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Characteristics of Small ORC System for Low Temperature Waste Heat Recovery*

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Abstract

This paper describes fundamental characteristics of a small organic Rankine cycle (ORC) system to be used for power generation from low temperature heat sources such as waste heat and solar energy. The aim of the study was to develop an ORC system with a small power output of less than 1 kW with a hot source with temperature ranging from 60 to 100°C and a cold source with temperature ranging from 10 to 30°C. An ORC system with a potential to produce a turbine/expander power of 250 W was built and its fundamental characteristics were elucidated. A turbine/expander was not actually installed but was simulated by controlling two expansion valves. First, steady-state energy balance of the system was examined and the required turbine/expander efficiency was estimated in consideration of pump power of the working fluid. Then, the relationship between the expansion ratio and thermal efficiency was elucidated. The most important result of the study was that for maintaining high thermal efficiency in the case that the temperature difference between hot and cold sources varies during operation, it is indispensable to employ a variable expansion mechanism by which the expansion ratio of the turbine/expander can be adjusted to fit the optimal ratio at the operating temperature level.

Key words: Organic Rankine Cycle, Thermodynamic Cycle, Thermal Efficiency, Turbine Efficiency, Expander Efficiency, Pump Efficiency, Heat Exchanger Efficiency, Waste Heat Recovery, Power Generation

1. Introduction

To mitigate the world's energy problems and global warming, we must use renewable energies. Waste heat is one such renewable energy. In industries around the world, a large amount of low temperature heat is wasted. According to a report by the Energy Conservation Center of Japan [(1)], industrial waste heat in Japan amounts to 2.7×10^5 Tcal/year. This amount is equivalent to approximately 70% of the yearly commercial and residential energy consumption in Japan. The report also mentions that the temperature level of 45% of the total waste heat is 100°C and below. Therefore, it is important to develop an efficient waste heat recovery system to generate power and/or electricity from low temperature heat sources with temperatures of less than 100°C. Further, the size of the recovery system must be fairly small because waste heat is a highly distributed energy source.

Thus far, various waste heat recovery systems have been proposed and developed. The most feasible and common technique is an organic Rankine cycle (ORC), in which a low boiling point organic fluid is used as a working fluid of the Rankine cycle. For example, Yamamoto et al. [(2)] described the effect of thermal properties of an organic working fluid on

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the turbine power output of an ORC system. Freepower Co., Ltd., introduced a commercial ORC system which converts waste heat into electricity [(3)]. In addition, Yamaguchi et al. [(4)] invented a unique Rankine cycle system using supercritical carbon dioxide (CO_2) as the working fluid, and they elucidated its potential as a solar thermal energy conversion system. For power generation from a small temperature difference between hot and cold sources (e.g. 15 K to 25 K in ocean thermal energy conversion, or OTEC), advanced cycles such as the Kalina cycle and the Uehara cycle have been developed. Almost all past researches and developments have been carried out for power outputs over 10 kW. For example, turbine powers of the ORC developed by Ebara Co., Ltd., and Freepower Co., Ltd., are up to 50 kW and 120 kW, respectively; those of OTEC systems and geothermal plants are usually over 30 kW [(5), (6)]. Accordingly, an ORC with an output of less than 1 kW has not yet been extensively studied and developed. However, the current energy and environmental conditions worldwide are such that there will soon be a requirement for a small ORC system which can be easily installed close to the location where waste heat is generated.

With the above-mentioned background, the aim of this study was to develop an ORC system with a small turbine power of less than 1 kW, which was equipped with a hot source with temperature ranging from 60 to 100°C and a cold source with temperature ranging from 10 to 30°C. In this study, an experimental ORC having a potential to produce a turbine/expander power of approximately 250 W was built and its fundamental characteristics were elucidated. First, the steady-state energy balance of the system was examined and the required turbine/expander efficiency was estimated in consideration of the pump power of the working fluid. The effect of heat exchange efficiency on the heat loss of the system was also evaluated. Then, the relationship between the expansion ratio and thermal efficiency was examined and discussed. A theoretical model of the ORC was employed to calculate the energy balance of the system.

