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Research Article

Energy and exergy analyses of a two-stage organic rankine cycle with low pressure stage regeneration for IC engine waste heat recovery

Pereddy NAGESWARA REDDY[1,*]

[1]*Department of Mechanical Engineering, Gudlavalleru Engineering College, Gudlavalleru - 521 356, Krishna District, A.P., India.*

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ABSTRACT

A two-stage Organic Rankine Cycle (ORC) with Low Pressure Stage Regeneration (LPSR) proposed in this article is intended to utilize the engine coolant energy completely for vaporization of organic fluid in a Low Pressure stage Heat Exchanger (LPHE) and the engine exhaust energy for sensible heating, vaporization and super heating of organic fluid in a High Pressure stage Heat Exchanger (HPHE) besides utilizing the superheated vapor energy of exhaust from Low Pressure stage Turbine (LPT) in a regenerator. Since regeneration is used only at low pressure stage, the energy associated with the engine exhaust gases can be utilized to the maximum extent by lowering its temperature nearer to the temperature of liquid phase working fluid after High Pressure stage Pump (HPP), thereby maximizing the Waste Heat Recovery (WHR) potential of the bottoming two stage ORC. The WHR efficiency of two-stage ORC with and without LPSR is analyzed at a typical operating condition of the engine using a nearly dry fluid R123 and a nearly isentropic fluid R134a as the working substances. It is observed that the thermal efficiency of the two-stage ORC with R123 is higher than that with R134a. The LP stage regeneration has been found to be effective in increasing the thermal efficiency and, in turn, the WHR efficiency of the two-stage ORC with both R123 and R134a. The increase in the fuel efficiency of the IC engine due to the bottoming two-stage ORC is found to be 7.22% with R123 and 6.21% with R134a with LPSR and 6.58% with R123 and 5.51% with R134a without LPSR. The optimum pressure in HPHE is found to be about 2.5 MPa and 3.5 MPa with R123 and R134a respectively.

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***Corresponding author.**
*E-mail address: pnreddy6494@gmail.com

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INTRODUCTION

The brake power output of total fuel energy ranges from 30% to 45% in compression ignition engine and just 20% to 35% in spark ignition engine. One of the ways of improving the efficiency of an IC engine is to recover the exhaust and coolant energies as a major part of fuel energy is carried away by the engine coolant and exhaust. In the last two decades, several concepts have been proposed for engine WHR. Among these, the ORC technology has been considered to be the best because of its higher WHR efficiency, reliability and flexibility.

J.P. Liu et al. [1] analyzed direct and indirect ways of recovering the engine exhaust energy; direct recovery through exhaust gas expansion and indirect recovery through bottoming Rankine steam cycle. The results showed that indirect recovery bottoming Rankine cycle was more efficient in converting the exhaust gas energy into power. Gequn Shu et al. [2] compared the performance of steam Rankine cycle (RC) and ORC for WHR from a large gaseous fuel IC engine. The ORC received increased interest over the past decade as a promising technology, especially for small and medium engines, to convert thermal energy into power. Compared with the steam Rankine cycle, the ORC employs organic fluids characterized by low boiling points, making it possible to generate power from low-grade energy sources such as engine coolant and exhaust. Many scholars conducted in-depth research on the waste heat recovery of diesel engine by ORC. The ORC power systems for low, medium and high temperature applications were reviewed by Piero C. et al. [3]. Various ORC technologies meant for recovering the waste heat from diesel engines were reviewed earlier [4-6].

TC Hung [7] analyzed the performance of ORC with benzene, toluene, p-xylene, R113 and R123 and indicated that benzene, toluene, and p-xylene were better in utilizing a high temperature heat while R113 and R123 were better for low temperature heat. T.C. Hung et al. [8] investigated the performance of ORC with wet, dry, and isentropic fluids and claimed that heat to power conversion efficiency of wet fluids was higher than that of the dry fluids. J.P. Roy et al. [9] analyzed ORC with working fluids R12, R123 and R134a for utilizing low temperature heat and showed that ORC with R123 was a better choice in terms of conversion efficiency. E.H. Wang et al. [10] analyzed ORC with nine different pure organic fluids and showed that R11, R141b, R113 and R123 were efficient in recovering low grade waste heat. Dongxiang Wang et al. [11] investigated the influence of thermodynamic properties of working fluid on the thermal efficiency of an ideal ORC. Results showed that fluids with low critical temperature, low liquid specific heat and high latent heat of vaporization were well suited for using in ORC. The potential of ORC system for exhaust waste heat recovery from marine diesel engines using bio-ethanol was investigated by Z. Mat Nawi et al. [12].

ORC systems for engine waste heat recovery are broadly classified into single loop and dual loop. In a single loop ORC, the working fluid is heated and vaporized by the engine coolant and later on superheated by the engine exhaust at the same pressure. In contrast to a single loop ORC, a dual loop system has a high temperature loop for recovering the exhaust energy and low temperature loop for recovering the remaining energy of exhaust gas and the coolant energy and can significantly reduce the exergy loss in the heat absorption process compared with conventional single-pressure evaporation ORCs. A single loop ORC system, combining with a gasoline engine, was investigated by Yung M.K. et al. [13], wherein the working fluid was preheated and vaporized by the engine coolant, and superheated by the engine exhaust. E.H. Wang et al. [14] analyzed a dual loop ORC for engine WHR. It showed that the relative power output of the engine improved from 14 to 16%. A dual loop ORC for recovering waste heat from the engine exhaust, intake air, and coolant was analyzed by H.G. Zang et al. [15]. Fubin Yang et al. [16] analyzed a dual loop ORC system to recover exhaust and coolant energies besides heat released in the intercooler of a turbo-charged diesel engine and showed that the increase in fuel efficiency could be up to 5.4%. Eunkoo Y. et al. [17] simulated single and dual parallel ORC systems for marine engine WHR. Sung and Kim [18] analyzed a novel dual loop ORC for engine waste heat using n-pentane and R125 as working fluids.

Recently, Guohui Zhu et al. [19] has investigated a combined ORC with double modes for engine WHR. The results indicated that the fuel efficiency increased from 4.7% to 5.8% depending on the BMEP with WHR efficiency of 7.3% to 10.7% when engine was coupled with ORC. Parametric investigation of four different waste heat recovery ORC systems was carried out by Yiji Lu et al. [20] using R245fa. Anandu S. et al. [21] analyzed two different two- stage architectures: Series two-stage ORC and Parallel two-stage ORC and compared their performances against a single stage pre-heated ORC at sub-critical conditions in the utilization of high temperature exhaust gases (573-773K) and low temperature secondary jacket water (353-373K) representing IC engine waste heat conditions. A novel cascade-Organic Rankine Cycle (C-ORC) system was proposed by Gequn Shu et al. [22] to recover multi-grade waste heat from a typical heavy-duty diesel engine. The C-ORC comprises a high-temperature ORC loop and a low temperature ORC loop to recover waste heat from an engine's exhaust gas, exhaust gas recirculation, jacket water and charge air in a cascaded pattern. Antonio Mariani [23] carried out a numerical and experimental analysis of an ORC system bottoming a CI engine powered passenger car with N-pentane (R601) and R134a as the working fluids to estimate the recoverable mechanical power and the consequent increase in the efficiency of engine at different torque-speed conditions.

