Performance Evaluation of Two Medium-Grade Power Generation Systems with CO_2 Based Transcritical Rankine Cycle (CTRC)

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Abstract

As CO_2 is emerging as an environment friendly working fluid its application in high temperature engine's waste heat recovery systems is found to be more suitable than other hydrocarbons. This paper presents a performance comparison of two systems based on transcritical Rankine cycle using CO_2 as working fluid. A heavy-duty truck is opted for analysis in which coolant is used to preheat CO_2 and further the engine exhaust is used to transfer the heat to main heater. The System-1 and System-2 having single and dual loop based transcritical Rankine cycle are analysed. The independent parameters taken for the investigative analysis are turbine inlet temperature (TIT), pressure ratio and effectiveness of heat exchangers. Comparison results show that System-2 is producing 11.8 kW more power than system-1 at 12 MPa pressure ratio and at 489°C TIT. However, under same conditions, system-1 is having 16.88% of thermal efficiency which is higher than system-2 by around 3%. Further, the Engine coolant utilization rates when compared are nearly same in both the systems, the exhaust gas utilization rate came out higher for System-2. In respect of exergy destruction, system-1 shows maximum destruction in regenerator and System-2 in heater-2.

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Keywords: CO_2 – Transcritical rankine cycle, multiple waste heat, utilization rate, exergy destruction.

1 Introduction

The extensive usage of fossil fuels, such a coal, petroleum products etc., worldwide in fulfilling the increasing power needs has been major contributor towards global warming, air pollution and indirect risk to human health. One of the biggest challenges in this 21st century is to reduce the risks arising due to excessive CO2 emissions from fossil fuels. This can be done by recovering waste heat or renewable energy. According to temperature ranges of heat source used, the waste heat sources are mainly divided into three categories: low temperature ($<$230°C), medium temperature (230–650°C) and high temperature ($>$650°C) [1]. Renewable energy sources such as solar radiation can be utilised as low grade heat source to produce power. Jiangfeng et al. [2] proposed a combined system in which solar radiations are used as heat source and an ejector expansion device is used to combine transcritical CO_2 refrigeration cycle with a Brayton Cycle. This system provides combined cooling and heating functions for different applications. Results showed that increase in inlet temperature of ejector and inlet pressure of turbine reduces overall efficiency of system. Others i.e. solar energy, fuel cells [3] and geothermal [4] are also having enough potential to be used as heating source for thermal cycles. Using traditional cooling water in conventional CO_2 based transcritical Rankine cycle (CTRC) system makes the condensation of CO_2 difficult. Pan Lisheng et al. [5] tried to solve this problem. They used solar energy as heat source and introduced a novel CTRC system which consisted of gas-liquid separator and expander device. This only sends liquid CO_2 to pump instead of mixture of gas and liquid states. But initial cost is increased due to more components added to system. However, these type of solar energy based systems are successful only in those regions where solar radiations are available most of the time during a year. Also during night time we can't use such systems as it solely depends on solar radiations.

Globally, wide variety of medium to low-grade waste heat sources from automobiles and industries are present which can be used to produce electricity by using advanced thermodynamic power cycles and suitable working fluids. But here using conventional thermal working fluids such as hydrocarbons, ammonia, Chlorofluorocarbons (CFCs) etc. can damage our environment, hence an environmental friendly fluid which is suitable for temperature range of cycle and also keep the system compact is needed.

The CO2 based transcritical thermodynamic cycles (T-CO2) is found more suitable for generating power from heat source [6]. One of the known method for utilising waste heat of a system is by use of an Organic Rankine Cycles (ORC) based system. The ORC systems are good in term of system size and thermal efficiency [7, 8]. But ORC systems have one of the limitation of constant evaporation temperature, it further leads to increase of irreversibility during a heat addition process when using a heat source which have sensible heat in it such as waste heat [9]. Antti Uusitalo et al. [10] proposed the super critical Brayton cycle as a good option for large scale heat recoveries as well as reducing emissions from power plants.

Nomenclature		Acronyms	
cp	Specific heat capacity, kJ/kg.K	CFCs	Chlorofluorocarbons
h	Specific enthalpy kJ/kg	CTPC	CO_2 based transcritical power cycle
I	Exergy destruction rate, kJ/kg	CTRC	CO_2 based transcritical Rankine cycle
m	Mass flow rate, kg/min	E-WHR	Engine waste heat recovery
P	Pressure, MPa	GWP	Global warming potential
Q	Heat transfer kJ/min	NPO	Net power output
s	Specific entropy, kJ/kg.K	ODP	Ozone depletion potential
t	Temperature,$^\circ$C	P-CTRC	CO_2 based transcritical thermodynamic cycles with preheater
U_c	Engine coolant utilization rate	PR-CTRC	CO_2 based transcritical thermodynamic cycles with preheater and regenerator
U_g	Exhaust gas utilization rate	T-CO_2	CO_2 based transcritical thermodynamic cycles
V	Volume, m^3	TE	Thermal efficiency
W	Work done, kJ/min	TRC	Transcritical Rankine cycle
ξ	Effectiveness of heat exchanger	WHR	Waste heat recovery
η	Efficiency		
$\triangle\ h_{is}$	Change in isentropic enthalpy		

In transcritical Rankine cycle (TRC) systems, CO_2 based cycle yields more power output than hydrocarbons, such as Ethane, Propane and Propene, based cycles. Also a system based on CO_2 not only can absorb more heat in regenerator and engine coolant heat exchanger as compared to hydrocarbons with comparable total heat transfer area but also have turbine size advantage over hydrocarbons [11]. Furthermore CO_2 is non-toxic and abundantly available in atmosphere. Hence using CTRC is more suitable in place of other hydrocarbon based cycle in such WHR system.

