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FARM WASTE UTILISATION FOR POWER PRODUCTION THROUGH A NOVEL BIOGAS OPERATED MICRO GAS TURBINE INTEGRATED WITH HEAT RECOVERY SYSTEM

Rupinder Pal Singh* and R S Gill

Department of Mechanical Engineering, Punjab Agricultural University, Ludhiana-141004, Punjab

Punjab is a leading state in the agricultural sector and its economy primarily depends upon it. The share of state in all India production for rice and wheat is 10% and 16.71%, respectively (Pocket Book of Agricultural Statistics, 2017). The other major crops grown are barley, maize, sugarcane, bajra, jowar, etc. along with other fruits and vegetables. The vast production of these crops, fruits and vegetables generates problem of waste utilization in the farm itself. There must be an effective mechanism to utilize the agricultural waste product known as biomass in a sustainable manner. There has been an increasing interest in the biochemical conversion of biomass to improve the combustion characteristics. The microorganism are used to break down the molecules of biomass with/without catalyst in the absence of oxygen (Wang *et al.*, 2019). Recently, there has been a growing interest in the biogas plants for cooking and power production by using biogas produced by anaerobic digestion of agricultural organic waste derived from different sources like crops, animal, wood, food industry, aquatic plants, etc (Yagli *et al.*, 2016). The biogas plants use biochemical conversion process to produce methane gas from biomass. Conventionally, the produced biogas was only used for cooking and heating applications. Nowadays, the produced biogas (methane enriched) is fed into the internal combustion engines and micro gas turbines for cogeneration applications. The 4-stroke spark-ignition engine and dual fuel engines are frequently used to burn the biogas for power production. The electric efficiency of these engines varies between 34%-40% with engine capacity in the range of 0.1 MW_{el} and 1 MW_{el}. The part of thermal energy is diverted into the digester for conducive anaerobic digestion conditions by maintaining an isothermal temperature conditions (40-50℃) (Benato and Macor, 2017). The large amount of heat is expelled into the atmosphere from the biogas driven IC engines which are responsible for environmental pollution and lower efficiencies. About 20-50% of the energy consumption is discharged into the atmosphere of waste heat (Karellas *et al.*, 2013). Bendig *et al.* (2013) gave a formal definition of waste heat with the explanation of avoidable and unavoidable waste heat. The strict environmental legislations to conserve our environment and high energy costs call for the energy efficient sustainable technologies.

Tapping of the unused heat is of utmost importance in the context of decreasing margins in the sale of electric power. The rankine cycle can be integrated to extract this waste heat for power production. However, the working fluid is water in the rankine cycle which requires a higher heat source temperature (500℃). Fluids with lower boiling temperature (organic fluids) are more suitable to recover the low temperature waste heat through organic rankine cycle (ORC). The working fluid in ORC evaporates and is expanded in the turbine in the same way as the conventional steam power cycle (Hung *et al.*, 1997). ORC has been widely used to recover low grade waste heat from gas turbines, industrial processes and engines (Kaska, 2014; Wang *et al.*, 2012; Hoang, 2018). Uusitalo *et al.* (2018) evaluated a number of organic fluids and their properties on the system efficiency. The study includes different working fluid groups as hydrocarbons, siloxanes and fluorocarbons. It was concluded that the hydrocarbons and siloxanes with high critical temperature are most suitable candidates for higher heat source temperature. On the other hand, fluorocarbons are best suited for low grade heat recovery. Bacciolo *et al.* (2019) proposed a heat recovery system for a biogas plant by integrating an ORC. The produced biogas is used to power a small gas turbine with 600 kWe capacity and exhaust gases (280 ℃) are expelled into the atmosphere. It is found that ORC system generates about 8.6% of the electricity produced by the small gas turbine. The system efficiency is enhanced by 2.3% with the ORC integration in the existing biogas plant.

Based on the scientific literature, very few studies have been conducted on the farm waste utilization using biogas plants especially from the exergetic analysis perspective. In the present work, an improved plant configuration using ORC is implemented with an existing biogas unit. The effect of varying heat source temperature on the energy and exergy efficiency is further investigated. The proposed analysis would contribute in improving the efficiency of the existing

*Corresponding author : rupinderpal@pau.edu
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biogas plant by effectively recovering the waste heat.

