SEMARAK ILMU PUBLISHING

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage: https://semarakilmu.com.my/journals/index.php/fluid_mechanics_thermal_sciences/index
ISSN: 2289-7879

Development of a Small-Scale Electricity Generation Plant Integrated on Biomass Carbonization: Thermodynamic and Thermal Operating Parameters Study

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ARTICLE INFO

Article history:
Received 10 November 2021
Received in revised form 5 March 2022
Accepted 7 March 2022
Available online 3 April 2022

Keywords:
Biomass; combustion; pyrolysis; stove; hot flue gas; ORC

ABSTRACT

Biomass has been known as a source of energy with a thermochemical process that produces heat and can be converted further into electricity. However, thermal energy losses are a huge problem during combustion. To overcome this problem, a system based on organic Rankine cycle (ORCs) was developed to recover and utilize them to generate electricity. The proposed ORCs include an evaporator, a turbine, a condenser, and a pump coupled with a biomass carbonizing system to create a promising technology for small-scale electricity generation. In this work, a thermodynamic modelling equation based on energy and exergy balances was briefly expressed for each subcomponent of the system. A case study with R134a as the working fluid is being investigated to validate the system's performance. In addition to the effects of R134a on temperature at the turbine exit, the suitable operating pressures has been specially adopted from several valid journals that focused on the effects of a wide range of possible operating pressure on the working fluid characteristics, which have a significant effect on the system performance. Finally, the theoretical analysis shows that the turbine work is profitable at an inlet pressure of 5 bar and an outlet pressure of 2 bar. This system is recommended to be integrated into the thermochemical biomass process. Recommendations have been made for the future development of small-scale biomass-fuelled power generation systems. This study shows that the thermal losses of the biomass thermochemical processes can be theoretically recovered in the form of electricity by using ORC efficiently.

1. Introduction

Until now, the utilization of biomass in the world as an energy source has increased significantly. Several researchers have also stated that biomass energy utilization can reduce greenhouse effects, gas emissions, and climate change [1-3]. Turning biomass into heat using

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https://doi.org/10.37934/arfmts.94.1.7995

various types of furnaces, such as stoves, usually happens at a high temperature of about 800–1000 °C. However, the performance of this kind of stove is still under investigation by researchers in order to increase its performance and reduce energy losses. In fact, high-temperature combustion is associated with huge energy losses, usually from the wall of the stove or even from outflowing hot flue gases. The exhaust flue gas with a temperature above 200 °C is usually sacrificed freely into the ambient, which results in both energy losses and extreme air pollution. According to the related regulations, the exhaust flue gas can be released to the atmosphere at a temperature of up to 60 °C. In conclusion, not all the considerable heat can be converted into another energy form.

In general, there are two ways of converting biomass into other forms of energy; the biochemical process [4], and the thermochemical process [5]. In this study, thermochemical processes are discussed with the conversion routes as pictured in Figure 1. Combustion is a very attractive method of producing high thermal energy or heat, which includes including heat in the exhaust gas streams. This thermal energy can be stored and utilized further for drying or heating purposes. A simple way of storing thermal energy has been theoretically studied by using a water tank with a thermocline model [6]. The thermal energy can also be utilized to produce electricity, which can be integrated into the grid or even be used privately.

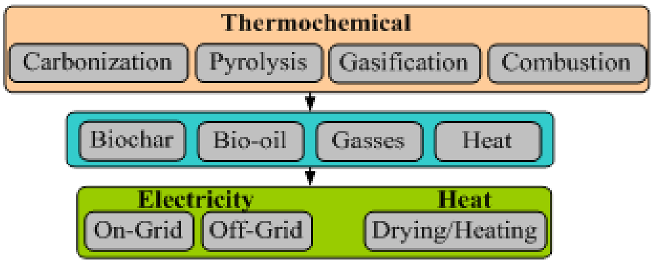

Fig. 1. State of the art of thermochemical way on biomass conversion process

Another energy source is in the form of flammable gas, which can also be generated by the gasification process. Gasifiers with high efficiency, however, are still in development. A novel fluidized gasifier with concentric cylinders has been hydrodynamically simulated and demonstrated the ability to produce high-calorific flammable gas [7]. This new gasifier can produce flammable gas with a heating value of about 7 MJ/m^3 higher than the conventional gasifier [8]. This flammable gas is very appealing as a replacement for fossil fuel in fuelled engines. Syngas of about 3 MJ/m^3 of heating value can easily be produced by means of a downdraft gasifier, which has the potential to replace the old fuel [9].

The thermochemical process is the most preferred method because it has a few advantages compared to the biochemical process. The equipment used is also not too complicated. Using a modified stove, for instance, can achieve an efficiency of up to 35% [10]. This has been confirmed by performing an experimental study with an internal air distributor placed in the centre of a modified stove. They obtained that the stove efficiency is about 25% [11] higher than the conventional stove. However, this is still a very low conversion efficiency associated with the energy contained in the biomass. The stove could not stand alone to convert all the stored energy in the biomass completely. Other supporting systems and equipment are essential in order to achieve maximum conversion efficiency [12]

A system with a combination of several equipment working together to generate electricity and reuse rejected heat is called a combined heat and power system (CHP). By considering heat waste, an internal combustion engine, for instance, converts only 25–38% of the fuel's thermal energy into shaft work. Some of it is rejected through the cooling system, the engine wall, and the exhaust gases as dispersed energy [13]. While used as the prime mover of a generator, internal combustion engines of small size can achieve electrical efficiencies of around 20 to 30% [14].