Many experimental and theoretical researches on engine waste heat recovery (E-WHR) by using CTRC are done in recent years. Theoretical research most of the time focus on selection of different type of systems and comparison between them. Jian et al. [12] tried to find out the potential of using preheater in supercritical CO_2 (S-CO_2) cycle based systems. Further they have compared two systems, first system consisting of single regenerator and the second improved version of same system consisting of additional regenerator. Their results indicated that the improved system utilised the regenerator waste heat load more efficiently and improved system performance. The improved system got 7.4% more net power output over first system. On utilising improved system overall engine power output is increased by 6.9%. Gequn et al. [13] found out the effect of variable engine conditions on system performance of a CO2 transcritical power cycle (CTPC) system. The results show that partial load condition of engine is more beneficial for variety of variations and continuous operation. By implementing mass flow rate guided strategy the optimal engine performance can be obtained. Mostly the focus of researchers in a CTRC system is on performance parameters such as thermal efficiency, net power output and utilization rate is hardly touched as an important subject but it is really significant. Higher utilization rate of engine coolant and exhaust gas results in higher energy input for a CTRC, hence will result in more power output. Also high utilization rate of an engine coolant results in lower load on cooling fan/ radiator system of vehicle because the return temperature of coolant already reduced in pre heater. For a system which has only engine exhaust gas recovery, utilization rate depends on temperature of exhaust gas leaving the heat exchanger. However system which have both exhaust gas and coolant heat exchanger have complex mechanism of utilization rate [14]. Tao et al. [15] explained six standard methods that are traditionally used by the researchers to find out the regenerator input output conditions. Results proposed Pinch point temperature difference method is best suited for transcritical cycles.

Apart from thermal analysis, exergy analysis is equally important and it shows the energy flow and energy in a system. Aklilu et al. [16] performed exergy analysis of CTRC based system having regenerator. He explained the method of doing exergy analysis on each part of a CTRC system. The results showed the highest exergy losses are experienced in main heater and regenerator and the lowest exergy losses are observed in pump and turbine. Various authors discussed the parametric investigations upon the combined cycles and waste heat recovery steam boilers are also mentioned in references [21–23].

In this paper performance of two systems based on CTRC cycles are described and compared for multiple WHR from a diesel engine. In both the systems a preheater and a main gas heater is used to absorb heat from engine coolant and engine exhaust gas respectively. The mathematical simulation models are developed for both systems. Pinch point temperature difference (PPTD) is followed at each heat exchanger while doing performance analysis. Effect of variation of turbine inlet temperature (TIT), pressure ratio (PR), effectiveness of heat exchangers and mass flow rate on system parameters are studied. The performance parameters compared in this research are: a. net power output on the basis of first law of thermodynamics; b. Thermal efficiency; c. Utilization rate. Also the exergy destruction at each component is calculated and a graph plot is made.

2 System Description

In comparison to hydrocarbon based working fluids CO_2 is environmental friendly, cheap, non-flammable, non-toxic and safer to use. Table 1 shows Ozone depletion potential (ODP) and Global warming potential (GWP) of CO_2 and some hydrocarbon fluids [17]. It shows GWP of hydrocarbons are much higher than CO_2. Using hydrocarbons as working fluid is more risky to our environment as compared to CO_2. Dai et al. [18] showed in research that most of hydrocarbons have decomposition temperature around (260–320°C) and they are not suitable for high temperature WHR. Hence in this research CO_2 has been chosen as a working fluid.

Table 1 ODP and GWP values of some important working fluids

Working Fluid	ODP	GWP
Ethane	0	5.5
Propane	0	3.3
Isobutane	0	3
Propene	0	1.8
CO_2	0	1

Table 2 Some important parameters of heavy-duty diesel engine used

Parameters	Values
Exhaust gas temperature (°C)	519
Exhaust gas mass flow rate (kg/min)	2.401
Engine coolant outlet temperature (°C)	88.4
Engine coolant return temperature (°C)	77.9
Engine coolant mass flow rate (kg/min)	211.3

2.1 The Chosen Diesel Engine

Automobiles such as heavy duty trucks generally have engines of size 8L to 16L. These engines loose large portion of their combustion energy in the form of waste heat into the atmosphere. One can utilise this waste heat from its coolant and exhaust gas in order to increase overall efficiency of engine without burning more fuel. From a heavy duty truck a 9.5 L turbocharged heavy duty diesel engine with intercooler is chosen as a waste heat source. 276.5 kW and 1900 N-m are rated power and maximum engine torque of engine respectively. All detailed parameters of engine are taken from reference [19]. Engine exhaust gas and engine coolant are used as heating sources. Here engine coolant is low temperate heat source whereas engine exhaust gas is medium temperate heat source. These heat sources consist of major portion of engine's combustion energy. Table 2 shows basic important parameters of heavy-duty diesel engine.

2.2 PR-CTRC Based System for Recovery of Waste Heat

Figure 1(a) shows schematic diagram of a PR-CTRC system (system-1). The main process of PR-CTRC include: Pumping process (1-2), heating process (2-3-4-5), expansion process (5-6), condensing process (6-7-1). The working fluid i.e. CO_2 is first passed through pump in order to raise its pressure. As the temperature is still below critical point so CO_2 is still in liquid phase. Then CO_2 is passed through pre-heater in order to absorb waste heat energy of engine coolant. Engine coolant return temperature cannot be lower than 77.9°C (a boundary condition). Then it is passed through regenerator in order to absorb heat from fluid that is returning from turbine as it is still having enough potential to heat the fluid. The fluid is then passed through gas heater where it absorbs heat from engine exhaust gas. The exhaust gas minimum temperature is kept as 120°C which is acidic dew point temperature of exhaust gas [20]. This helps in protecting the heat exchanger from acidic damage or rusting. The fluid is then injected to turbine at defined temperature and pressure. After producing power at turbine the fluid is sent to regenerator and finally to condenser. Cooling water at condenser is set at 15°C. Here fluid is cooled to 20°C before sending back to pump.

2.3 Dual Loop P-CTRC Based System for Recovery of Waste Heat

Figure 1(b) shows layout of a dual loop P-CTRC system (System-2). The main process of P-CTRC include:

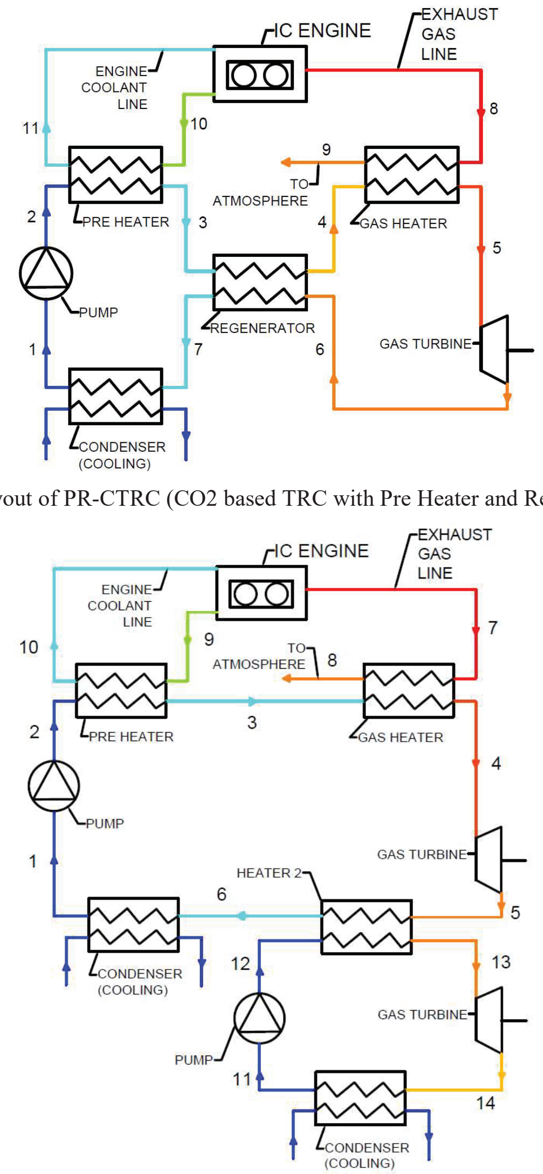
For loop 1: Pumping process (1-2), heating process (2-3-4), expansion process (4-5), condensing process (5-6-1). For loop 2: Pumping process (11-12), heating process (12-13), expansion process (13-14), and condensing process (14-11). The first loop processes are similar to PR-CTRC system but in second in the place of regenerator a condenser/secondary heater has been used. This condenser acts as a heater for secondary loop and as a cooling device for primary loop. By same heat input in this arrangement power is generated from two turbines. However as this system doesn't have regenerator so the utilization rate will vary as compared to previous system.

3 System Modelling

In order to perform thermodynamic and exergy analysis of systems, mathematical equations are developed for each component. Temperature entropy diagrams for systems are also produced.

3.1 Thermodynamic Modelling

T-s diagram of systems are shown in Figures 2 and 3. Calculations are done with the help of REFPROP 10.0 and Engineering Equation Solver (EES) software. The pinch point temperature differential (PPTD) method is used to measure the mass flow rate of the working fluid [19]. For CO_2 the maximum cycle temperature, i.e. inlet temperature of turbine, is defined by exhaust gas temperatures minus PPTD that is needed to maintain at main gas heater. By REFPROP we can find out that critical temperature of CO_2 is nearly 30.5°C. For pump to operate at maximum efficiency the temperature of working fluid is required to be kept below critical values during this operation. In order to maintain the same the lowest temperature of cycle as 20°C has been chosen.



(a). Schematic layout of PR-CTRC (CO2 based TRC with Pre Heater and Regenerator) system

(b). Schematic layout of dual loop P-CTRC (CO2 based TRC with Pre Heater) system

Figure 1 Schematic diagram of CTRC systems for engine waste heat recovery (E-WHR).

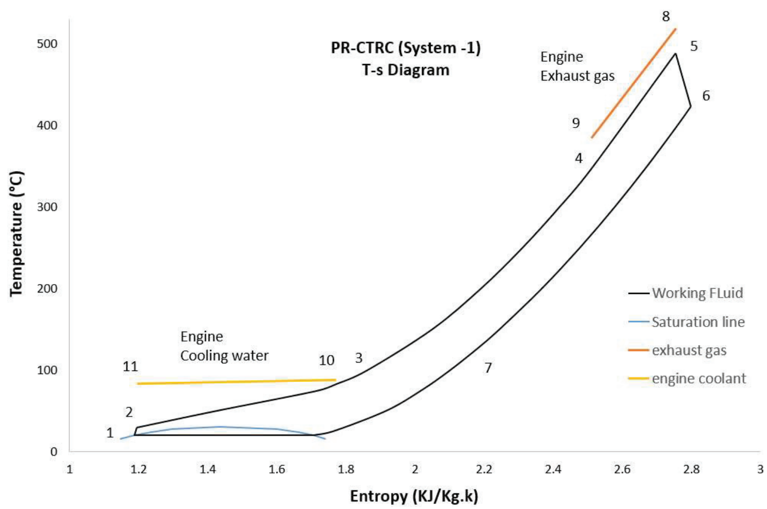


Figure 2 T-s diagram of PR-CTRC system (system-1).

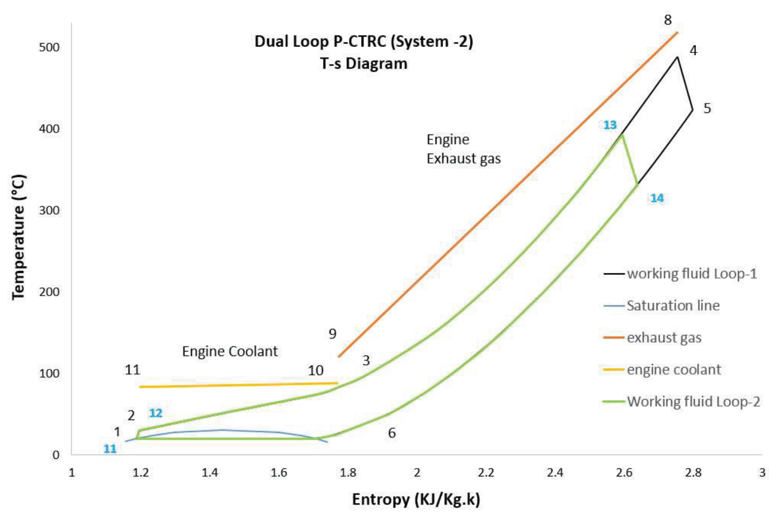


Figure 3 T-s diagram of Dual loop P-CTRC system (system-2).

Assumptions considered for the analysis are given below:

- Temperature of exhaust gas during heat recovery process should be kept above acidic dew point temperature i.e. 120°C.
- Each process is in stable state.

- Negligible pressure loss in Heat exchangers.
- Always achieves specified state at pump inlet.
- Heat exchangers efficiency are fixed.
- No pressure or heat loss in pipes.
- Pinch point temperature difference is always maintained in each heat exchanger.
- Engine coolant temperature can't fall below coolant return temperature.
- Pump and turbine have fixed efficiency.
- Environment temperature is fixed at 20°C for exergy calculations.
- Environment temperature and pressure are constant.

The mathematical equations obtained by mass, energy and exergy balance for the two systems are given below.

Process 1-2 is the pumping process and it is same for both systems. Work done by pump is given by:

$$W_{pump} = \frac{m \cdot (h_{2s} - h_1)}{\eta_{pump}} = m \cdot (h_2 - h_1) \tag{1}$$

Where m is the mass flow rate of working fluid, h_1 is enthalpy of fluid entering the pump, and h_2 is enthalpy of fluid at exit of turbine. h_{2s} is the enthalpy at the exit of pump if we consider pressurizing process by pump is isentropic. η_{pump} is the efficiency of pump.

