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A Hybrid Solar-Biomass ORC-Based CHP System Using a Novel Scroll Expander

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ABSTRACT

Micro Combined heat and power generation (Micro-CHP) based on the organic Rankine cycle (ORC) presents a flexible and reliable technology that can help in saving the environment, achieving energy security and flourishing the economic growth especially if it is driven by a renewable energy resource. However, the majority of ORC expanders employed recently suffer from key problems including excessive working fluid leakage, thermal losses and low isentropic efficiency, with no commercial micro-scale scroll expanders available in the market and applicable for ORC applications. In this study, a hybrid solar-biomass driven micro-CHP system based on the organic Rankine cycle technology is theoretically and experimentally investigated to provide the thermal needs and part of the electrical demands for a residential application. The micro-CHP system employs a new type micro-expander with environmentally friendly working fluid. A Numerical model was developed using the Engineering Equation Solver (EES) software to simulate the thermodynamic behaviour of the ORC unit predicting the thermal and electrical performance of the overall CHP system. In addition, an experimental setup was built to test the whole ORC-CHP system performance under different operational conditions using HFE-7100 fluid. The maximum electric power generated by the expander was in the range of 500 W at a pressure differential of about 4.5 bars. The expander isentropic efficiency has attained more than 80% at its peak operating conditions with no fluid leakage observed.

KEYWORDS: *Combined Heat and Power, Organic Rankine Cycle, Scroll Expander, Solar Energy, Biomass Boiler*

1 INTRODUCTION

The world primary energy demand has increased drastically in the last two decades due to the large technological and industrial development accompanied by a tremendous population growth. The total primary energy consumption in the world has progressively increased from 283 Quadrillion Btu in 1980 to about 510 Btu in 2010 [1]. The increase in the demand for energy resources has been accompanied by a progressive rise in the conventional resources prices, mainly petroleum products. In addition, the consumption of such traditional energy resources (gas, oil, coal…) contributes to a large portion of the world's greenhouse gases emissions leading to the devastating warming global warming problem. The conventional energy resources prices increase along with the imminent energy shortage and the global warming problem have urged the change in energy production and consumption patterns and the switch towards alternative and renewable energy resources. (solar, hydro, biomass, wind, geothermal…). This switch towards renewables allows flourishing the economic growth, fostering the rural development and saving the environment.

Combined heat and power generation system (CHP) is a flexible technology that can help in achieving energy security, flourishing the economic growth and saving the environment. CHP systems offer high reliability with the simultaneous production of heat and power in the same process in addition to the possibility of being driven by alternative and clean renewable energy resources including solar thermal energy, biomass combustion heat and waste heat. Using CHP systems for electricity and heat production is very cost effective for both industrial and residential consumers and can save up to 35% of the primary energy generated compared to conventional heat and power systems [2]. A framework for the development of highly efficient combined heat and power generation systems was provided in the European Directive 2004/8/EC in order to attain primary savings in the internal energy market [3]. The total CHP systems capacity installed in the EU has exceeded 100 GW where Germany is leading with 22% of the EU overall capacity followed by Poland and Denmark with 9%. Moreover, 46% of the electricity generation in Denmark is provided by CHP systems with 30% in each of Latvia, Finland and Netherlands. On the other hand, the good quality installed CHP capacity has increased from 5,614 MWe in 2009 to about 6,111 MWe at the end of 2011 with medium-scale CHP systems with capacity ranging between 100 kW and 1 MW dominating the market. Fig.1 presents the progressive increase in the installed CHP capacity in the UK in the recent 15 years [4]. In addition, 68% of the combined heat and power systems installed in UK run on natural gas, where the share of renewable energy-driven CHP systems was only 6% of the total installed CHP systems in 2010.

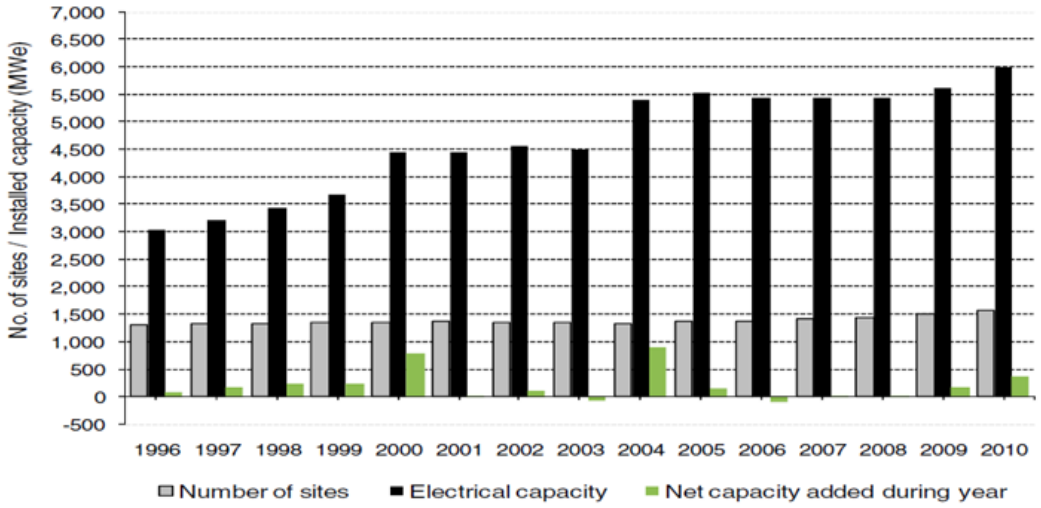

Fig. 1. Installed CHP capacity in UK [4]