Nomenclature

$\dot{m}$	: Mass flow rate, kg/s
W	: Work, kJ/kg
h	: Enthalpy, kJ/kg
Q	: Heat, kJ/kg
P	: Pressure, MPa
T	: Temperature, °C or K
$\dot{W}_P$	: Pump power, W
$\dot{W}_T$	: Turbine power, W

Subscripts

C	: Condenser
E	: Evaporator
P	: Pump
T	: Turbine/Expander
WF	: Working fluid
cs	: Condenser inlet
co	: Condenser outlet
cw	: Cold water
hs	: Evaporator inlet
ho	: Evaporator outlet
hw	: Hot water
pi	: Pump inlet
po	: Pump outlet
ti	: Turbine inlet

to	: Turbine outlet

Greek symbols

η_R	: Thermal efficiency
η	: Efficiency
γ_{int}	: Internal heat loss
γ_{ext}	: External heat loss

2. Theoretical ORC model

Figure 1 shows a schematic of the operation of a closed Rankine cycle. The ORC is a Rankine cycle in which an organic working fluid is used. The Rankine cycle consists of five main components: a pump, evaporator, turbine/expander, condenser, and working fluid. The evaporator and condenser are heat exchangers which absorb heat into the cycle and release it from the cycle [7]. The cycle is started by the pump pushing the working fluid to the evaporator. In the evaporator, the hot source water heats the working fluid up to the saturated or superheated vapour state. Then, the vapour expands and rotates the turbine/expander to produce power. After the vapour leaves the turbine, the cold source water cools and condenses the working fluid into the liquid state in the condenser. Then, the pump re-circulates the fluid.

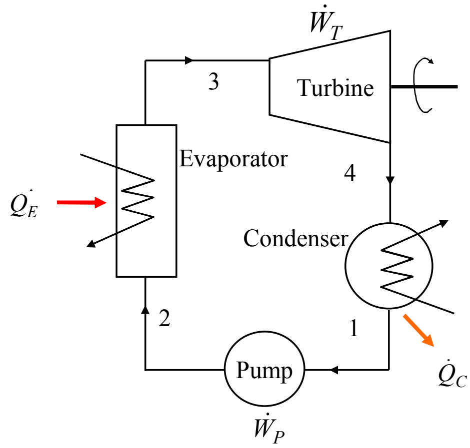

Fig.1 Schematic diagram of operation of closed Rankine cycle

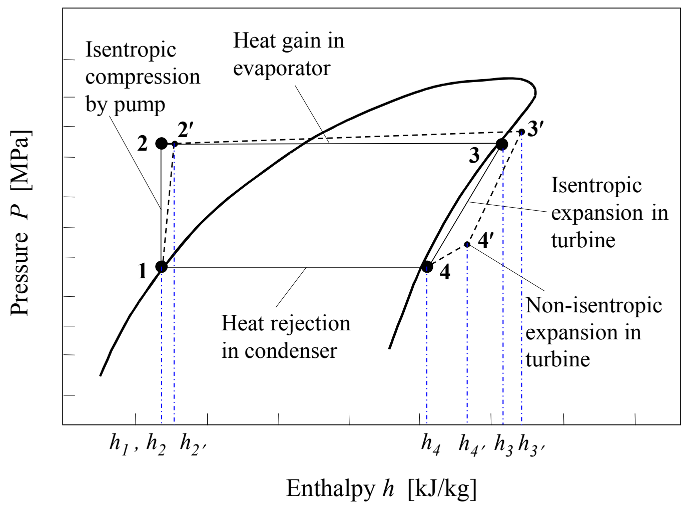

Fig.2 p–h diagram of closed Rankine cycle

Figure 2 shows the pressure–enthalpy (p–h) diagram corresponding to Fig.1. It also shows an ideal superheated Rankine cycle and an actual cycle.