Until now, investigations into the reuse of wasted energy, including waste heat from engines, using thermodynamic studies have been widely used in research and development related to renewable energy [15,16,17,18,19,20,21,22]. As shown in Table 1, the Rankine cycle with water as the working fluid has been used widely to convert mostly energy from fossil fuels into electricity. However, the Rankine cycle is available only for high contained energy sources such as from fuel combustion. For low to medium heat sources, for instance, waste energy contained in the exhaust flue gases of industry, the Rankine cycle is no longer effective and efficient.

Table 1
Large-Scale Electric Power Generation through 2050 from Renewable and Non-renewable [23]

Power plant type	Non-renewable source	Renewable source	Thermodynamic cycle
Coal-fuelled	Yes		Rankine
Natural gas-fuelled	Yes		Brayton
Nuclear fuelled	Yes		Rankine
Oil-fuelled	Yes		Rankine
Biomass-fuelled		Yes	Rankine
Geothermal		Yes	Rankine
Solar-concentrating		Yes	Rankine
Hydroelectric		Yes	None
Wind		Yes	None
Solar-photovoltaic		Yes	None
Fuel cells	Yes		None
Currents, tides, and waves		Yes	None

For low grade or even moderate temperature energy sources (less than 300 ^oC), an organic Rankine cycle (ORC) is the most potential technology to recover waste energy [24,25,26]. ORC has been received as an attractive way to absorb the rejected heat in the range of low, medium, and high energy content and be converted into electrical energy. ORC employs organic substances as working fluids instead of water, including R123, R245fa, and R134a. The organic working fluid is typically selected to meet the requirements of the application. For instance, the relatively low boiling point of these substances allows the ORC to produce power from low temperature sources, including industrial waste heat, geothermal hot water, and fluids heated by concentrating solar collectors.

The number of published articles on waste energy harvesting based on ORC is rapidly increasing. Hung *et al.,* [27], Maizza *et al.,* [28,29], Liu *et al.,* [30], Hung [31], Saleh *et al.,* [32], Li *et al.,* [33], Li *et al.,* [34], Uusitalo *et al.,* [35] proposed and analysed various ORCs for waste heat recovery systems. Meanwhile, He *et al.,* [36], Shu *et al.,* [37], Wang *et al.,* [38], Wu *et al.,* [39], Islam *et al.,* [40], Saadon *et al.,* [41] showed their interest in the waste heat recovery from internal combustion engine or gasoline engine. Pantaleo *et al.,* [42] uses ORC integrated with steam and gas turbine. Tchanche *et al.,* [43], Hung *et al.,* [44], Trifon *et al.,* [45], Petrollese *et al.,* [46] shown their interest in the application of ORCs in solar organic Rankine cycle systems. Vittorini, *et al.,* [47], Dijoux, *et al.,* [48] also considering the system efficiency of ORC using ocean thermal energy conversion (OTEC) system as heat source.

Numerous ORC units have been installed in industries such as cement, steel, glass, and oil-gas industries to recover wasted energy from their machines. However, research concerning waste energy harvesting from thermochemical biomass processing, which is focused on the smoke condensation of wood carbonizing manufacture, is still rare. Rahbar *et al.,* [21] described a feasibility study of power generation through waste heat recovery from wood-burning stoves using the ORC technology. Velez *et al.,* [49] concerned a technical, economic, and market review of ORC for the conversion of low-grade heat for power generation. Carraro *et al.,* [50] performed cogeneration based on biomass and ORC systems and successfully obtained 2.2 kW of electricity. Cao *et al.,* [51] used biomass to fuel an externally fuelled gas turbine. Algieri *et al.,* [52] demonstrate a cycle for micro–scale CHP using subcritical and transcritical biomass–fired ORC systems. Mascuch *et al.,* [53] investigated woodchip combustion fuelled ORC for use as a biomass boiler replacement. Meanwhile, Germany is the biggest country in the world by producing electricity at a rate of about 40.1 MW based on biomass-powered ORC [54]. Another potential application has been identified as exhaust gas recovery from gas turbines [55,56,57]. In addition, ORC exhibits unique advantages such as small size, low capital and maintenance cost, simplicity, and high reliability when combined with renewables such as biomass, solar energy, or geothermal heat.

The ORC can also be integrated with charcoal–based biomass and condensate smoke manufacture where the torrefaction process is employed. In this process, the thermal decomposition of wood is carried out under controlled conditions, usually without or with very limited oxygen. With relatively long residence times at temperatures (200–300^{o}C), up to 35% of char can be obtained [58]. Other products are liquids and gases. During the process, lots of smoke is generated due to the incomplete combustion in the reactor. The smoke can be condensed to a liquid state instead of being reused for drying. A heat exchanger of the shell and tube type can be employed with water as the cooling medium instead of air. During the smoke condensing process, the cooling water adsorbs heat, causing its temperature to increase. The hot water is then used in the ORC to evaporate the organic working fluid. In some industries, the traditional condensation processes may still be in use where the condensing process using a simple batch and ambient air as the cooling medium causes the condensate produced to have low quality and quantity.