A comprehensive review for the recent developments in the field of micro CHP systems has been presented by Jradi et al [5]. Different prime movers with various technologies can be considered for micro CHP applications. Some of these technologies are commercially mature and are widely available such as reciprocating and micro turbine small-scale engines, while other techniques are still in the research and development phase with very limited commercial systems in the market including organic Rankine cycle (ORC)-based systems and fuel cell-based units. Based on their flexibility, low emissions and acceptable efficiencies, Stirling and Rankine engines tend to be the two most favourable and feasible technologies for micro CHP systems in buildings driven by renewable energy resources. However, Stirling engines need high quality heat which is accompanied with high investment costs. Therefore, Rankine-based micro CHP units provide an effective technology to supply heat and electricity for residential applications with an acceptable efficiency. ORC-based micro CHP systems work using the same technology as the ordinary steam-based units but utilize an organic working fluid as the main driver instead of water. They have received an increasing attention in the recent years in the field of micro-scale power generation and heating in residential and building applications with various advantages compared to the ordinary steam cycle [6].

2 EXPANDER SELECTION

Different types of expanders were investigated and evaluated throughout the literature for ORC applications including radial turbine expanders [7], screw expanders [8] and scroll expanders [9]. Qiu et al. examined various types of expanders including turbine, scroll, screw and van expanders, and indicated that scroll expanders and vane expanders have a high potential to serve for fluid expansion and power generation in micro-scale ORC applications due to their simplicity, high expansion ratios and cost effectiveness [10]. In addition, Lemort et al. identified volume type expanders including screw, scroll and reciprocal expanders as favourable candidates for ORC applications since they provide lower flow rates, lower rotational speeds and higher pressure ratios compared to turbine expanders [11], and they can adopt to the two-phase conditions that may appear at the end of the expansion process is some applications. They recommended the use of scroll expanders for low-temperature ORC applications due to the reduced number of moving parts, reliability and availability, taking into account the scroll compressors long successful record in refrigeration and air conditioning applications. Different studies have been conducted to model and optimize the performance of scroll expanders over a wide range of applications especially for ORC systems [12-15]. Peterson et al. studied the performance of a regenerative ORC-based system for electricity generation using scroll expander and R-123 as a working fluid with a power output in the range of 187-256 W and ORC cycle efficiency of 7.2% [16]. The recorded expander efficiency was between 45 and 50 % with an excessive fluid leakage across the expander all through the operation. To avoid fluid leakage problems, Clemente et al. reported that scroll expanders derived from HVAC systems compressors provide a very attractive choice for small-scale ORC-based systems with a reduced cost and acceptable efficiency [17]. The same suggestion was provided by Brasz et al. who carried out the modification of a centrifugal compressor into a radial-in flow turbine [18]. Wang et al. have carried out an experimental procedure to test a scroll expander modified from a compliant scroll compressor utilizing R-134 and a maximum expander isentropic efficiency attained was in the range of 77% with about 1 kWe power output [19].

Based on the background and review provided, it is shown that the majority of the ORC expanders employed recently for ORC applications have suffered from key problems, mainly excessive working fluid leakage, thermal losses and low isentropic efficiency, with no commercial micro-scale scroll expanders available in the market and applicable for ORC applications hindering the development of commercial small-scale ORC-based micro-CHP systems. In this study, a small-scale ORC-based micro-CHP system is theoretically and experimentally investigated to provide the thermal needs and part of the electrical demands for a residential application. The hybrid micro-CHP system is driven by renewable solar and biomass resources and employs a new type scroll micro-expander with environmentally friendly working fluid. Using the Engineering Equation Solver (EES) software, the micro-CHP system was modelled to simulate the thermodynamic behaviour of the ORC unit predicting the thermal and electrical

performance of the overall CHP system. In addition, an experimental setup was built to test the whole ORC-CHP system performance under different operational conditions using HFE-7100 fluid.

3 ORC- CHP MATHEMATICAL FORMULATION

The ideal ORC cycle comprises four main components; evaporator, pump, turbine and condenser. Due to their durability, cost effectiveness, high levels of reliability and simplicity, micro CHP systems with ORC have received increasing attention for domestic and building applications. ORC-based micro CHP systems are still in the phase of research and development and many companies are working on developing such systems but currently no viable small-scale products are available commercially. Fig. 2 shows the configuration of a regenerative Rankine cycle driven by a hybrid solar-biomass renewable energy heating source and employing a thermal recuperator in order to improve the performance of the simple Rankine cycle. Water heated by an external heat resource (solar collector, biomass boiler) will be circulated to heat the ORC fluid through the evaporator to produce saturated vapour supply to the expander. The saturated vapour undergoes an expansion through the expander allowing electricity generation, and then cooled to saturated liquid in the condenser. The working fluid condensation is accompanied with heat rejection which can be utilized for beneficial thermal purposes. The recuperator allows utilizing additional heat from the hot vapour coming out at the expander exhaust to pre-heat the condensed ORC fluid pumped to the evaporator.

