Diesel engine waste heat recovery using a CO_2-Propane mixture based trans-critical power cycle: Optimization using Energy, Exergy and Economic Analyses

A thesis submitted in partial fulfillment of the requirements for the degree of Master in Mechanical Engineering

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Contents

1. Abstract

Though CO_2 power cycles are preferred for diesel engine waste-heat recovery, cost of equipments and complexity of equipments for a CO_2 power cycle is an issue of concern. For mitigating this issue, in my ME thesis work, CO_2/ Propane mixtures with various CO_2 mass fraction ratios are used as working fluid for a regenerative trans-critical power cycle diesel power plant waste heat recovery. We have also used an optimization algorithm to ensure the best performance – both power and cost wise. To avoid accidental flame propagation, the minimum permissible CO_2 mass fraction is restricted to 0.3. At a lower turbine inlet pressure, we can have a higher output power by reducing CO_2 mass fraction, and lesser levelized electricity cost (LEC). By using the presented waste heat recovery scheme, we can also enhance the output power of the diesel power plant by 8%. The LEC of the presented optimized cycle is about 6.36% lower compared to that of the optimized supercritical CO_2 power cycle recovering the same waste heat. Turbine inlet pressure corresponding to the minimum LEC of the optimized mixture-based cycle is close to 33% lower than that of the optimized supercritical CO_2 power cycle. Finally, while recovering diesel power plant waste-heat, the optimized mixture-based cycle can ensure a safe, environment friendly and economically feasible operation at a significantly lower operating pressure.

Key words:

Diesel power plant waste heat; CO_2/Propane mixture; Transcritical power cycle; Output power; Levelized electricity cost.

2. Nomenclature

Symbol	
$\dot{m}$	Mass flow rate (kg/s)
C_p	Specific heat capacity (kJ/kgK)
t	Temperature (^{o}C)
T	Temperature (K)
h	Specific enthalpy (kJ/kg)
s	Specific entropy (kJ / kg K)
u	Fluid velocity (m /s)
$\dot{W}$	Power (kW)
$\dot{E}_{in}$, $\dot{E}_D$, $\dot{E}_L$	Exergy inflow, destruction and loss (kW)
$LMTD$	Log mean temperature difference (K)
q	Heat transfer at each HX section (kW)
N	Number of HX sections
Q	Heat transfer (kW)
A	Area of heat exchange (m^2)
F	Correction factor of HX
U	Overall heat transfer coefficient (W/m^{2}K)
α	Heat transfer coefficient (W/m^{2}K)
Re	Reynolds number
Pr	Prandtl number
Nu	Nusselt number
C_p^0	Purchase cost
K_1, K_2, K_3	Purchase cost coefficients
B_1, B_2	Bare module coefficients
F_m, F_p, F_{BM}	Material, Pressure, Bare module factors
CRF	Capital Recovery Factor

C_{OM}	Cost of Operations and Management ($)
AOH	Annualized Operating Hours (h)
LEC	Levelized Energy Cost ($/kWh)
i	Interest rate
LT	Lifetime (years)
P	Pressure (MPa)
S_T, S_L	Transverse and longitudinal pitch (m)
mf	Mass fraction
Greek symbol	
η	Efficiency (Dimensionless)
Subscript	
CO_2	Carbon dioxide
cond	Condenser
WF	Working fluid
gi, g, fg	Flue gas inlet or flue gas
Jcw	Jacket coolant water
cw	Coolant water
w	Water
in	Inlet
coolant	coolant
i	Section number i
si	Shell side at section i
ti	Tube side at section i
II	2^{nd} law
I	1^{st} law
1-1e	Flue gas states
2-2c	Jacket coolant water states

3-7,7c,8	Working fluid cycle states
0	Ambient state
Abbreviation	
FGHRU	Flue gas heat recovery unit
ORC	Organic Rankine Cycle
HRU	Heat recovery unit
ODP	Ozone depletion potential
GWP	Global Warming Potential
LMTD	Log Mean Temperature Difference
CEPCI	Chemical Engineering Plant Cost Index
CWHRU	Coolant water heat recovery unit
LEC	Levelized Energy Cost
CRF	Capital Recovery Factor
AOH	Annual Operating Hours
TIP	Turbine inlet pressure
TIT	Turbine inlet temperature

3. Introduction

A huge amount, around 2/3rd of heat input to a diesel engine in the form of calorific value of fuel is usually rejected to the immediate surroundings as the waste heat. Hence, systematic waste heat recovery can enhance the energy utilization efficiency of a diesel engine. Turbocompounding, CO_2 based power cycle, organic Rankine cycle (ORC) are some commonly employed methods for diesel engine waste heat recovery [1].

Turbocompounding is preferred for a moving vehicle operating at a higher load over a substantially higher time period. The yielded efficiency of a marine diesel engine with turbocompounding is above 50% [2]. However, turbocompounding produces a back pressure due to which the exhaust stroke encounters significant adverse pressure gradient, hampering the process therein [3].

Recently, ORC has been proposed by many researchers. Yu et al. [4] proposed to use R245fa as the working fluid of an ORC cycle driven by the waste heat of a diesel engine coupled with a generator. Shu et al. [5] reported that an ORC driven by diesel engine waste heat using Cyclohexane as the working fluid can reduce brake specific fuel consumption by 10%. Yang et al. [6] proposed an ORC layout which recovers waste heat simultaneously from the exhaust gas, jacket cooling water, scavenging air cooling water and lubricating oil of a large marine diesel engine. Boodaghi et al. [7] reported that a dual loop ORC delivers 310 kW power at an engine speed of 1800 RPM when it is coupled with a heavy-duty diesel engine. R134a and R245fa had been working fluids for the low temperature and high temperature loops respectively. Lion et al. [8] revealed that the annual fuel cost of the marine diesel engine with a waste heat recovery system using ORC was around 5% lower when compared to that of the marine diesel engine without using waste heat recovery.