Regeneration was shown to be an effective means of improving the thermal efficiency of steam Rankine cycle and ORC. Pedro J. Mago et al. [24] investigated a regenerative ORC with dry organic refrigerants such as R113, R245ca, and R123 as working fluids and concluded that the regenerative ORC was better than the basic ORC in utilizing low temperature heat source. Maoqing Li [25] conducted experiments on regenerative and basic ORC systems with R123 and showed that the thermal efficiency of the regenerative ORC was about 3.2 to 4.36% more when compared to basic ORC depending on superheating temperature at inlet to the turbine. Ozdemir et al. [26] presented a thermodynamics examination of basic ORC and regenerative ORC for waste heat recovery applications using dry organic fluids. R113, R114, R227ea, R245fa and R600a. Results showed that regenerative ORC had higher thermal efficiency compared with basic ORC. Enhua Wang et al. [27] examined a regenerative dual loop ORC system using a pair of environmentally friendly refrigerants, R1233zd and R1234yf, as working fluids, to recover energy from the waste heat of compressed natural gas (CNG) engine.

The exergy method of analysis is based on the Second Law of thermodynamics and the concept of irreversible production of entropy while the energy-based analysis is based on the First Law of thermodynamics and the concept of conservation of energy. Exergy analysis is a powerful and effective tool for analyzing and optimizing energy systems by combining the conservation of mass and energy principles with the Second Law of thermodynamics. Min-Hsiung Yang et al. [28] investigated the thermodynamic and economic performances optimization for an ORC system recovering the waste heat of exhaust gas from a large marine diesel engine. The maximum net power output index and thermal efficiency were obtained and the corresponding turbine inlet pressure, turbine outlet pressure, and effectiveness of pre-heater of the ORC system were also evaluated using R1234ze, R245fa, R600, and R600a. Seyedali et al. [29] performed exergy analysis of a two-parallel-step organic Rankine cycle (ORC) for waste heat recovery from an internal combustion engine with R-123, R-134a, and water as the working fluids. The net output power and the exergy efficiency were used as the objective functions with a goal of maximizing them. Jian Li et al. [30] investigated a typical dual-pressure evaporation ORC driven by the 100–200 °C heat sources without a limit on the outlet temperature. Evaporation pressures and evaporator outlet temperatures of the single-pressure and dual-pressure evaporation ORCs were optimized, and their optimized system thermodynamic performance was compared. Guillermo Valencia et al. [31] presented the energy and exergy analysis of three ORC–WHR configurations, viz. a simple ORC, an ORC with a recuperator, and an ORC with double-pressure configuration with cyclohexane, toluene, and acetone as ORC working fluids. Energy and exergy thermodynamic balances were employed to investigate the effect of evaporating pressure on the net power output, thermal efficiency and exergy destruction. Peng Liu [32] carried out an off-design performance analysis of an ORC fed by the waste heat from an IC engine. The effect of operational variables and engine load on system performance was analyzed from the aspect of energy and exergy to show its maximal working potential. Guanglin Liu [33] studied the subcritical saturated organic Rankine cycle system with four different organic working fluids. The variations in the exergy efficiencies for the single-stage/two-stage systems were analyzed. It was concluded that the exergy efficiency of the two-stage system was larger than that of the single-stage system.

From the above literature review, it is observed that two stage ORC systems are better choice for the conversion of heat energy of low temperature heat sources like IC engine waste heat to power when compared to single stage and dual loop ORC systems. Further, with an increase in vapor pressure and temperature in the HP stage heat exchanger (boiler), particularly with isentropic and dry working fluids, the vapor exhausted from the turbine is very much a super heated vapor. If this super heated exhaust vapor from the turbine is released into the condenser, it not only increases the cooling requirements of the condenser but also increases the exergy destruction. The higher the exergy destruction, the lower the thermal efficiency of the ORC plant. To mitigate this problem and improve the thermal efficiency of ORC plant further, the energy available with the turbine exhaust vapor is utilized in the proposed two-stage ORC to raise the temperature of liquid before LP stage heat exchanger by incorporating the regenerator. This will improve the mass flow rate of vapor generated in the LP stage heat exchanger as nearly saturated liquid enters it and thus increases the net power output of ORC plant. This will also increase the average temperature of heat addition in the LP stage heat exchanger, thereby increasing the thermal efficiency of the cycle.

Even though this two-stage ORC is not a new concept, utilizing the exhaust vapor energy for preheating of liquid before the LP stage heat exchanger and utilizing the energy of exhaust gases to the maximum extent by lowering its temperature nearer to the temperature of the liquid phase working fluid after HPP so as to increase the mass flow rate of working fluid vaporized in both the LPHE and the HPHE and thereby increasing the net power output of the ORC plant is the prime objective of the present work. To the best knowledge of the author, regeneration or preheating of liquid phase working fluid by turbine exhaust vapor energy has been applied so far to single-stage ORC systems only, but not to two-stage ORC cycles. Further, besides the energy analysis, exergy analysis of two-stage ORC has also been carried out in the present work to determine the optimum operating pressure in the HPHE for the minimum exergy destruction for the working fluids under consideration.

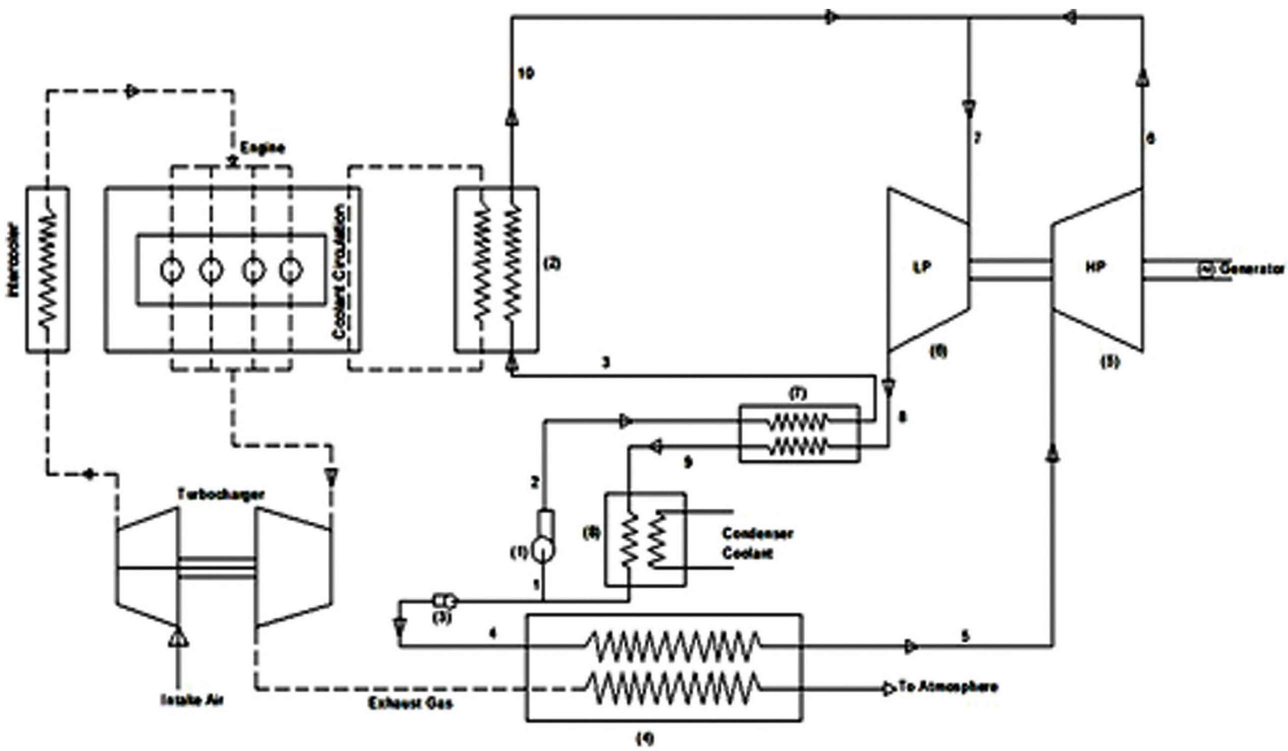

Figure 1. Schematic diagram of a two-stage ORC with LPSR for IC engine WHR.