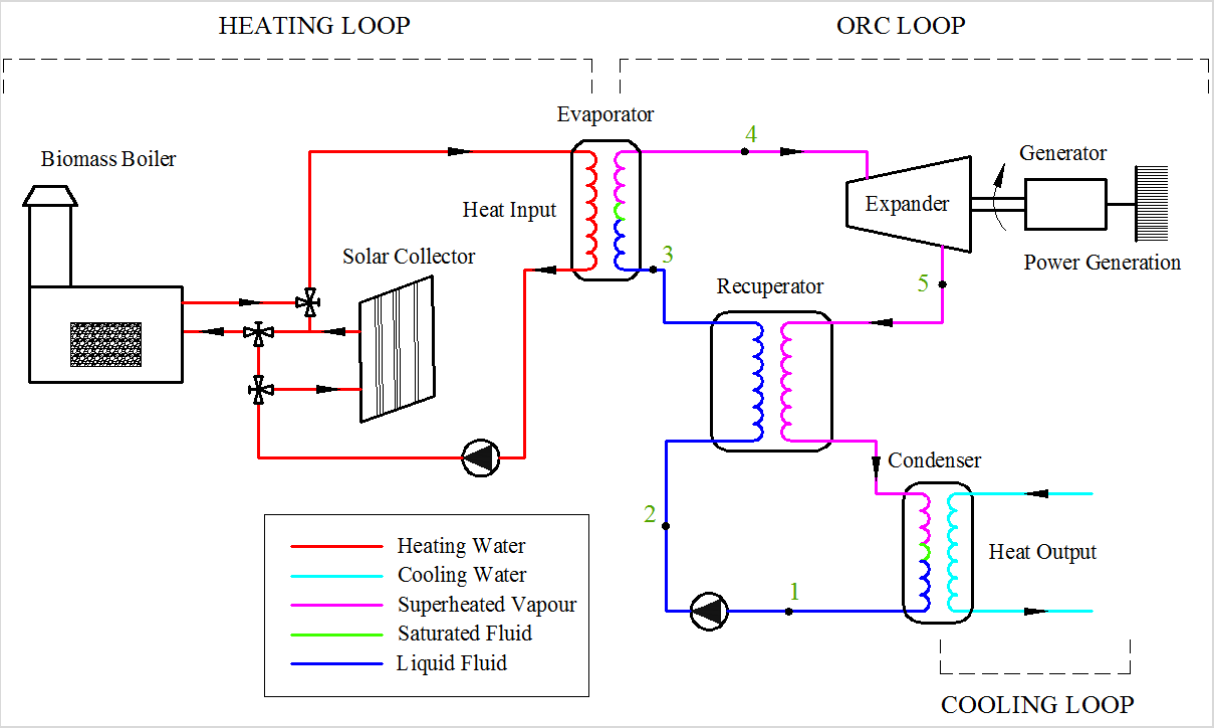

Fig. 2. Configuration of a regenerative Rankine cycle with a thermal recuperator

Applying the first law of thermodynamics for the ideal organic Rankine cycle and taking into account the conservation of mass of the working fluid employed in the cycle, the work of the pump can be given by:

$$W_P = \dot{m} \times (h_2 - h_1) \tag{1}$$

In the same manner, the ideal work output provided by the expander is given by Eq.(2):

$$W_{Ex} = \dot{m} \times (h_4 - h_5) \tag{2}$$

h is the enthalpy of the working fluid in J/kg and $\dot{m}$ is the fluid mass flow rate in kg/s. Then the net work of the organic Rankine cycle can be expressed as the difference between the expander and the pump work as:

$$W_{net} == \dot{m} \times [(h_4 - h_5) - (h_2 - h_1)] \tag{3}$$

On the other side, the external heat input at the evaporator can be given by the following equation:

$$Q_{in} = \dot{m} \times (h_4 - h_3) \tag{4}$$

Thus the overall organic Rankine cycle thermal efficiency can be written as the ratio of the net *ORC* work over the total input external heat at the evaporator as:

$$\eta_{ORC} = \frac{W_{net}}{Q_{in}} \tag{5}$$

The mechanical efficiency of the expander can be expressed as the electric power output W_{elec} divided by the expander actual work output (shaft power):

$$\eta_{mech} = \frac{W_{elec}}{\dot{m} \times (h_4 - h_5)} \tag{6}$$

The isentropic efficiency of the expander is defined as the ratio of the actual work produced by the expander over the ideal work and can be expressed as follows:

$$\eta_{is} = \frac{(h_4 - h_5)}{(h_4 - h_{5'})} \tag{7}$$

$h_{5'}$ is the isentropic enthalpy of the working fluid at the expander outlet. On the condenser side, the heat rejected and absorbed by the cooling water can be given by Eq.(8):

$$Q_{cond} = \dot{m}_{cond} \times (h_{cond_out} - h_{cond_in}) \tag{8}$$

$\dot{m}_{cond}$ is the mass flow rate of the cooling water at the condenser side in kg/s. The useful heat absorbed by the heating fluid in the solar collector and the boiler can be given by:

$$Q_{useful} = \dot{m}_{hf} \times cp_{hf} (T_{hf_out} - T_{hf_in}) \tag{9}$$

$\dot{m}_{hf}$ is the mass flow rate of fluid heated in the solar collector and the biomass boiler in kg/s, cp_{hf} is the specific capacity of the collector fluid in J/kg.K, and T is the temperature of the fluid in °C. The total amount of solar energy available at the solar collector is expressed as:

$$Q_{Solar} = I_{Solar} \times A_{col} \tag{10}$$

I_{Solar} is the total solar radiative flux available at the collector in W/m^2 and A_{col} is the solar collector aperture area in m^2. Then the solar collector overall efficiency can be denoted as the ratio of the useful heat absorbed by the heating fluid to the total amount of solar energy available at the collector:

$$\eta_{Col} = \frac{\dot{m}_{hf} \times cp_{hf} \Delta T_{hf-Col}}{Q_{Solar}} \tag{11}$$