Recently, CO_2 has been considered as the working fluid in many waste heat recovery schemes for power cycles as it is environment friendly and non-flammable [9-11]. CO_2 is also preferred for engine waste heat recovery as it is stable at higher temperature and the design of CO_2 turbomachinery is very compact [12]. Song et al. [13] proposed a supercritical CO_2 power cycle driven by diesel engine waste heat with two pre-heaters. By incorporating the 2nd pre-heater output power was enhanced by 6.7%. Pan et al. [14] showed that recompression CO_2 power cycle, driven by diesel engine waste heat, with dual turbine would substantially reduce auxiliary fuel consumption. Mondal et al. [15] maximized the heat recovery from a marine diesel engine using a CO_2 organic cascade cycle. Sakalis [16] showed that the efficiency improvement of a standalone marine diesel engine would be 6.6% to 7.25%, due to waste heat recovery through supercritical CO_2 power cycles**.** Zhang et al. [17] revealed that the maximum thermal efficiency that could be achieved by an optimized recompression CO_2 power cycle driven by engine waste heat would be about 35.86%.

The drawback of CO_2 power cycles is very high turbine inlet pressure, in spite of studies conducted to show its efficacy. A mixture of CO_2 and organic fluid as the working fluid of power cycles had been recommended by many researchers, to reduce this limitation [18]. Liu et al. [19] showed experimentally that a power cycle with (0.6/0.4) CO_2/ R134a mixture exhibited better performance in engine waste heat recovery compared to the pure CO_2 power cycle.. Shu et al. [20] reported that the optimum operating pressure of a power cycle driven by engine waste heat could be decreased by 1.4 MPa, using (0.3/0.7) CO_2/R32 mixture as the working fluid compared to that of the pure CO_2 power cycle.

It appears that waste heat recovery from engines is an emerging subject in the field of energy study. In the present study, a regenerative trans-critical power cycle with CO_2/ Propane mixture as the working fluid is employed for a stationary diesel power plant waste heat recovery. Quality and quantity of exhaust gas waste heat differs substantially from that of the engine coolant waste heat. The fraction of waste heat from the two sources can be optimized under temperature and compositional constraints to deliver the maximum output power. Thus, we find out the turbine inlet temperature using an optimization algorithm corresponding to this operating condition. The optimization algorithm also helps to determine the operating condition to minimize the levelized cost of electricity. The use of CO_2/ propane mixture ensures operation in an environment-friendly manner and also provides a safeguard for the flame propagation by accident.

4. Selection of working fluid

Selection of a suitable working fluid is a critical factor in design of a waste heat recovery system. Good thermodynamic performance with a minimal adverse environmental impact is an important selection criterion for a suitable working fluid. Flammability and commercial availability are also factors which determine the selection of a working fluid. As already discussed, many researchers propose CO_2 as a working fluid for waste heat recovery from engine as it is commercially available and stable at a higher temperature. CO_2 is non-flammable, has zero ozone depletion potential (ODP). Global warming potential of CO_2 is 1. Temperature profile of supercritical CO2 streams has been observed to match well with that of many heat carrying streams. In spite of so many advantages, a very high operating pressure of the CO_2 power cycle is an issue in proper equipment design [21]. Thus, we have tried to reduce the operating pressure of the supercritical CO_2 power cycle in the present study using different compositions of CO_2/ propane mixture as working fluids of cycle driven by waste heat.

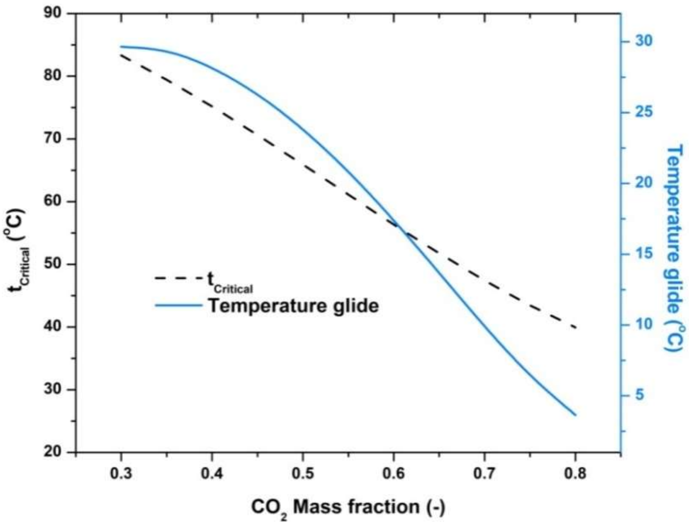

Fig. 1: Effect of different CO_2 mass fraction in (CO_2/ Propane) mixture of critical temperature and temperature glide at condenser operating condition

It is worthwhile to mention that ODP and GWP of propane are 0 and 3 respectively. Thus, the mixture working fluid ensures an environment friendly operation for all compositions. An inert working fluid can be mixed with a hydrocarbon to reduce the chances of accidental fire due to burning of the hydrocarbon [22-24]. CO_2 acts as the diluent to suppress the Propane from accidental flame propagation. To ensure safety against the high flammability of propane, the mass fraction of CO_2 should not be allowed to fall below 0.3 [24].

In case of a mixture working fluid, it is not desirable to have a very high glide temperature as it would lead to a composition shift [25]. From Fig. 1, it appears that the temperature glides of the considered CO_2/ propane mixture for all the considered mass fraction ratios remain well below 50°C.