Process 1→2 shown in Fig.1 and Fig.2 is the isentropic compression by the pump. The ideal pump power is given by

$$\dot{W}_P = \dot{m}_{WF}\,(h_2 - h_1) \tag{1}$$

In contrast, in the actual cycle, the compression by the pump is not exactly isentropic. Moreover, some losses also occur; hence, the pump power is given by

$$\dot{W}_P = \dot{m}_{WF}\,(h_{2'} - h_1)\,/\,\eta_P \tag{2}$$

Process 2→3 is the heating of the working fluid at a constant pressure in the evaporator. The heat absorbed by the working fluid is given by

$$\dot{Q}_E = \dot{m}_{WF}\,(h_3 - h_2) \qquad \{(h_{3'} - h_{2'}) \text{ in the actual cycle}\} \tag{3}$$

Process 3→4 is the isentropic expansion in the turbine/expander. The ideal turbine/expander power is given by

$$\dot{W}_T = \dot{m}_{WF}\,(h_3 - h_4) \tag{4}$$

On the other hand, process 3′→4′ is non-isentropic expansion from a certain state in the turbine/expander, which is generally observed in the actual cycle. Using the measured pressures and temperatures at the inlet and outlet of the turbine/expander, the turbine/expander power is given by

$$\dot{W}_T = \dot{m}_{WF}\,(h_{3'} - h_{4'})\;\eta_T \tag{5}$$

Process 4→1 is the cooling of the working fluid at a constant pressure in the condenser. The heat released from the working fluid is given by

$$\dot{Q}_C = \dot{m}_{WF}\,(h_4 - h_1) \qquad \{(h_{4'} - h_1) \text{ in the actual cycle}\} \tag{6}$$

The thermal efficiency of the ORC is calculated as follows:

$$\begin{aligned} \eta_R &= \frac{\text{Net power}}{\text{Total heat input}} \\ &= \frac{(\text{Turbine power } \dot{W}_T) - (\text{Pump power } \dot{W}_P)}{\text{Heat gain in Evaporator } \dot{Q}_E} \end{aligned} \tag{7}$$

In the above equations, enthalpies are calculated using the pressure and temperature measured in the experiment. REFPROP ver. 7 developed by the NIST [8] is used in the calculation. The mass flow rate of the working fluid is also measured in the experiment.

3. Experiment on 250 W ORC System

3.1 Experimental Setup

A small ORC system, which has the potential to produce a turbine power of approximately 250 W, was built to elucidate the cycle characteristics. Figure 3 shows a schematic diagram of the built ORC system. Table 1 shows the principle specifications of the system. HFC245fa was used as the working fluid because of its characteristic of being a dry liquid, which provides relatively higher efficiency than other fluids in low temperature ranges [9]. The critical temperature of HFC245fa is 427.16 K (approximately 154°C), which is considerably higher than the highest temperature expected in the present system. The hot source was water heated by an electric heater by circulation. Similarly, the cold source was water cooled by a chiller by circulation. Plate-type heat exchangers were used as both the evaporator and the condenser. The measuring points of temperature and pressure shown in Fig.3 correspond to the numbered points shown in Fig.1 and Fig.2. To elucidate the required turbine efficiency in consideration of the pump power, a clamp-on power meter was used to measure the pump power. All measured data were recorded to and monitored using an acquisition PC.

Currently, there is no suitable turbine/expander that can function efficiently under the conditions of the present system. Therefore, we installed two expansion valves instead of the turbine/expander, in order to simulate its functions. It is possible to adjust the expansion ratio in a certain range using these expansion valves. In contrast, a conventional small turbine/expander has a fixed expansion ratio. The simulated turbine power $\dot{W}_T$ is calculated from Eq.(5). The ideal turbine power is calculated from Eq.(4). Thus, the effect of the expansion ratio on the thermal efficiency was experimentally elucidated.