$\Delta T_{hf\text{-}Col}$ is the heating fluid temperature rise in the solar collector. The heat of combustion provided by the biomass boiler is given in terms of the biomass pellets consumption rate $\dot{m}_{Biomass}$ and the lower heating value LHV of the pellets:

$$Q_{Biomass} = \dot{m}_{Biomass} \times LHV \tag{12}$$

The biomass boiler efficiency can be given by Eq.(13):

$$\eta_{Boiler} = \frac{\dot{m}_{hf} \times cp_{hf} \Delta T_{hf-Boiler}}{Q_{Biomass}} \tag{13}$$

$\Delta T_{hf\text{-}Boiler}$ is the heating fluid temperature rise in the biomass boiler. Therefore the electrical efficiency of the *ORC*-based micro-*CHP* system will be given by:

$$\eta_{Ele} = \frac{W_{output}}{Q_{solar} + Q_{Biomass}} \quad (14)$$

The thermal efficiency of the micro-*CHP* system is given by:

$$\eta_{Th} = \frac{Q_{cond}}{Q_{solar} + Q_{Biomass}} \quad (15)$$

4 CASE STUDY

A numerical simulation model was developed using the Engineering Equation Solver (EES) software to simulate the thermodynamic behaviour of the ORC unit predicting the thermal and electrical performance of the overall ORC-CHP system. A case study has been considered employing the environmentally friendly HFE7100 as an organic working fluid in the Rankine cycle. The biomass boiler was considered as the only heating resource in the case study through burning wood pellets of 4.8 kWh/kg calorific value. The model developed in the EES software comprises 183 equations which are solved simultaneously to predict the overall performance of the ORC-CHP system. Table 1. presents various electrical and thermal parameters as predicted by the simulation model with a 0.15 kg/s HFE7100 mass flow rate, 0.1 kg/s water mass flow rate, expander and pump efficiencies of 85 and 80%. Utilizing 24.23 kW input heat at the evaporator, the predicted ORC efficiency is 16.11% with CHP system thermal efficiency of 75.96% and an electrical efficiency of 10.89%.

Table 1. ORC-CHP electrical and thermal parameters predicted by the simulation model

Evaporator Water Inlet Temperature (°C)	120
Evaporator Water Outlet Temperature (°C)	62.42
Condenser Water Inlet Temperature (°C)	20
Condenser Water Outlet Temperature (°C)	68.59
Expander HFE7100 Supply Temperature (°C)	115
Expander HFE7100 Exhaust Temperature (°C)	82.65
Evaporator HFE7100 Supply Temperature (°C)	58.73
Expander HFE7100 Supply Pressure (kPa)	400.13
Expander HFE7100 Exhaust Pressure (kPa)	27.74
Heat Input at Evaporator (kW)	24.23
Expander Electric Power Generation (kW)	2.82
ORC Efficiency (%)	16.11
Thermal Efficiency (%)	75.96
Electrical Efficiency (%)	10.83
Overall CHP Efficiency (%)	86.79

Table 2 presents the ORC-CHP system electrical and thermal efficiencies in addition to the wood pellets consumption rate and the solar collector area needed for different cases based on the share of solar energy and biomass heat in the overall heating input to the ORC unit. The Thermomax DF100 evacuated tube collector was employed in the simulation model to capture solar energy and heat the water circulated [20]. It is shown that the CHP electrical and thermal efficiencies are inversely proportional to the solar energy share in the heating input. The overall CHP efficiency has decreased from 86.79% to about 69.72% with the increase in the solar energy share in the heating from 0 to 100%. Moreover, to supply the whole ORC heating input from solar energy, a solar collector area of 36.58 m2 is required. On the other hand, the burning of 5.608 kg/h of wood pellets in the biomass boiler can supply all the heating input required. It is concluded that the higher the biomass boiler share in the heating input, the more efficient the ORC-CHP unit.

Table 2. Effect of the solar energy and biomass heating share on the overall CHP performance

Total Heat Input (kW)	Solar Energy Share (%)	Biomass Energy Share (%)	Solar Collector Area (m^2)	Wood Pellets Consumed (kg/h)	Thermal Efficiency (%)	Electrical Efficiency (%)	CHP Efficiency (%)
24.23	100	0	36.58	0	61.23	8.49	69.72
24.23	80	20	29.56	1.122	63.54	8.81	72.35
24.23	60	40	22.17	2.243	66.16	9.18	75.34
24.23	50	50	18.48	2.804	67.55	9.37	76.92
24.23	40	60	14.78	3.365	69.78	9.57	79.35
24.23	20	80	7.39	4.487	72.11	10.09	82.2
24.23	0	100	0	5.608	75.96	10.83	86.79

5 EXPERIMENTAL SETUP

A novel-type expander modified from an air conditioning scroll compressor was considered in this study to test and assess the feasibility of using such new-type expander in low-temperature ORC applications. The considered compressor is the SHS-33B4150 Sanden scroll compressor introduced and used very recently as air compressor to fulfill the air conditioning demands in the new Mercedes S400 electric hybrid cars [21]. The compressor has a fix suction and discharge volumes of approximately 33cc and 19cc respectively, and it is enclosed in a semi-hermetic housing decreasing the risks of any working refrigerant leakages and thus allowing more environmental friendly operation. A compressor was provided by Sanden to the University of Nottingham and it has been modified to work as an expander by removing the internal discharge valve and disconnecting the integrated inverter. The compressor was designed to utilize R-134a as a refrigerant, but will be tested in this study with HFE7100 working fluid to eliminate any refrigerant environmental negative impacts.

