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Development of a hybrid compact organic Rankine cycle micro-CHP system

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Abstract— This paper presents an innovative hybrid and compact organic Rankine cycle (ORC) system for micro combined heat and power utilizing solar thermal energy and natural gas. The ORC investigated in this work consists of heat sources, an evaporator, a scroll expander, a condenser, a fluid pump, a recuperator, a control unit and solar heater simulator. The experimental ORC unit has an electric power capacity of 1100 We. Its advantages include short start-up time, clean and automatic operation, low maintenance and very low noise level. It is well suited for hybrid solar and natural gas energy systems. A mathematical model was established to assess the thermodynamic performance of the ORC system under various conditions. Power output and thermal efficiency were calculated using the established model.

Keywords— ORC; solar energy; natural gas; power; model

I. INTRODUCTION

In recent years, there has been an increased interest in distributed energy production based on local energy resources including both fossil fuels and renewable energy. Micro combined heat and power (micro-CHP) is considered an effective approach to achieving higher overall energy efficiency while reducing greenhouse gas (GHG) emissions as compared to conventional electrical and thermal energy production, and fits well to the strategy for distributed energy production. Micro-CHP systems (<10 kWe) simultaneously produce heat and power for an individual dwelling on site and they can be recognized as a household appliance. Micro-CHP systems are particularly attractive for cold climate areas and remote communities where connection to the grid is not cost-effective. The growing interest in micro-CHP is attributed to several incentives: first, the liberalization of energy markets is changing the approach to energy production; second, environmental pressures, particularly to minimize CO_2 emissions, are increasing considerably; and third, the technology is becoming available. Rising energy costs, volatile fuel prices, electrical power blackouts and increasing environmental concerns are in favor of the acceptance of the micro-CHP concept. Organic Rankine cycle (ORC) can be driven by low temperature heat sources and renewable energy. It is particularly suitable for micro-CHP. ORC has been successfully applied to waste heat recovery. Unlike in the steam power cycle, ORCs employ refrigerants, hydrocarbons, solvents or other organic substances. The ORC process allows the use of low temperature heat sources, offering an advantageous efficiency for small-scale applications. The heat source can be of various origins, such as solar thermal energy, gas or biomass combustion, ground heat source or waste heat. High-temperature solar thermal power systems such as single axis and two axis tracking technologies were widely explored and developed [1]. The high-temperature systems turn out not to be very competitive due to their complexity and high costs. Thus, low-temperature solar thermal power systems using ORC are attracting more and more technical attention [2-4]. For instance, a low temperature solar thermal electric system was investigated [3], which consisted of compound parabolic concentrators with a low concentration ratio and an ORC unit with the working fluid being R123. A solar ORC system consisting of parabolic trough collectors, a thermal storage tank, and a small-scale ORC unit using a scroll expander was presented by Quoilin et al. [4]. mCHP units using ORC as a prime mover for domestic applications are not commercially available, but they are under active development. ORC systems designed for mCHP have been explored a number of authors [5-12]. Investigations have shown that there are developmental issues requiring attention in implementation of this technology. New developments concern small systems with acceptable performance, new working fluids with low environmental impact, scroll expander and advanced heat transfer components. For instance, the small size of expansion devices is a demanding challenge to overcome as the rotating shaft attains high velocities which make it difficult to select appropriate electricity generator. In other words, the key component in a domestic-scale ORC system is the expansion device. Qiu et al. [9] reviewed the state-of-the-art of various ORC expansion devices and concluded that scroll expanders could be suitable for mCHP systems of 1-10 kWe. Peterson et al. [11] studied a micro-scale scroll expander using refrigerants R123 and R134a and achieved different testing results. Liu et al. [9] simulated a scroll expander and a generator in the ORC to convert thermal energy from high-pressure vapor into electricity. Residential mCHP using ORC is showing potential, and is quickly becoming identified as a promising endeavor [12-17]

In this paper, we studied an integrated ORC system for gas and solar thermal power applications. The ORC performance was investigated under various operating conditions. A mathematical model for the ORC was established and calculations of its thermodynamic performance were made.