(1. LP stage pump (LPP), 2. LP stage heat exchanger (LPHE), 3. HP stage pump (HPP), 4. HP stage heat exchanger (HPHE), 5. HP stage turbine (HPT), 6. LP stage turbine (LPT), 7. regenerator, and 8. Condenser)

TWO-STAGE ORC WITH LPSR FOR IC ENGINE WHR

Working Principle: The schematic diagram of a two-stage ORC with LPSR is shown in Figure 1.

Firstly, a part of the working fluid is pressurized from condenser pressure p_1 to intermediate pressure p_2 in LP stage pump, and then it flows through the regenerator where it is pre-heated by the turbine exhaust vapor. The preheated working fluid flows through LPHE where it is further heated and vaporized by utilizing engine coolant energy. The remaining working fluid is pressurized from pressure p_1 to p_3 in HP stage pump, and then flows through HPHE where it captures engine exhaust energy and turns into superheated vapor. The high pressure superheated vapor is expanded from pressure p_3 to p_2 in the HP turbine (HPT). The exhaust vapor from HPT mix with vapor coming out of LPHE before LP turbine (LPT) and together is expanded from pressure p_2 to p_1 in LPT. The power output of both HPT and LPT is used to drive the generator. Finally, the exhaust from LPT passes through the regenerator and then enters the condenser where it is condensed by giving up heat to the coolant. In this two-stage ORC with LPSR, both IC engine coolant and exhaust gas energies, besides the superheated vapor energy of turbine exhaust, are utilized effectively. T-s diagram of a two-stage ORC with LPSR is presented in Figure 2.

THERMODYNAMIC PROCESSES AND ANALYSIS

Since the application of LP stage regeneration is a new concept as far as two-stage ORC is concerned, a detailed energy and exergy analysis of two-stage ORC with LPSR is presented below with respect to Figure 2 based on the First and Second laws of thermodynamics. Steady state operation with no pressure drop and no heat loss in each component of the system is assumed.

Energy based analysis:

The specific enthalpies of working fluid at the exit of LP and HP stage pumps are calculated as

$$h_2 = h_1 + \frac{v_1(p_2 - p_1)X1000}{h_p} \tag{1}$$

$$h_4 = h_1 + \frac{v_1(p_3 - p_1)X1000}{h_p} \tag{2}$$

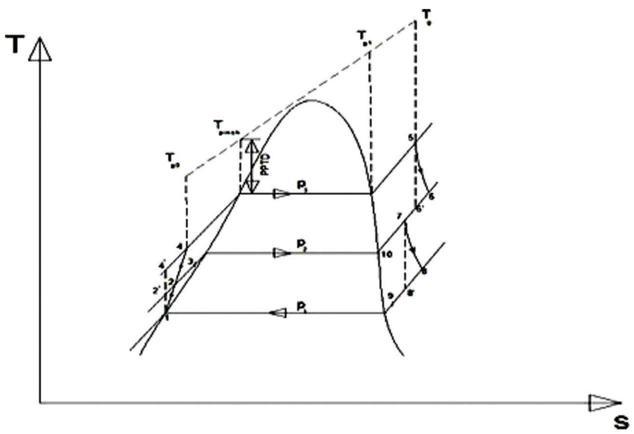

Figure 2. T-s diagram of a two-stage ORC with LPSR for IC engine WHR.

The working fluid from HP stage pump is vaporized and super heated at pressure p_3 in the HPHE from state point 4 to state point 5 by utilizing exhaust gas energy. The mass flow rates of working fluid through HPHE and LPHE are determined from the first law of thermodynamics as

$$m_1(h_5 - h_4) = m_g \cdot c_{pg}(T_g - T_{g0}) \tag{3}$$

$$m_2(h_{10} - h_3) = m_w \cdot c_{pw}(T_{w1} - T_{w2}) \tag{4}$$

where m_1 and m_2 are the mass flow rates of working fluid in kg/s through HPHE and LPHE, respectively; m_g and m_w are the mass flow rates of engine exhaust gas and coolant in kg/s, respectively; T_g and T_{g0} are exhaust gas temperatures at inlet and outlet of HPHE, respectively; T_{w1} and T_{w2} are engine coolant temperatures at inlet and outlet of LPHE, respectively; and c_{pg} and c_{pw} are isobaric specific heats of exhaust gas and coolant, respectively.

The temperature of exhaust gas, T_{g1} after superheating section of HPHE is calculated by energy balance as

$$m_g c_{pg}(T_g - T_{g1}) = m_1(h_1 - h_{gb}) \tag{5}$$

where h_{gb} is the enthalpy of saturated vapor at pressure p_3.

The pinch-point temperature, T_{pinch}, that is the temperature of exhaust gas in the HPHE at the beginning of vaporization of working fluid,and T_{g0} have been calculated by solving the equations (6) and (7) as given in [12].

$$T_{pinch} = \frac{T_{g1} - T_{g0}}{h_{gb} - h_4}\{(h_{fb} - h_4) - (h_{gb} - h_4)\} + T_{g1} \tag{6}$$

$$m_g c_{pg}(T_{pinch} - T_{g0}) = m_1(h_{fb} - h_4) \tag{7}$$

The pinch point temperature difference (PPTD) in HPHE is given by

$$PPTD = T_{pinch} - T_{sat,b} \tag{8}$$

where $T_{sat,b}$ is the saturation temperature at pressure p_3.

The super heated vapor from the HPHE is expanded firstly in the HPT. The specific enthalpy of working fluid after HP stage expansion, h_6, is calculated as

$$h_6 = h_5 - \eta_t(h_5 - h_{6'}) \tag{9}$$

The exhaust vapor from HPT mix with vapor leaving the LPHE at intermediate pressure p_2. The specific enthalpy of vapor after mixing process at state point 7 is calculated by energy balance as

$$h_7 = \frac{m_1 h_6 + m_2 h_{10}}{(m_1 + m_2)} \tag{10}$$

After mixing process, the vapor at state point 7 is expanded in the LPT. The specific enthalpy of working fluid after LP stage expansion, h_8, is determined as

$$h_8 = h_7 - \eta_t(h_7 - h_{8'}) \tag{11}$$

The exhaust vapor from LPT flows through the regenerator. The temperature of pressurized working fluid, T_3, after regeneration is calculated from the effectiveness of the regenerator as given in [16].

$$\in_{reg} = \frac{(T_3 + T_2)}{(T_8 + T_2)} \tag{12}$$

where $\in_{reg}$ is the effectiveness of regenerator and is set to 0.8 [16]; and T_2 and T_8 are the temperatures of working fluid at state points 2 and 8 respectively.

The specific enthalpy of working fluid, h_3, after LPSRis calculated at temperature T_3.