The thermodynamic performance of a novel small–scale electricity generation system based on heat harvested from smoke condensation and integrated with the ORC system is investigated in this paper. The proposed system provides several advantages, such as fuel flexibility, saving fuel during the whole process, and increasing the enterprise revenue. The aim of this paper is to study the appropriate operational parameters of the ORCs system proposed to better understand the factors that lead to variability and uncertainty in this system due to the heat availability from the condensation system. Discussions have been focused on the effect of the exhaust temperature, condenser pressure, working fluid mass, and flow rate in the whole system on the cycle performance. The parameters used are based on numerous previous researchers and confirmed by Quoilin *et al.,* [59], Morrone *et al.,* [60], and Ambarita *et al.,* [61], and Wang *et al.,* [62].

2. System Description

The system developed in this study is illustrated in Figure 2. The system is a kind of simple one with the basic components consisting of (1) the working fluid pump, in which the working fluid is circulated and adjusted through the control valve (CV) and measured by means of a rotameter (R), (2) the working fluid evaporator, where the working fluid of organic type is boiled, (3) the steam turbine, which expands the working fluid in vapour form to gain useful shaft power, and (4) the working fluid condenser, where the working fluid is cooled and changed into liquid form. The features of this system are small size, no emissions, and environmentally friendly. The organic fluid evaporator receives heat from the hot water which is gained during hot smoke condensation. The hot water temperature is approximately (T_1 = 95 °C) entering the evaporator where the selected working fluid in the liquid state is evaporated. After that, the water leaves the evaporator at approximately T_2 = 35 °C, as designed, and is pumped back into the water tank. A throttle valve was adopted in the bypass line to protect the turbine during the starting and closing processes.

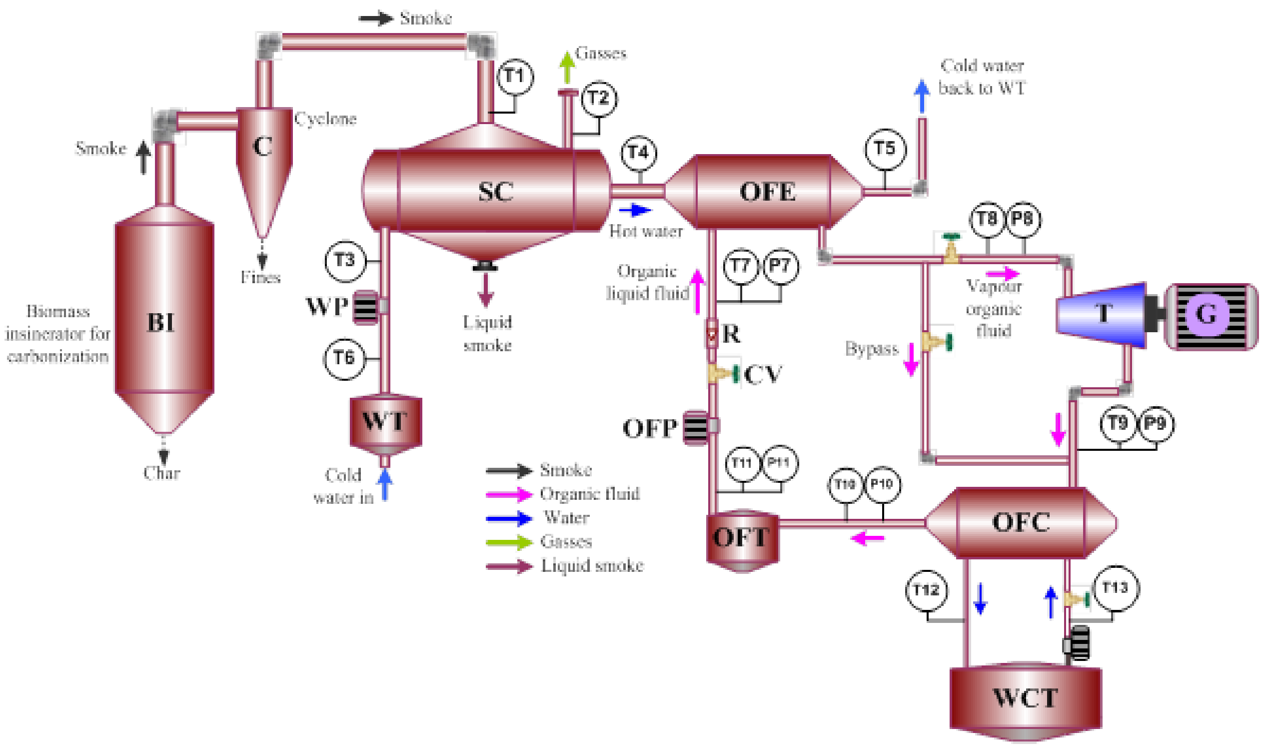

Fig. 2. Schematic diagram of ORC system integrated with wood carbonization process; Biomass carbonizer (BC), Cyclone (C), Smoke condenser (SC), Organic fluid evaporator (OFE), Turbine (T), Organic fluid condenser (OFC), Organic fluid feed pump (OFP), Water cooling tank (WCT), Generator (G)

3. Organic Fluid Selection

Selection of the working fluids in an ORC system is a crucial aspect. Lots of working fluid candidates can be found in the literature. However, in this study, the working fluid used refers to previous studies by numerous researchers, as shown in Table 2. The exact working fluid, which is suitable to be implemented in this ORC, is shown in Figure 3. It is also used as the guide.

