EXPERIMENTAL STUDY ON PARALLEL EXPANDER ORGANIC RANKINE CYCLE WITH TWO DIFFERENT CAPACITY EXPANDERS

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ABSTRACT

Recently, an organic Rankine cycle (ORC) with dual expanders in parallel called as parallel-Expanders ORC (PE-ORC) has been proposed for more efficient waste heat recovery in applications in which there are large heat variations such as distributed energy system with multiple internal combustion engines. This study describes a PE-ORC adopting two expanders with different capacities. The system could have three operating modes according to the operated expanders. An ORC loop with 1 kW and 3kW class expanders in parallel are prepared for the test. The ORC test bench has a 100kW heater as the heat source, and an air-cooled chiller as the heat sink. R245fa was used as working fluid. In order to evaluate the performance characteristics of the system for each operating mode, efficiencies and shaft powers were obtained under various evaporative heat transfer conditions. The appropriateness of utilizing proposed ORC system and optimal operation mode which can produce higher power output will be discussed with experimental results.

1. INTRODUCTION

Some applications including internal combustion engine (ICE) have large heat variation of sources. Single-expander ORC system in those fields could be operated at off-design points, and thus the efficiency could be severely reduced, or the ORC system might not be operated (Choi and Kim, 2013). Recently, an organic Rankine cycle (ORC) with multiple expanders in parallel called as parallel-Expanders ORC (PE-ORC) has been proposed for more efficient waste heat recovery in applications in which there are large heat variations such as distributed energy system with multiple internal combustion engines (Yun *et al.*, 2015). They showed the feasibility of the PE-ORC by experimental evaluation of simple PE-ORC which had two identical scroll expanders. The study showed the PE-ORC with two identical expanders could have two design points which can achieve the maximum performance. The number of the design points and the number of operating mode depend on the number of expander and each expander capacity.

In this study, a PE-ORC adopting two expanders with different capacities is tested, and thus the tested system has three design points and three operating modes. The performance for each operating mode has been evaluated.

2. EXPREIMENTAL SETUP

A schematic diagram of ORC test bench adopting two expanders with different capacities is given in Figure 1. The ORC loop consists of two expander modules, an evaporator, a condenser, a working fluid pump, and a liquid receiver. Two scroll expanders, Expander 1 and Expander 2, with different capacities were assembled in parallel between distribution and mixing chambers. The separated working fluid through the distribution chamber flows into both expanders, and merges in mixing chamber after expansion, and then is fed into the condenser. Manual open-close valves for switching between operating modes are installed at the entrances of both expanders. Brazed plate heat exchangers were used for the evaporator and condenser, respectively. The feed pump for working fluid was a volumetric Diaphragm type pump whose rotational speed was controlled using a frequency drive. Pressurized hot water heated by an electric heater was used as heat source, and an air-cooled chiller was connected with condenser in order to supply cooling water at a constant temperature. A 3-D schematic model of ORC system and Expander 1, and a photograph of Expander 2 are shown in Figure 2.

Technical overview of each expander unit is summarized in Table 1. Expander 1 was an oil-free open-drive scroll air compressor has been adapted to run in an expander mode by reversing the flow direction. A container for preventing leakage was designed as shown in Figure 2 (Wang *et al.*, 2011). Expander 2 is a commercially available oil-free scroll expander unit manufactured by Air Squared, Inc. The shaft of each expander is directly coupled with torque sensor to measure the shaft power and permanent magnetic motor to control the rotational speed in series. The specifications of sensors are listed in Table 2. R245fa was used as the working fluid because of its thermodynamic suitability for low-temperature heat recovery, non-flammability, and lack of toxicity (Declaye *et al.*, 2013).