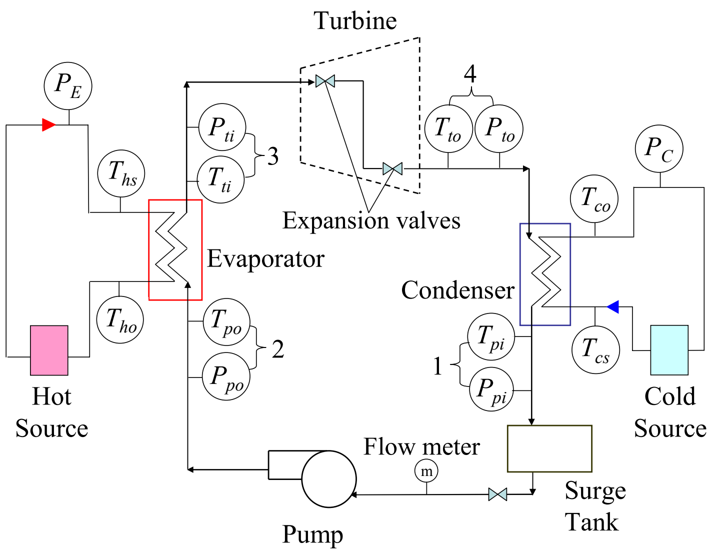

Fig.3 Schematic diagram of built ORC system

Table 1 Principle specifications of built ORC system

Working Fluid	R245fa ($CF_3CH_2CHF_2$), molecular weight:134.05 Boiling temperature: 14.9°C
Heat Exchangers (Evaporator and Condenser)	Brazed plate heat exchanger Heat conduction area: 0.4 m^2
Hot source	Water; temperature range: 60~100°C
Heater	Electrical heater (adjustable by variable resistor) Maximum output: 3.5 kW
Cold source	Water; temperature range: 10~20°C
Turbine/Expander	Simulated by expansion valve control
Working fluid pump	Diaphragm pump, maximum pressure: 0.9 MPa Maximum volume flow rate: 0.52 L/min