The power output of two-stage ORC is calculated as

$$P_{orc} = m_1[(h_5 - h_6) - (h_4 - h_1)] + (m_1 + m_2)(h_7 - h_8) - m_2(h_2 - h_1) \tag{13}$$

The thermal efficiency of two-stage ORC, η_{orc}, is given by

$$h_{orc} = \frac{P_{orc}}{\{m_1(h_s - h_4) + m_2(h_{10} - h_3)\}} \times 100\% \tag{14}$$

The WHR efficiency i.e. the percentage of engine waste heat converted into power output, η_{WHR}, of this two-stage ORC is given by

$$h_{WHR} = \frac{P_{orc}}{\{dQ_{coolant} + dQ_{exhaust}\}} \times 100\% \tag{15}$$

where $\delta Q_{coolant}$ is the rate of coolant energy, and $\delta Q_{exhaust}$ is the rate of available energy of engine exhaust gas.

$$\delta Q_{coolant} = m_w \cdot c_{pw} (T_{w1} - T_{w2}) \quad (16)$$

$$\delta Q_{exhaust} = m_g \cdot c_{pg} (T_g - T_{air}) \quad (17)$$

where T_{air} is the temperature of air at inlet to the engine and is taken as 288.15 K.

The cumulative fuel efficiency of IC engine i.e. the percentage of fuel energy converted into useful power output when the engine is coupled with the ORC plant, η_{cum}, can be calculated as

$$h_{cum} = \frac{BP_{ice} + P_{orc}}{m_f \cdot LHV} \times 100\% \quad (18)$$

where BP_{ice} is the brake power output of IC engine without bottoming ORC, m_f is the rate of fuel consumption, and LHV is the lower heating value of fuel.

The fuel efficiency of IC engine without bottoming two-stage ORC, η_{ice}, is calculated as

$$h_{ice} = \frac{BP_{ice}}{m_f \cdot LHV} \times 100\% \quad (19)$$

The increase in fuel efficiency of IC engine, η_{inc}, after coupling with two-stage ORC is determined as

$$\eta_{inc} = \eta_{cum} - \eta_{ice} \quad (20)$$

Exergy based analysis:

Exergy destruction in HPHE,

$$E_{dist,HPHE} = m_1 T_{air} (s_5 - s_4) - m_g c_{pg} T_{air} Ln \frac{T_g}{T_{g0}} \quad (21)$$

Exergy destruction in LPHE,

$$E_{dist,LPHE} = m_2 T_{air} (s_{10} - s_3) - m_w c_{pw} T_{air} Ln \frac{T_{w1}}{T_{w2}} \quad (22)$$

Exergy destruction in regenerator,

$$E_{dist,Regenerator} = m_2 T_{air}(s_3 - s_2) - (m_1 + m_2) T_{air}(s_8 - s_9) \quad (23)$$

Exergy destruction in condenser,

$$E_{dist,Condenser} = m_w c_{pw} T_{air} Ln \frac{T_{w1}}{T_{w2}} (m_1 + m_2) T_{air} (s_9 - s_1) \quad (24)$$

Exergy destruction in HP turbine,

$$E_{dist,HPT} = m_1 T_{air} (s_5 - s_6) \quad (25)$$

Exergy destruction in LP turbine,

$$E_{dist,LPT} = (m_1 + m_2)\, T_{air} (s_7 - s_8) \quad (26)$$

Exergy destruction in HP pump,

$$E_{dist,HPP} = m_1 T_{air} (s_4 - s_1) \quad (27)$$

Exergy destruction in LP pump,

$$E_{dist,LPP} = m_2 T_{air} (s_2 - s_1) \quad (28)$$

Net exergy destruction,

$$\begin{aligned} E_{dist,Net} = E_{dist,HPHE} + E_{dist,LPHE} + E_{dist,Regenerator} + \\ E_{dist,Condenser} + E_{dist,HPT} + E_{dist,LPT} + E_{dist,HPP} + E_{dist,LPP} \end{aligned} \quad (29)$$

SELECTION OF WORKING FLUIDS

The selection of working fluids depending upon turbine inlet pressure has a significant influence on the performance of a basic and regenerative ORC. The working fluid of an ORC determines thermal efficiency, safety, stability, environmental impact, and economic profitability of the system. Hence, a proper selection of working fluid is essential in improving the waste heat recovery potential of ORC cycle. Working fluids are classified as dry, isotropic, or wet fluids depending on the slope of the saturation vapor curve on a T–s diagram (dT/ds). The slope of the saturation curve of a working fluid in a T–s diagram can be positive, negative or vertical, and the fluids are accordingly called 'wet', 'dry' and 'isentropic' fluids. Isentropic or dry fluids are suggested for organic Rankine cycle to avoid liquid droplet impingent in the turbine blades during the expansion. The physical and chemical properties of the nearly dry fluids R123 and R245fa and a nearly isentropic fluid R134a are presented in Table 1.

OPERATING PARAMETERS OF A TWO-STAGE ORC

The operating pressures in the LPHE and HPHE are selected such that the saturation temperature of working fluid at a given pressure is below the temperature of heat source. The condenser pressure is set by considering the coolant temperature as 288 K. The operating parameters of a two-stage ORC for R123 and R134a are given in Table 2.

The PPTD in LPHE is taken as 10 K [19] and vapor leaving the LPHE is considered to be just dry and saturated as there is no scope for superheating. By taking the pressure (p_3) and temperature (T_5) at inlet to the HPT as independent variables, two-stage ORC with and without LPSR is

Table 1. Physical and chemical properties of R123, R245fa and R134a

Property name (Unit)	R123	R245fa	R134a
Chemical formula	CF_3-$CHCl_2$	$CF_3CH_2CHF_2$	CH_2FCF_3
Molecular weight (kg/kg mol)	152.93	134.0	102.03
Normal boiling point (°C)	27.85	15.3	–26.1
Freezing point (°C)	–107.0	–107.0	–103.0
Critical temperature (°C)	183.68	154.05	101.1
Critical pressure (MPa)	3.668	3.640	4.060
Density (Liquid) at 25°C (kg/m^3)	1463.0	1339.0	1206.0
Heat of vaporization at normal boiling point (kJ/kg)	170.0	196.7	217.2
Liquid specific heat at 25°C (kJ/kg-K)	0.965	1.360	1.440
Vapor specific heat at 1 atm and 25°C (kJ/kg-K)	0.721	0.8931	0.852
Thermal conductivity (Liquid) at 25°C (W/m-K)	0.081	0.081	0.0824
Thermal conductivity (Vapor) at 1 atm and 25°C (W/m-K)	0.0112	0.0125	0.0145
Viscosity (Liquid) at 25°C (m-Pa-s)	0.456	0.4027	0.202
Viscosity (Vapor) at 1 atm and 25°C (m-Pa-s)	0.011	0.0103	0.012
Ozone depletion potential (ODP)	0.02	0.0	0.0
Global Warming Potential (GWP)	93	1030	1200

Table 2. Operating parameters of a two-stage ORC

Parameter name (Unit)	R123	R245fa	R134a
Pump efficiency (both LP and HP stage), η_p	0.75	0.75	0.75
Turbine efficiency (both LPT and HPT), η_t	0.72	0.72	0.72
Pressure in the HPHE, p_3 (MPa)	1.0 – 3.0	1.0 – 3.5	3.0 – 4.0
Pressure in the LPHE, p_2 (MPa)	0.43114	0.6953	2.3643
Pressure in the condenser, p_1 (MPa)	0.09827	0.15965	0.7067

simulated at a typical operating condition of the engine. The waste heat energy-recovery potential of a two-stage ORC with and without LPSR is presented in the subsequent section and the results are compared with those presented in [19].

TESTED RESULTS OF IC ENGINE

Since Gasoline Direct Injection (GDI) is the state-of-the-art technology for the advanced gasoline engine, the inline, four cylinder, four stroke, water cooled, turbo-charged, GDI engine has been chosen as the study object for this paper. The rated technical specifications of a typical GDI engine are given in Table 3.