Table 2
Working fluid and operating parameters of some ORCs system

No	Working fluid	Component conditions	Values	Reff
1	R123	Condenser outlet	23-30°C	[19]
		Evaporation	80-110°C	
		Turbine inlet	130°C	[59]
		Heat source	50-90°C	[60]
		Turbine inlet	101.7-165.2°C	[26, 27]
		Heat source	85.95, 110, 130°C	[61, 62]
		Turbine inlet	101.7-165.2°C	
		Heat source	72°C	[63]
		Condenser outlet	90-220°C	
		Heat source	90-220°C	[64]
2	R245fa	Heat source	100°C	[65]
		Condenser temperature	21.86-43.63°C	
		Heat source	77.9°C	[66]
		Cooling water	11.5-21°C	
		Evaporation	77.1-82.3°C	[67]
		Condenser outlet	37.4-40.3°C	
		Expander inlet	50-115°C	[20]
		Heat source	90°C	[68]
		Heat source	100-150°C	[69]
		Heat source	125°C	[67]
		Cooling glycol water	14-43°C	
3	R245fa	Heat source	100°C	[58]
		Condenser	27°C	
4	R227ea	Heat source	66.8°C-110°C	[70]
5	HFE7000	Heat source	117.8-128.9°C	[71]
6	SES36	Heat source	125°C	[67]
		Cooling glycol water	14-43°C	
7	R245fa/R365 (48.5%/51.5%)	Heat source	100-150°C	[69]
8	R11	Heat source	100°C	[58]
		Evaporation	70°C	

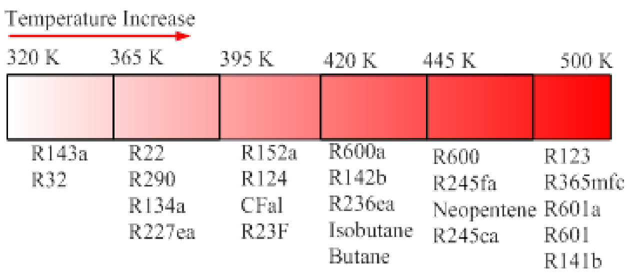

Fig. 3. Working fluids selection hint according to the heat sources temperature [59]

4. System Analysis

Thermodynamic performance analysis of the system proposed is performed based on the characteristic parameters of the selected working fluid and criterion, which refers to the previews researchers used as shown in Table 3. The R134a working fluid was selected to study the system's performance. The thermal properties of the working fluid as a function of the pressure and temperature of each component are used as an input to the thermodynamic performance equations and correlations related to the system considered. Some technical data and information from the preview articles are used to compare the results of this study.

Table 3
The considered input parameters of the 10 kW_e power system proposed with R134a as the working fluid

No	Parameter	Units	Value	Ref.
1	Smoke temperature, T_1	°C	150-220	Measured
2	Hot water temperature	°C	70-95	Measured
3	Organic fluid evaporation temperature, T_{evap}	°C	60-90	[60]
4	Organic fluid evaporation pressure, P_{evap}	bar	5	[58,61]
5	Turbine inlet pressure, P_8	bar	5	[25,58,62]
6	Turbine inlet temperature, T_8	°C	60-90	[25,58]
7	Turbine inlet temperature, T_9	°C	20-40	Measured
8	Turbine inlet pressure, P_9	bar	1-2	[58]
9	Organic fluid condensation pressure, P_{con}	bar	1-2	[58]
10	Organic working fluid flow rates, $\dot{m}_7$	kg/s	0.5-1	[58,63]
11	Isentropic adiabatic pump and turbine efficiency, $\eta_{P,s}$ and $\eta_{T,s}$	%	0.80 and 0.70	64]
12	Pressure ration, P_r	-	3-7	Measured

5. Basic Expression State of the Proposed ORC

The ORC sub-system in the carbonization process is schematically simplified as shown in Figure 4. Each component is assumed as a control volume with the following idealizations; OFE as an isobaric exothermal process, OFC as an isobaric endothermic process, the turbine as an isentropic expansion process, and the OFP as an isentropic compression process.