Biogas plant

The actual plant data of the biogas unit which is installed in Italy (Baccioli *et al.*, 2019) is considered in the present analysis. It consists of receiver tank, two digester, gas holders and a biogas operated micro gas turbine. The biogas production rate is 276.6 kg/h (65% methane concentration) from the organic waste of 10.8 tons per hour. The produced gas is burnt inside the gas turbine of 600 kWe capacity (3 modules each with a capacity of 200 kWe). The part of waste heat is used to heat the sludge using a water loop, before putting into the anaerobic digester. The remaining heat at 280 ℃ with a flow rate of 4 kg/s is directly expelled into the atmosphere. The digester is maintained at a constant temperature (37 ℃). There is an urgent requirement to extract the waste heat through an effective heat recovery system. The recovered heat from the biogas engine is best suited for power production in comparison with other uses like hot water production, etc. (Poschl *et al.*, 2010). In the present study, an ORC system using benzene as a working fluid is proposed to tap the rejected heat. The exhaust flue gases vaporize the working fluid which is further expanded in the turbine for power production. The ORC turbine generator has a capacity of 50 kW.

ORC thermodynamic cycle

The ORC system works on both Trans and supercritical zone. However, the working fluid should be selected in such a manner that it will not get decomposed. Moreover, the system piping and components must be robust enough to sustain the higher pressure & temperature that arise due to the working in trans/super critical zone. In this study, subcritical ORC with regenerator integrated with a biogas fired turbine is considered and schematics diagram of the thermodynamic cycle is plotted in the Fig. 1. The system consists of an evaporator, condenser, liquid pump, turbine/ expander and a regenerative heat exchanger. The working fluid is heated up in the evaporator by the exhaust gases coming out of the gas turbine. The high pressure heated vapors (at point 2) are expanded in a radial turbine which is connected with a generator. The fluid vapors are passed through a regenerator heat exchanger to heat the incoming liquid from the pump. The vapors enter at point 4 inside the condenser and coming out as liquid as shown by the point 5. The liquid is then pumped into the evaporator via a regenerator. The produced biogas enriched with methane is burnt in the micro gas turbine and the exhaust gases are used to heat the working fluid. The exhaust gases enter the evaporator at a temperature of 280 ℃ (T_{in}) and expelled into the atmosphere (T_{out}).

Thermal model

Working fluid selection

There are different working fluids for recovering waste heat using ORC. However, working fluids play a significant role in the thermal energy investigations. The ideal fluids must have higher decomposition temperature and desirable environmental properties. The safety, low freezing temperature, higher heat of vaporization, low cost, availability are some other aspects of a best organic fluid (Braimakis *et al.*, 2018; Song and Gu, 2015; Dai *et al.*, 2016). Based on these parameters, benzene possesses greater thermal performance for high temperature waste heat recovery (Kumar and Shukla, 2016; Shu *et al.*, 2014). Benzene is a dry working fluid with critical temperature of 289 °C and critical pressure is 49 bar. It is a colourless compound and has an aromatic odour with melting temperature of 5.5 °C. Saha *et al.* (2019) concluded that benzene and toluene possess the highest efficiency among the fifty two selected working fluids. The other properties of the working fluid are given in Table 1. (Ge *et al.*, 2019). The proposed subcritical ORC has been analyzed with these three working fluids to evaluate the energy efficiency.

Assumptions

1. The steady state conditions prevail among all the system components.
2. The pinch point temperature difference in the evaporator and condenser is 30 K & 10 K, respectively.
3. The dead state temperature and pressure are 25℃ & 1 bar, respectively.
4. The isentropic efficiencies of both pump and turbine are 85% & 80% respectively. The effectiveness of the regenerator is 0.8.
5. The pressure drop, potential and kinetic energy, frictional losses are neglected.
6. The cycle is operated in the subcritical zone and the evaporator pressure is nearer to the critical point.

Energy analysis

The energy analysis is used to evaluate the thermal efficiency of the system, based on the first law of thermodynamics. Mathematically, it is the ratio of new work done to the total heat supplied in a thermodynamic cycle.

$$\eta_{energy} = \frac{W_{net}}{Q_e} = \frac{W_t - W_p}{Q_e} \tag{1}$$

where, W_{net}, W_t & W_p represent net work done, turbine work and pump work done ,respectively. Q_e is the heat supplied to the working fluid in the evaporator.