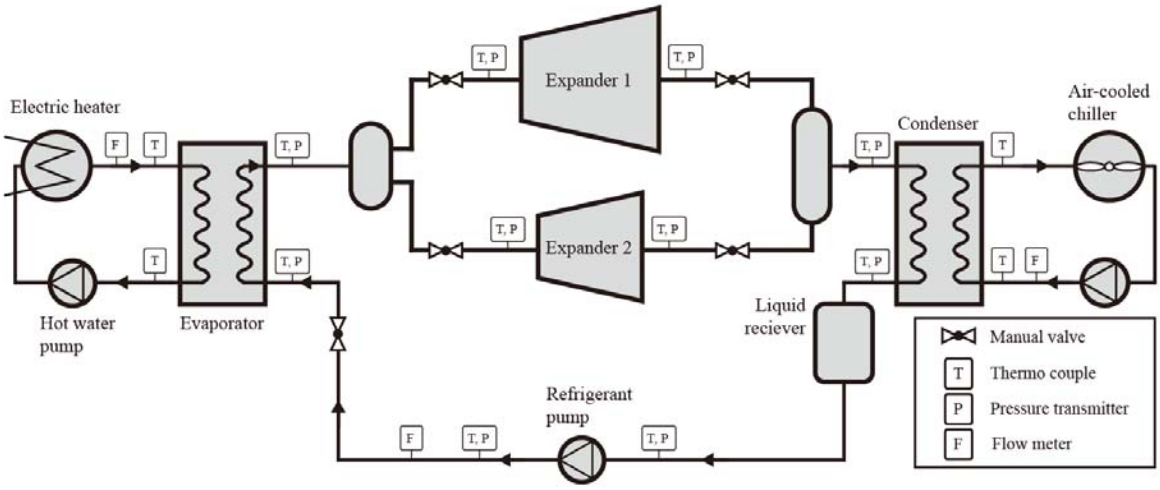

Figure 1: Schematic diagram of Dual-expander ORC test rig

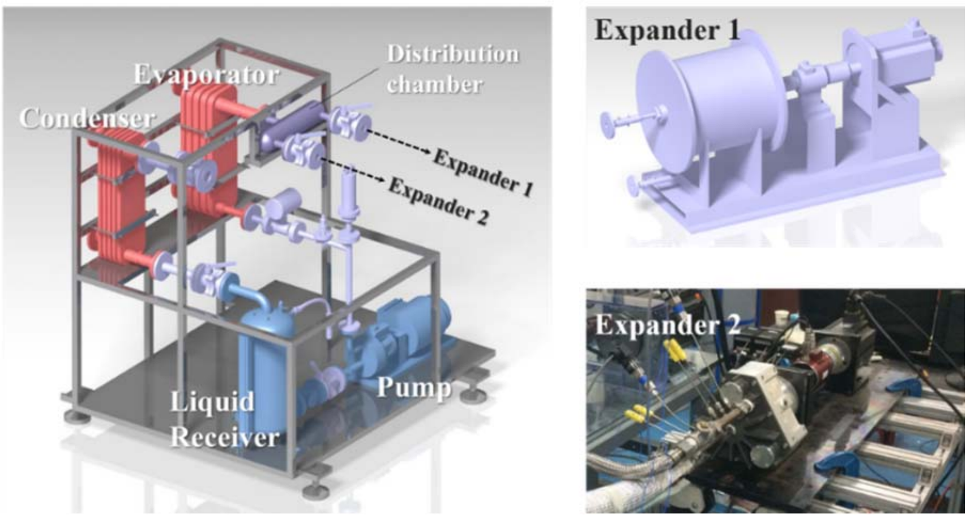

Figure 2: Schematics of ORC system and Expander 1, and a photograph of Expander 2

Table 1: Technical overviews of Expander 1 and 2

Component	Expander 1 (Modified from oil-free scroll air compressor)	Expander 2 (Commercial oil-free scroll expander)
Model numbers	BC-KL52H	E15H22N4.25
Manufacturer	Kyungwon Machinery Co, LTD.	Air Squared, Inc.
Max. pressure	10 bar (Compressed air)	13.8 bar
Expansion ratio	4	3.5
Rated speed	2900 rpm	3600 rpm
Output (nominal)	3.7 kW (Motor spec.)	1 kW
Swept volume/rev	30.34 cm^3/rev	12 cm^3/rev

Table 2: Specifications of sensors used in test rig

Measurement	Type	Range	Accuracy
Pressure	Piezo resistive	0-20 bar	± 0.5 % F.S.
Flow rate	Oval gear type	0.5-30 l/min	± 0.5 % F.S.
Rotational speed	Magneto type	0-10000 rpm	± 1 rpm
Torque	Strain gauge	0-50 N-m	± 1 % F.S.

3. RESULTS AND DISCUSSION

In this study, only Expander 2 has been tested because the Expander 1 is in modification for improving its tightness. The preliminary tests were conducted according to various expander inlet pressure conditions under the fixed shaft rotational speed. The preliminary test conditions for Expander 2 summarized in Table 3. For assessment of expander performances, several factors such as filling factor, power output, and isentropic efficiency should be considered with expander model used in previous studies (Lemort *et al.*, 2009).
The filling factor represented the volumetric performance of the expander, and is defined as given in Equation (1).

$$FF = \frac{\dot{m}_{wf,meas} \cdot v_{ex,in}}{\dot{V}_s} \tag{1}$$

Measured shaft power and internal expansion power can be calculated by using Equation (2) and (3).

$$\dot{W}_{ex,meas} = \tau \cdot \omega \tag{2}$$

$$\dot{W}_{int} = \dot{m}_{int}[(h_{ex,in} - h_{ad}) + v_{ad}(P_{ad} - P_{ex,out})] \tag{3}$$