Process 2-3 is preheating of working fluid and it is same for both the systems. Heat gained by working fluid is:

$$Q_{preh} = m \cdot (h_3 - h_2) = m_1 \cdot cp_c \cdot (t_{c\ in} - t_{c\ out}) \cdot \xi \tag{2}$$

Where, h_2 and h_3 are the enthalpy of working fluid before entering and after exit the pre heater respectively. m_1 is mass flow rate of engine coolant, cp_c is specific heat capacity of coolant, $t_{c\ in}$ is inlet temperature of coolant, $t_{c\ out}$ is temperature of coolant at exit of pre heater, ξ is heat exchanger efficiency.

Process 3-4 is regeneration process. This component is only used in system 1. Heat gain in regenerator is given by:

$$Q_{reg} = m \cdot (h_4 - h_3) = m \cdot (h_6 - h_7) \cdot \xi \tag{3}$$

For system 1, process 4-5 and for system 2, process 3-4 is heat addition to the working fluid in main gas heater. Engine exhaust gas acts as heating source for working fluid. Heat gained by working fluid is given by:

$$Q_{gh} = m \cdot (h_5 - h_4) = m_2 \cdot cp_g \cdot (t_{g\ in} - t_{g\ out}) \cdot \xi \tag{4}$$

$$Q_{gh} = m \cdot (h_4 - h_3) = m_2 \cdot cp_g \cdot (t_{g\ in} - t_{g\ out}) \cdot \xi \tag{5}$$

Here m_2 and cp_g is the mass flow rate and specific heat capacity of engine exhaust gas and $t_{g\ in}$ and $t_{g\ out}$ is the inlet and outlet temperature of exhaust gas.

For system 1 process 5-6 and for system 2 process 4-5 is expansion process in the turbine. In this process power is developed by turbine.

$$W_{turb} = m \cdot (h_5 - h_6) = m \cdot (h_5 - h_{6s}) \cdot \eta_{turb} \tag{6}$$

$$W_{turb} = m \cdot (h_4 - h_5) = m \cdot (h_4 - h_{5s}) \cdot \eta_{turb} \tag{7}$$

Here h_{6s} and h_{5s} are the entropy at outlet of turbine if process is taken as isentropic expansion. Also η_{turb} is the efficiency of turbine.

Process 6-7-1 is condensing process for system 1. Here 6-7 is already shown above in regenerator. Process 7-1 is cooling by external source in a condenser to bring the temperature down to 20°C. Heat loss in condenser:

$$Q_{cond} = m \cdot (h_7 - h_1) \cdot \xi \tag{8}$$

Process 5-6-1 is condensing process for system 2 loop-1. Also heating process for system 2 loop-2. Loop 2 follow similar set of equations of pump, turbine and heating as given above. Heat loss in condenser during process 6-1:

$$Q_{cond} = m \cdot (h_6 - h_1) \cdot \xi \tag{9}$$

Net Power produced by is given by:

$$W_{net} = W_{turb} - W_{pump} \tag{10}$$

However for system 2 net power produced in both loops are added to get overall power produced by system.

$$W_{net} = W_{net\ loop1} + W_{net\ loop2} \tag{11}$$

Total Heat absorbed for system 1 and 2 is given by:

$$Q_{total} = Q_{preh} + Q_{gh} \tag{12}$$

Thermal efficiency of system can be given by:

$$\eta_{ther} = \frac{W_{net}}{Q_{total}} \tag{13}$$

Utilization rate for engine exhaust gas (U_g) and engine coolant (U_c) are given by:

$$U_g = \frac{m_2 \cdot cp_g \cdot (t_{g\ in} - t_{g\ out})}{m_2 \cdot cp_g \cdot (t_{g\ in} - t_{g\ dew})} \tag{14}$$

$$U_c = \frac{m_1 \cdot cp_c \cdot (t_{c\ in} - t_{c\ out})}{m_1 \cdot cp_c \cdot (t_{c\ in} - t_{c\ dew})} \tag{15}$$

Here $t_{c\ dew}$ is coolant return temperature and $t_{g\ dew}$ is engine exhaust gas dew point temperature. Tables 3 and 4 are showing system-1 and System-2 parameters at specified points while keeping mass flow rate at 22 kg/min and effectiveness of heat exchangers at 0.95.

Here, the key step is to calculate the mass flow rate of working fluid (CO_2) first. Pinch point temperature difference (PPTD) method is first adopted to calculate the ideal mass flow rate for each system. For two waste heat recovery sources, we need to calculate two separate mass flow rate for condition of complete recovery of each waste heat, represented below:

$$m_{f,1} = \frac{m_1 \cdot cp_c \cdot (t_{c\ in} - t_{c\ dew})}{(h_3 - h_2)} \tag{16}$$

$$m_{f,2} = \frac{m_2 \cdot cp_g \cdot (t_{g\ in} - t_{g\ dew})}{(h_5 - h_4)} \tag{17}$$

Table 3 Thermodynamic parameters at various state points for system 1 (PR-CTRC)

State Point	Temperature (°C)	Pressure (MPa)	Entropy (kJ/kg-K)	Enthalpy (kJ/kg)
1	20	5.7291	1.1877	255.87
2	30.052	12	1.1942	265.71
3	83.4	12	1.7722	455.32
4	347.76	12	2.5112	799.88
5	489	12	2.7543	967.47
6	422.97	5.7291	2.7983	897.32
7	98.4	5.7291	2.0973	534.63

Table 4 Thermodynamic parameters at various state points for system 2 (dual loop P-CTRC)

State Point	Temperature (°C)	Pressure (MPa)	Entropy (kJ/kg-K)	Enthalpy (kJ/kg)
1	20	5.7291	1.1877	255.87
2	30.052	12	1.1942	265.71
3	83.4	12	1.7722	455.32
4	489	12	2.7543	967.47
5	422.97	5.7291	2.7983	897.32
6	50	5.7291	1.9167	472.15
11	20	5.7291	1.1877	255.87
12	30.052	12	1.1942	265.71
13	392.97	12	2.5939	853.10
14	331.22	5.7291	2.6374	792.85

Where $m_{f,1}$ and $m_{f,2}$ are ideal mass flow rate for maximum heat recovery of engine coolant and engine exhaust gas respectively. We have to choose minimum of both mass flow rate to get complete utilization of one waste heat source.