The experimental system overall configuration is shown in Fig. 3 with all system components. Basically, the system operation is governed by three main loops, hot water heating loop, cooling water loop and the main organic Rankine cycle loop. As shown in Fig. 3, the external heat input source (biomass boiler) provides heat input to the ORC-CHP system through supplying hot water to heat the ORC fluid at the level of the system evaporator. In addition, a compact plate heat exchanger/superheater is integrated in the system to allow additional heating of the working fluid before the expander to produce hot superheated vapour at the inlet of the expander. After the expansion phase through the expander accompanied with electric power generation, the expanded vapour is cooled to saturated liquid in the condenser through heat exchange with the cooling water loop that absorbs the rejected heat to be used in beneficial thermal applications. Thus, the saturated liquid goes back to the evaporator again under the action of the compression pump. A recuperator is introduced in the system to utilize additional heat from the expanded vapour from the expander to pre-heat the condensed ORC fluid pumped to the evaporator. In addition, a sight glass in included to keep the expander in proper operation by monitoring the quality of the ORC fluid vapour quality introduced. The receiver vessel employed in the cycle is used as a storage container for the ORC fluid and the liquid level in the cycle can be monitored through the receiver vessel liquid level indicator. The reservoir integrated before the expander helps in stabilizing the working fluid pressure and hinders any flow of ORC fluid condensate with the heated vapour to the expander.