Table 3. Engine specifications

Parameter name (Unit)	Value
Bore (mm)	82.5
Stroke (mm)	92.8
Compression ratio	9.6
Rated brake power (kW/rpm)	155/5300
Maximum torque (N.m/(rpm))	280/1700

Since the speed of 2000 rpm is the common speed of GDI engine, it was taken as the target speed for conducting load and energy balance tests. All tests were conducted with 250 kW AC dynamometer equipped with a torque meter. The orifice meter was used to measure the rate of air consumption. The amount of fuel supplied to the engine was measured accurately with Coriolis flow-meter, which could measure very wide ranges of fueling rates from 0.2 to 125 kg/h with the measurement uncertainty lower than 0.12%. A piezoelectric pressure transducer (Kistler 6065A) mounted in the cylinder head was used to measure in-cylinder pressure. Chromel-Alumel K-type thermocouples

Table 4 . Tested results of the engine

Parameter name (Unit)	Value
Brake mean effective pressure (MPa)	1.4
Speed (r/min)	2000
Brake power, BP_{ice} (kW)	47.6
Rate of air consumption, m_a (kg/s)	0.04494
Rate of fuel consumption, m_f (kg/s)	0.0032139
Temperature of coolant at outlet, T_{w1} (°C)	88.9
Temperature of coolant at inlet, T_{w2} (°C)	84.2
Mass flow rate of coolant, m_w (liter/s)	1.5128
Temperature of exhaust gas after turbo-charger, T_g (°C)	795.2

Table 5. Comparison of present results with those presented in [19].

Parameter name	Present with LPSR		Present without LPSR		Ref. 19
	R123	R134a	R123	R134a	R123
WHR efficiency (%)	13.39%	10.91%	12.33%	9.16%	10.7%
Improvement in fuel efficiency	7.04%	5.73%	6.48%	4.81%	5.8%

were connected to a 12 channel digital panel meter to measure the temperatures of exhaust gas and jacket cooling water. An 'MTC' make digital panel tachometer was used for the measurement of engine speed. The tested results of the engine at 1.4 MPa BMEP and 2000 rpm are presented in Table 4.

SIMULATION RESULTS AND PARAMETRIC OPTIMIZATION

Based on the thermodynamic analysis presented in the "THERMODYNAMIC PROCESSES AND ANALYSIS" section, a simulation code is developed in MATLAB to simulate the performance of a "Two-Stage Organic Rankine Cycle with Low Pressure Stage Regeneration". The comparison of WHR efficiency and the improvement in fuel efficiency of the present analysis with those presented in [19] is given in the Table 5 at 3.0 Mpa pressure in the HPHE and 252°C temperature at HPT inlet.

Simulation results show variation in the performance of two-stage ORC with operating pressures and temperatures. The PPTD in the HPHE, and the exhaust gas temperature at the HPHE outlet (T_{g0}) with R123 and R134a as ORC working fluids are plotted against the vapor temperature at HPT inlet (T5) at different operating pressures of HPHE in Figure 3. With an increase in vapor temperature at HPT inlet, both PPTD and T_{g0} decrease as more energy of the engine exhaust gas is utilized for superheating of vapor in the HPHE. With an increase in pressure in the HPHE, both PPTD and T_{g0} increase due to corresponding increase in vaporization temperature. With an increase in vaporization temperature, more sensible heat is to be added to increase the temperature of the liquid to the saturated state and hence higher pinch point temperature (T_{pinch}) is needed. For a given effectiveness of the heat exchanger, the higher the T_{pinch}, the greater the T_{g0}. The greater the T_{g0}, the lower the utilization of available energy of exhaust gas. Further, since the vaporization temperatures are low in the case of R134a due to low critical temperature as compared to that of R123, the PPTD and T_{g0} are also low.

Figures 4 and 5 show the variation in exhaust gas temperature at HPHE outlet (T_{g0}) and exhaust vapor temperature at LPT outlet (T_8), with and without LPSR, with the vapor temperature at HPT inlet (T5) at different operating pressures of HPHE, with R123 and R134a respectively. With an increase in T5, the exhaust gas temperature at HPHE outlet decreases. This results in better utilization of the available energy of exhaust gases. LPSR does not influence the high pressure stage vaporization and hence, Tg_0 remains the same with and without LPSR. However, with LPSR, the mass flow rate of vaporization in LPHE increases (30 to 45% of mass of total working fluid flows through HPHE and the remaining flows through LPHE depending on the vapor temperature at HPT inlet) and the state point of vapor after mixing of HPT exhaust vapor and vapor coming out of LPHE, i.e. state point (7) moves towards saturation point. As a result, the temperature at state point (8) after expansion in the LPT is also low. Hence, T_8 with LPSR is relatively low as compared to T_8 without LPSR. With an increase in T_5, the temperature of exhaust vapor from HPT increases. As a result, the temperature of vapor after mixing (T_7) also increases. With an increase in T_7, the exhaust vapor temperature at LPT outlet (T_8) increases.

The expansion ratio of HPT increases with an increase in pressure (p_3) of HPHE, and as a result, the temperature T_7 decreases. This causes decrease in exhaust vapor temperature at LPT outlet (T_8). The lower the temperature T_8, the smaller the influence of LPSR is on the improvement of efficiency. This can be observed clearly in subsequent figures.

The thermal efficiency of this two-stage ORC demonstrates the heat recovery potential of the proposed arrangement. Figures 6 and 7 show the variation in thermal and WHR efficiencies of the two-stage ORC with operating pressure (p_3) and vapor temperature (T_5), respectively with R123 and R134a as ORC working fluids. With an increase in pressure (p_3) and vapor temperature (T_5), both the thermal and WHR efficiencies of the cycle increase. With LPSR,

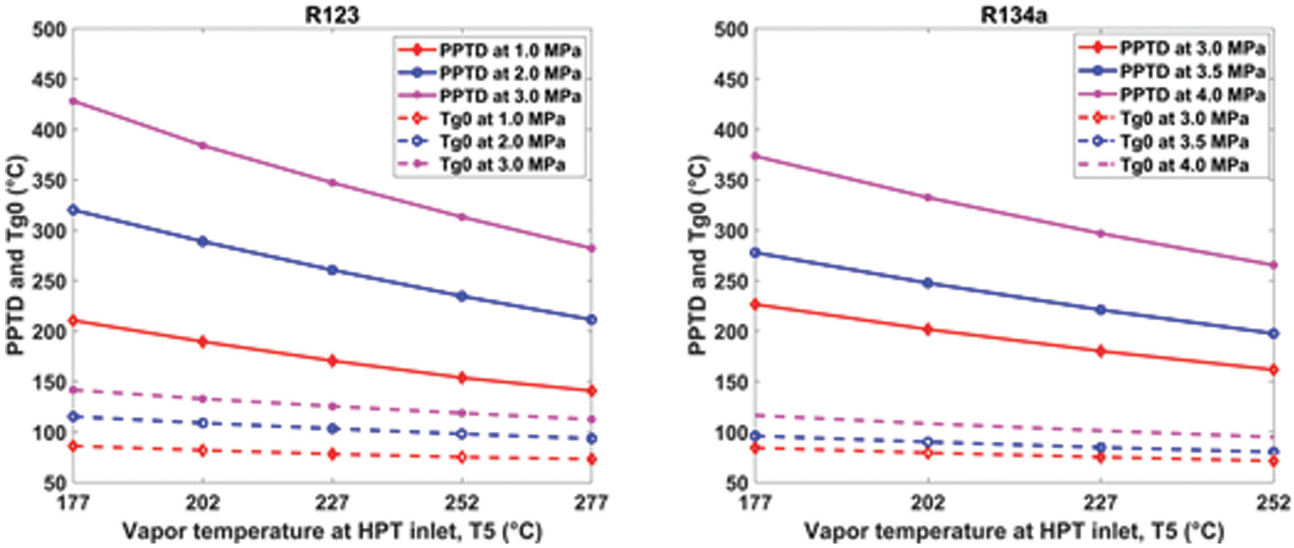

Figure 3. PPTD and exhaust gas outlet temperature with R123 and R134a.

