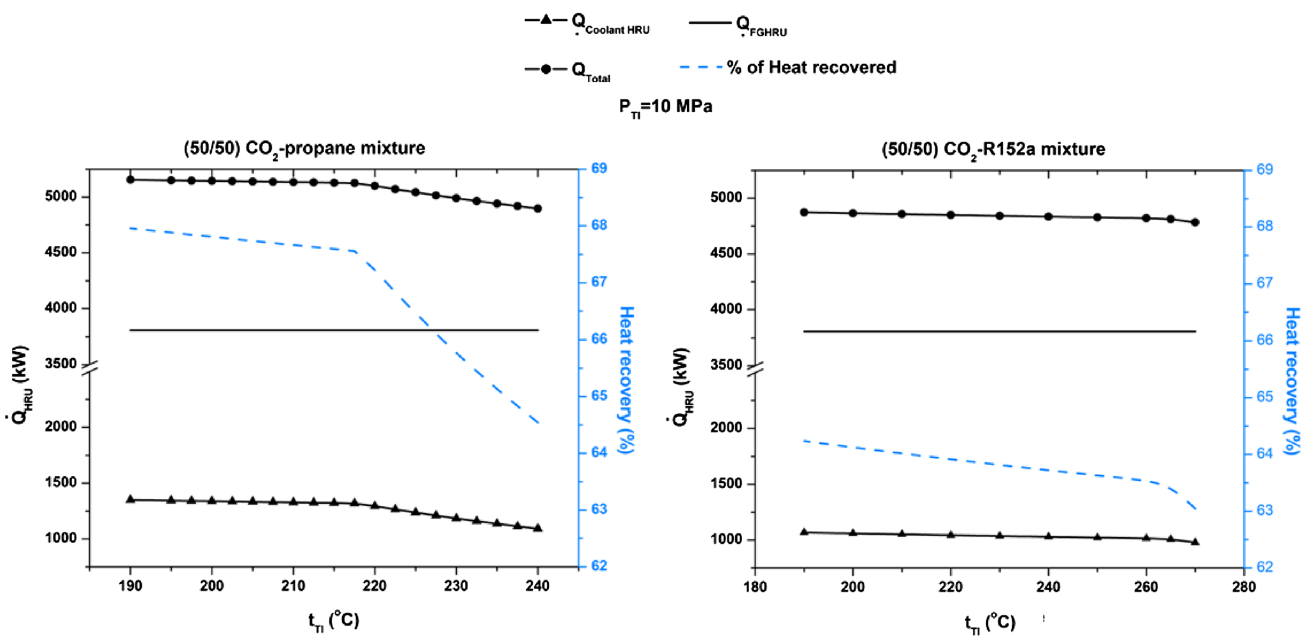


Fig. 6 Effects of varying turbine inlet temperature of waste heat recovery

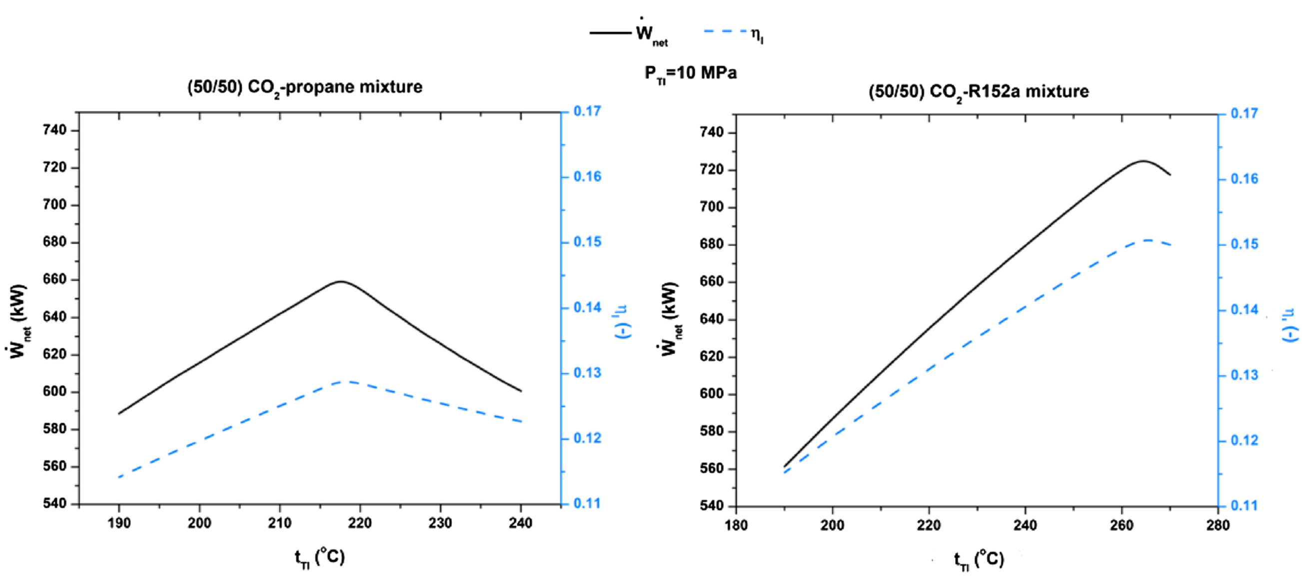


Fig. 7 Effects of varying turbine inlet temperature in net work output and 1st law efficiency

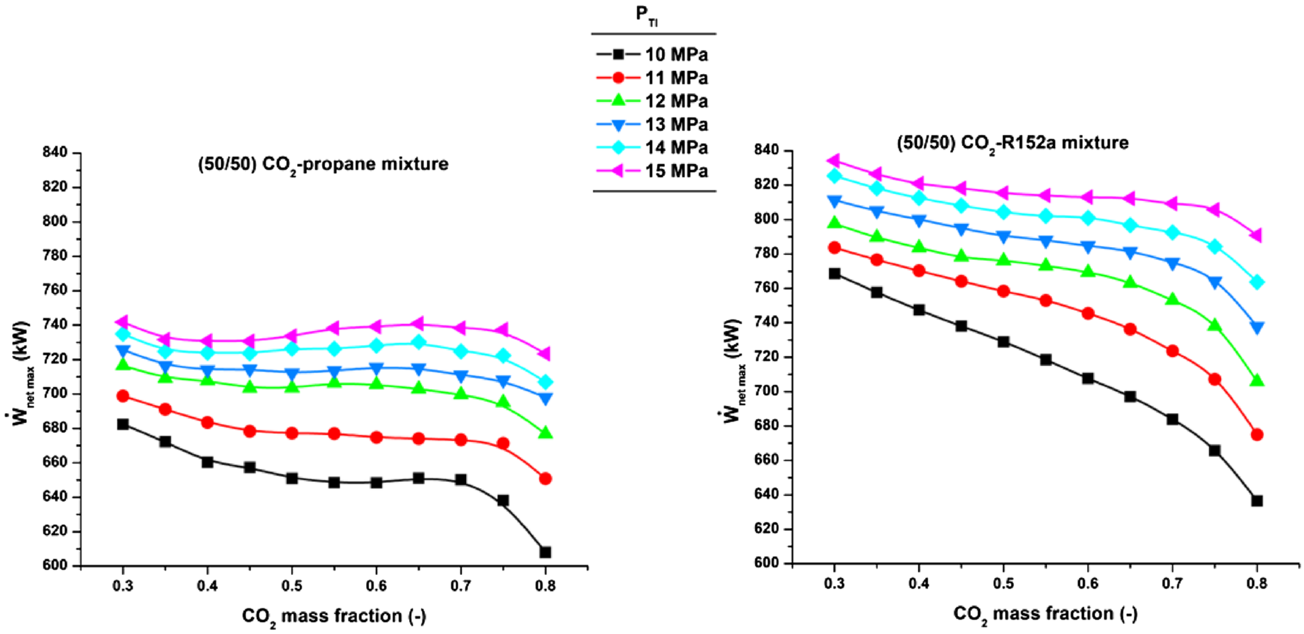


Fig. 8 Effects of varying CO_2 mass fraction on maximum cycle power output at different turbine inlet pressure

In Eq. (22), annual operating hour (AOH) is assumed to be 8000 h. Annual operation and maintenance cost is 5% of the total capital investment. The capital recovery factor (CRF) is estimated from the following equation:

$$\mathrm{CRF} = \frac{i(1+i)^{\mathrm{LT}}}{(1+i)^{\mathrm{LT}} - 1} \tag{23}$$

While estimating the CRF, life of the plant and interest rate are assumed 25 years and 5%, respectively.