The turbine efficiency is given by the ratio of actual work done to the theoretical work produced and given by:

$$\eta_{turbine} = \frac{(h_2 - h_3)}{h_2 - h_{3s}} \tag{2}$$

Similarly, pump efficiency, $\eta_{pump} = \frac{(h_{6s} - h_5)}{h_6 - h_5}$ (3)

The heat exchanger effectiveness;

$$\varepsilon = \frac{Q_{actual}}{Q_{ü}} = \frac{(h_3 - h_4)\, or\, (h_1 - h_6)}{(h_3 - h_{4,\min})} \tag{4}$$

In the regenerator, energy balance gives:

$$h_3 - h_4 = h_1 - h_6 \tag{5}$$

Mathematically, the turbine work, $W_t = m_f.(h_2 - h_3)$ (6)

Pump work, $W_p = m_f.(h_6 - h_5)$ (7)

Heat supplied to the evaporator, $Q_e = m_f.(h_2 - h_6)$ (8)

Exergy based on the entropy analysis

The energy analysis based on the first law of thermodynamics is not sufficient enough to evaluate the quality of the work potential and the associated irreversibilities in the system components. Hence, the exergy analysis given by the 2nd law of thermodynamics is needed (Koca *et al.*, 2008; Zhang *et al.*, 2015). The exergy analysis quantifies the available energy among different system components and their associated irreversibilities. It identifies the components with highest irreversibilities and provides a scope for improving their work potential.

Irreversibility is given as,

$$I = T_o s_{gen} \tag{9}$$

where, T_o and s_{gen} are the dead state temperature & entropy generation, respectively.

$$s_{gen} = \Delta s_{system} + \Delta s_{surr} \geq 0 \tag{10}$$

$$\Delta s_{system} = m_f.(s_o - s_i) \tag{11}$$

$$\Delta s_{surr} = \sum \frac{\dot{Q}}{T} \tag{12}$$

Using the above equations, irreversibilities of individual components are calculated as:

$$I_{cond} = m_f.T_o \left\{ (s_4 - s_5) - \left(\frac{Q}{T_{\sin k}} \right) \right\} \tag{13}$$

$$I_{evap} = m_f.T_o \left\{ (s_2 - s_6) - \left(\frac{Q}{T_s} \right) \right\} \tag{14}$$

$$I_{recup} = m_f.T_o \left\{ (s_1 - s_6) - (s_3 - s_4) \right\} \tag{15}$$

$$I_{turbine} = m_f.T_o (s_3 - s_2) \tag{16}$$

$$I_{pump} = m_f.T_o (s_6 - s_5) \tag{17}$$

$$\sum I_{total} = I_{cond} + I_{evap} + I_{turbine} + I_{recup} + I_{pump} \tag{18}$$

$$\eta_{exergy} = \frac{E_{in} - \sum I_{total}}{E_{in}} \tag{19}$$

The ORC is integrated for power production from the rejected heat of the biogas fired micro gas turbine. The evaporator pressure is lower than the critical pressure of the working fluid in the proposed ORC and the thermodynamic cycle is working in the subcritical zone. The pressure at the turbine inlet varies between 2 bar to 25 bar and the heat source temperature changes in the range of 250 ℃ to 400 ℃. The maximum temperature of the working fluid is restricted below the physical/chemical decomposition temperature.

Evaporator pressure and temperature

The effect of evaporator pressure on the energy and exergy efficiency of the ORC is shown in Fig. 2. The energy efficiency rises abruptly up to a pressure of 9 bar. However, this increase in energy efficiency is not substantial at higher evaporator pressures. This is due to the fact that total heat supplied to the ORC turbine also increases with the inlet pressure of turbine. The maximum thermal efficiency of 32.7% is obtained at highest heat source temperature of 400 ℃. The lower temperature of the exhaust gas (250 ℃) produces an energy efficiency of 19% due to the reduction of turbine work. It is concluded that the efficiency based on the first law of thermodynamics increases with heat source temperature and evaporator pressure. However, the other factors like decomposition temperature and cost parameters must be taken into consideration while proposing a limit to the turbine inlet pressure & temperature for maximizing the efficiency. The exergy efficiency is plotted against evaporating pressure for different heat source temperatures as shown in Fig.3. The irreversibilities reduce with the rise in temperature of exhaust gases which improves the exergy efficiency of the proposed ORC. The maximum exergy efficiency of the cycle is obtained as 64% at 28 bar and 400 ℃. The exergy efficiency increases rapidly with lower heat source temperature up to a pressure of 6 bar. The curve flattens out for higher evaporator pressure

Table 1. ORC working fluid properties

Working fluid	Chemical formula	Critical temperature (°C)	Critical pressure (bar)	Boiling temperature (°C)	ODP	GWP
Benzene	C6H6	289.1	48.94	80.2	0	Low