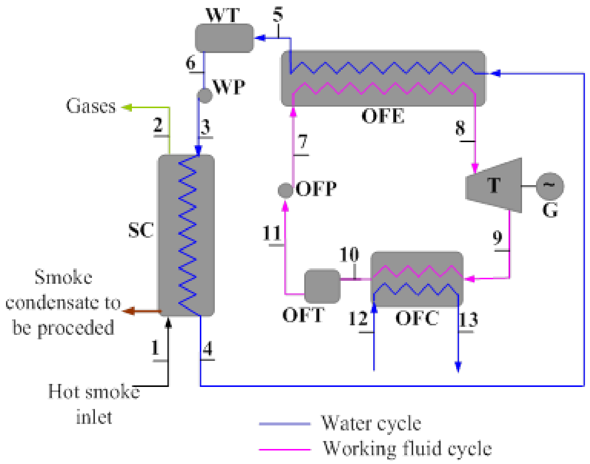

Fig. 4. Schematic flow diagram of the ORC integrated to smoke condensation process

By assuming steady state condition, kinetic and potential changes are ignored, the general principle of continuity, energy rate of change, and exergy rate of change of flow stream at the inlet (i) and exit (e) for each component of the ORC can be expressed according to the Eq. (1) to (3) as follows

$$\sum_{i} \dot{m}_i - \sum_{e} \dot{m}_e = \dot{m} \tag{1}$$

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \tag{2}$$

$$\frac{d\Phi_{cv}}{dt} = \left(1 - \frac{T_0}{T_b}\right)\dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_i \psi_i - \sum_e \dot{m}_e \psi_e - \dot{I}_d \tag{3}$$

Where $\dot{m}$ is the mass flow rate of working fluid, h is the working fluid enthalpy at the specified state, ψ is the specific stream exergy flows at inlet and exit site respectively, T_0 is the dead state temperature where the exergy is zero, T_b is the boundary temperature where the heat transfer to surroundings occurred, $\dot{I}_d$ is the exergy destruction during process. T—s diagram of saturated vapor curve of the organic fluid on the ORC subsystem considered is shown in Figure 5. The carbonization product smoke of high temperature entering the SC, immediately undergoes condensation using water as cooling medium. The hot water from the SC then is utilized to evaporate the hydrocarbon working fluid in the OFE.

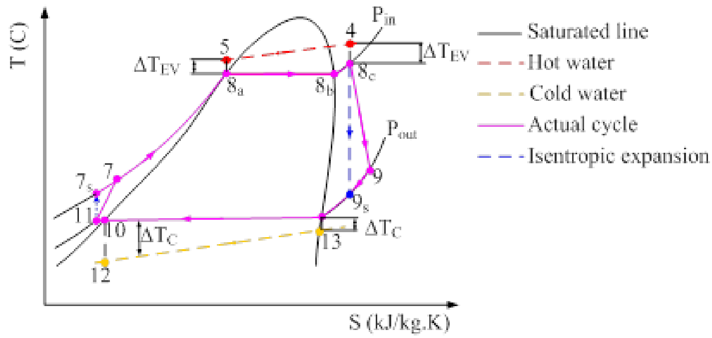

Fig. 5. T—s diagram of the working fluid curve of the designed cycle

5.1 Smoke Condensation (SC)

During smoke condensation, the heat absorbed by the cooling water is assumed as the total heat input to the OFE which is expressed as

$$\dot{Q}_{OFE} = \dot{m}_1 C_{p,sm}(T_1 - T_2) = \dot{m}_3 C_{p,w}(T_4 - T_3) = \dot{m}_3(h_4 - h_3) \tag{4}$$

5.2 Organic Fluids Evaporation (OFE)

There are two fluid streams entering and living the evaporator; the hot water (*hw*) as the hot stream and the working fluid (*wf*) as the cold streams. Assume the heat transfer to the surrounding is ignored and there is no work across the OFE, therefore, mass flow rate ratio in the OFE can be written by solving the energy balance rate equation, Eq. (2) as

$$\frac{\dot{\mathbf{m}}_7}{\dot{\mathbf{m}}_4} = \frac{(\mathbf{h}_4 - \mathbf{h}_5)}{(\mathbf{h}_{8b} - \mathbf{h}_7)} \tag{5}$$

Exergy destruction in the OFE is determined by simplifying and solving Eq. (3) as

$$\dot{\mathbf{I}}_{d,OFE} = \dot{\mathbf{m}}_4 \dot{\mathbf{X}}_{hw} + \dot{\mathbf{m}}_7 \dot{\mathbf{X}}_{wf} \tag{6}$$

where $\dot{X}_{hw}$ and $\dot{X}_{wf}$ are the exergy rate change of streams in the OFE and are expressed as

$$\dot{\mathbf{X}}_{hw} = \dot{\mathbf{m}}_4[(\mathbf{h_4} - \mathbf{h_5}) - \mathbf{T_0}(\mathbf{s_4} - \mathbf{s_5})] \tag{7}$$

$$\dot{\mathbf{X}}_{wf} = \dot{\mathbf{m}}_7[(\mathbf{h}_{8b} - \mathbf{h_7}) - \mathbf{T_0}(\mathbf{s}_{8b} - \mathbf{s_7})] \tag{8}$$

Exergy efficiency of the OFE is written as

$$\eta_{\mathbf{OFE,rev}} = \frac{\dot{\mathbf{X}}_{wf}}{\dot{\mathbf{X}}_{hw}} \times \mathbf{100}\% \tag{9}$$