Results and discussion

In the present study, CO_2-propane and CO_2-R152a mixtures with different CO_2 mass fractions are used as the working fluid of a transcritical power cycle recovering the diesel power plant waste heat. Initially, in this section, the effects of different input parameters on cycle performance are discussed for a specified mixture composition of each considered working fluid pair. Effects of varying mixture composition on cycle performance are also discussed in the subsequent stages of the study. Finally, the optimum levelized electricity cost (LEC) and associated operating parameters are presented and compared with those of the optimized supercritical CO_2 power cycle recovering the waste heat of a similar source. It is important to note that the proposed power cycle consists of two heat recovery units and a regenerator. The quantity of recoverable high grade waste heat differs appreciably compared to that of the recoverable low-grade waste heat. While optimizing, maximum usable high grade waste heat is recovered due to its greater exergy content.

Effects of varying turbine inlet temperature on working fluid mass flow rate and inlet temperature of working fluid to the FGHRU are presented in Fig. 5. For both the considered working fluid pairs (i.e., CO_2-propane and CO_2-R152a), working fluid mass flow rates initially decrease at a slower rate with an increase in turbine inlet temperature. However, mass flow rates of both the working fluid pairs start to decrease at a faster rate as the turbine inlet temperature is raised above a certain value. With an increase in turbine inlet temperature, the specific enthalpy change of each working fluid pair in the FGHRU increases. However, with the initial increase in the turbine inlet temperature, this increase is not very rapid as the inlet temperature of each considered working fluid to the FGHRU increases due to the increasing heat duty of the regenerator. However, as a reasonable pinch point temperature difference (i.e., 10 °C or above) is to be maintained in the FGHRU, the temperature of the high-pressure working fluid exiting the regenerator is not allowed to go above 120 °C, as shown in Fig. 5. As the temperature of the high-pressure working fluid stream reaches 120 °C, the mass flow rate of each working fluid pair through the FGHRU starts to decrease rapidly with a further increase of the turbine inlet temperature. It is mostly due to the rapid increase in specific enthalpy change of the working fluid.

Effects of varying turbine inlet temperature on heat recoveries in different HRUs are presented in Fig. 6. Always the entire recoverable heat of the exhaust gas is desired to

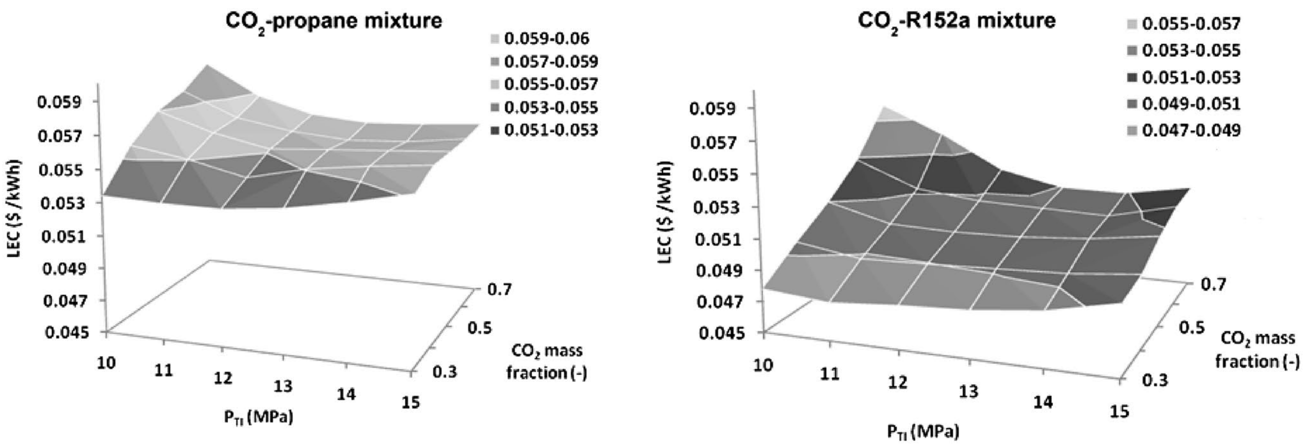


Fig. 9 Effects of varying CO_2 mass fraction and turbine inlet pressure on levelized electricity cost