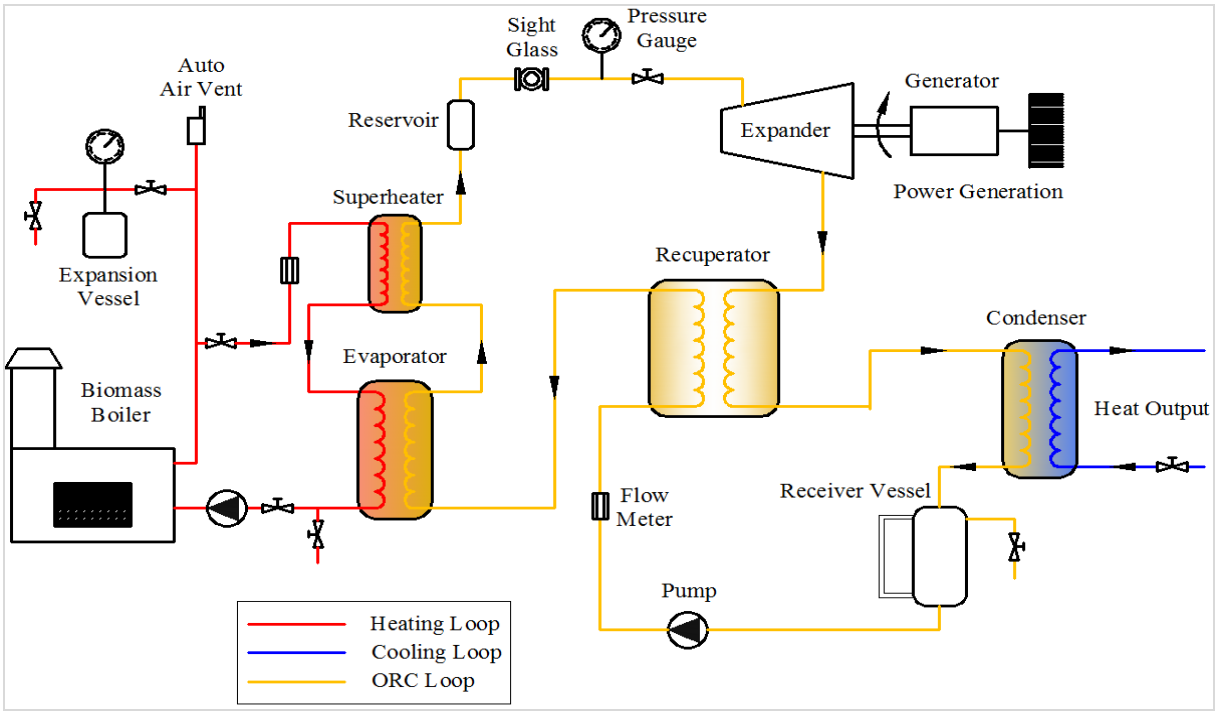

Fig. 3. ORC-CHP experimental system overall configuration

As shown in Fig. 4-a, the modified scroll expander was fitted in the experimental unit using welded refrigerant copper connections and piping and placed at the highest possible position in the rig with the scroll axis in a vertical position allowing vertical downward flow of the vapour and avoiding excess bending in the pipes connecting the expander to the main evaporator and to the other system components. The whole system pipe-work including the different heat exchangers at different levels were tightly insulated to minimize the thermal losses and improve system performance as much as possible. The environmentally friendly HFE7100 was used as an organic working fluid in the Rankine cycle, instead of conventional HFC and HCFC fluids which have negative impacts on the environment. A 50 kW Ashwell wood pellet biomass boiler shown in Fig. 4-b, was utilized as an external heat input source to heat the water to be introduced to the evaporator of the micro-CHP system in order to drive the ORC-based CHP system. A monitor panel is used to control the boiler operation including set temperature and wood pellets feeding. The boiler combustion is controlled by a primary air supply from the holes on the furnace bed walls and a secondary air supply from the vent tubes around the flame. The biomass boiler hopper was loaded with wood-pellets to launch the heating process, and the auger screw conveyor feeds the pellets form the hopper to the furnace allowing continuous and stable combustion. A bank of 12V, 20W and 50W GE halogen lamps was integrated in the system and used as a load. Different recording and monitoring devices were installed at various positions in the rig including pressure transducers, K-type thermocouples, water and refrigerant flow meters, electric DK current transducer. All these recording devices are connected to a Datataker DT 80 data logger which is directly linked to a computer for instant data logging and monitoring of all system operational parameters.

Fig. 4. Scroll expander fitted in the experimental unit (a), Experimental rig and biomass boiler

Fig. 5 shows the variation in the HFE7100 pressure difference across the expander and the electric power generated by the expander throughout the experimental session carried out employing various operational settings for the boiler setting temperature and HFE7100 and water mass flow rates. As shown in Fig. 5, there is a direct proportionality between the pressure differential across the expander and the amount of electric power generated. The pressure difference across the expander has ranged between 1.75 and 4.52 bar where the maximum electric power produced by the expander was about 500 W with a pressure differential of 4.31 bar and a pressure ratio P_{ex_in}/P_{ex_out} of 4.64. All through the experimental sessions, the expander was operating in a smooth and quite mode under different operational conditions with very low noise level and no fluid leakage observed.

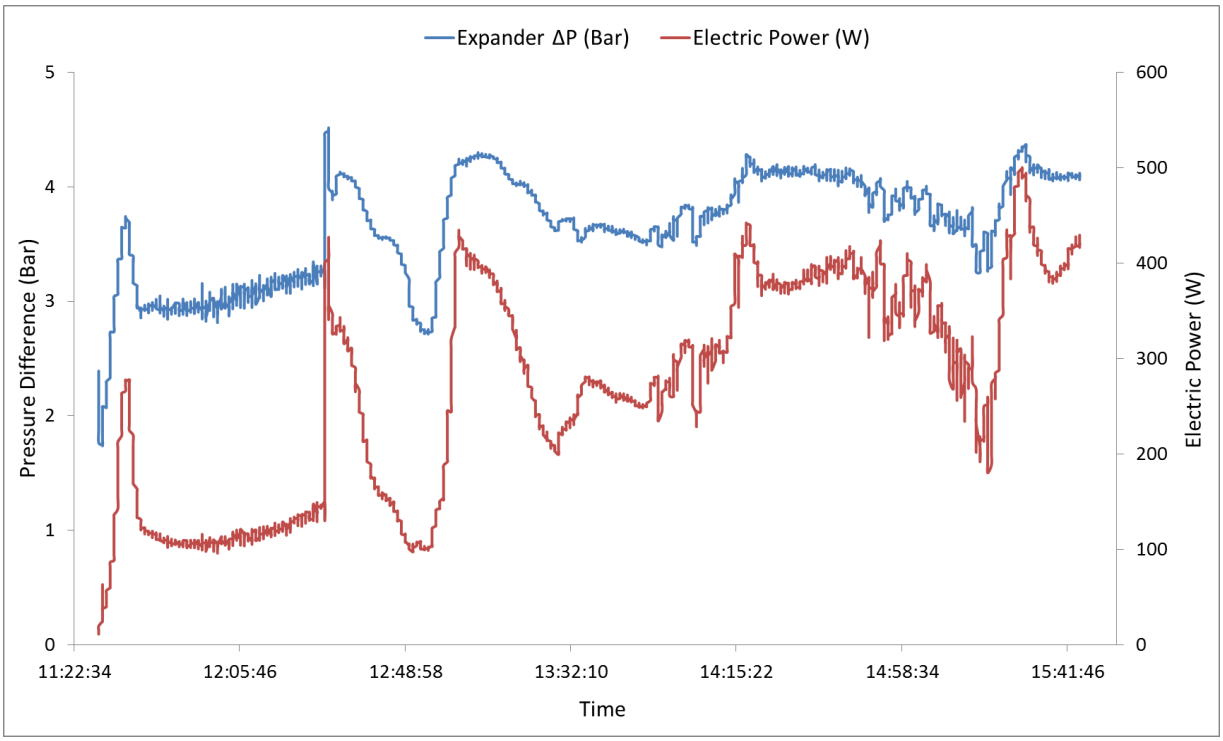

Fig. 5. Electric power generated and HFE7100 pressure difference across the expander

K-type thermocouples were placed at different points throughout the ORC cycle including the expander inlet and exhaust, superheater outlet, recuperater outlet and evaporator outlet. Fig. 6 presents the variation in the HFE7100 temperature at different points in the cycle for one experimental session with 0.047 kg/s HFE7100 mass flow rate. In addition, the figure shows the biomass boiler heating water supply and return temperatures with a mass flow rate of about 0.13 kg/s and a boiler set temperature of 135°C. As shown in Fig. 6, the biomass boiler water supply temperature ranges between 119.54 and 140.77°C with HFE7100 leaving the superheater with temperature between 112.95 and 133.82°C. This demonstrates a good heat exchange between the heating water supply from the biomass boiler and the HFE7100 working fluid at the level of the superheater/ heat exchanger unit. Moreover, there is a slight difference between the temperature of the HFE7100 at the superheater outlet and the expander inlet indicating an acceptable heat loss through the superheater-expander pipe line with an average temperature difference of 1.34°C. The HFE7100 temperature at the expander outlet has varied between 93.68 and 112.14°C with an average temperature difference across the expander of about 18.82°C.