II. MODELING

A. Model description

The schematic diagram of the ORC system is illustrated in Figure 1. The system consists of heat sources, a working fluid evaporator including a preheater, a turbine (scroll expander) coupled with a generator, a condenser, a feed pump, a recuperator a water pump and a water-circulating loop. The ORC uses N-pentane as a working fluid that is heated in the evaporator to produce vapor, rotating the expander. The vapor then passes through a condenser that transfers the fluid's heat content for heating needs. The condensed fluid is pumped back to the evaporator.

The working fluid is pumped from low to high pressure in Process 1 to 2 (Figure 2). The work done by the pump is given by

$$W_{pump} = \dot{m}(h_2 - h_1)/\eta_{pump} \tag{1}$$

where η_{pump} is the pump efficiency, h is the working fluid enthalpy and $\dot{m}$ is the working fluid mass flow rate.

Process 3 to 4 is the heating process in the evaporator where the high pressure liquid is heated at constant pressure by the heat source to become vapor. The input energy, i.e. the heat transferred to the fluid is

$$Q_{in} = \dot{m}(h_4 - h_3) \tag{2}$$

During Process 4 to 5, the high pressure vapor expands through the scroll expander, generating power. Ideally, this should be an isentropic process 4–5s. However, in practice, the process is not an isentropic one. In other words, the efficiency of the energy conversion in the expander device can't reach 100%. The state of the working fluid at the exit of the turbine is indicated by 5. The power generated by the expander is

$$W_t = \dot{m}(h_4 - h_{5s})\eta_s\eta_{me} = \dot{m}(h_4 - h_5)\eta_{me} \tag{3}$$

where η_s is the isentropic efficiency of the expander device and η_{me} is the expander's mechanical efficiency. η_s can be expressed as

$$\eta_s = \frac{h_5 - h_4}{h_{5s} - h_4} \tag{4}$$

The vapor after the expansion process enters a recuperator where part of the working fluid enthalpy is recovered. The energy balance in the recuperator is expressed as

$$Q_{re} = \dot{m}(h_3 - h_2) = \dot{m}(h_5 - h_6) \tag{5}$$

Finally, the vapor after the expansion process enters a condenser where it is condensed at a constant low pressure to become a saturated liquid (Process 6 to 1).

The cycle thermal efficiency is then obtained from

$$\eta_{cyc} = \frac{W_{out}}{Q_{in}} = \frac{W_t - W_{feedpump} - W_{waterpump}}{\dot{m}(h_4 - h_3)} \tag{5}$$

For known or given operating conditions, heat transfer coefficients, working fluid properties and expander parameters, the aforementioned equations can be solved for power output and electrical efficiency.

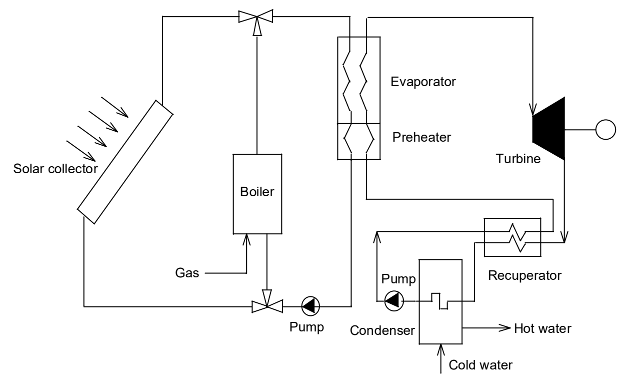

Figure 1. Schematic diagram of the ORC system

B. Modeling results

A model for the ORC system was established and the ORC system performance was simulated under varying conditions. Figure 2 presents the variations of power output with the heat input for various ratios of pump pressure. It is shown that the power output of the ORC cycle increases linearly with increasing the heat input for a given pressure ratio. This is due to the fact that the working fluid mass flow rate increases proportionally as the evaporator heat input is increased. Accordingly the relationship between the power output and the evaporator heat input is expressed as a straight line. For a given heat input, the power output increases with the pressure ratio since a rise in pressure ratio results in an increased ($h_4 - h_5$).