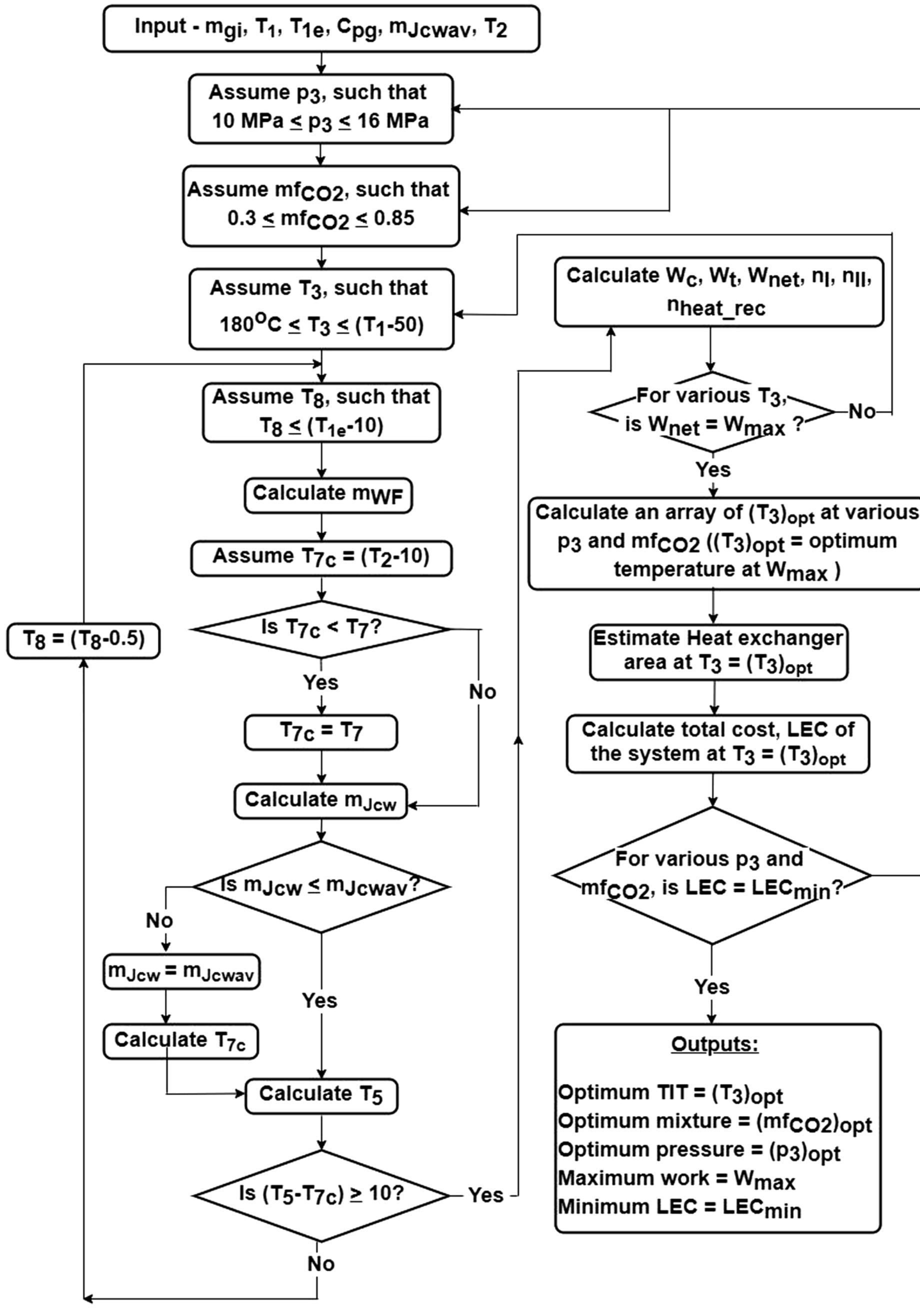
be recovered due to its higher exergy content. Hence, the heat duty of the FGHRU is set to be independent of the turbine inlet temperature. It is evident from Fig. 6 that for each of the considered working fluid pairs, heat recovery in the coolant HRU and the total % of heat recovery rapidly decrease above a certain turbine inlet temperature due to the rapid decrease in the working fluid mass flow rate. It also appears from Fig. 6 that the heat recovery percentage starts to decrease rapidly beyond a turbine inlet temperature of 215 °C for the CO_2-propane mixture-based system. For the CO_2-R152a-based transcritical cycle, a similar rapid decrement is observed beyond a turbine inlet temperature of 260 °C. The achievable total heat recovery percentage is also somehow smaller for the CO_2-R152a-based working fluid mixture compared to that achievable with the same percentage of the CO_2-propane mixture.