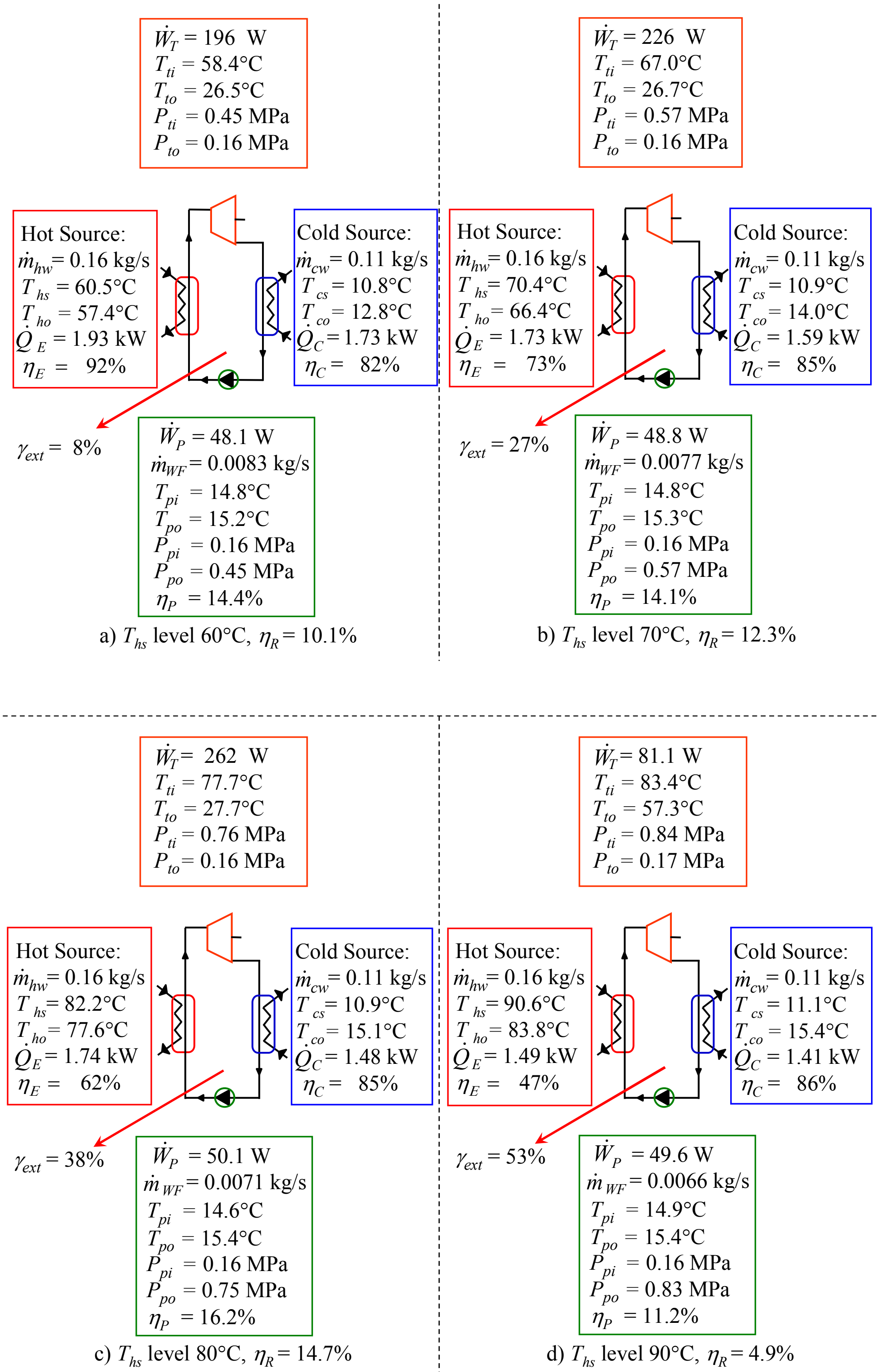

Fig.4 Steady-state energy balance of built ORC system

3.2 Energy Balance in Steady State

First, we examined the energy balance of the ORC system in the steady state. Figure 4 shows the energy balance at four levels of the hot source temperature T_{hs}: 60°C, 70°C, 80°C, and 90°C; each level is referred to as the 'T_{hs} level'. The temperature of the cold source was set to 10°C throughout the experiment. It should be noted that the actual average temperatures of both the sources fluctuated within ±2°C. The flow rates of the working fluid and hot and cold sources were constant during the experiment at each T_{hs} level. Under the given conditions, the working fluid absorbed heat of up to approximately 2 kW in the evaporator and released heat of up to approximately 2 kW in the condenser.

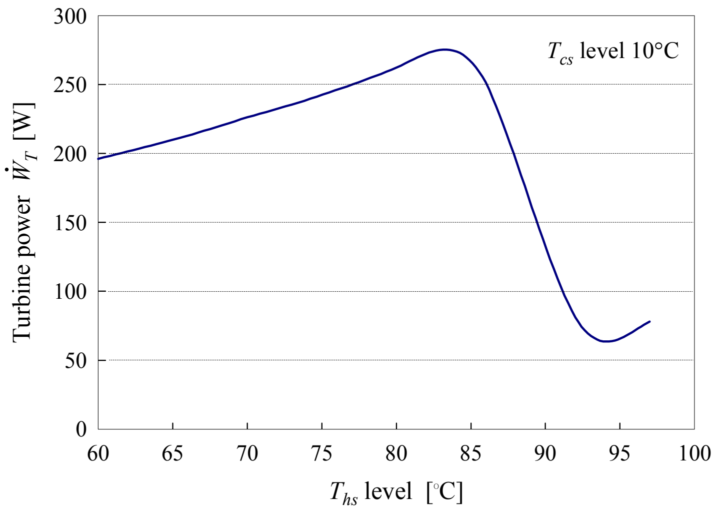

Fig.5 Relationship between turbine power and hot source temperature level under given conditions of present experiment

Figure 5 shows the turbine power calculated from Eq.(5) for the given hot source temperatures in the experiment. Here, it should be noted that the turbine efficiency η_T was assumed to be 100%. The maximum turbine power $\dot{W}_T$ = 262 W was obtained at the T_{hs} level of 80°C. The power tended to decrease for T_{hs} greater than 85°C, and the minimum turbine power $\dot{W}_T$ = 81 W was obtained at the T_{hs} level of 90°C. This drastic decrease was mainly caused by an inappropriate expansion, as discussed in a subsequent section.

Under the given conditions, the measured pump power ranges from 48 W to 50 W, while the ideal pump power calculated at the T_{hs} level of 80°C using Eq.(1) is only $\dot{W}_P$ = 8.2 W. The pump efficiency calculated from Eq.(2) results in a rather small η_P value of 16.2%. Thus, the thermal efficiency of the present system with an ideal turbine is estimated to be η_R = 14.7%, while the ideal thermal efficiency with both an ideal turbine and an ideal pump is calculated to be η_R = 16.4% from Eqs.(1), (3), (4), and (7). A mismatch in pressure and in the flow rate between those required by the system and those resulting from the pump would result in low pump efficiency. Moreover, a small pump generally tends to result in low pump efficiency. Consequently, this fact seriously aggravates the thermal efficiency of a small ORC system.