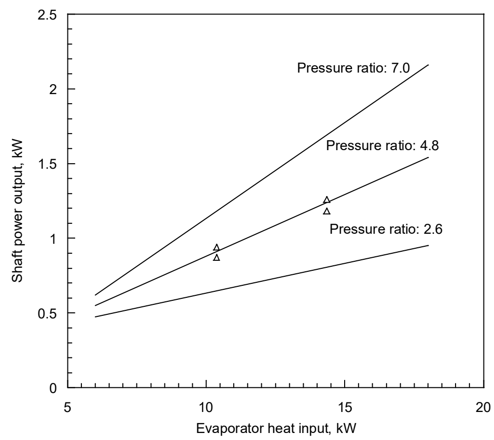

Figure 2. Modeling results of power output vs. evaporator heat input. Expander inlet temperature: 145°C. Δ: Experimental data

Figure 3 shows the system cycle efficiency as a function of the expander inlet temperature for various pressure ratios for 14.5 kW heat input. The system efficiency increases with rising inlet temperature. Obviously, the higher grade of heat source improves the system power output and the efficiency. Modeling results show that, a system cycle efficiency of 10% could be achieved at the typical operating conditions.

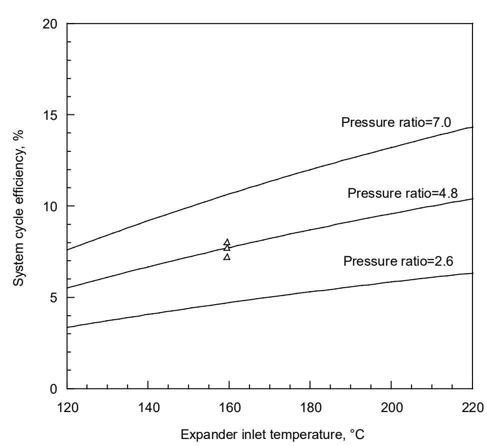

Figure 3. System cycle efficiency vs. expander inlet temperature for various pressure ratios. Heat input: 14.8 kWt. Δ: Experimental data

Figure 4. Prototype ORC unit

III. EXPEREMENTS

A. Experimental setup

Figure 4 shows the prototype ORC unit. It consists of a gas fired boiler, a programmable electrical heater that was used as a simulator of a solar thermal collector, an evaporator, a scroll expander, a condenser, a generator, a feed pump, a recuperator, control valves, electrical loads and a control unit. An efficient expander is the key to successfully developing a domestic-scale ORC system. It converts the kinetic energy in a stream of gas into mechanical energy in the ORC system. A recently-developed cost-effective scroll expander combined with a generator was used in the prototype unit for this project, as shown in Figure 5. The scroll expander is a positive displacement machine consisting of two interleaving spiral-shaped vanes or scrolls fixed on back plates. Figure 6 show the structure of the scroll expander. It consists of a fixed scroll and an orbiting scroll that are at a relative angle of π. Figure 7 shows the schematic diagram of the scroll expander. The profile of the scrolls is a circle involute. The high-pressure gas enters the central chamber of the scrolls through an intake port to expand, driving the orbiting scroll to rotate, and flows out from an exhaust port. This involves three stages: suction, expansion and discharge, as presented in Figure 8. The suction begins at the orbiting angle $\theta=0°$ and ends at the orbiting angle $\theta=\theta_s$ where the suction chamber is sealed. Then the expansion takes place from $\theta=\theta_s$ to $\theta=\theta_d$. When the orbiting angle reaches the discharge angle θ_d, the discharge chamber opens up to the discharge region and starts to discharge. The scrolls can be defined by a number of geometric parameters, including the radius of the basic circle of the scroll, the orbiting radius of the rotating scroll, the height of scroll vanes, the initial angle of the outer involute, the initial angle of the inner involute, the rolling angle (or involute ending angle) and the built-in volume ratio. The built-in volume ratio of an expander is defined as the volume of the expansion chamber at the end of the expansion process over the one at the beginning of the expansion