It is observed in Fig. 7 that for any specified turbine inlet pressure of each CO_2-based transcritical power cycle, there is an optimum turbine inlet temperature corresponding to the maximum cycle power output (or maximum 1st law efficiency). As the regenerator exit temperature of the higher pressure working fluid stream is not allowed to go above 120 °C, the turbine inlet temperature corresponding to this particular operating condition results in the highest 1st law efficiency and power output. It also appears from Fig. 7 that for any specified CO_2 mass fraction, the achievable output power for the CO_2-R152a mixture is substantially higher than that can be achieved with the CO_2-propane-based working fluid mixture. As the optimum turbine inlet temperature of the CO2-R152a mixture-based cycle is much higher than that of the CO2-propane-based cycle, noticeably higher 1st law efficiency is achieved using the CO2-R152a-based mixture. Obviously higher 1st law efficiency indicates higher power output from the CO_2-R152a-based power cycle.

Effects of varying CO_2 mass fraction in CO_2-propane mixture as well as in CO_2-R152a mixture on maximum achievable power outputs corresponding to different turbine inlet pressures of the transcritical power cycle are presented in Fig. 8. It is observed that for any specified working fluid composition of each mixture, cycle power output increases with an increase in turbine inlet pressure. However, above a certain turbine inlet pressure, this increment is not very significant. It appears from Fig. 8 that at a lower turbine inlet pressure, reducing the CO_2 mass fraction in each considered CO_2-based mixture significantly enhances cycle power output. For a turbine inlet pressure above a certain value, net cycle power output remains more or less constant for the CO_2 mass fraction of each working fluid pair varying between 0.3 and 0.75. For all considered turbine inlet pressures, raising the CO_2 mass fraction above 0.75 results in a rapid decrease in cycle power output for each of the considered CO_2-based working fluid mixtures.

For the two considered CO_2-based mixture pairs, the effects of simultaneous variations of the CO_2 mass fraction and the turbine inlet pressure on the levelized electricity cost (LEC) are presented in Fig. 9. It appears from Fig. 9 that LEC is a minimum if the cycle is operated with a CO_2 mass fraction of 0.3 in each of the considered CO_2-based working fluid mixtures. It is also observed that for each specified CO_2-based mixture pair, LEC at a higher turbine inlet pressure is a weak function of the working fluid composition. It is observed in Fig. 9 that for all specified working fluid compositions, variation in turbine inlet pressure significantly affects the LEC. For each specified composition of the working fluid, there exists an optimum turbine inlet pressure corresponding to a minimum LEC of the proposed waste heat recovery scheme using any of the considered CO_2-based mixtures.

Fig. 10 Optimization algorithm



As the objective of the present study is to select the superior working fluid pair between the CO_2-propane and CO_2-R152a, the optimum operating condition of the diesel power plant waste heat-driven transcritical power cycle with each considered working fluid pair is to be explored. Thus, to serve this purpose, an optimization algorithm is proposed as presented in Fig. 10. While recovering the maximum usable high grade waste heat, the presented optimization algorithm of Fig. 10 initially estimates the optimum turbine inlet temperature corresponding to the maximum power output at any specified turbine inlet pressure. The optimization algorithm also iterates the turbine inlet pressures and mixture composition to find out their optimum value corresponding to the minimum levelized electricity cost (LEC). Pinch point temperature differences of all heat exchangers, maximum usable low-grade waste heat in the coolant HRU, acid condensation temperature of the flue gas and minimum allowable mass fraction of CO_2 (depending on fire safety issue and GWP value) are applied constraints of the optimization algorithm.