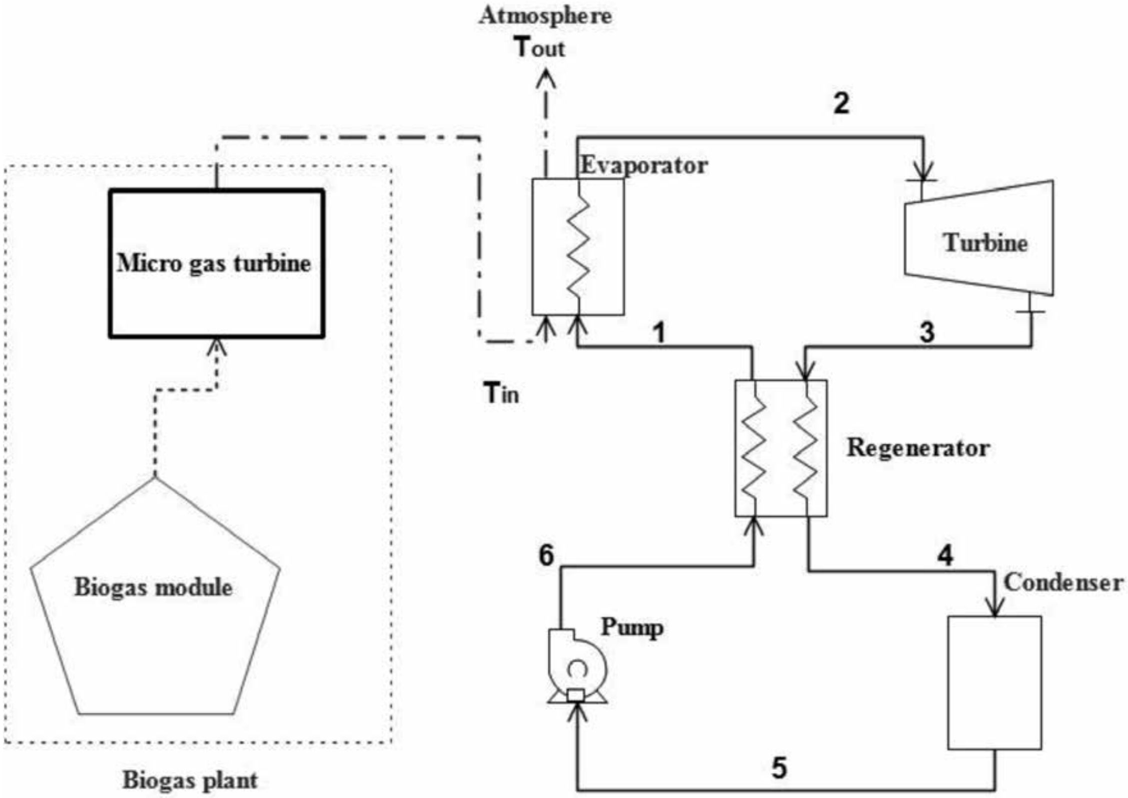

Fig. 1. Schematic layout of ORC integrated with biogas plant

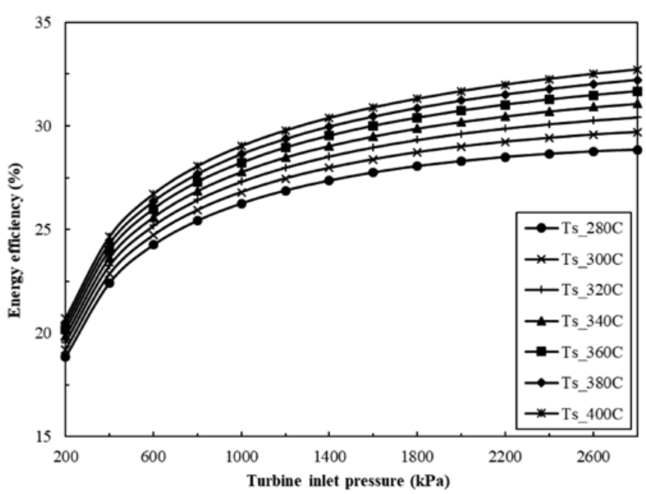

Fig. 2. Effect of turbine inlet pressure at different heat source temperatures on the energy efficiency