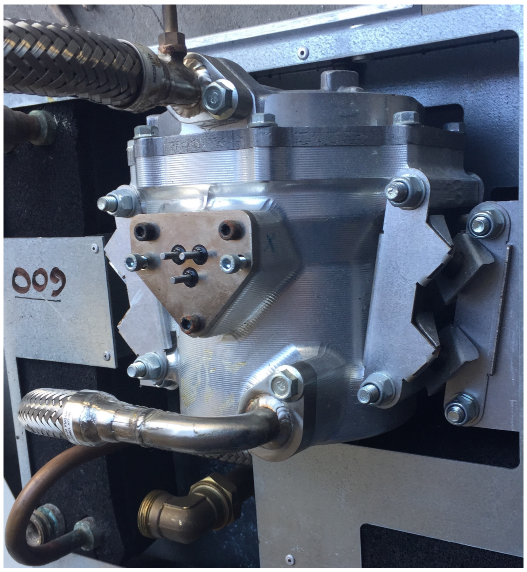

Figure 5. Scroll expander used in the prototype ORC

Figure 6. Structure of the scroll expander

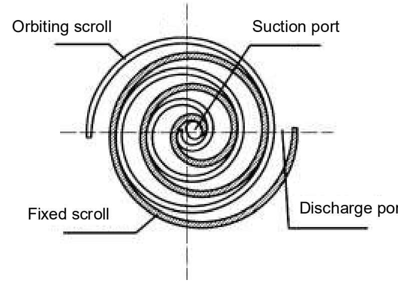

Figure 7. Schematic diagram of the scroll expander consisting of a fixed scroll and an orbiting scroll that are at a relative angle of π

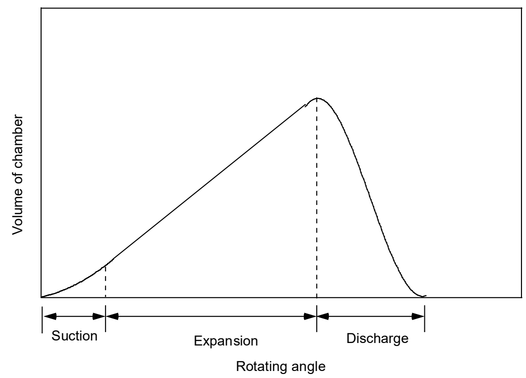

Figure 8. Volume of chamber vs. orbiting angle

In the ORC unit, the liquid working fluid, N-Pentane, boils in the evaporator that is a brazed plate heat exchanger as heat is transferred from the heat sources. The produced N-Pentane vapor passes through a control valve to the scroll expander inlet. The shaft output of the expander runs a permanent magnet asynchronous generator. Low-pressure vapor exhausted from the expander enters the water-cooled condenser where the vapor's heat content is transferred to circulating water for the central heating. The liquid working fluid is compressed with the pump and is then directed to the evaporator. The water circuit operates on a closed loop. In addition, automatic air vents and safety valves were installed in the circulating water loop. Various measuring and control devices, including temperature sensors, flow meters, pressure sensors, a power analyzer and a data acquisition unit were installed. These instruments were connected to a PC. The PC-based data acquisition unit collects signals from all measuring and control devices. Figure 9 shows the prototype ORC-based micro-CHP system.