It also appears from Fig. 1 that mixing propane with CO_2 results in an increase in the critical temperature compared to that of pure CO_2, thus eliminating a low temperature heat sink (at temperatures less than 30°C) and permitting the operation of the cycle driven by exhaust gas heat in trans-critical mode even for a 25°C ambient temperature. Trans-critical mode of operation appreciably reduces the power consumption of the compressor when compared to that of a supercritical cycle.

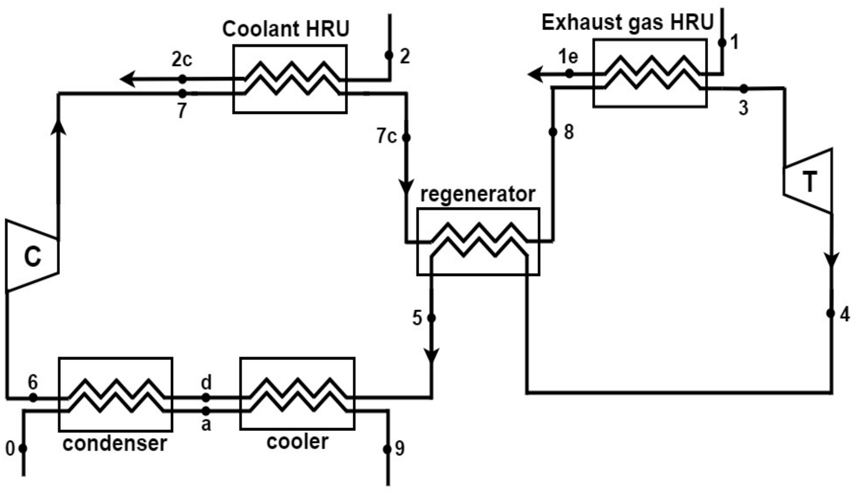

Fig. 2(a): Layout of the waste heat driven cycle

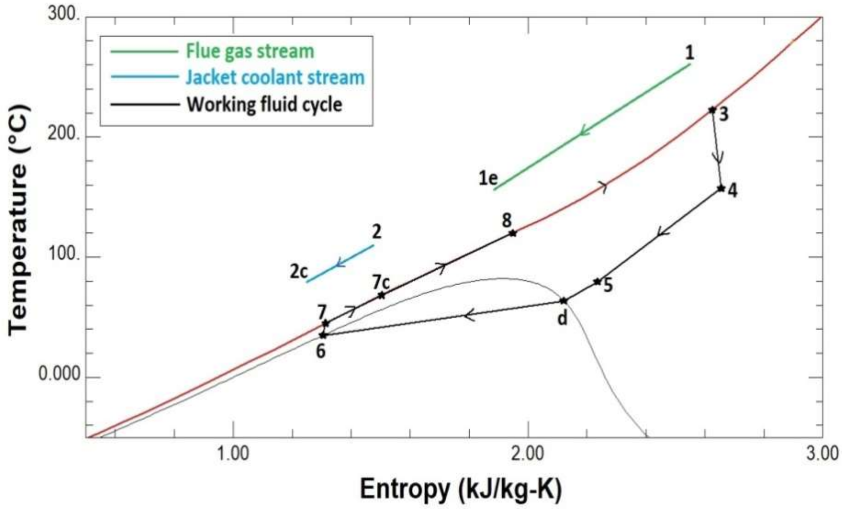

Fig. 2 (b): T-s diagram of the waste heat driven cycle

Table 1: Diesel power plant waste heat data [26]

Input parameters	Value
Engine shaft power (MW)	9
Electric power (MW)	8.73
Specific fuel consumption (g /kWh)	183
Exhaust gas temperature (^{o}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

5. System description

Fig. 2 (a) shows the layout of the power cycle recovering waste heat from a diesel power plant. Fig. 2 (b) is the temperature entropy diagram associated with the layout. From Table 1, we observe that exhaust gas and engine coolant are the main carriers of the exhaust waste heat of the diesel power plant under consideration. The power cycle presented, hence, consists of two heat recovery units (HRUs) recovering waste heat of both the exhaust gas and engine coolant. The working fluid is heated from the compressor exit state (i.e., 7) to the regenerator inlet state (i.e., 7c) in the engine coolant HRU. After undergoing a heating process in the regenerator (process 7c-8), the hot working fluid 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 portion of turbine power is expensed to run the compressor.

6. Modeling and methodology

The intention of this study is to systematically implement waste heat recovery in a diesel power plant using a CO_2/ propane mixture based transcritical power cycle. Thus, to analyze the performance of the implemented waste heat recovery scheme, a mathematical is developed based on the following simplified assumptions:

i. Steady flow operating conditions are applicable

ii. Isentropic efficiencies of the compressor and the turbine are both assumed to be 85%.

iii. The ambient condition is specified to be 100kPa and 25^{o}C.

iv. The chemical exergy has been ignored as the composition is constant throughout the cycle.

v. All the heat exchangers are assumed to be of shell and tube type.

vi. In both the HRU, the working fluid is assumed to flows through tubes

vii. Working fluid mass flux is assumed to be 350kg/m^{2}s

viii. Flue gas velocity over the tube bank is not allowed to go above 10m/s.

ix. Maximum velocity of engine coolant in the HRU is assumed to be 0.75m/s

x. In the regenerator, high pressure working fluid is taken on the tube side, while the low pressure working fluid is taken on the shell side.

xi. The thermo-physical properties of flue gas are assumed to be the same as that of air.

xii. The cooling water is available at 25^{o}C. Thus, the minimum cycle temperature is assumed to be 35^{o}C.

xiii. Flue gas contains SO_2. Thus, the acid dew point temperature of the flue gas is assumed to be 120°C (so the flue gas is cooled to just 130 °C for additional safety).

xiv. Due to a lower turbine power output (< 1.5 MW turbine power), radial flow turbines are used.

xv. The whole heat exchanger system is designed taking a pinch point temperature difference of 10°C.