For the comparison of performance of turbine, calculation and comparison of volumetric flow ratio (VFR) and size parameter (SP) is adopted [11]. It give idea of size of turbines used in both systems.

$$VFR = \frac{V_{out}}{V_{in}} \tag{18}$$

$$SP = \frac{\sqrt{V_{out}}}{(\triangle\ h_{is})^{0.25}} \tag{19}$$

Where V_{in} and V_{out} volume flowing inward and outward of turbine respectively. These volumes can be calculated by multiplying mass of working fluid with density. $\triangle\ h_{is}$ represent change in isentropic enthalpy during the expansion process in turbine.

3.2 Exergy Calculation for Components

Exergy is the energy in a system that is available to be used. The concept of exergy is first introduced in 1873 by J. Willard Gibbs. Exergy of a system will become zero if system reaches equilibrium with surroundings. Exergy destruction is the loss of energy in a system. According to "Second Law of

Thermodynamics" for an irreversible process there is always some exergy destruction, for example heat loss to the surroundings by a system.

For the pump exergy destruction is given by,

$$I_{pump} = (h_1 - h_2) - T_o(s_1 - s_2) + W_{pump} \tag{20}$$

Where I is exergy destruction rate due to irreversibility, and T_o is the surrounding temperature.

For the preheater exergy destruction rate is given by,

$$I_{preh} = \left(1 - \frac{T_o}{T_3}\right) \cdot (h_3 - h_2) + (h_2 - h_3) - T_o(s_2 - s_3) \tag{21}$$

For the regenerator exergy destruction rate is given by,

$$I_{reg} = (h_6 - h_7) - (h_4 - h_3) + T_o[(s_7 - s_6) + (s_4 - s_3)] \tag{22}$$

For the gas heater exergy destruction rate is given by,

$$I_{gh} = \left(1 - \frac{T_o}{T_5}\right) \cdot (h_5 - h_4) + (h_4 - h_5) - T_o(s_4 - s_5) \tag{23}$$

For the turbine exergy destruction rate is given by,

$$I_{turb} = (h_5 - h_6) - T_o(s_5 - s_6) - W_{turb} \tag{24}$$

For the condenser exergy destruction rate is given by,

$$I_{cond} = (h_7 - h_1) - T_o(s_7 - s_1) \tag{25}$$

In second system (Dual loop P-CTRC) similar mathematical equations as similar components are used.

Table 5 shows the exergy destruction values for each component of system 1 and System 2 respectively.

The adopted boundary conditions for the simulation are given in Table 6. All properties of CO_2 at different temperature and pressure conditions are obtained from REFPROP.

4 Result and Discussion

In this work effect of variation of parameters turbine inlet temperature (TIT), pressure ratio (PR) and effectiveness of heat exchanger on performance of turbine are studied. Also the exergy destruction of each component is obtained.

Table 5 Component wise exergy destruction values of system 1 and system 2

System 1		System 2	
Component	Exergy Destruction (kJ/kg)	Component	Exergy Destruction (kJ/kg)
Pump	1.905	Pump 1	1.905
Pre-heater	13.538	Pre-heater	13.538
Regenerator	29.562	Gas heater 1	90.91
Gas heater	6.803	Turbine 1	12.898
Turbine	12.898	Heater 2	151.82
Condenser	12.110	Turbine 2	12.752
		Pump 2	1.905
		Condenser 2	112

Table 6 Boundary conditions for simulation [14]

Parameters	Value Taken
Turbine efficiency	0.7
Pump efficiency	0.8
Inlet temperature of cooling water	15°C
Exhaust gas acid dew point	120°C
Maximum temperature of cycle	489°C
PPTD of gas heater/regenerator/preheater/condenser	30/15/5/5°C
Condensing temperature	20°C
Maximum pressure of cycle	12 MPa
Minimum pressure of cycle	5.729 MPa
Environment temperature	20°C

4.1 Effect of Parametric Analysis

Figure 4 describes the effect of variation of turbine inlet temperature (TIT) on thermal efficiency and net power output of both the systems. For this comparison mass flow rate at 22 kg/min and effectiveness of heat exchanger at 0.95 has been kept constant. Net power output of System 2 is nearly 11–12 kW (53–55%) more than that of system-1 for variation in TIT from 449 to 489°C. On the other hand thermal efficiency of system-1 is higher than System-2 due to use of regenerator. As power difference is much higher between two systems, system 2 is considered as better choice in comparison with system-1.

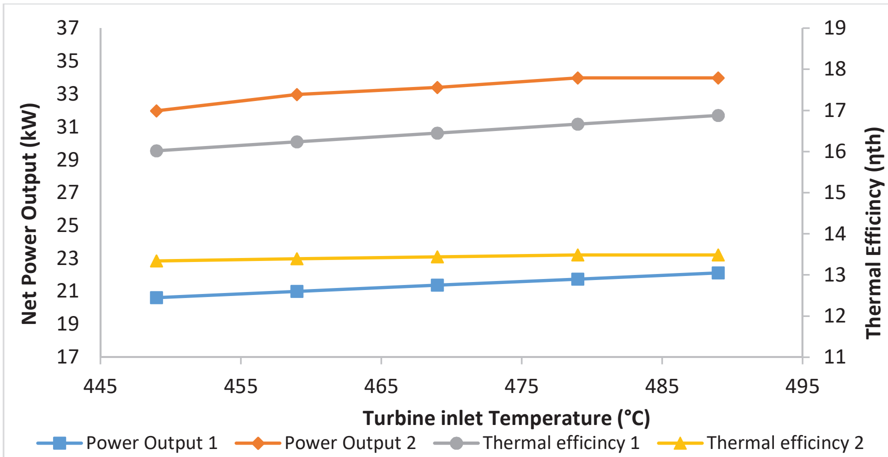


Figure 4 Comparison of net power output and thermal efficiency of systems while changing TIT.