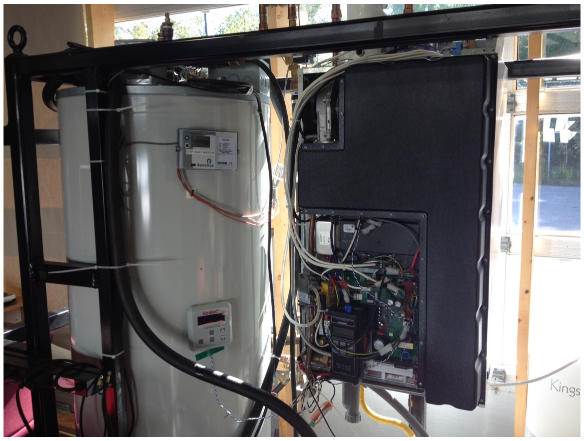

Figure 9. Prototype ORC-based micro-CHP system

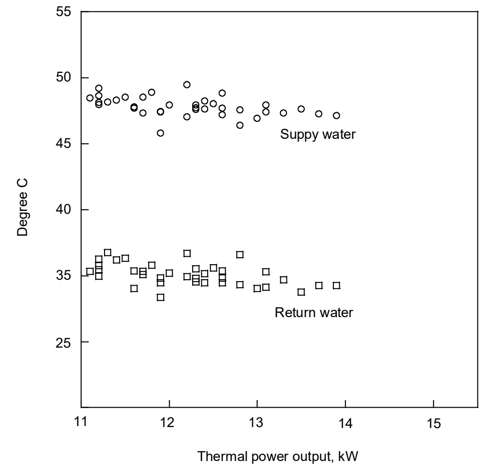

Figure 10. Thermal power output as functions of the temperature of return water to the condenser and supply hot water temperature.

B. Experimental results

A series of tests was conducted to examine the performance of the ORC system under various operating conditions. The power output as a function of evaporator heat input is presented in Figure 2. Table 1 presents a typical set of experimental data. The cycle thermal efficiency is defined by Eq. (1). Table 1 shows that the experimental result of the cycle thermal efficiency reaches approximate to 7.4% when the temperature of return water to condenser is lowest, while the modeling result of the cycle thermal efficiency 8.42% (Pressure ratio=4.8, Figure 3). Obviously, the experimental results obtained are in fairly good agreement with the modeling results presented above. The system cycle efficiency is considerably lower compared to the Carnot efficiency (23%) for the expander inlet and condenser temperatures being considered. This is partly due to the low scroll expander's isentropic efficiency. Improvements in the expander device design could improve its isentropic efficiency significantly and hence improve the system efficiency.

Figure 10 shows the thermal power output as functions of the temperature of return water to the condenser and supply hot water temperature in the micro-CHP system.

TABLE 1 EXPERIMENTAL RESULTS OF ORC UNIT

Parameter	**Value**
Evaporator heat input	14.8 kW
Evaporator pressure	1.02 MPa
Inlet temperature of expander	145°C
Working fluid flow rate	0.12kg/s
Power output*	950-1100 We
Max cycle thermal efficiency	7.4%
Ambient temperature	22°C
Return water temperature (to condenser)	24-38°C
Outlet temperature of circulating water	42°C

* The power output and thus the cycle thermal efficiency depend on the temperature of return water to the condenser

IV. SUMMARY

An ORC system was investigated for gas and solar thermal power applications. N-Pentane was used as the cycle working medium in the ORC. The prototype ORC unit has a thermal efficiency of 8.1% and an electric power output of up to 1100We under typical operating conditions. The advantages of the ORC system include clean and automatic operation, low maintenance and very low noise level and it is suited for small-scale solar thermal power applications. A mathematical model was established to simulate the system. Thermodynamic simulations of the ORC were conducted to assess the system performance under various conditions. Power output and cycle efficiency were obtained using the established model. Simulating results show that a cycle efficiency of 10% could be achieved under the operating conditions being considered. The values of the ORC integrated system provided to consumers would be both the heating system reliability and a reduction in electric power consumption.

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