The total mass flow rate entering the expander can be defined as sum of internal and leakage mass flow rate, and is expressed in Equation (4)

$$\dot{m} = \dot{m}_{int} + \dot{m}_{leak} = \frac{\dot{V}_s}{v_{ex,in}} + \dot{m}_{leak} = \frac{N \cdot V_s}{v_{ex,in}} + \dot{m}_{leak} \tag{4}$$

The leakage mass flow rate can be calculated by mass and energy conservation equations through isentropic converging nozzle, which has a lumped leakage area (A_{leak}) at the nozzle throat, as given in Equation (5).

$$\dot{m}_{leak} = \frac{A_{leak}}{v_{leak,thr}} \sqrt{2(h_{ex,in} - h_{leak,thr})} \tag{5}$$

Table 3: Preliminary testing condition for Expander 2

Unit	N_{ex} [rpm]	$P_{ex,in}$ [bar]	$P_{ex,out}$ [bar]	$T_{H,in}$ [°C]	$T_{C,in}$ [°C]
Expander 2	3600	6.84-13.8	1.7-1.98	120	18

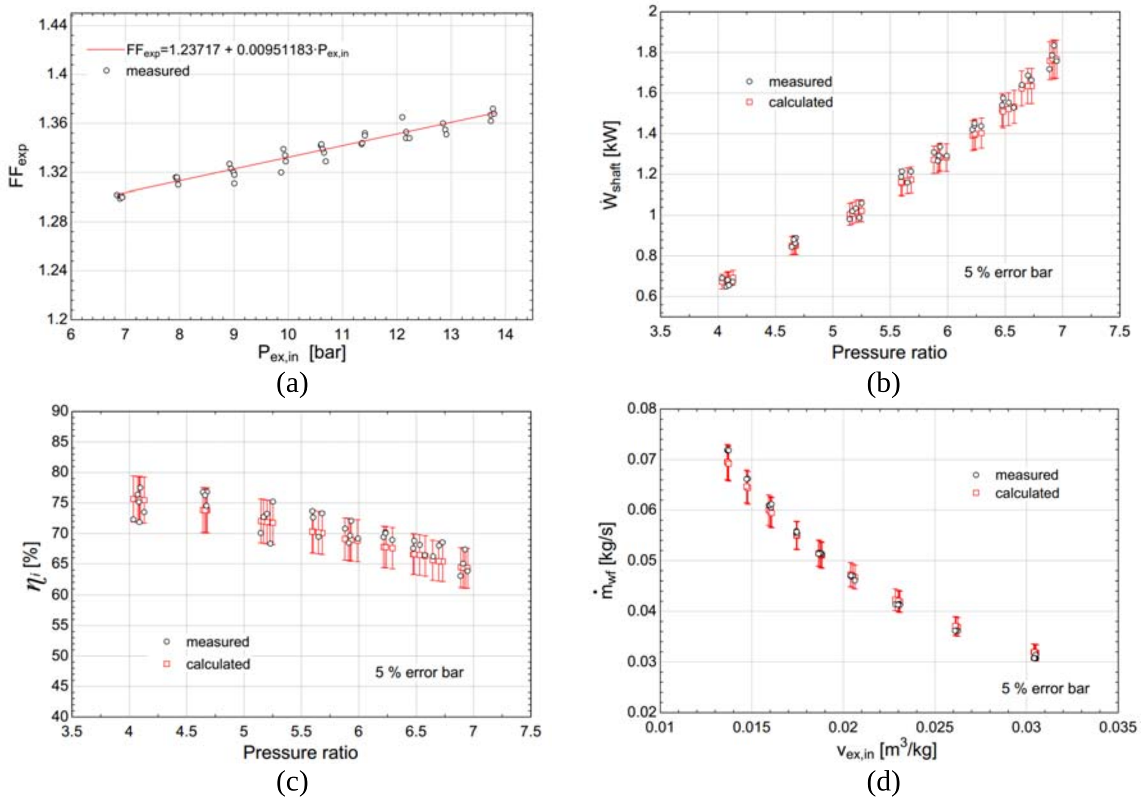

Figure 3: Measured and calculated results of performance characteristics for Expander 2

The lumped leakage area (A_{leak}) of Expander 2 is empirically identified to 2.85 mm^2. The nozzle outlet pressure should reach to the critical pressure due to the narrow throat area. The critical pressure at the throat can be calculated by Equation (6).

$$P_{leak,thr} = P_{ex,in}\left[\left(\frac{2}{\gamma+1}\right)^{\left(\frac{\gamma}{\gamma-1}\right)}\right] \quad (6)$$

In this study, the thermal losses by heat transfer could be neglectable because the temperatures and pressures was measured direct after inlet and outlet of expander. The expander isentropic efficiency is defined as given in Equation (7).