6.1. Thermodynamic performance estimation

Cycle power output and 2nd law efficiency are the parameters to depict 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:

$$\dot{m}_{WF} = \frac{\dot{m}_{gi} C_{pg} (T_1 - T_{1e})}{h_3 - h_8} \quad (1)$$

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

$$\dot{m}_{Jcw} = \frac{\dot{m}(h_{7c} - h_7)}{C_{pJCw}(T_2 - T_{2c})} \quad (2)$$

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

$$\dot{W}_t = \dot{m}_{WF}(h_3 - h_4) \quad (3)$$

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

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \tag{5}$$

$$\eta_I = \frac{\dot{W}_{net}}{\dot{m}_{cw}C_{pw}(T_2 - T_{2c}) + \dot{m}_{gi}C_{pg}(T_1 - T_{1e})} \tag{6}$$

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

Table 2: component exergy destruction equations

Component	Exergy destruction (kW)
Turbine	$\dot{m}_{WF}T_o(s_4 - s_3)$
Compressor	$\dot{m}_{WF}T_o(s_7 - s_6)$
Condenser	$\dot{m}_{WF}T_o(s_6 - s_d) + \dot{m}_{cond}T_o(s_a - s_o)$
Cooler	$\dot{m}_{WF}T_o(s_d - s_5) + \dot{m}_{cond}T_o(s_9 - s_a)$
Regenerator	$\dot{m}_{WF}T_o\{(s_8 - s_{7c}) + (s_5 - s_4)\}$
Flue gas HRU	$\dot{m}_{gi}C_{pg}T_o ln\left(\frac{T_{1e}}{T_1}\right) + \dot{m}_{WF}T_o(s_3 - s_8)$
Coolant HRU	$\dot{m}_{Jcw}C_{pJcw}T_o ln\left(\frac{T_{2c}}{T_2}\right) + \dot{m}_{WF}T_o(s_{7c} - s_7)$

Now, 2nd law efficiency is expressed as

$$\eta_{II} = \frac{\dot{W}_{net}}{\dot{E}_{infg} + \dot{E}_{inJcw}} = \frac{\dot{E}_{infg} + \dot{E}_{incw} - \Sigma\dot{E}_D - \dot{E}_{L\,fg} - \dot{E}_{L\,Jcw} - \dot{E}_{L\,coolant}}{\dot{E}_{infg} + \dot{E}_{inJcw}} \tag{7}$$

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

$$\dot{E}_{infg} = \dot{m}_{gi}C_{pg}(T_1 - T_o) - \dot{m}_{gi}C_{pg}T_o ln\left(\frac{T_1}{T_o}\right) \tag{8}$$

$$\dot{E}_{in\ Jcw} = \dot{m}_{Jcwav}\, C_{pw}(T_2 - T_o) - \dot{m}_{Jcwav} C_{pJCw} T_o ln\left(\frac{T_2}{T_o}\right) \quad (9)$$

Exergy losses with different outgoing streams are presented as follows:

$$\dot{E}_{Lfg} = \dot{m}_{gi} C_{pg}(T_{1e} - T_o) - \dot{m}_{gi} C_{pg} T_o ln\left(\frac{T_{1e}}{T_o}\right) \quad (10)$$

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

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

6.2. Economic assessment

The intention of the economic assessment is to estimate the levelized electricity 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 area and associated operating pressure. As shown in Fig. 3 the heat exchanger has been discretised into N subsections assuming an equal enthalpy drop is occurring across the each subsection. Finally, the temperatures of the tube side fluid are evaluated accordingly at each of the subsections.

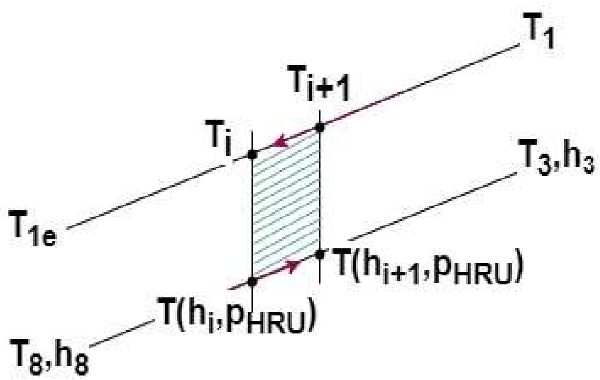

Fig. 3: Discritzation of the flue gas HRU

The LMTD at each section has been calculated as

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

The heat recovery at each and every subsection is estimated as

$$q = \frac{Q_{FGHRU}}{N} \tag{14}$$

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

$$A_i = \frac{q}{FU_i LMTD_i} \tag{15}$$

In above equation *F* is the correction factor and $\boldsymbol{U_i}$ is the overall heat transfer coefficient. The overall heat transfer coefficient is expressed as

$$\frac{1}{U_i} = \frac{1}{\alpha_{si}} + \frac{1}{\alpha_{ti}} \tag{16}$$

Shell side heat transfer coefficient is estimated by using following correlation of fluid flowing over the tube bank [27]:

$$Nu_{si} = 0.35\left(\frac{S_T}{S_L}\right)^{0.2} {Re_{si}}^{0.6} {Pr_{si}}^{0.36} \tag{17}$$

For supercritical CO_2 flowing inside the tube, Protopopov correlation (without wall correction factor) is applied to estimate the associated Nusselt number [28]. 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 following equation [29]

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10} Z + K_3[\log_{10} Z]^2 \tag{18}$$

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

$$C_{bm} = C_p^0\left(B_1 + B_2 F_m F_p\right) = C_p^0 F_{BM} \tag{19}$$

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:

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

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).

$$C_{Tot} = \left(\sum C_{BM,eq}\right) \times \left(\frac{CEPCI_{current\ year}}{CEPCI_{2001}}\right) \tag{21}$$

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

Table 3: Coefficient for total cost estimation [29]

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	____	____	____

Finally, the levelized electricity cost is estimated as follows

$$LEC = \frac{CRF \times C_{Tot} + C_{OM}}{W_{net} \times AOH} \tag{22}$$

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

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

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

7. <u>Results and discussion</u>

In the present study, CO_2/Propane mixtures with different CO_2 mass ratios are employed as the working fluid of a trans-critical power cycle recovering the diesel power plant waste heat. Initially in this section, effects of different input parameters on cycle performance are discussed for a specified mixture composition. Effects of varying mixture composition on cycle performance are also discussed in the subsequent stage 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.