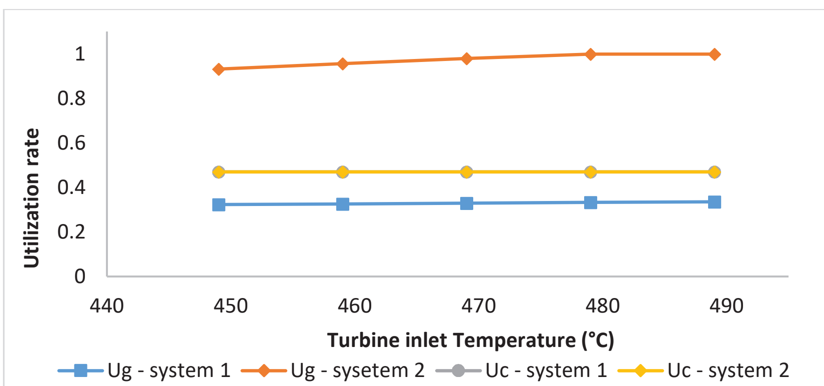


Figure 5 Comparison of exhaust gas and engine coolant utilization rates (u_g, u_c) of systems while changing TIT.

Comparison of utilization rate of both the systems is shown in Figure 5. Utilization rate shows how much part of waste heat energy is utilized by system from total available waste heat. In other words utilization rate shows heat recovery capability of a system. Utilization rate of engine exhaust gas (U_g) of system-1 and System-2 is showing slight increasing trend with the increase of TIT. However, System-2 is having very high utilization rate around 0.93 to 1 and in comparison to system 1 which has lower utilization rate of around

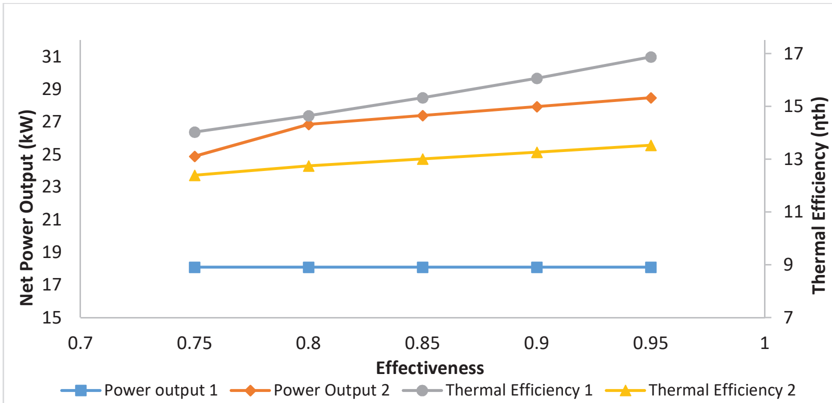


Figure 6 Comparison of net power output and thermal efficiency of systems while increasing effectiveness of heat exchanger.

0.32 to 0.34. The use of regenerator lowers down utilization rate of exhaust gas and in return increases thermal efficiency of system. Utilization rate of engine coolant (U_c) is showing increasing trend with increase in temperature and is having same values for both the systems.

For heat exchangers effectiveness variation related study the values of effectiveness from 0.75 to 0.95 have been varied. Figure 6 shows the comparison of both systems for variation of effectiveness of heat exchangers. For this comparison the mass flow rate at 18 kg/min and TIT as 489°C has been kept fixed. It is observed that power output of system-1 remains constant whereas of System-2 increases with increase of effectiveness because effectiveness is not affecting pump and turbine input-output conditions for system-1 and system-2 (loop-1). In system-2 due to loop-2 variation in power output is showing. At 0.95 effectiveness the maximum difference in net power output is obtained around 10.4 kW. Which means System-2 is having 57.5% higher net power output than system-1. This is due to dual loop structure of System-2 which is not present in system-1. However here also thermal efficiency of system-1 is higher than System-2 due to use of regenerator and both systems are showing increasing trend for thermal efficiency.

As the engine coolant waste heat utilization rate of both are system are same so the comparison of utilization rate of exhaust gas is treated as deciding parameter for comparison. Figure 7 shows the comparison between both systems. Engine coolant as well as engine exhaust gas utilization rate shows the decreasing trend here for both the systems while increasing effectiveness

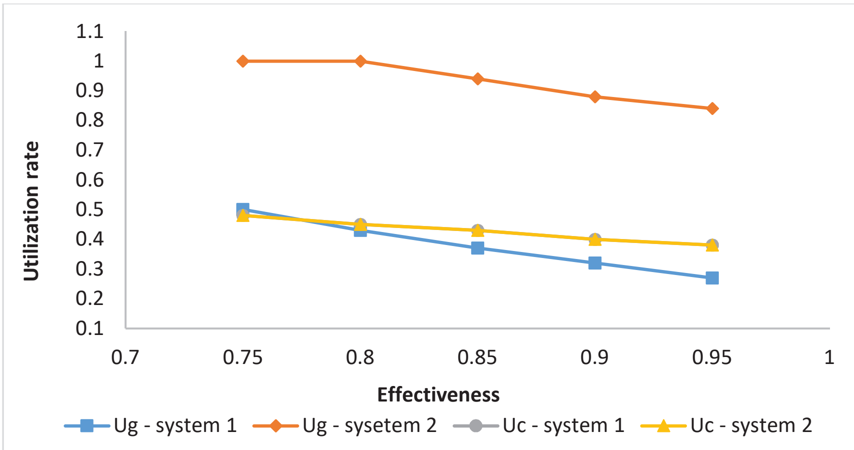


Figure 7 Comparison of exhaust gas and engine coolant utilization rates (U_g, U_c) of systems while changing effectiveness of heat exchanger.

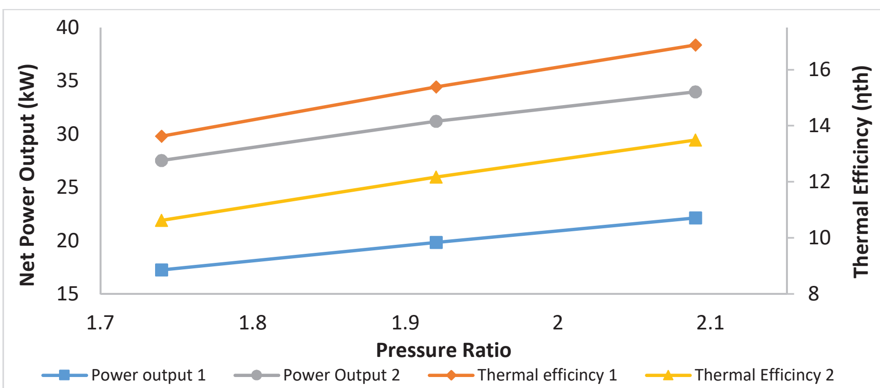


Figure 8 Comparison of net power output and thermal efficiency of systems while increasing Pressure ratio.

of system. The reason for this is due to increase in effectiveness of heat exchanger the heat exchanger is absorbing lesser heat from heat source to increase the temperature of working fluid to a fixed point and it results in higher temperature of heating source at exit of exchanger. Exhaust gas utilization rate of System-2 is very high in comparison to system-1. From figure we can see system 2 is absorbing nearly maximum possible energy from exhaust gas.