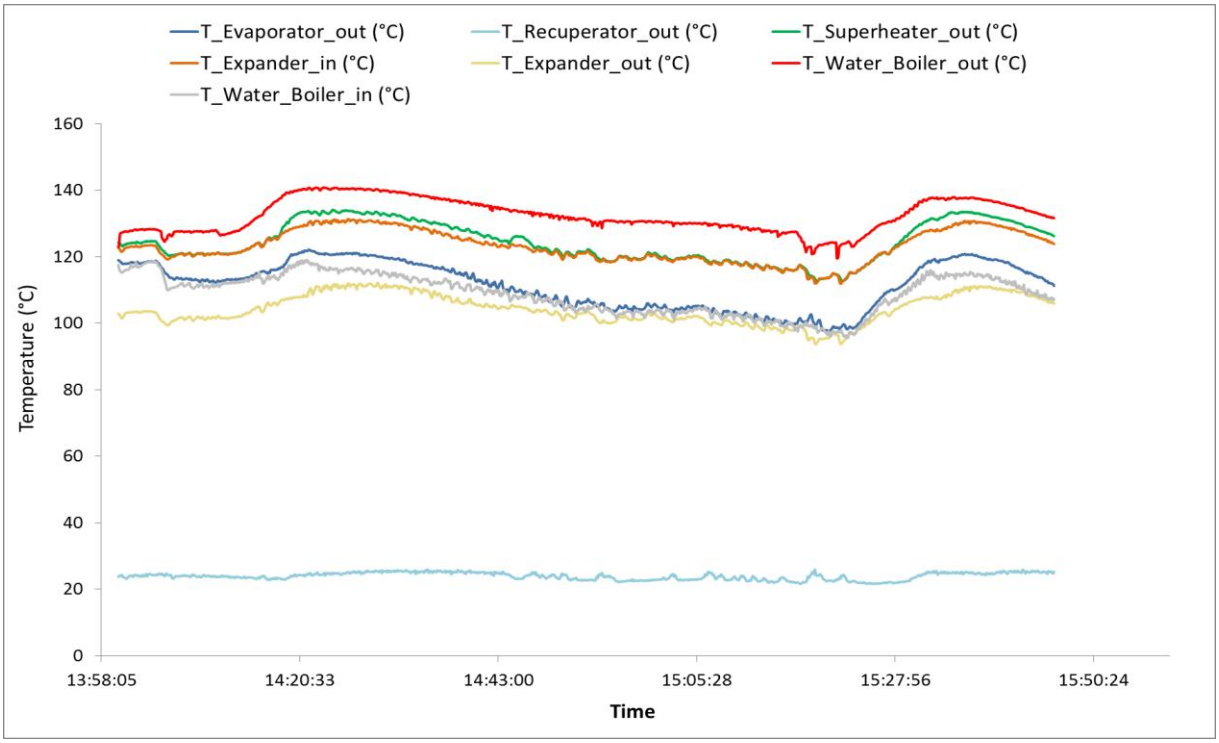

Fig. 6. HFE7100 temperature variation at different points in the cycle

Fig. 7 demonstrates the working fluid pressure change at the inlet and outlet of the expander in addition to the variation in the temperature of the cooling water at the condenser inlet and outlet. The average temperature of the water supply to the condenser was 7.73°C with about 0.0542 kg/s mass flow rate. The temperature of the water at the condenser outlet has attained 53.13°C with an average of 47.7°C. The working fluid inlet pressure at the expander inlet was between 3.84 and 5.93 bar and the recorded pressure at the expander outlet was between 0.48 and 1.77 bar.