Minimum levelized electricity cost (LEC) and associated other parameters for each of the considered CO_2-based mixture pair, obtained by applying the optimization algorithm are summarized in Table 4. For the CO_2-propane mixture,

Table 4 Parameters corresponding to minimum LEC

Type of cycle	t_{TI}(°C)	P_{TI}(MPa)	P_{Te} (MPa)	CO_2mass fraction	$\dot{W}_{net}$(kW)	η_{II}(%)	LEC_{Min}($/kWh)
Transcritical cycle with CO_2-propane mixture	222.50	12.00	56	0.3	711.25	30.93%	0.053
Transcritical cycle with $CO_2{}^{-}$R152a mixture	260.05	11.00	2.85	0.3	782.25	34.02%	0.048

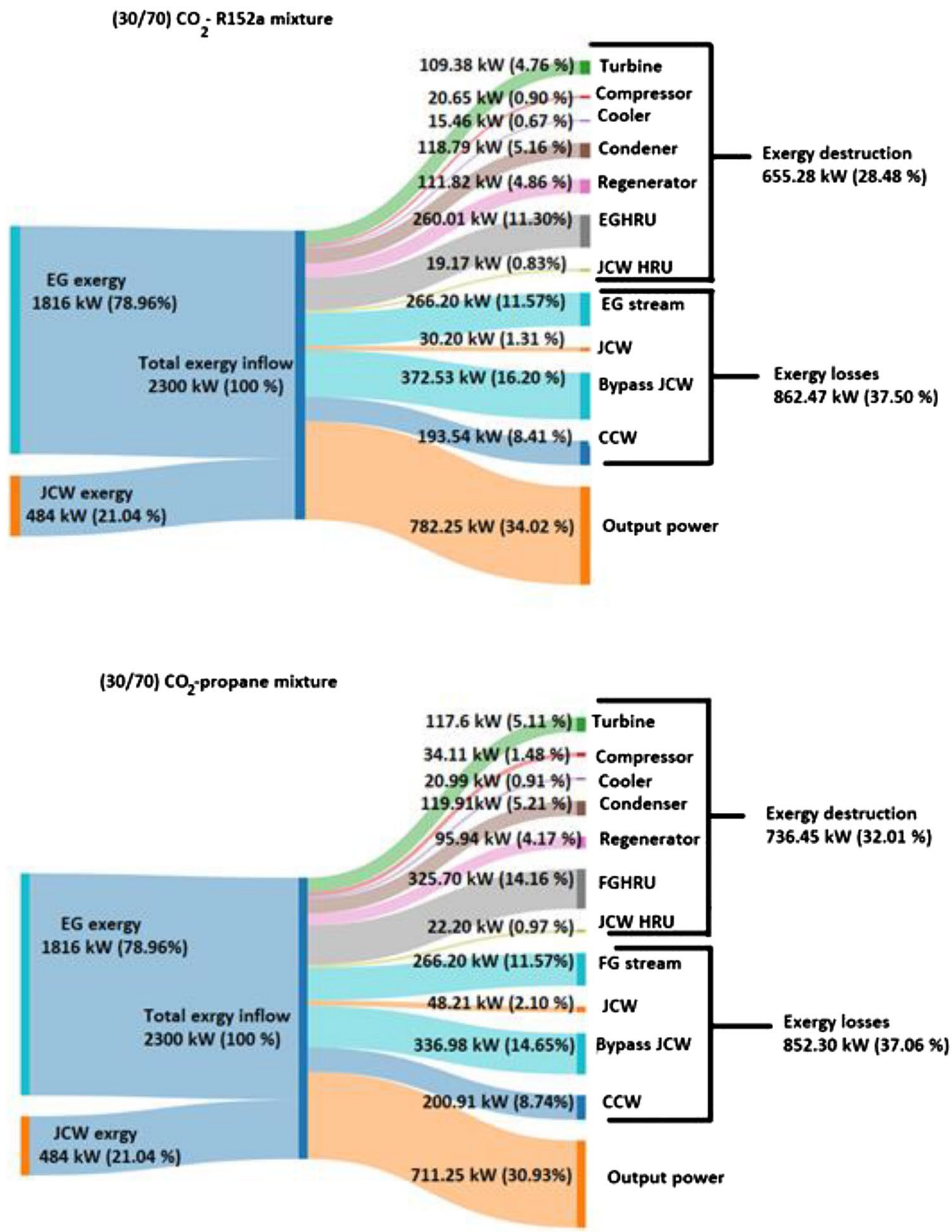


Fig. 11 Grassmann diagrams for optimized cycles

the estimated 2nd law efficiency corresponding to this optimized state is close to 31%. On the other hand, 2nd law efficiency corresponding to the minimized LEC of the transcritical cycle using CO_2-R152a mixture is close to 34%.

For both the considered CO_2-based mixture working fluids, Grassmann diagrams corresponding to the operating conditions of Table 4 are presented in Fig. 11. These Grassmann diagrams depict exergy destructions of different components and exergy losses associated with different fluid streams exiting the waste heat recovery system. It is observed that for each considered working fluid pair, the largest exergy destruction occurs in the exhaust gas heat recovery unit (EGHRU). It appears that using the CO_2-R152a mixture instead of the CO_2-propane mixture as the working fluid of the transcritical