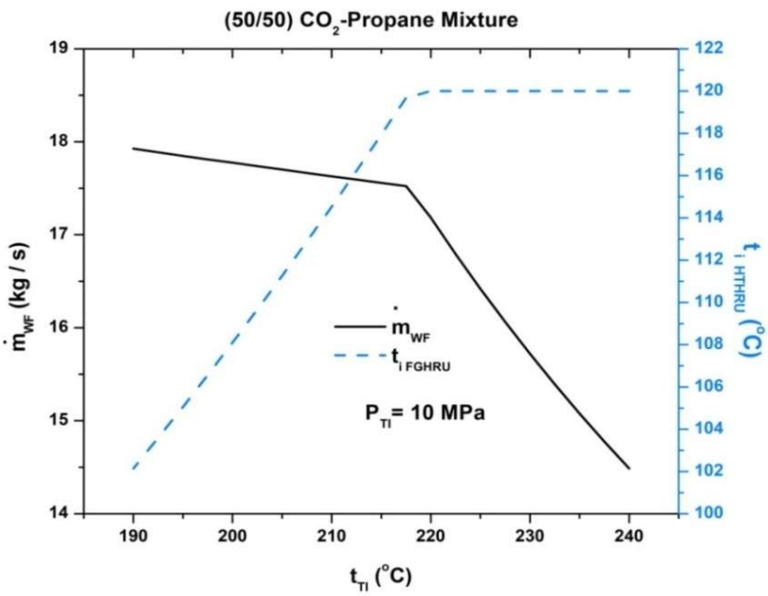

Fig. 4: Effects of turbine inlet temperature on working fluid mass flow rate and working fluid inlet temperature to FGHRU

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. 4. Working fluid mass flow rate initially slowly decreases with an increasing turbine inlet temperature. However, the mass flow rate of working fluid starts to decrease much rapidly as turbine inlet temperature is raised above a certain value. With an increasing turbine inlet temperature, specific enthalpy change of the working fluid in the FGHRU increases. However, with initial increments of the turbine inlet temperature, this increment is not very rapid as working fluid inlet temperature to the FGHRU increases due to the increasing heat duty of the regenerator. However, as a reasonable pinch point temperature difference (i.e., 10^{o}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^{o}C, as shown in Fig. 4. As temperature of the high pressure working fluid stream reaches to 120^{o}C, mass flow rate of the working fluid through the FGHRU starts to decrease rapidly with a further increment in turbine inlet temperature (mainly, due to the rapid increment in specific enthalpy change of the working fluid).

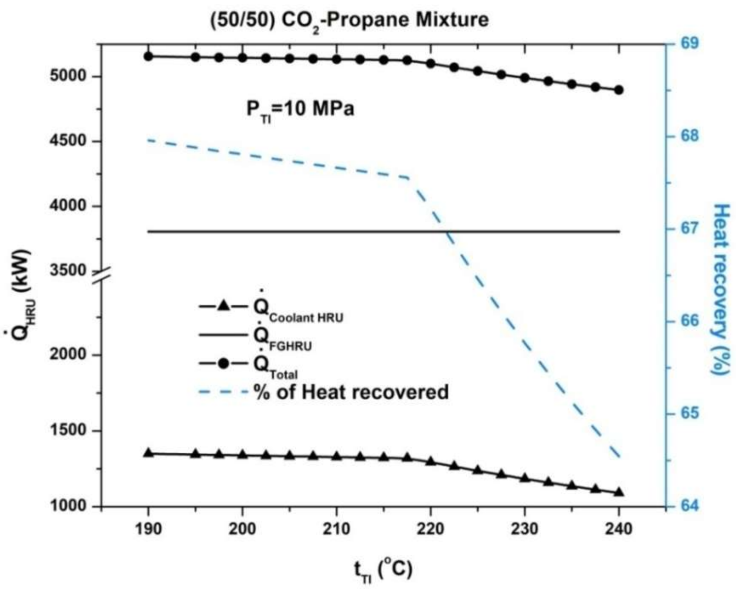

Fig. 5: Effects of varying turbine inlet temperature of waste heat recovery

Effects of varying turbine inlet temperature on heat recoveries in different HRUs are presented in Fig. 5. Always the entire recoverable heat of the exhaust gas is recovered due its higher exergy content. Thus, the heat duty of the FGHRU is independent of turbine inlet temperature. Heat recovery in the coolant HRU as well as % of heat recovery rapidly decreases above a certain turbine inlet pressure due to the rapid reduction of the working fluid mass flow rate.