Considering the heat balance of the system, the heat exchange efficiency of the evaporator was significantly influenced by the hot source temperature T_{hs}. It decreased from η_E = 92% at the T_{hs} level of 60°C to η_E = 47% at the T_{hs} level of 90°C. The authors expected that this decrease was caused by the decrease in the flow rate of the working fluid due to a pressure rise in the evaporator. Because of this effect, the heat transfer area of the evaporator must be sufficiently large at all expected T_{hs} levels. However, the heat exchanger efficiency does not affect the thermal efficiency according to the present definition of 'internal' thermal efficiency, which accounts only for heat successfully transferred to the working fluid. For example, in the case of the maximum thermal efficiency at the T_{hs} level of 80°C, the heat released by the hot water in the evaporator was 2.81 kW, while the heat absorbed by the working fluid was $\dot{Q}_E$ = 1.74 kW. From this heat gain, heat energy of 262 W can be maximally converted into power by the turbine. Further, heat energy of 1.48 kW was released into the cold water in the condenser. Consequently, a hot source energy of 1.1 kW was lost due to heat exchange in the evaporator. In this case, the internal heat loss defined by

$$\gamma_{int} = (\dot{Q}_E - \dot{W}_T - \dot{Q}_C) / \dot{Q}_E \tag{8}$$

is approximately zero; however, the external heat loss defined by

$$\gamma_{ext} = ((\dot{Q}_E / \eta_E) - \dot{W}_T - \dot{Q}_C) / (\dot{Q}_E / \eta_E) \tag{9}$$

is 38%. In the other cases, γ_{ext} = 8%, 27%, and 53% at T_{hs} levels of 60°C, 70°C, and 90°C, respectively. The condenser also loses energy due to heat exchange. At all T_{hs} levels in the experiment, approximately 15% of the cold source energy was not transferred to the working fluid. Although previous studies on ORC, except for those on OTEC, mostly tended to treat the heat exchange efficiency as an external phenomenon of the Rankine cycle, it is obvious that the decrease in the heat exchange efficiency aggravates the 'external' thermal efficiency, i.e. the waste heat recovery rate in an actual system, particularly at low temperature levels.

3.3 Required Turbine and Pump Efficiencies

The observed energy balance shown in Fig.4 implies that the turbine efficiency η_T and pump efficiency η_P dominate the thermal efficiency of the small ORC system for low temperature waste heat recovery. In the built ORC system, the turbine power $\dot{W}_T$ must be greater than 50 W in order to achieve more than zero thermal efficiency, because the measured pump power $\dot{W}_P$ is approximately 50 W. Figure 6 shows the relationship between the pump power and the turbine power calculated from Eq.(2) and Eq.(5), respectively, at given efficiencies in the present system. It is obvious that even if the pump efficiency is considered to be greater than 10%, the turbine efficiency should be more than η_T = 70% to achieve positive waste heat recovery. Obviously, at high η_P, required $\dot{W}_P$ is low. For example, in the case of $\dot{W}_P$ = 37 W at η_P = 15%, the critical turbine efficiency for achieving positive waste heat recovery is $\dot{W}_T$ = 40 W at η_T = 50%.