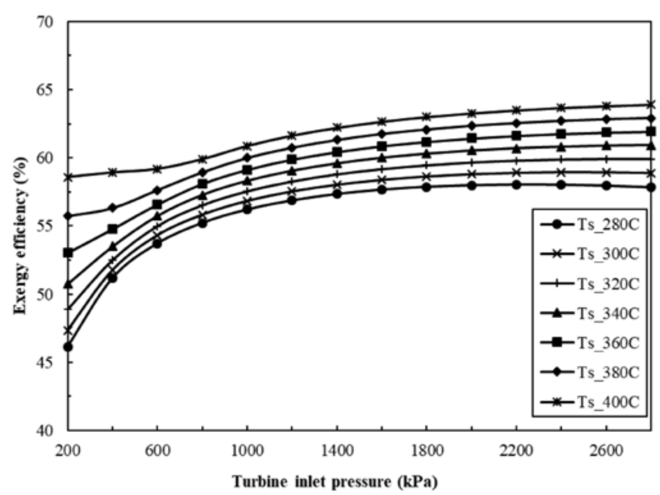

Fig. 3. Effect of turbine inlet pressure at different heat source temperatures on the exergy efficiency

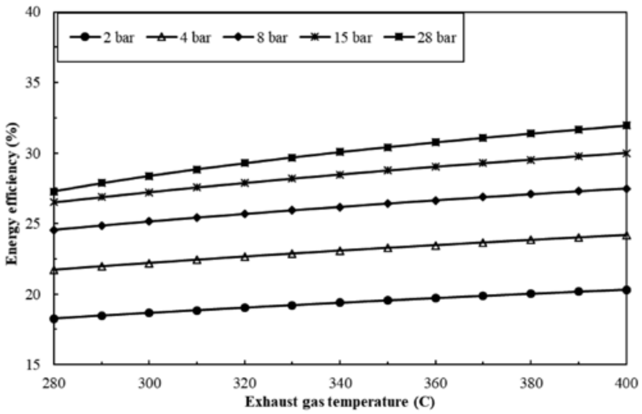

Fig. 4. Effect of exhaust gas temperature at different evaporating pressure levels on the energy efficiency

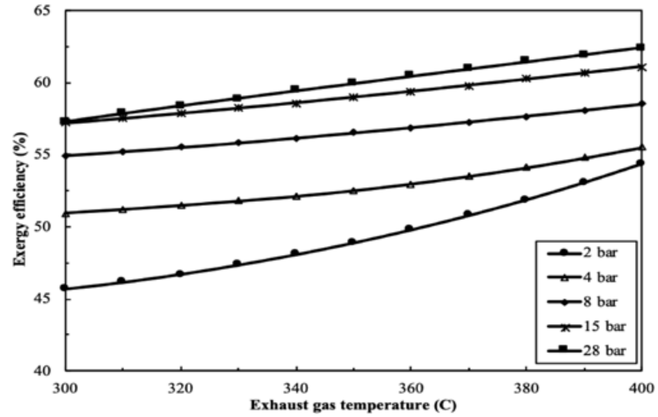

Fig. 5. Effect of exhaust gas temperature at different evaporating pressure levels on the energy efficiency

for all the heat sources. The network produced by the turbine reduces with lowering the inlet pressure and temperature of turbine. Hence, the lowest efficiency of the ORC is calculated as 46% at a pressure of 2 bar and temperature (280 ℃).

The effect of varying the heat source temperature on energy efficiency at different pressure is shown in Fig. 4. The rise in the exhaust gas temperature increases the net work produced by the cycle which improves the thermal efficiency. At 2 bar evaporating pressure of Benzene, thermal efficiency is 18% & 20% respectively at 280 ℃ and 400 ℃. The pressure increment from 2 bar to 28 bar rises the thermal efficiency by 50% at a constant heat source temperature of 280 ℃. The reason for this thermal efficiency enhancement is due to the large pressure drop which produces high turbine output. The large pressure drop also increases the pump work. However, the increased pump work is very nominal in comparison with the produced turbine output. Hence, overall efficiency of the ORC increases with rise in evaporator pressure. The maximum efficiency of 32% is obtained at a pressure of 28 bar and 400 ℃. The exergy efficiency variations with exhaust gas temperature are shown in Fig. 5. The highest exergy efficiency of 62% is obtained at the maximum pressure (28 bar) and temperature (400 ℃). The increase in the evaporator pressure rises the exergy efficiency at a particular temperature of the exhaust gas. However, the local increment in the exergy efficiency reduces at higher evaporator pressures. The total irreversibilities in the system components diminish with evaporator pressure which cause an increase in the cycle efficiency.