5.3 Organic Fluid Condensation (OFC)

Condensation is assumed as an endothermic adiabatic process which rejects heat. Assume the heat transfer to the surrounding is ignored and there is no work across the OFC, therefore, the mass flow rate ratio in the OFE can be written by solving the energy balance rate equation, Eq. (2) as

$$\frac{\dot{\mathbf{m}}_9}{\dot{\mathbf{m}}_{12}} = \frac{(\mathbf{h}_{13} - \mathbf{h}_{12})}{(\mathbf{h}_{10} - \mathbf{h}_9)} \tag{10}$$

Exergy destruction in the OFC is determined by simplifying and solving Eq. (3) as

$$\dot{\mathbf{I}}_{\mathbf{d,OFC}} = \dot{\mathbf{m}}_9\dot{\mathbf{X}}_{wf} + \dot{\mathbf{m}}_{12}\dot{\mathbf{X}}_{cw} \tag{11}$$

Where $\dot{X}_{wf}$ and $\dot{X}_{cw}$ are the change exergy rate of streams in the OFC and expressed as

$$\dot{\mathbf{X}}_{wf} = \dot{\mathbf{m}}_9[(\mathbf{h_9} - \mathbf{h}_{10}) - \mathbf{T_0}(\mathbf{s_9} - \mathbf{s}_{10})] \tag{12}$$

$$\dot{\mathbf{X}}_{cw} = \dot{\mathbf{m}}_{12}[(\mathbf{h}_{12} - \mathbf{h}_{13}) - \mathbf{T_0}(\mathbf{s}_{12} - \mathbf{s}_{13})] \tag{13}$$

$$\eta_{\mathbf{OFC,rev}} = \frac{\dot{\mathbf{X}}_{cw}}{\dot{\mathbf{X}}_{wf}} \times \mathbf{100}\% \tag{14}$$

5.4 Organic Fluid Feed Pump (OFP)

A feed pump is required to increase the liquid form of working fluid pressure to the evaporation pressure. The pump performances expressions are determined after simplifying and solving Eq. (2) and (3) as follows

$$\dot{\mathbf{W}}_{\mathbf{OFP,act}} = \dot{\mathbf{m}}_7(\mathbf{h_7} - \mathbf{h}_{11}) \tag{15}$$

$$\dot{\mathbf{W}}_{\mathbf{OFP},s} = \frac{\dot{\mathbf{m}}_7(\mathbf{h}_{7s} - \mathbf{h}_{11})}{\eta_{\mathbf{P}}} \tag{16}$$

where, $\mathbf{h_7}$ and $\mathbf{h}_{11}$ are the working fluid enthalpy in the inlet and outlet of the pump, and $\mathbf{h}_{7s}$ is the working fluid enthalpy after the isentropic pressurization process. Pump reversible efficiency can be written as follows

$$\boldsymbol{\eta_{OFP,s} = \frac{\dot{W}_{OFP,act}}{\dot{W}_{OFP,s}} \times 100\%} \tag{17}$$

$$\boldsymbol{\dot{I}_{d,OFP} = \dot{W}_{OFP,rev} + \dot{m}_7[(h_7 - h_{11}) - T_0(s_7 - s_{11})} \tag{18}$$

$$\boldsymbol{\dot{W}_{OFP,rev} = \dot{m}_7[(h_7 - h_{11}) - T_0(s_7 - s_{11})] - \dot{I}_{OFP}} \tag{19}$$

$$\boldsymbol{\eta_{OFP,rev} = \frac{\dot{W}_{OFP,act}}{\dot{W}_{OFP,rev}} \times 100\%} \tag{20}$$

5.5 Turbines

Turbine as an adiabatic isentropic expansion process (state 7-8s) generates energy in the form of shaft work. Work produced ($\dot{W}_T$) and the turbine performances are expressed by simplify Eq. (2) as

$$\dot{W}_{T,act} = \dot{m}_{8c}(h_{8c} - h_9) \tag{21}$$

$$\dot{W}_{T,s} = \frac{\dot{m}_{8c}(h_{8c} - h_{9,s})}{\eta_T} \tag{22}$$

$$\eta_{T,s} = \frac{\dot{W}_{T,act}}{\dot{W}_{T,s}} \times 100\% \tag{23}$$

Maximum power output or reversible power ($\dot{W}_{T,rev}$) is determined by simplifying Eq. (3) where exergy destruction in the turbine is set to zero due to the turbine reversibility process ($\dot{I}_d = 0$), therefore

$$\dot{W}_{T,rev} = \dot{m}_{8b}[(h_{8c} - h_9) - T_0(s_{8c} - s_9)] \tag{24}$$

Exergy efficiency of the turbine is determined by

$$\eta_{T,rev} = \frac{\dot{W}_{T,act}}{\dot{W}_{T,rev}} \times 100\% \tag{25}$$

In this system, the net work done by the working fluid or the work output ($\dot{W}_{NET}$) should be equivalent to the power output from the turbine minuses the power consumption by the OFP is written as

$$\dot{W}_{NET} = \dot{W}_{T,act} - \dot{W}_{OFP,act} \tag{26}$$

6. Performances Analysis Expression

Thermal efficiency indicates the extent to which the energy input to the working fluid passing through the OFE subcomponent is converted to the network output. Using the expressions just introduced, the energetic and exergetic efficiencies of the proposed ORC system can be evaluated by using the following expression [39]