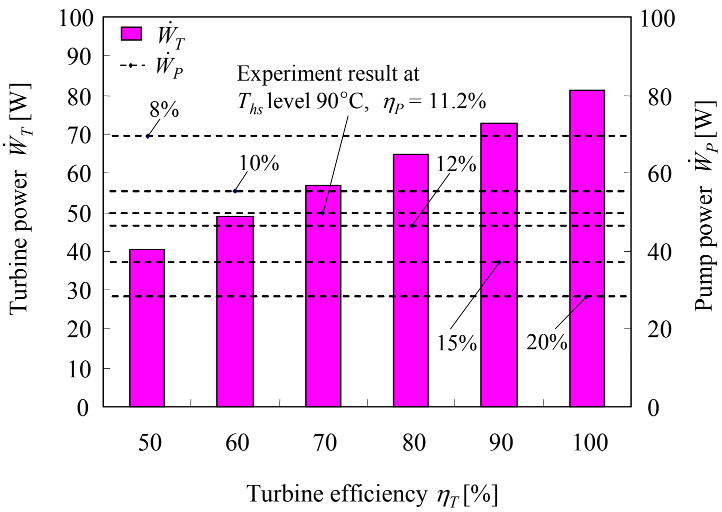

Fig.6 Relationship between turbine power and pump power at given efficiencies

3.4 Effect of Expansion Ratio on Thermal Efficiency

An ideal expansion of the vapour of the working fluid in the turbine/expander contributes to higher turbine efficiency and thermal efficiency of the system. As mentioned above, the expansion process must be isentropic for achieving an ideal efficient cycle. However, in an actual system, the expansion process tends to be non-isentropic due to heat loss and insufficient expansion.

Figure 7 shows the comparison between the expansion ratio for isentropic expansion and experimental expansions at various T_{hs} levels. The ideal expansion ratio was calculated from the ratio of the turbine inlet and outlet pressures to achieve η_T = 100% under the following assumptions: $T_{ti} = T_{hs}$; $T_{to} = T_{cs}$. P_{ti} is the saturated vapour pressure at T_{ti} and P_{to} is the pressure at T_{to} with the same entropy at the turbine inlet.

While carrying out the experimental expansions, we manually tuned two expansion settings—referred to as Exp. I and Exp. II—by controlling the two expansion valves. The experimental expansion ratio was estimated as the ratio of the measured turbine inlet and outlet pressures, P_{to}/P_{ti}. It should be noted that the valve settings for Exp. I and Exp. II were fixed during the experiment at each T_{hs} level ranging from 60°C to 90°C. In other words, in order to simulate a turbine/expander with non-variable expansion, which has been traditionally employed in a conventional ORC system, the valve openings were maintained constant at every T_{hs} level. As shown in Fig.7, the experimental expansion ratios are obviously lower than the ideal expansion ratio. The ideal expansion ratio increased from 5 up to 12 with an increase in the T_{hs} level from 60°C to 90°C. On the other hand, for Exp. I, the ratio increased from 3 to 5 with an increase in the T_{hs} level. The ratio for Exp. II was even lower than that for Exp. I, because in Exp. II, the valve openings were tuned to be relatively larger than those in Exp. I.

Figure 8 shows the comparison of thermal efficiencies shown in Fig.7. Exp. I and Exp. II show the peak thermal efficiency at different hot source temperatures—15% at 85°C for Exp. I and 10% at 70°C for Exp. II. In contrast, the ideal expansion ratio increases linearly with an increase in the hot source temperature. When the hot source temperature exceeds T_{hs} = 85°C, the experimental thermal efficiency for Exp. I decreases drastically from η_R = 15% at T_{hs} = 85°C to η_R = 5% at T_{hs} = 90°C. The thermal efficiency for Exp. II also decreases from η_R = 10% at T_{hs} = 70°C to η_R = 7% at T_{hs} = 80°C. This decrease is because of the fact that the expansion ratio at the turbine is not as high as the ideal expansion ratio shown in Fig.7.

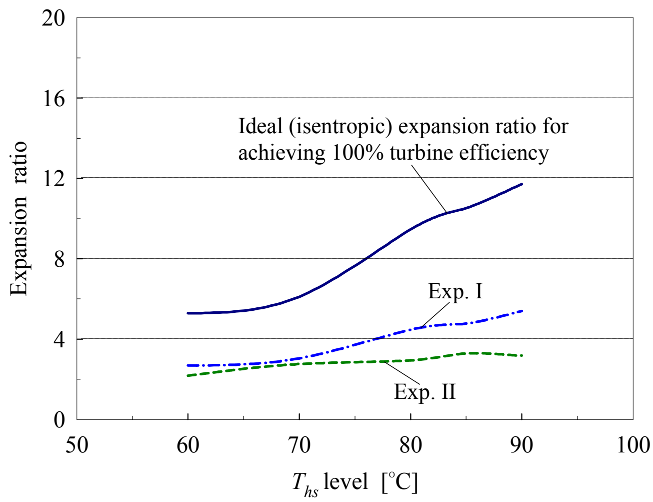

Fig.7 Relationship between expansion ratio and hot source temperature level for settings Exp. I and Exp. II in comparison to theoretical ideal expansion