Exergy destruction

The exergy destruction in a thermodynamic system is equivalent to irreversibility which is defined as the difference between the reversible and useful work. In other words, irreversibility provides an accurate representation of the wasted work potential in a particular component and gives a scope for improving the overall exergy efficiency. In an actual system, irreversibilities are responsible for lowering the cycle efficiencies and

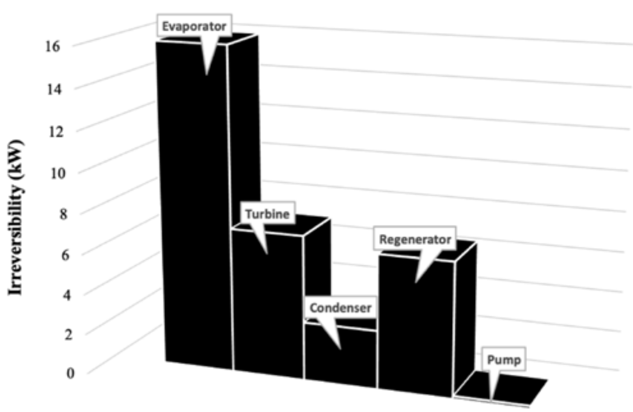

Fig. 6. Irreversibilities inside the various components of ORC

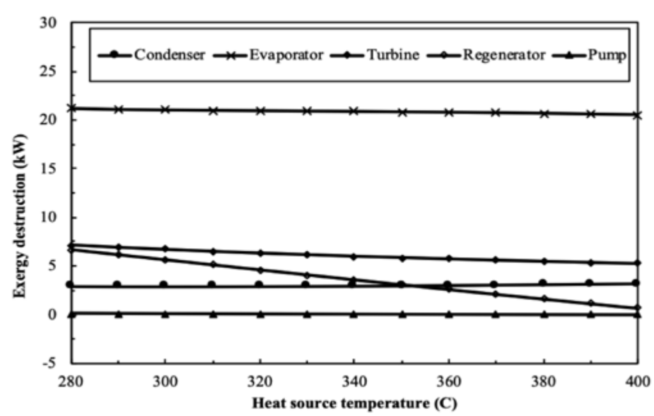

Fig. 7. Effect of heat source temperature on the exergy destruction in each component.

mainly occurs during the turbine expansion or in the heat exchanging process (evaporator/condensers). The exergy destruction needs to be evaluated in each components of the ORC as shown in Fig. 6. The highest exergy destructed is inside the evaporator followed by turbine, condenser and regenerator. Pump shows minimum irreversibility of less than 1 kW at a constant heat source temperature. The irreversibility associated with evaporator and condenser are 21.2 kW & 6.7 kW, respectively. The exergy destruction in the evaporator is predominantly due to the finite temperature difference between the system and surroundings. The consequence of the higher irreversibilities in the evaporator is the loss of exergy efficiency which further increases the overall cost component. Hence, it is concluded that heat source inlet temperature of the evaporator has to be precisely analysed for maximizing the efficiency of ORC systems. The individual exergy destruction at different temperature of the heat source is shown in Fig. 7. The exergy destruction decreases slightly for evaporator, pump and turbine with the rise in temperature. However, condenser shows a minimal increase in the rate of exergy destruction due to the increased temperature differentials of the cooling air. The irreversibilities associated with Regenerator go away with the increase in exhaust gas temperature.

To conclude, the present work proposes a novel farm waste utilization system using regenerative Organic rankine cycle (ORC) integrated with a biogas plant. The cycle evaporator pressure is varied between 2 bar to 28 bar and heat source temperature from 250 to 400 ℃. The maximum thermal efficiency of 32.7% is obtained at highest heat source temperature of 400 ℃. The lower temperature of the exhaust gas (250 ℃) produces an energy efficiency of 19% due to the reduction of turbine work. It is found that the maximum exergy is destructed inside the evaporator followed by turbine, condenser and regenerator. This observation provides a scope of reducing irreversibilities inside the evaporator. The highest exergy efficiency of 62% is obtained at the maximum pressure (28 bar) and temperature (400 ℃). It is concluded that the increase in the evaporator pressure raises the exergy efficiency. However, the pressure increment from 2 bar to 28 bar raises the thermal efficiency by 50% at a constant heat source temperature of 280 ℃. The study is further extended to include a detailed case study of a farm for year round production of electricity.

Authors' contribution

Conceptualization and designing of the research work (RPS,RSG); Execution of field/lab experiments and data collection (RPS, RSG); Analysis of data and interpretation (RPS,RSG); Preparation of manuscript (RPS,RSG).

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