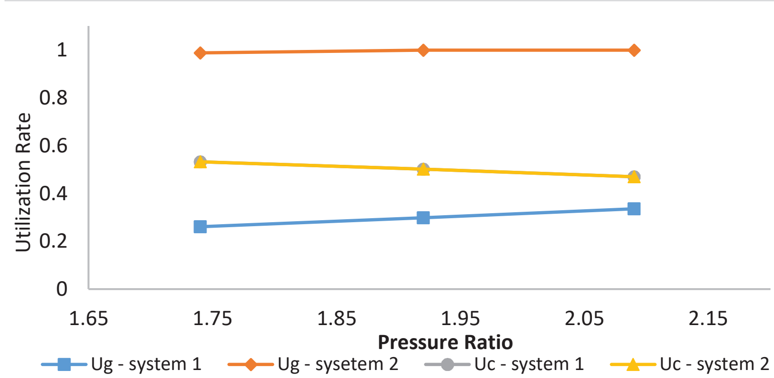


Figure 9 Comparison of exhaust gas and engine coolant utilization rates (U_g, U_c) of systems while changing PR.

For analysing effect of variation of Pressure ratio on systems, effectiveness of heat exchanger is fixed at 0.95, mass flow rate at 22 kg/min and TIT at 489°C. Lower pressure side of system is fixed at 5.7291 MPa and higher pressure side is varied from 10MPa to 12 MPa. For both the systems net power output and thermal efficiency are increasing with increase in Pressure ratio. The comparison of net power output results show, System-2 is producing 33.978 kW net power at 2.09 pressure ratio which is 53.6% higher than system-1 for same pressure ratio. On the other hand similar to TIT variation results here also Thermal efficiency of system-1 is found higher than System-2 for all pressure ratio by a considerable margin.

Figure 9 shows the effect of variation of pressure ratio on utilization rate of systems. Engine coolant utilization rate (U_c) is same for both the systems and is decreasing with increasing pressure ratio (PR). However exhaust gas utilization rate (U_g) is showing increasing trend for both the systems. For System-2 U$_g$ is nearly 1 for all the pressure ratios on the other hand U_g of system-1 is very low at 0.26 to 0.33. System-1 is not even able to utilize 50% of exhaust gas energy.

Figure 10 shows the effect of increasing mass flow rate on net power output and thermal efficiency of system-1 and 2. For this result the effectiveness is 0.95 and pressure ratio is 2.09 kept fixed. With increase in mass flow rate net power output of both the systems increase. It is observed that at 21.37 kg/min mass flow rate, graph shows a sudden change in trend of net power output of System-2. At this mass flow rate System-2 reaches its ideal point, and maximum difference in net power output of both systems

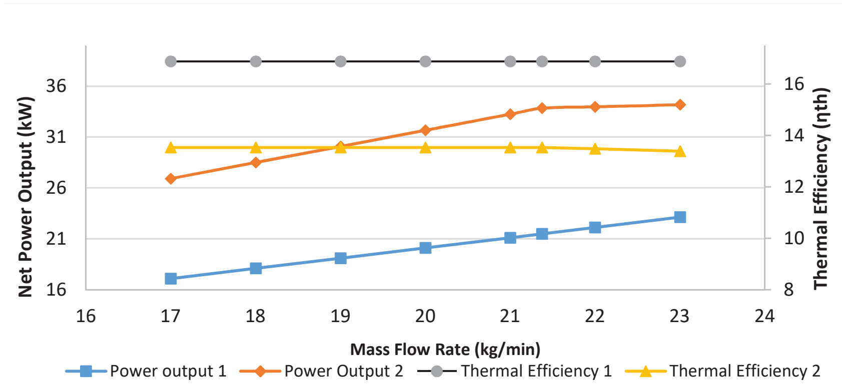


Figure 10 Comparison of Net Power Output & Thermal Efficiency of systems on basis of CO_2 mass flow rate.

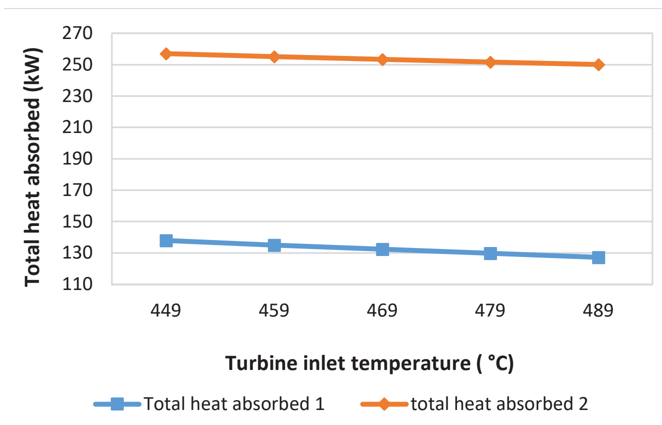
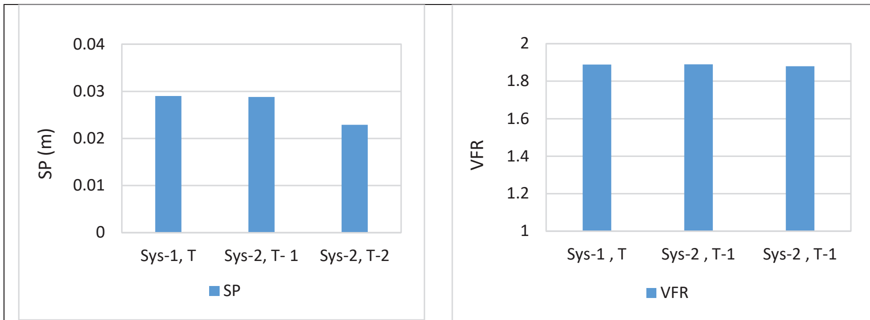


Figure 11 Comparison of Total heat absorbed on the basis of TIT.

is obtained i.e. System-2 is having 12.34 kW (57.42%) more power output than system-1. Thermal efficiency of system 1 is found more than System-2. However both systems have nearly constant thermal efficiency throughout the mass flow rate. In all the above comparison we found that System-2 is having nearly 50% more power output than system-1 under all different conditions, whereas system-1 is having higher thermal efficiency in all cases due to use of regenerator.