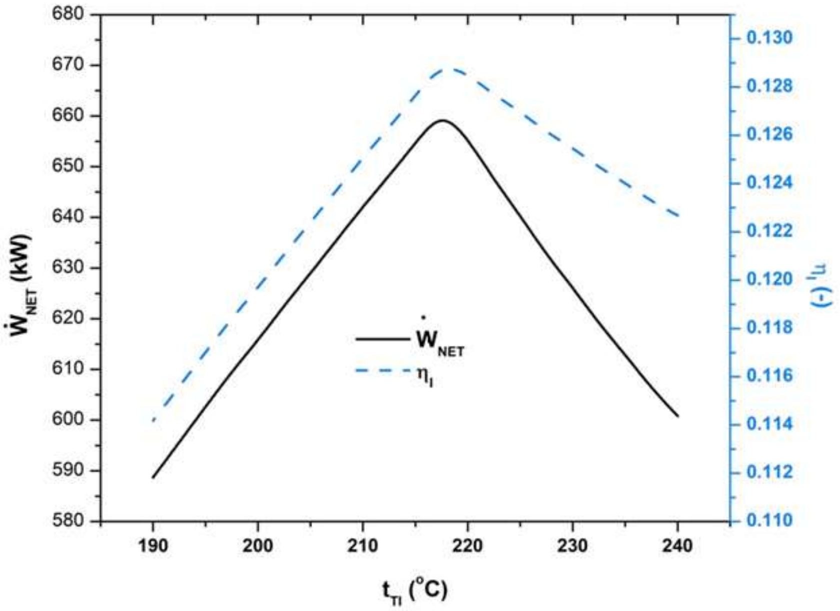

Fig. 6: Effects of varying turbine inlet temperature in net work output and 1st law efficiency

It is observed in Fig. 6 that for any specified turbine inlet pressure; there exists an optimum turbine inlet temperature corresponding to the maximum cycle power output (or maximum 1st law efficiency). As 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.

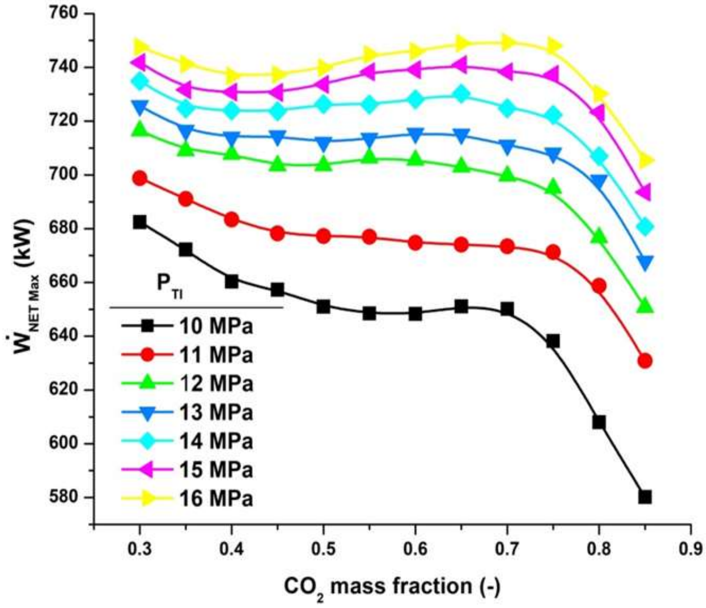

Fig. 7: Effects of varying CO_2 mass fraction on maximum cycle power output at different turbine inlet pressure

Effects of varying CO_2 mass fraction in CO_2/ propane mixture on maximum achievable power outputs corresponding to different turbine inlet pressure are presented in Fig. 7. It is observed that for any specified working fluid composition, cycle power output increases with an increasing turbine inlet pressure. However, above a certain turbine inlet pressure this increment is not very significant. It appears from Fig. 7 that, for a lower turbine inlet pressure (13 MPa or less); reducing the CO_2 mass fraction of the mixture working fluid significantly enhances cycle power output. For a turbine inlet pressure above 13 MPa, net cycle power output remains more or less constant for CO_2 mass fraction varying between 0.3 and 0.75. For all considered turbine inlet pressures, raising the CO_2 mass fraction above 0.75 results rapid decrement in cycle power output.

Effects of varying the CO_2 mass fraction in the working fluid and the turbine inlet pressure on levelized electricity cost (LEC) is presented in Fig. 8. It appears from Fig. 8 that LEC is a minimum if the cycle is operated with the 0.3 mass fraction of CO_2 in the working fluid. It is also observed that for a higher turbine inlet pressure, LEC is a weak function of the working fluid composition. It is observed in Fig. 8 that for all specified working fluid compositions, varying 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 presented waste heat recovery scheme.

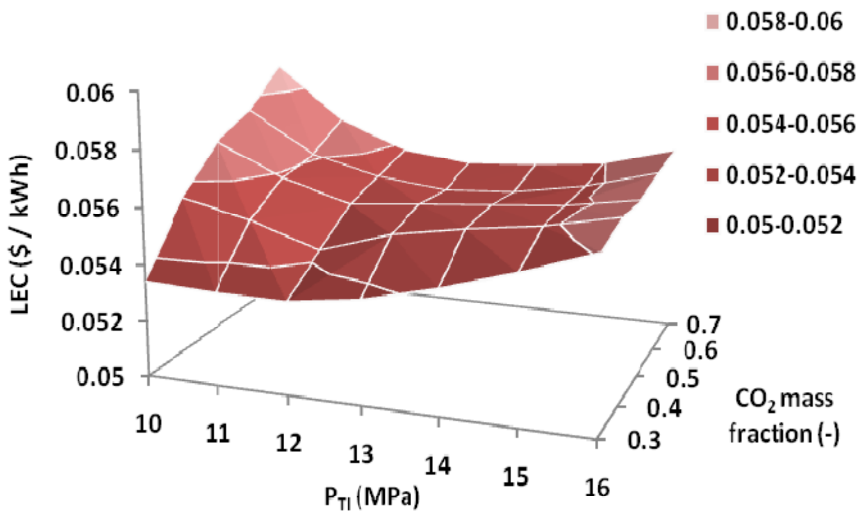

Fig. 8: Effects of varying CO_2 mass fraction and turbine inlet pressure on levelized electricity cost

As the intention of the present study is to explore the optimum operating condition for the CO_2/ propane mixture based transcritical power cycle recovering the waste heat of a diesel power plant, an optimization algorithm is proposed as presented in Fig. 9. While recovering the maximum usable high grade waste heat, the presented optimization algorithm of Fig. 4 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 safety issue) are applied constraints of the optimization algorithm.