$$\eta_i = \frac{\dot{W}}{\dot{m}_{wf,meas}\cdot\Delta h_i} \quad (7)$$

Figure 3 shows the measured and calculated results of performance characteristics for Expander 2. The filling factor linearly increases as the expander inlet pressure increases. As shown in Figure 3 (b)-(d), the results calculated through the expander model closely match the experimental results. The maximum shaft power measured was 1.8 kW at the maximum pressure ratio. The maximum isentropic efficiency measured was about 77 % at a pressure ratio of 4.1. The calculated mass flow rates of working fluid by expander model are in good agreement with the experimental results.
With the confidence of the analytical expander model, the performances of Expander 1 and Expander 2 have been determined by using the expander model, respectively. The leakage area of Expander 1 is assumed to 5.86 mm^2. In order to predict the performance characteristics of PE-ORC system cycle

analysis was conducted and, main parameters used for the cycle analysis are summarized in Table 4. Figure 4 shows the calculated isentropic efficiencies against evaporative heat transfer using the expander model and simple cycle model.
In simple cycle model, the cycle efficiency was calculated by Equation (8).

$$\eta_c = \frac{(\dot{W}_{in} - \dot{W}_{pp})_{total}}{\dot{Q}_{ev,total}} \tag{8}$$

The simulated results of power outputs and cycle efficiencies against the evaporative heat transfer are graphically represented in Figure 5. The selected mode changing points were for maximization of the total power output and cycle efficiency.

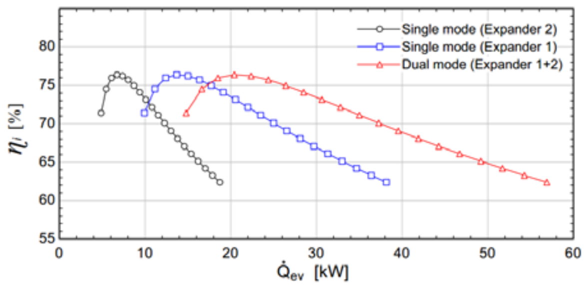

Figure 4: Predicted isentropic efficiencies for three operating modes (Single modes and Dual mode)

Table 3: Parameters for cycle model

$P_{ex,in}$ [bar]	$P_{ex,out}$ [bar]	$T_{superheat}$ [°C]	$T_{subcool}$ [°C]	η_{pp} [%]
4.5-13.8	1.8	5	3	80

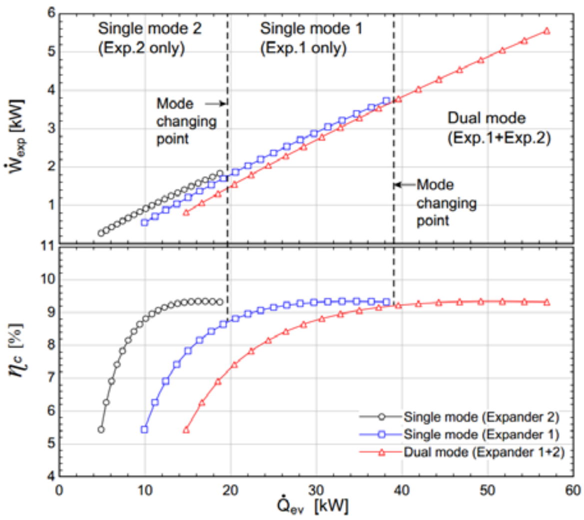

Figure 5: Predicted power outputs and cycle efficiencies for three operating modes (Single modes and Dual mode) and mode changing points for maximizing cycle efficiency

5. CONCLUSIONS

In this study, the preliminary test of a PE-ORC adopting two expanders with different capacities is carried out, and the performance for each operating mode has been predicted by the expander model and the simple cycle model.

- Expander 2 has been tested according to various inlet pressure conditions for validation of expander model.
- The calculations by the expander model closely match the experimental results.

- The performance characteristics of PE-ORC were predicted by simple cycle model, and mode changing points selected for maximizing cycle efficiency.

NOMENCLATURE

h	enthalpy	(kJ/kg)
η	efficiency	(–)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(bar)
$\dot{Q}$	heat	(kW)
T	temperature	(^{o}C)
v	specific volume	(m^3/kg)
$\dot{V}$	volumetric flow rate	(m^3/s)
$\dot{W}$	work	(kW)

Subscript

ad	adapted
c	cycle
ev	evaporator
ex	expander
i	isentropic
in	inlet
int	internal
meas	measured
out	outlet
pp	pump
wf	working fluid

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ACKNOWLEDGEMENT

This research was supported partially by Basic Science Research Program through the National Research Foundation of Korea (NRF) funded by the Ministry of Education (NRF-2013R1A1A2012173), partially by the Energy Efficiency and Resources of the Korea Institute of Energy Technology Evaluation and Planning (KETEP) granted financial resource from the Ministry of Trade, Industry & Energy, Republic of Korea (No. 20112010100030).