power cycle, the exergy destruction of the EGHRU can be reduced by 2.86%. The higher turbine inlet temperature is responsible for this reduction of exergy destruction. Due to the reduced exergy destruction in the EGHRU, the achievable 2nd law efficiency of the CO_2-R152a-based transcritical power cycle is somehow higher than that of the CO_2-propane-based transcritical power cycle. For both the working fluid pairs, about 11.57% exergy loss occurs with the exhaust gas stream exiting the HRU. However, this exergy loss with the exhaust gas stream is unavoidable due to the constraint of maintaining the final gas exit temperature above certain value to avoid any acid condensation. It is necessary to mention that at the optimum operating condition, the entire recoverable waste heat of the engine coolant (or jacket cooling water) is not possible to utilize. Thus, about 14.65% exergy loss occurs with the jacket cooling water (JCW) mass bypassing the engine coolant HRU of the CO_2-propane-based transcritical cycle. For the CO_2-R152a mixture-based transcritical cycle, exergy loss with the bypass jacket cooling water is about 16.2%.

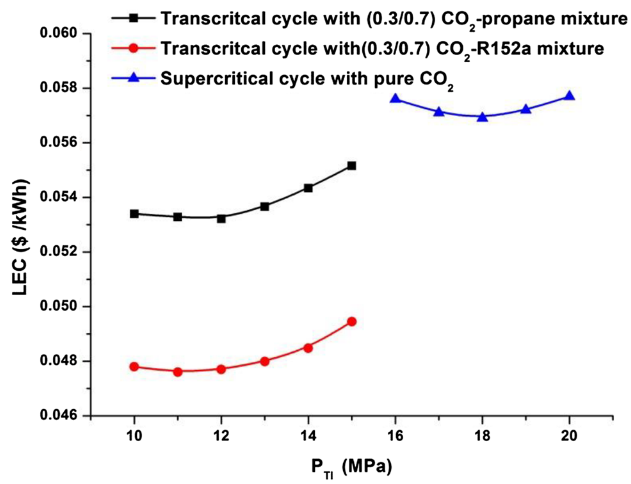


Fig.12 Comparison of minimum achievable LEC of the mixture-based cycle with the LEC of the supercritical CO_2 power cycle

For a better understanding, the state point properties of the optimized transcritical power cycle with each considered CO_2-based mixture pair are also summarized in Table 5.