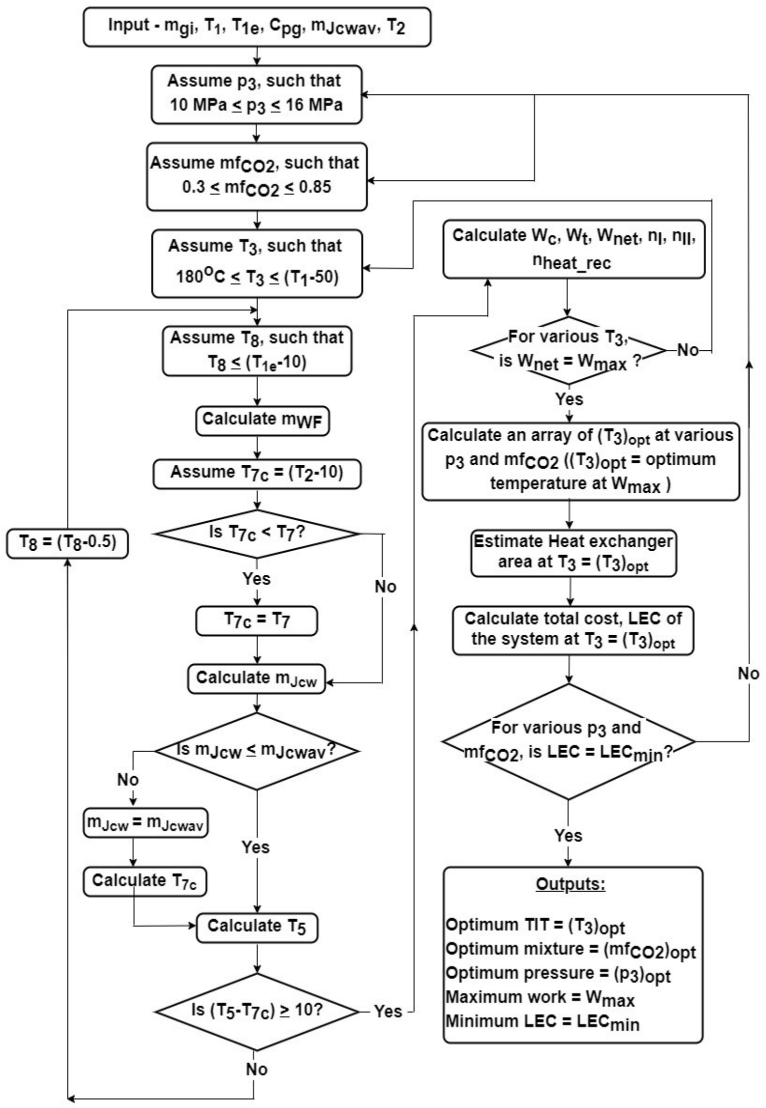

Fig. 9: Optimization algorithm

Minimum levelized electricity cost (LEC) and associated other parameters obtained by applying the optimization algorithm are summarized in Table 4.

Table 4: Parameters corresponding to minimum LEC

Type of cycle	t_{TI} (°C)	P_{TI} (MPa)	P_{Te} (MPa)	CO_2 mass fraction	W_{NET} (kW)	η_{II} (%)	LEC_{Min} ($/kWh)
Trans-critical cycle with CO_2-propane mixture	222.5	12.00	3.5606	0.3	712.90	36.32	0.053

A Grassmann diagram corresponding to the operating condition of Table 4 is presented in Fig. 10 to depict exergy destructions of different components and exergy losses associated with different fluid streams exiting the waste heat recovery system. It is observed that the largest exergy destruction occurs in the FGHRU. The maximum exergy loss occurs with the flue gas stream exiting the HRU. However, this exergy loss with the flue gas stream is unavoidable due to the constraint of acid condensation.

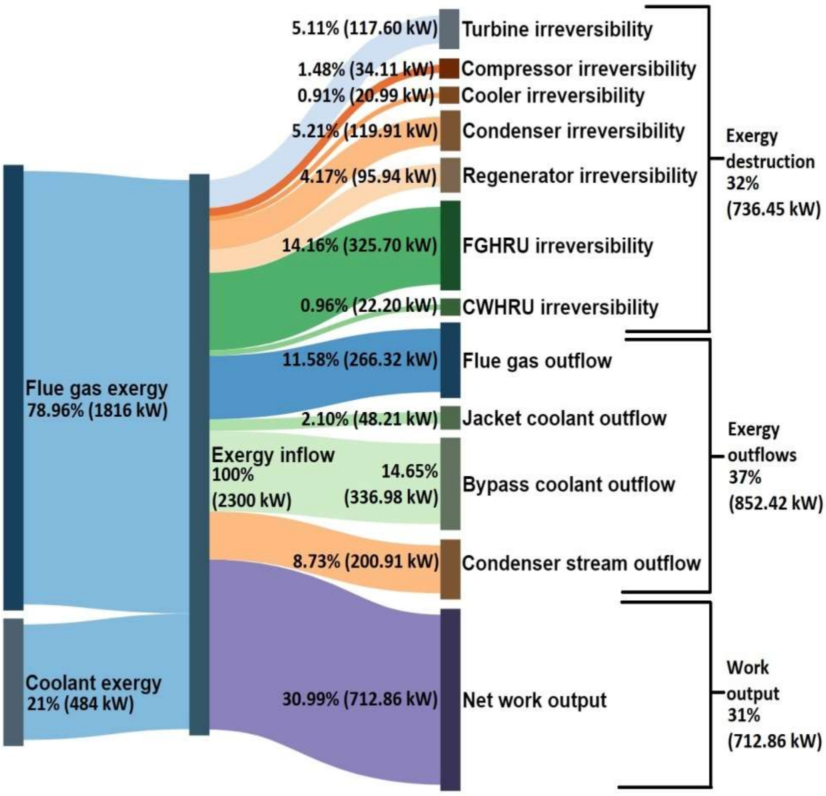

Fig. 10: Grassmann diagram

For a better understanding state point properties corresponding to the optimized operating condition are summarized in Table 5.