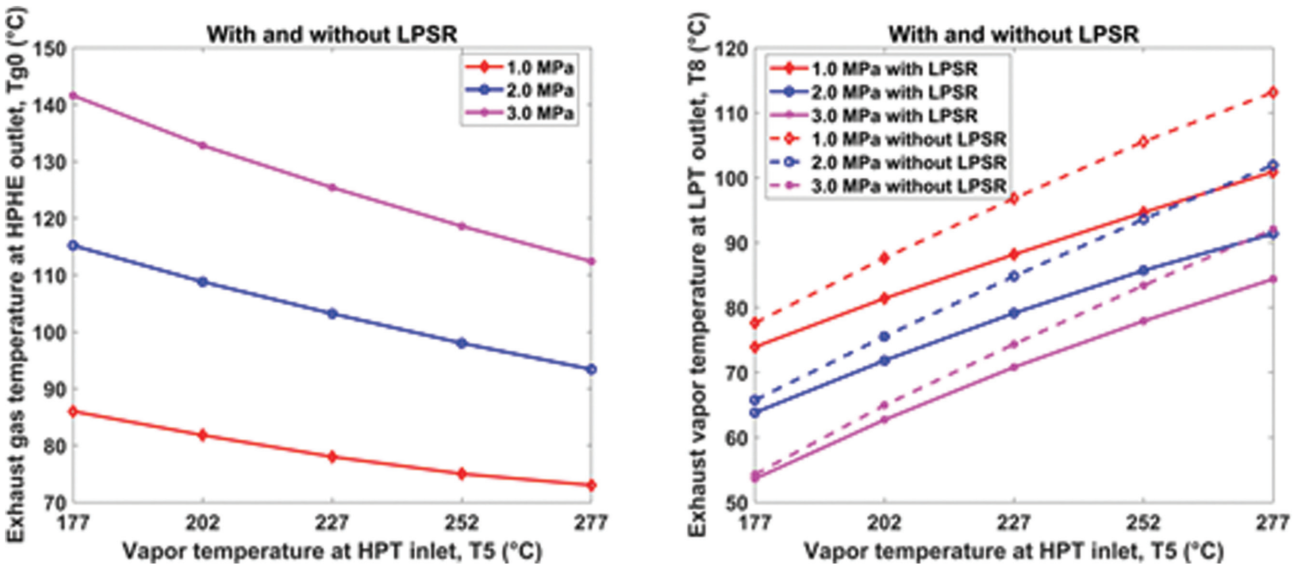

Figure 4. Exhaust gas outlet and exhaust vapor outlet temperatures with R123.

the thermal and WHR efficiencies of ORC are considerably higher than that of without LPSR as a part of the enthalpy of exhaust vapor from the LPT is recovered in the regenerator and is used to raise the temperature of working fluid before LPHE.

The maximum thermal efficiency of the two-stage ORC at 3.0 MPa pressure and 277^{0}C temperature is 14.78% with LPSR and 13.46% without LPSR while the corresponding WHR efficiency is 13.79 % with LPSR and 12.52% without LPSR with R123. The maximum WHR efficiency of 10.7% is estimated by Guohui Zhu et al. [19] at similar operating conditions and ORC design parameters, which is 3% lower than that predicted with the two-stage ORC under investigation. Hence, the two stage ORC system is a better choice for an IC engine waste heat recovery when compared to the single stage or dual loop ORC systems. The heat recovery potential of two stage ORC system can be further increased with LPSR proposed in the present work, especially with dry and isentropic working fluids.

Even though T_{g0} is lower with R134a than that of R123, both thermal and WHR efficiencies are lower with R134a than those with R123 because of smaller expansion ratio in the case of R134a. The LPSR has a greater influence on both the thermal and WHR efficiencies in the case of R134a as

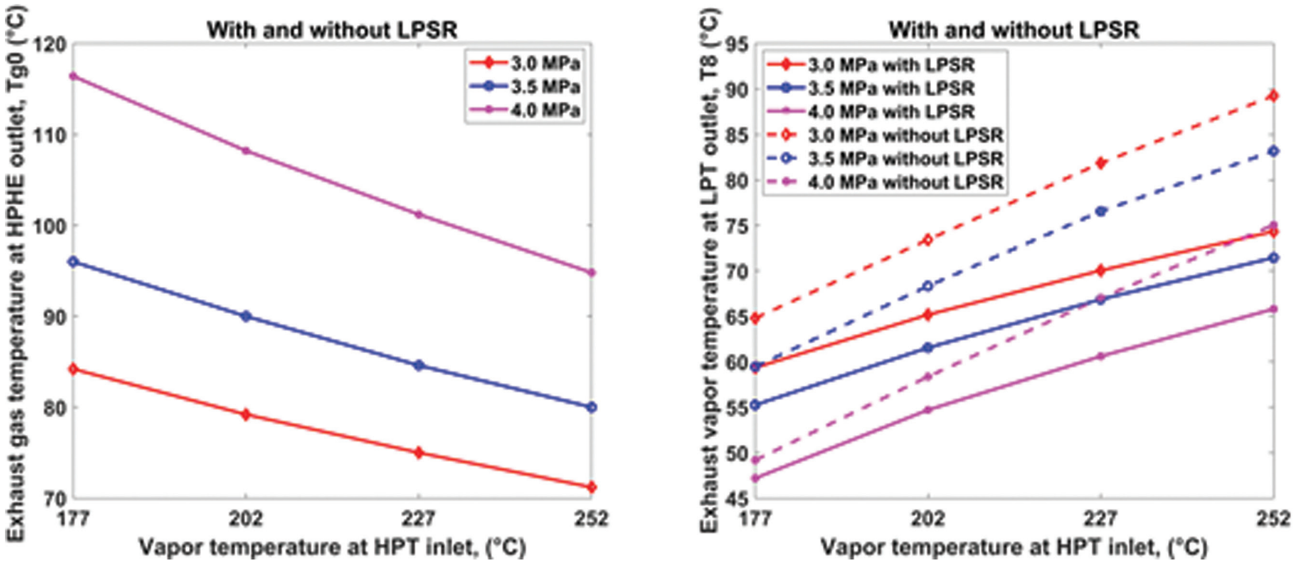

Figure 5. Exhaust gas outlet and exhaust vapor outlet temperatures with R134a.