In Fig. 12, minimum levelized electricity costs for different turbine inlet pressures of the two considered mixture-based cycles are compared with those of the optimized supercritical CO_2 power cycle. It appears from Fig. 12 that the minimum achievable LEC of the transcritical cycle with the CO_2-propane mixture working fluid is 6.36% lower

Table 5 State points of the optimized transcritical cycles

Streams	State points	Transcritical cycle with CO_2–propane mixture				Transcritical cycle with CO_2–R152a mixture			
		$t(°C)$	P(MPa)	$h\left(\frac{kJ}{kg}\right)$	$s\left(\frac{kJ}{kg.\ K}\right)$	t(°C)	P(MPa)	$h\left(\frac{kJ}{kg}\right)$	$s\left(\frac{kJ}{kg.K}\right)$
Working fluid	3	222.50	12.000	834.19	2.6248	260.05	11.000	755.80	2.3992
	4	157.38	3.5606	759.611	2.6556	178.77	2.8521	684.67	2.4273
	5	79.54	3.5606	595.991	2.2358	82.21	2.8521	551.522	2.0953
	D	63.72	3.5606	556.52	2.1212	65.19	2.8521	524.59	2.0176
	6	35.00	3.5606	291.73	1.3046	35.00	2.8521	267.45	1.2262
	7	44.95	12.000	310.64	1.3135	41.94	11.000	278.64	1.2315
	7c	68.40	12.000	373.35	1.5037	68.40	11.000	331.54	1.3926
	8	120.00	12.000	536.97	1.9486	120.00	11.000	464.69	1.7533
Exhaust gas	1	345.00	0.1013	–	–	345.00	0.1013	–	–
	1e	130.00	0.1013	–	–	130.00	0.1013	–	–
JCW	2	78.40	0.1013	–	–	78.40	0.1013	–	–
	2c	54.95	0.1013	–	–	51.94	0.1013	–	–
CCW	0	25.00	0.1013	–	–	25.00	0.1013	–	–
	A	53.71	0.1013	–	–	55.19	0.1013	–	–
	9	57.99	0.1013	–	–	58.36	0.1013	–	–

compared to that of the supercritical power cycle using pure CO_2 as the working fluid. For the CO_2-R152a mixture-based cycle, the corresponding achievable reduction in LEC is about 15.20%. It is also apparent that turbine inlet pressures corresponding to the minimum LECs of the optimized CO_2-propane and CO_2-R152a mixture-based cycles are, respectively, close to 33% and 39% lower than that of the optimized supercritical CO_2 power cycle.

Conclusions

The present study explores the optimum operation of a transcritical regenerative power cycle recovering waste heat of a diesel power plant. CO_2-propane and CO_2-R152a mixtures with different CO_2 mass ratios are used as the working fluid of the power cycle to ensure a safe and environment-friendly operation. An optimization algorithm is also employed to ensure the best possible performance of the presented power cycle operating with each consider CO2-based mixture working fluid. The major findings of the study are summarized as follows:

- It is observed that, at a lower turbine inlet pressure, cycle power output significantly increases with a decreasing CO_2 mass fraction in each of the considered CO_2-based mixture working fluids. The corresponding reduction in levelized electricity cost (LEC) is also found to be significant. For both the considered working fluid pairs, at a higher turbine inlet pressure, effects of varying mixture composition are having less significant effects on cycle power output and LECs.
- The minimum LEC for the presented transcritical cycle operating with the CO_2^{-}propane mixture working fluid is about 6.36% lower compared to that of an optimized supercritical CO_2 power cycle recovering waste heat from a similar source. The corresponding achievable reduction in LEC with the CO_2-R152a mixture is about 15.2%. Turbine inlet pressures corresponding to minimum LECs of the transcritical power cycles using CO_2-propane mixture and CO_2-R152a mixture are, respectively, 33% and 39% lower compared to that of the supercritical CO_2 power cycle.

Finally, diesel power plant waste heat recovery through a transcritical power cycle with the CO_2-propane mixture working fluid yields about 8% extra output power. About 8.7% extra power output is achievable if the transcritical power cycle is operated with the CO_2-R152a mixture working fluid. As R152a is less flammable than propane, an R152a-based mixture working fluid also ensures a safer operation compare to a propane-based mixture.

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Declarations

Conflict of interest The authors declare no competing interests.

Data availability Not applicable.

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