Table 1: State points of the optimized cycle

Working fluid				
State points	*t (°C)*	*P (MPa)*	*h (kJ/kg)*	*s (kJ/kgK)*
3	222.50	12.000	834.19	2.6248
4	157.38	3.5606	759.611	2.6556
5	79.54	3.5606	595.991	2.2358
d	63.72	3.5606	556.52	2.1212
6	35.00	3.5606	291.73	1.3046
7	44.95	12.000	310.64	1.3135
7c	68.40	12.000	373.35	1.5037
8	120.00	12.000	536.97	1.9486
Flue gas stream				
State points	*t (°C)*	*P (MPa)*	*h (kJ/kg)*	*s (kJ/kgK)*
1	345.00	0.1	_______	_______
1e	130.00	0.1	_______	_______
Jacket coolant water stream				
State points	*t (°C)*	*P (MPa)*	*h (kJ/kg)*	*s (kJ/kgK)*
2	78.40	0.1	_______	_______
2c	54.95	0.1	_______	_______
Condenser and cooler water streams				
State points	*t (°C)*	*P (MPa)*	*h (kJ/kg)*	*s (kJ/kgK)*
0	25.00	0.1	_______	_______
a	53.71	0.1	_______	_______
9	57.99	0.1	_______	_______

In Fig. 11, minimum levelized electricity costs for different turbine inlet pressures of the mixture based cycle are compared with those of the supercritical CO_2 power cycle.

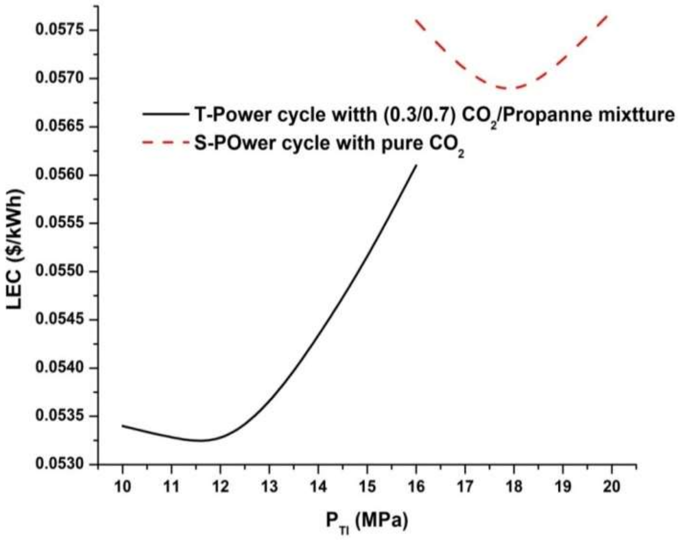

Fig.11: Comparison of minimum achievable LEC at different TIP of the transcritical power cycle with mixture working fluid with the LEC at different TIP of the supercritical CO_2 power cycle.

It appears from Fig. 11 that the minimum achievable LEC of the transcritical cycle with the mixture working fluid is 6.36% lower compared to that of the supercritical power cycle using pure CO_2 as the working fluid. It is also apparent that the turbine inlet pressure corresponding to the minimum LEC of the waste heat driven cycle using CO_2/ Propane mixture as the working fluid is close to 33% lower compared to that of the waste heat driven supercritical power cycle using pure CO_2 as the working fluid.

8. Conclusions

Intention of the present study is to employ a transcritical regenerative power cycle to recover waste heat of a diesel power plant. CO_2/propane mixture with different CO_2 mass ratio is used as the working fluid of the power cycle to ensure safe and environment friendly operation simultaneously. An optimization algorithm is also employed to ensure the best operating performance of the presented power cycle. The principal findings of the study are summarized as follows:

- It is observed that, at a lower turbine inlet pressure of working fluid, cycle power output significantly increases with reducing CO_2 mass fraction in the mixture working fluid. Corresponding reduction in levelized electricity cost (LEC) is also found to be significant. For a higher turbine inlet pressure, effects of varying mixture composition are having less significant effects on cycle power output and LECs.
- For any specified mixture composition net cycle power output increases with an increasing turbine inlet pressure. However, this increment is not very appreciable above a certain turbine inlet pressure (i.e., 14 MPa).
- For any specified mixture composition, LEC becomes minimum corresponding to a specific turbine inlet pressure. The minimum LEC of the cycle is found to be at 12 MPa turbine inlet pressure, for 0.3 mass fraction of CO_2.
- The minimum achievable LEC for the presented transcritical cycle operating with mixture working fluid is close to 6.36% lower compared to that of an optimized supercritical CO_2 power cycle recovering waste heat from a similar source.

Corresponding turbine inlet pressure for the transcritical mixture based cycle is close to 33% lower compared to that of the supercritical CO_2 power cycle.

Finally, diesel power plant and the presented waste heat recovery scheme together can yield 8% higher power output compared to the yielded power output of the diesel power plant without waste heat recovery.

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