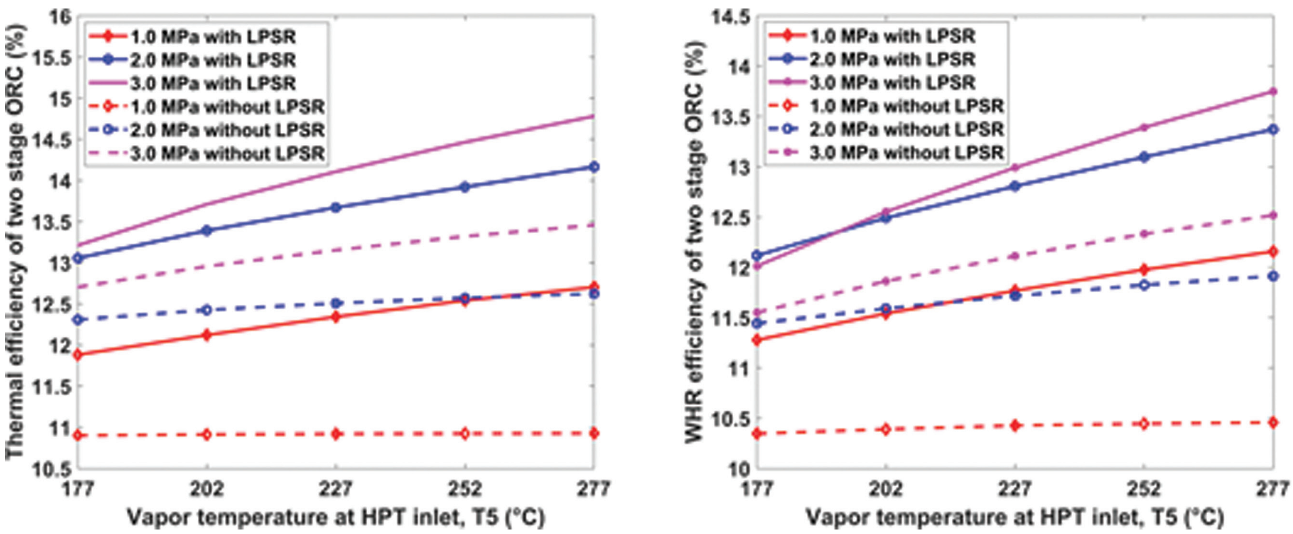

Figure 6. Thermal and WHR efficiencies of a two-stage ORC with R123 at 1.4 MPa BMEP.

compared to that in the case of R123. Further, it is observed that the rate of increase in thermal and WHR efficiencies of the two-stage ORC with LPSR decreases with an increase in operating pressure (p3). With an increase in operating pressure (p3), the temperature of vapor at the exit of HPT (T_6) and in turn at the exit of LPT (T_8) decreases. This lowers the rate of heat transfer during regeneration.

Figure 8 shows the improvement in IC engine fuel efficiency, when the IC engine is coupled with the bottoming two-stage ORC, with operating pressure (p_3) and vapor temperature (T_5) with R123 and R134a as ORC working fluids. As the thermal and WHR efficiencies of a two-stage ORC increase with an increase in cycle pressure and vapor temperature, the IC engine fuel efficiency also increases. With LPSR, the improvement in fuel efficiency of IC engine due to two-stage ORC is considerably higher as a part of the enthalpy of exhaust vapor from LPT is recovered. The increase in fuel efficiency of an IC engine due to bottoming two-stage ORC at 3.0 MPa pressure and 277°C is found to be 7.22% with LPSR and 6.58% without LPSR in the case of R123, and 6.21% with LPSR and 5.51% without LPSR in the case of R134a. The maximum improvement in fuel

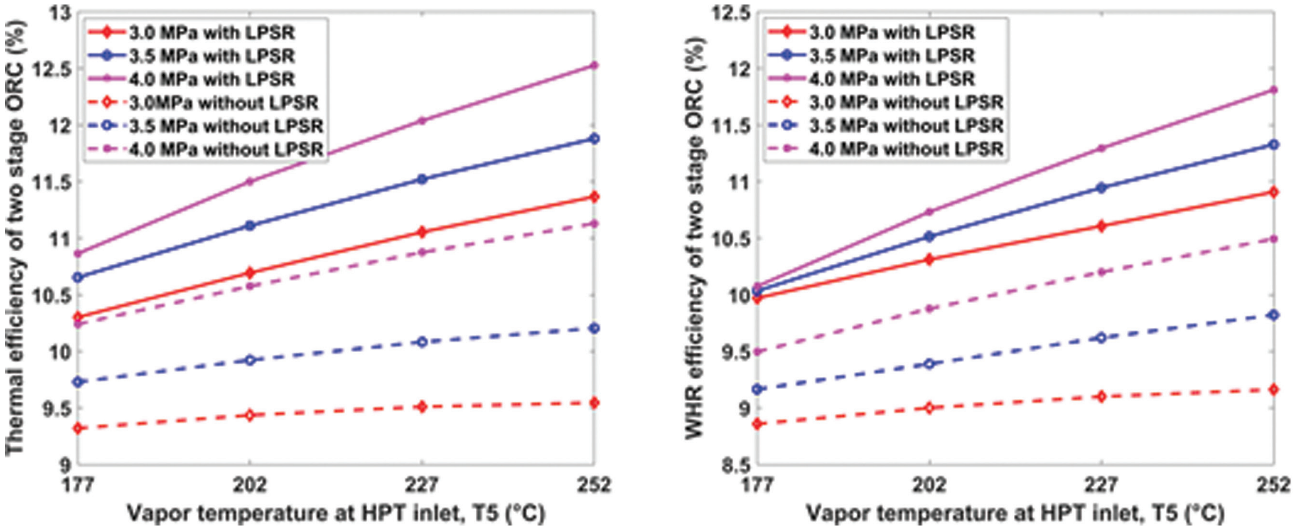

Figure 7. Thermal and WHR efficiencies of a two-stage ORC with R134a at 1.4 MPa BMEP.

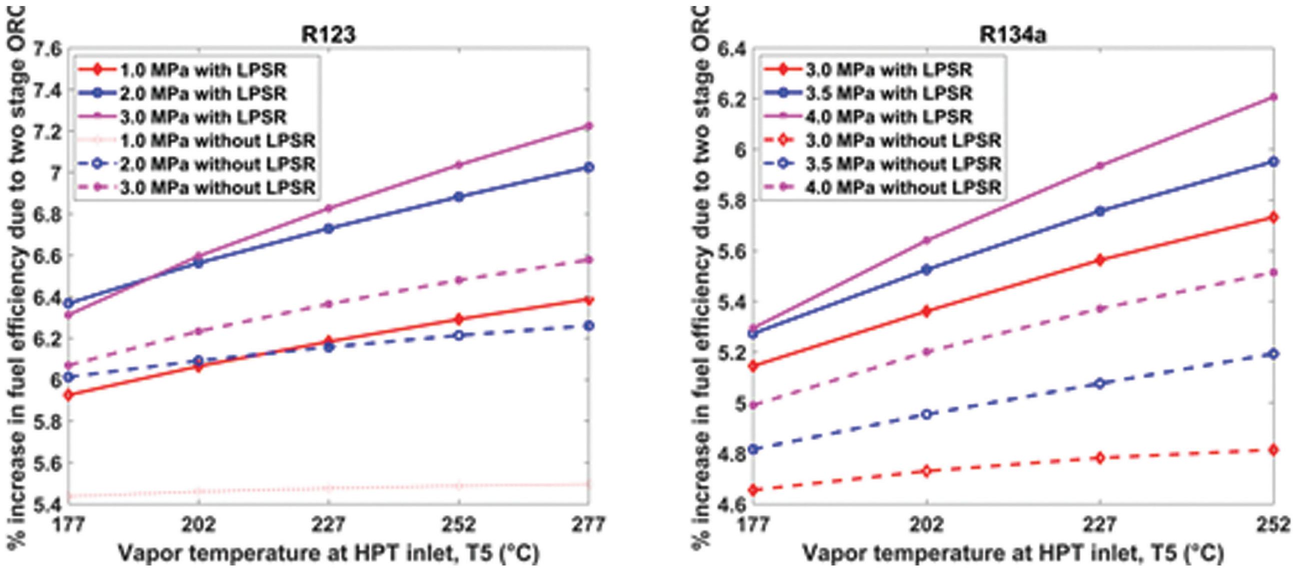

Figure 8. The improvement in fuel efficiency of IC engine due to a two-stage ORC at 1.4 MPa BMEP.

efficiency of 5.8% was claimed by Guohui Zhu et al. [19] with R123 with similar ORC design parameters, which is 1.4% lower than that estimated in the present research work.