$$\eta_{th} = \frac{\dot{W}_{NET}}{\dot{Q}_{OFE}} = \frac{\dot{W}_{T,act} - \dot{W}_{OFP,act}}{\dot{Q}_{OFE}} \times 100\% \tag{27}$$

$$\eta_X = \frac{\dot{W}_{NET}}{\dot{Q}_{OFE} \times \left(1 - \frac{T_0}{T_1}\right) - \dot{Q}_{OFC} \times \left(1 - \frac{T_0}{T_2}\right)} \times 100\% \tag{28}$$

where η_{th} and η_X are the energetic and exergetic efficiencies of the ORC system respectively, $\dot{W}_{NET}$ is the net shaft power output, $\dot{Q}_{OFE}$ is the heat into the OFE, $\dot{Q}_{OFC}$ is the heat rejected in the OFC. T_1 and T_2 are the corresponding input and output smoke temperature. Other important parameters such as the back–work ratio (bwr) as defined the ratio of the pump work input to the work developed by the turbine, the pressure ratio or expansion ratio in the turbine (P_r) are also utilize to quantify the system performance as the following expressions

$$\mathrm{bwr} = \frac{W_{P,act}}{W_{T,act}} \tag{29}$$

$$P_r = \frac{P_{T,in}}{P_{T,out}} \tag{30}$$

7. Special Case

Here are some theoretical simulation results using the R143 as working fluid. The properties of this fluid are available on the table. Figure 6 shows the feed pump's required power to achieve the required evaporation pressure, which is determined based on Eq. (15). From the graph, it is known that the higher the output pressure of the feed pump, the higher the input power required, but that it decreases when the input pressure increases. This information will greatly determine the amount of network generated by the system. This power will be supplied by the turbine outlet power, therefore the turbine output should be higher than the feed pump power required.

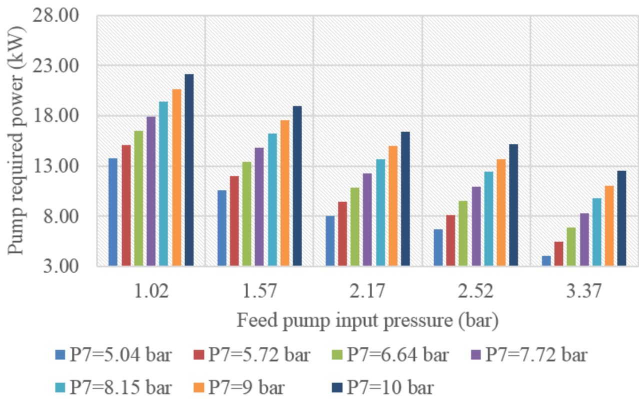

Fig. 6. Pump power required based on input and output pressure

Figure 7 shows the output power calculated by Eq. (21) with respect to the turbine inlet temperature. The output power varies with the outlet temperature of the turbine (T_9). The highest output power is obtained when the temperature difference between the inlet (T_{8c}) and turbine outlet (T_9) is quite high.

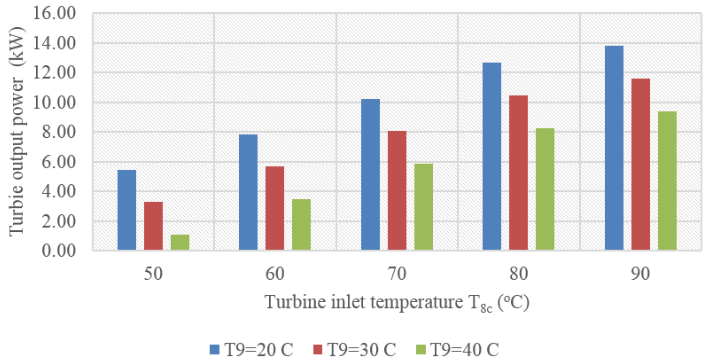

Fig. 7. Effect of turbine inlet temperature on the output power with turbine outlet pressure of 2 bar

Figure 8 shows the output power affected by the turbine outlet pressure (P_9). The greater the pressure outlet of the turbine, the greater the output power obtained. However, the maximum power obtained depends on the ratio of the inlet pressure (P_{8c}) and outlet of the turbine (P_9).

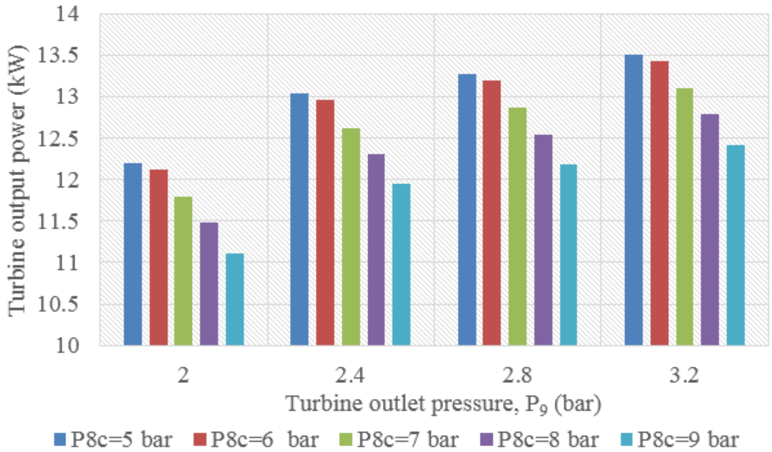

Fig. 8. Effect of turbine outlet pressure on the output power at turbine outlet temperature of 20 °C