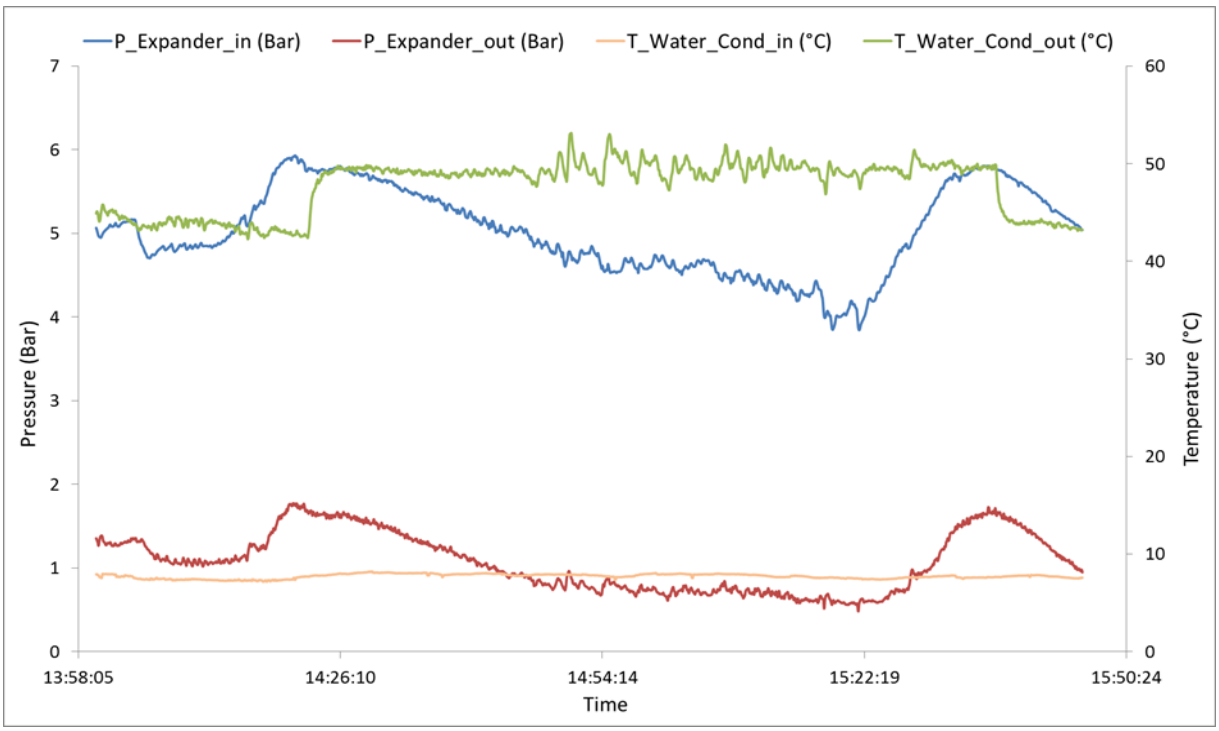

Fig. 7. HFE7100 pressure across the expander and water temperature variation across the condenser

Table 3 presents various ORC-CHP system operational parameters as recorded by the measuring devices including temperatures, pressures, flow rates and different efficiencies at the point of expander maximum electric power generation. The boiler water supply and return temperatures are 135.15 and 112.76°C respectively with a water mass flow rate of 0.13 kg/s. The cooling water

temperature at the condenser level has increased from 7.72 to 49.97°C with a mass flow rate of 0.0542 kg/s and 9581.72 W total heat generated at the condenser side. The temperature and pressure of the HFE7100 working fluid at the inlet of the expander were 126.53°C and 5.5 bar respectively. In addition, the electric power generated by the expander was about 500 W under a pressure ratio of 4.64. The voltage measured across the halogen lamps bank load was 43.16 V and the electric current produced through the load was about 11.59 A. The measured temperatures, pressures and flow rates throughout the cycle were introduced to the numerical simulation model developed in the EES software to simulate the thermodynamic behaviour of the ORC unit predicting the thermal and electrical performance of the overall CHP system. Based on the results, the expander attained good mechanical and isentropic efficiencies of 81.03 and 74.23% respectively with an ORC efficiency of 5.64%. The CHP system electrical efficiency achieved was 4.16% with 78.92% thermal efficiency, providing an overall ORC-CHP system efficiency of 83.08%.

Table 3. ORC-CHP system various operational parameters as recorded in the experimental session

Boiler Water Supply Temperature (°C)	135.15	HFE7100 Mass Flow Rate (kg/s)	0.047
Boiler Water Return Temperature (°C)	112.76	Heat Generated at Condenser (W)	9581.73
Heating Water Mass Flow Rate (kg/s)	0.13	Voltage Across the Load Bank (V)	43.16
Condenser Water Inlet Temperature (°C)	7.72	Electric Current through the Load Bank (A)	11.59
Condenser Water Outlet Temperature (°C)	49.97	Expander Electric Power Generation (W)	500.28
Cooling Water Mass Flow Rate (kg/s)	0.0542	Expander Mechanical Efficiency (%)	81.03
Expander HFE7100 Supply Temperature (°C)	126.53	Expander Isentropic Efficiency (%)	74.23
Expander HFE7100 Exhaust Temperature (°C)	107.14	ORC Efficiency (%)	5.64
Expander HFE7100 Supply Pressure (kPa)	550.32	CHP Thermal Efficiency (%)	78.92
Expander HFE7100 Exhaust Pressure (kPa)	118.45	CHP Electrical Efficiency (%)	4.16
Expander Pressure Ratio	4.64	Overall ORC-CHP System Efficiency (%)	83.08

6 CONCLUSION

Different problems accompanied with small-scale ORC expanders including excessive fluid leakage, high thermal losses and low isentropic efficiency, have hindered the development of commercial small-scale ORC-CHP units. In this study, a micro-scale ORC-based CHP system driven by a hybrid biomass/solar energy resource is considered to provide the thermal and part of the electrical demands for a residential application. A new-type expander modified from an air-conditioning scroll compressor has been employed using the environmentally friendly HFE7100 as an ORC working fluid. A Numerical model was developed in the EES software to predict the thermal and the electrical performance of the overall CHP system. In addition, an experimental setup was built to test the whole ORC-CHP system performance under different operational conditions utilizing a 50 kW biomass boiler. The maximum electric power generated by the expander was about 500 W with 9581.72 W total heat generated at the condenser side under a pressure ratio of 4.64. The expander isentropic efficiency was found to be 74.23% with ORC efficiency of 5.64% and overall CHP system efficiency of 83.08%. The employed modified scroll expander has proved its reliability with an acceptable isentropic efficiency over a wide range of operational conditions accompanied with smooth and quiet operation and no working fluid leakage. This can open the door for such types of scroll expanders to be further investigated and developed to serve different ORC small-scale applications including CHP units.

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