Parametric optimization was made based on the minimum exergy destruction. Figure 9 shows the variation in exergy destruction with vapor pressure and temperature in the HPHE. With an increase in temperature of vapor, the rate of exergy destruction decreases in both the cases of R123 and R134a. With an increase in vapor pressure, the rate of exergy destruction decreases considerably in the case of R123 up to 2.5 MPa, and then onwards it starts increasing. But, the vapor pressure in the HPHE has a little effect on the rate of exergy destruction in the case of R134a. From the exergy analysis, it is observed from the Figure 9 that the optimum pressure in HPHE with R123 is 2.5 MPa while it is 3.5 MPa with R134a.

Figure 10 shows the comparison between two different dry organic fluids, viz. R123 and R245fa, in terms of exergy destruction. At lower vapor pressure in the HPHE, R245fa has considerably higher exergy destruction when compared to that of R123. With an increase in vapor pressure, the rate

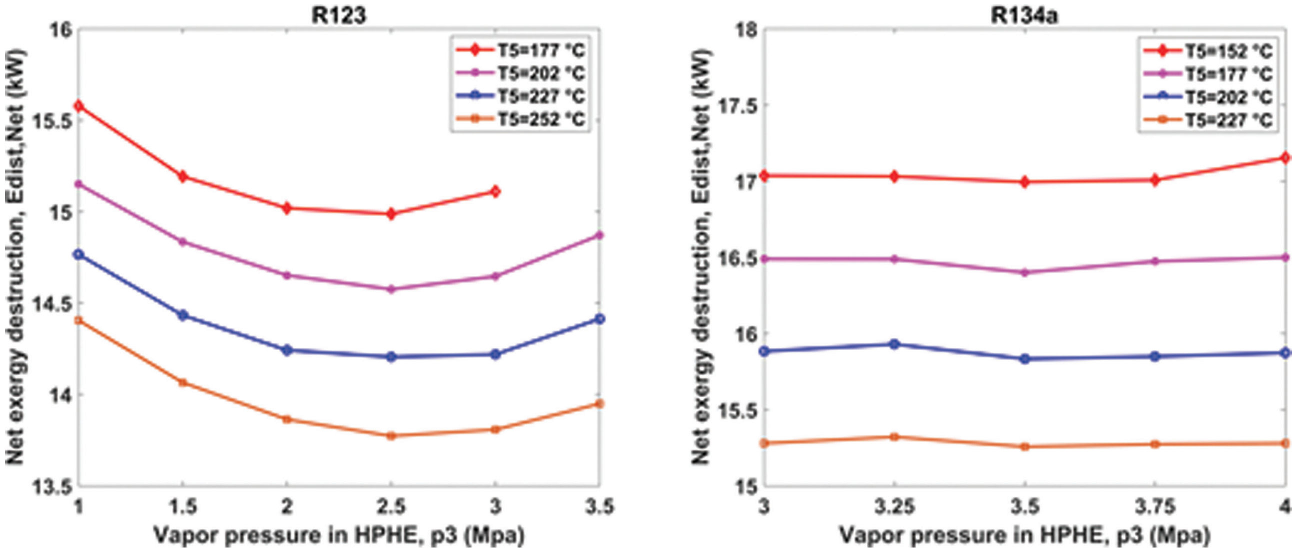

Figure 9. Net exergy destruction as a function of vapor pressure and temperature at 1.4 MPa BMEP.

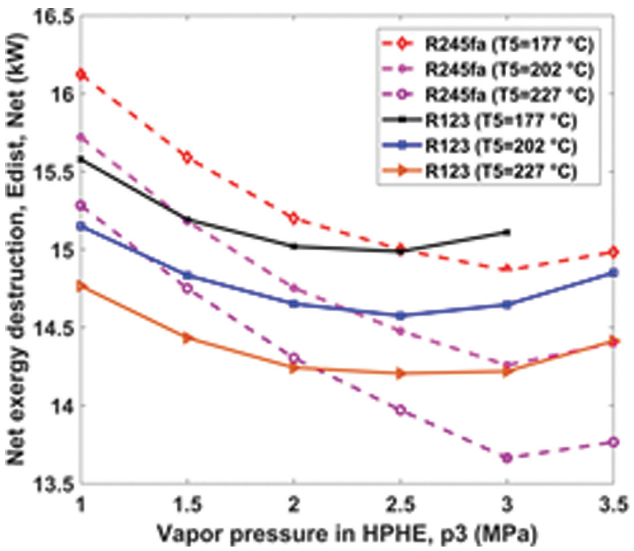

Figure 10. Net exergy destruction as a function of vapor pressure and temperature with R245fa and R123 at 1.4 MPa BMEP.

of exergy destruction decreases more sharply with R245fa compared to R123. At around 2.25 MPa pressure, both the fluids have nearly the same exergy destruction and hence, R245fa can be a better substitute to R123 as R123 does not meet the environmental concerns. The optimum operating pressure in the HPHE is 3.0 MPa with R245fa while it is 2.5 MPa with R123.

CONCLUSION

The two-stage ORC for recovering the engine waste heat, both the coolant energy and exhaust gas energy, besides the superheated vapor energy of exhaust from the ORC turbine has been analyzed. The influence of ORC operating pressures and temperatures on the cycle thermal efficiency and the percentage of increase in the fuel efficiency of the engine when it is coupled with the two-stage ORC with and without LPSR have been investigated by conducting energy analysis. An exergy analysis of two-stage ORC has also been performed to find the net exergy destruction and in turn to determine the optimum operating pressures in the HPHE with R123, R134a and R245fa. The major research findings are listed below.

- It is observed that the LPSR has a considerable influence on the thermal and WHR efficiencies of the two-stage ORC and is found to be effective in improving the IC engine fuel efficiency with both dry and isentropic working fluids.
- The thermal efficiency of the two-stage ORC varies from 10.3% to 14.8% with LPSR and from 9.3% to 12.9% without LPSR depending on the type of working fluid and the pressure and temperature at the inlet to HPT.
- The WHR efficiency of the two-stage ORC varies from 10.0% to 13.75% with LPSR and from 8.85% to 12.5% without LPSR and the corresponding increase in fuel efficiency of an IC engine varies from 5.15% to 7.22% with LPSR and from 4.65% to 6.58% without LPSR depending on the type of working fluid and the pressure and temperature at the inlet to HPT.
- It is also concluded that dry fluids, like R123 and R245fa, are better choice over isentropic fluids like R134a for the use in two-stage ORC to recover the engine waste heat effectively and improve the fuel efficiency of IC engine substantially.

The heat losses and pressure drop in various components of ORC plant are not included in the present analysis.

Hence, the correlations developed in this paper would rather give slightly higher values of WHR efficiency.

NOMENCLATURE

BP	brake power, kW
c_{pa}	constant pressure specific heat of air, kJ/kg-K
c_{pg}	constant pressure specific heat of gases, kJ/kg-K
c_{pw}	specific heat of water, kJ/kg-K
h	specific enthalpy, kJ/kg
HP	high pressure
HPHE	HP stage heat exchanger
HPP	HP stage pump
HPT	HP stage turbine
LP	low pressure
LPHE	LP stage heat exchanger
LPP	LP stage pump
LPSR	LP stage regeneration
LPT	LP stage turbine
m	mass flow rate, kg/s
ORC	organic Rankine cycle
p	pressure, MPa
δQ	rate of heat transfer, kW
T	temperature, K
v	specific volume, m^3/kg
WHR	waste heat recovery
η	efficiency
∈	effectiveness

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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