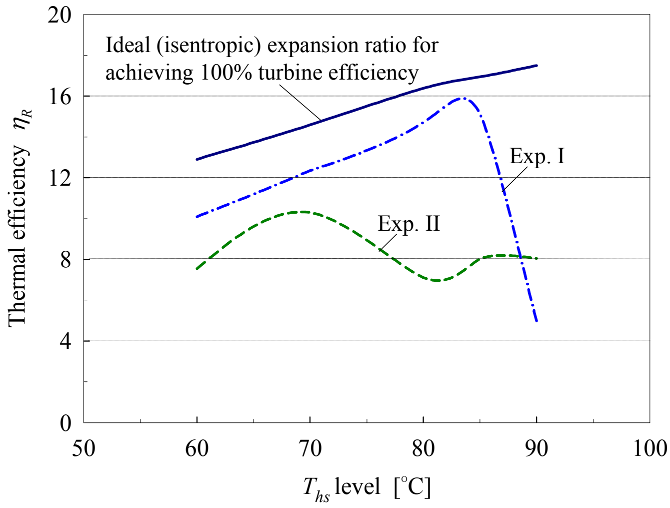

Fig.8 Effect of expansion ratio on thermal efficiency at various hot source temperature levels

The above-mentioned results indicate that for a particular fixed setting of an expansion mechanism, there is one appropriate temperature level at which the system achieves the highest thermal efficiency towards the theoretical limit. In other words, the variable expansion mechanism is indispensable for maintaining high efficiency at various levels of the hot source temperature, i.e. for various differences between temperatures of the hot and cold sources. Some of the expected hot sources such as waste heat from vehicles and solar thermal energy show time dependence and unstable characteristics with respect to their temperature level. Therefore, to maintain high thermal efficiency in the case of such hot sources, high efficiency of the turbine/expander must be achieved by adjusting the expansion ratio. For example, if we employ a velocity-type turbine such as a radial turbine in the ORC system, we should consider using a variable nozzle design at the turbine inlet. By this design, we can adjust the expansion ratio and possibly maintain high turbine efficiency according to the operating temperature level. If we employ a displacement-type expander such as a scroll expander or a screw expander, a variable inlet/outlet volumetric control mechanism should be designed to fit the optimal expansion ratio at the operating temperature level.

4. Conclusions

In this study, we built a small ORC system by using HFC245fa as a working fluid and examined its characteristics such as steady-state energy balance, required turbine and pump efficiencies, and expansion ratio. The conclusions of our study are as follows:

1. To maintain high thermal efficiency even when the temperature difference between the hot and cold sources varies during operation, it is indispensable to employ a variable expansion mechanism by which the expansion ratio of the turbine/expander can be adjusted to fit the optimal expansion ratio at the operating temperature level. The expansion ratio and temperature fluctuation should be taken into account before designing the turbine/expander mechanism of the small ORC system.
2. In a small ORC system with a turbine power of 250 W, the pump power reaches up to 50 W due to the rather low pump efficiency. Pump efficiency drop requires high turbine/expander efficiency. Assuming that the pump efficiency is greater than η_P = 10%, the turbine efficiency should be greater than η_T = 70% in order to achieve

positive waste heat recovery. When η_P is high, the required pump power $\dot{W}_P$ is low.

3. In the built 250 W ORC system, at hot source temperature levels of 60°C and 90°, approximately 8% and 53% of the heat, respectively, is released (heat loss) into ambient atmosphere. This heat loss occurs due to heat exchange loss in the evaporator.
4. The heat exchange efficiency of the evaporator decreases from 92% at a hot source temperature level of 60°C to 47% at a hot source temperature level of 90°C under the condition of constant volume flow rates of the hot source fluid and working fluid. This decrease does not affect the internal thermal efficiency of the built system; however, it adversely affects the external thermal efficiency.

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