Figure 11 shows total waste heat absorbed by both the systems, at a mass flow rate (estimating using equations 16/17), with variation turbine inlet



a). Size parameter (SP) plot for turbines. b). Volumetric flow ratio (VFR) plot for turbines

Figure 12 Variation in SP and VFR for turbines of system-1 and System-2.

temperature (TIT). With increase in inlet temperature of turbine ideal mass flow rate is also reducing, hence total heat absorption is also reducing. It also show that for reaching higher temperature range, working fluid mass flow rate is needed to reduce in system.

Size parameter (SP) of system 1 turbine and system2 turbine2 are found same due to same input-output conditions, however system2 turbine2 have different temperature conditions at input-output therefore SP value for this turbine 2 came out less, as shown in Figure 12(a). Which also shows that second turbine size is smaller than first turbine. As we know SP values depends on volume flow at exit of turbine, hence it shows in turbine2 (system2) volume flow at exit of turbine is less. Volumetric flow ratio (VFR) is found to be nearly same for all the turbines, as shown in Figure 12(b). It means ratio of volumes of fluids flowing out and in is similar in all these cases.

4.2 Exergy Analysis

Second law of Thermodynamics tell us that in any irreversible thermodynamic process there is always some amount of heat loss to surrounding. Exergy destruction shows that loss of heat happened at a component. Figure 13 shows the component wise exergy destruction trends observed for system-1 and System-2. Exergy destruction in System-2 came out higher than system-1, which means loss of energy is more in System-2. In system-1 plot, Figure 13(a), we can see regenerator is having maximum exergy destruction of 29.5 (kJ/kg) followed by preheater and on the other hand gas heater is having low value. The reason for this is the behaviour of constant pressure

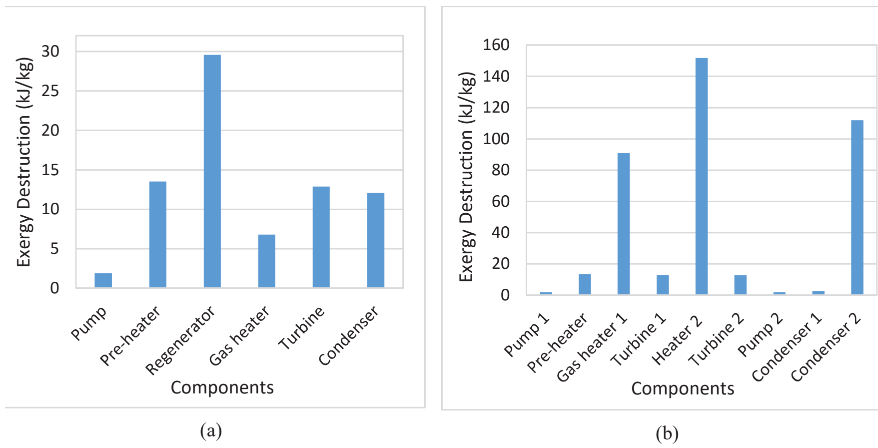


Figure 13 (a), (b) showing exergy destruction on each component of system-1 and System-2 respectively.

lines of CO_2 at higher temperature. By considering 12 MPa pressure line of CO_2 in T-s diagram (Figure 2), initially slight rise in temperature results in faster entropy increasing rate but later this rate falls and results in less exergy destruction at gas heater of system-1. In System-2 plot, Figure 13(b), the output temperature of working fluid is 393°C in heater2 where as in gas heater1 it is 489°C. The heater 2 is having maximum exergy destruction of 151 kJ/kg and gas heater 1 is having lower exergy destruction than heater 2. This is because that in gas heater1 with total temperature increase change in total entropy is very less in comparison to heater2. Pump and preheater are having almost same values of exergy destruction in both the systems. Therefore by comparing the two systems, exergy destruction provide a tool, which tells about the areas of major losses and require improvement.

5 Validation

In this study mathematical equations used for parametric analysis are taken from a CTRC based single loop system given in Ref. [14]. System results shows decline in total heat absorption with change in TIT at 12 MPa turbine inlet pressure which is found similar trend as we studied in a PR-CTRC system, mentioned in above reference, while changing temperature from 250 to 730°C at 15 MPa turbine inlet pressure. Turbine size parameter (SP) and volume flow ratio (VFR) equations are derived from a CTRC based waste heat recovery system used for comparison of different working fluids performance given in Ref. [11]. The values of SP and VFR obtained in our research for CO_2 turbine (both systems) is 0.029 m and 1.89 which very near to values found for CO_2 in mentioned research. Baheta et al. [16] provide the component wise exergy destruction equations that they derived for CTRC based system. In present work, trend for thermal efficiency with respect to increasing TIT at 12 MPa Turbine inlet pressure is showing similar trend as we studied in a PR-CTRC system results of above reference at 10 MPa turbine inlet pressure. Ge et al. [6] showed performance analysis of a CTRC based system, the results for power generation variation with increase in mass flow rate of CO_2 at 22 kg/min mass flow rate, 12 MPa pressure and 489°C temperature is showing similar trend studied in above mentioned work. Consequently these two systems results are similar to various systems that are based on CTRC concept and are published.

6 Conclusion

This study given the comparison of two systems based on CTRC for multiple waste heat recovery from a 9.5 L heavy duty truck engine. This type of systems not only increases power output and efficiency of engine without use of any extra fuel but also helps reducing environment pollution. Utilisation rate, net power output and thermal efficiency are main parameters for system performance comparison. Effect of mass flow rate variation on these parameters are also compared. Finally analysis of exergy destruction of each component is also performed. Based on results obtained following conclusions are drawn:

1. With the increase in TIT, pressure ratio or heat exchanger effectiveness, net power output of both the systems increases. As per the requirement system designer can choose system-1 for higher thermal efficiency and System-2 for higher power output under such varying parameters.
2. Exhaust gas utilisation rate for System-2 is nearly 1 and from 0.5 to 0.2 for System-2. We can conclude that for absorbing most of the heat energy of exhaust gas one should prefer System-2 over system-1.
3. Net power output of system-1 is 23.12 kW which increase the net power output of engine by 8.36% however System-2 which have 33.83 kW of power output increases the net power output of engine by 12.23%. This makes System-2 more attractive option to be used in multiple waste recovery of a diesel engine.

4. Exergy destruction of regenerator, gas heater and condenser of system 2 are found considerable higher than system2. It means energy loss to surrounding is higher in system2. System1 components will have less load and hence it makes system1 more compact and efficient than system2.

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