Figure 9 shows the output power with respect to the turbine inlet temperature at different turbine inlet pressures (P_{8c}). It is shown that the output power increases with the temperature (T_{8c}). The output power also depends on the turbine input pressure. The higher the inlet pressure and temperature, the greater the output power produced. However, at the same temperature, an increase in inlet pressure causes a decrease in output power. With an inlet temperature of 90 °C, about 14.5 kW of output power can be obtained. This finding is confirmed by Touaibi *et al.,* [63].

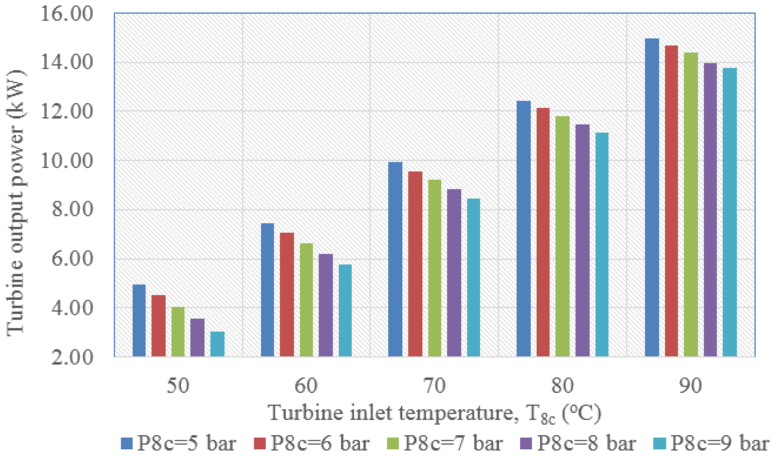

Fig. 9. Effect of turbine inlet temperature on the output power at 20 °C of outlet temperature

8. Discussion and Future Work

In this study, the energy and energy balance equations for the ORC plant powered by thermal energy from biomass carbonization have been briefly expressed. The governing equations of continuity, energy, and energy destruction rate balances have been applied to each subcomponent of the plant, which can be used to predict its thermodynamic performance. The operating temperatures and pressures of the evaporator and condenser are more important than the heat source's temperature. Because the high amount of heat available does not always ensure a high amount of net power output [64]. The most important operating parameter that has a strong influence is the operating pressure.

Several selected operating target parameters were adopted to perform the ORC proposed as described in Table 3. By applying all the parameters together with working fluid characteristics into these expressed equations, the ORC system performance could be predicted and should be comparable with other results of preview researchers. Generally, heat transfer in the evaporator is very sensitive to a change in the temperature and flow rate of the heat source. Increments in heat source temperature and flow rate result in a decrease in working fluid viscosity that has a positive impact on working fluid flow rate through the system. Good heat transfer resulted in the increase of both the turbine isentropic and thermal efficiencies [65]. From this study, it is recommended that the system performance reach optimum performance at turbine inlet pressure of 5 bar and turbine outlet pressure of 2 bar. It is hoped that the performance of the proposed system will be similar or even better compared to the results of previous researchers.

The estimated system performances using energy and energy efficiency analysis were also intended for educational purposes as a unit test for performing experiments on solid biomass combustion for energy production. It is necessary to perform modelling using computer simulation using related software to facilitate preliminary design and development of a real ORC system on a lab-scale basis, so it can then be validated with industrial data. Furthermore, to achieve the highest ORC system performance, the selection of equipment for subcomponents needs to be addressed in the future.

9. Conclusion

Thermal energy, which is regarded as waste heat, can be used to generate small-scale electrical power. A process that involves heat transfer or heat removal will waste thermal energy, as in the condensation process in a condenser. This wasted thermal energy can still be used as electrical power even on a small scale. However, before constructing a power plant system based on ORC, a preliminary thermal analysis is really required to estimate the electrical power that can be obtained and can be very helpful to select the appropriate organic fluid as the working fluid. In contrast to the water-based Rankine system, the ORCs system is really considered the operating parameters (temperature and pressure) at the inlet and outlet. The performance of an ORCs system in a carbonization application is presented in this study. The study focuses on the operating parameters of the plant with the major goal of determining the output power that can be obtained from the operating parameter study. Through the analysis, it is shown that the higher the turbine outlet pressure, the higher the output power obtained. However, this will require additional tools to condense the fluid so that it immediately becomes a liquid. According to this study, about 13 kW of output power can be obtained if the steam temperature can reach 90 °C at a pressure of about 5 Bar. The ORC is well suited to converting wasted thermal energy into electrical power.

Acknowledgements
The presented work is the result and contribution of the project: Development and testing of biomass combustion systems for power generation in remote areas based on the Organic Rankine Cycle, funded by the Universitas Negeri Medan on the Research Development Scheme with Contract Number: 13/UN33.8/PL-PNBP/2020.

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