Experimental study on power generation plant of a 1 kW small-scale Organic Rankine Cycle system using R290

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Abstract
This work presents an experimental investigation of a small-scale cryogenic Organic Rankine Cycle (ORC) power generation plant for cold energy utilization using scroll expander and R290. The fresh water and liquid nitrogen were used as heat source and heat sink, respectively. Furthermore, expander gross output power and isentropic efficiency, system thermal efficiency, and cold energy utilization efficiency were calculated as a principle to evaluate the overall system performance. The experimental results demonstrate that increasing pressure drop and heat source temperature will benefit the system performance in some extent, and the pressure drop exhibits a relative sensitivity to heat source temperature about the system performance. Moreover, the expander gross output power will get the peak value while the heat source and heat sink in the best matching. When the mass flow rate of liquid nitrogen at 120 kg/h, optimum pressure drop is 1.02 Mpa, and the corresponding expander gross output power, thermal efficiency, and cold energy utilization efficiency were 638.2W, 0.0586, and 0.0976, respectively. The optimum heat source temperature is 50°C under liquid nitrogen mass flow rate of 120 kg/h, and the corresponding expander gross output power, thermal efficiency, and cold energy utilization efficiency were 674.2 W, 0.0594, and 0.099, respectively.

KEYWORDS
cold energy utilization, LNG, operational parameters, ORC power generation, R290

1 | INTRODUCTION

Currently, the global warming effect has been approved worldwide, and the greenhouse gas emissions from conventional power generation plant have caused more and more concerns.\(^1\) The natural gas, as a clean energy, which has a characteristic of high calorific value and less greenhouse gas emission when it is burned as a fossil fuel.\(^2,3\) The best way to storage and long-distance transportation across ocean of the natural gas is usually liquefied into liquefied natural gas (LNG) at a temperature of approximately \(-162^\circ\)C and atmospheric pressure,\(^4,5\) which can increase the energy density and easy for transport. However, LNG must be gasified into natural gas at ambient temperature in LNG terminals prior to use.\(^6\) The regasification process of LNG will release a lot of 830 KJ/Kg cold energy by absorbing heat from the seawater,\(^7\) air, and industry waste heat in the LNG terminals, which will not

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only waste those valuable energy but also can cause “cold pollution” near the LNG terminals area. Consequently, how to utilize those LNG cold energy is gradually becoming a hot spot research field.

At present, the method of LNG cold energy includes power generation, air separation, cryogenic carbon dioxide capture, desalination, dry ice manufacture, rubber crushing, food freezing, data center cooling, and so on. Among these methods, power generation technology is one of the most efficient and economical solutions in the LNG cold energy industry. Cryogenic Organic Rankine Cycle (ORC) is considered to be one of the most effective ways to convert LNG cold energy to electrical power. The performance of ORC system will directly determine the utilization efficiency of LNG cold energy. As a result, the researches and studies about ORC to utilize LNG cold energy mainly focused on the system cycle configuration, low-boiling working fluid selection, and operational parameters optimum.

In order to improve the efficiency of LNG cold energy power generation system, some solutions were to be made on the cycle configuration of the conventional ORC system. Ma et al. based on the LNG cold energy release law and its gasification characteristics established a five-stage ORC stimulation model. Compared with the single-stage ORC, the exergy recovery rate, cold energy utilization, and net power generation of five-stage ORC are increased by 35.47%, 14.32%, 19.15%, and 281.7%, respectively. Li et al. proposed and analyzed the improved ORC system integrates a precooler and solar collections into a double-loop combined system. The simulation results proved that the net power output and system thermal efficiency are improved by 9.3% and 7.33% in comparison with the conventional double-loop combined cycle system. Bao et al. proposed a two-stage condensation Rankine Cycle (TCRC) system and used R290 as working fluid. Compared with the combined cycle in the conventional LNG cold energy power generation method, the net power output, thermal efficiency, and exergy efficiency of this new system are, respectively, increased by 45.27%, 42.91%, and 52.31%. Tomków et al. proposed a novel multi-stage power generation cycle using ethane-krypton binary mixture as working fluid. Their numerical simulations show that exergy efficiency can be raised to 19.3% and obtained thermal efficiency of 10.4%.

On the contrary, some of the solutions aim to find out the relationship between working fluid and ORC system performance. Yu et al. investigated 22 working fluids for ORC utilizing LNG cold energy. Adopted particle swarm optimization (PSO) algorithm to optimum 5- and 7-dimensional search spaces for the combined systems operated across and below ambient temperature. In addition, according to stimulation results, the most energy-efficient working fluids are R125, R134a, R290, and R1270. He et al. proposed a cryogenic ORC system using different pure or binary working fluids. The results indicated that R127 exhibited the highest net power output (NPO) of 89.34 kJ/kg and exergy efficiency of 18.96%, while the highest thermal efficiency of 14.51% at R1270 working fluid. Moreover, the overall performance was significantly improved by using binary mixture working fluid (R1270 of 30% and C3H8 of 70%) at 4000 kPa LNG vaporization pressure. Lee et al. proposed a ORC system considering the thermodynamic and safety aspects is explored with a multi-objective optimization methodology. Selected 6 working fluid in three categories including pure component (C2 and C3), binary component (NH3-H2O and R14-C3H8), and ternary component (R601-R23-R14) and R30-R23-R14). As a result, C3 provided higher exergy efficiency than C2. R14-C3H8 was more efficient and safer than NH3-H2O. In the case of ternary components, both working fluids showed similar ranges of exergy efficiency.

In addition, the operating parameters were also the key factors to ORC system. Ma et al. proposed a multi-stage Rankine Cycle and found that the heat source temperature has a very small effect on the exergy recovery rate, but it has a great influence on the net power generation and cold energy utilization. Jang et al. used a compact ORC system with R245fa working fluid and heat source temperature in the range of 100–140°C. They found that the mechanical efficiency of expander increased non-linearly when decreasing the heat source temperature and increasing the pressure difference of expander. Unamba et al. presented results from tests on a 1kW ORC system and used electric oil-heater acted as the heat source in the range of 120–140°C. Analysis of the results suggests that increased heat source temperatures will improve the overall cycle efficiency. Yang et al. studied on 3kW ORC using R245fa as working fluid and fixed heat source temperature at 100°C. Li et al. investigated the influent of R123 mass flow rate on the efficiency of a regenerative ORC. Minea analyzed a 50kW ORC machine, which using industrial waste or renewable energy at temperatures varying between 85°C and 116°C. Sun et al. established a supercritical ORC system with an unconventional condenser. They found that the highest efficiency at different heat source temperature was 17.89% for R290, 21.26% for R290, and 25.35% for R600a at 100°C, 150°C, and 200°C, respectively.

Through the above literature review, it can be found that most studies about LNG cold energy power generation based on ORC system still focused on the stage of theoretical research and numerical simulation. However, there is some doubt whether these findings are consistent with practical engineering applications. Therefore, the purpose of this work is to present experimental investigation of 1kW cold energy generation plant based on
cryogenic ORC under different operation parameters. We focus on the effects of the pressure drop and heat source temperature on system performance.

2 | EXPERIMENTAL SYSTEM INVESTIGATIONS

2.1 | Experimental assumption

Before the experiment, the following assumptions were made to simplify the investigation:

1. From the perspective of safety, fire, and explosion-proof consideration, the heat sink was selected liquid nitrogen instead of LNG
2. Steady-state was determined based on the evaporating pressure and condensing pressure that remain relatively stable within 2 min
3. The pressure drop in the pipe of the system was ignored
4. Heat transfer between the environment and the system was neglected

2.2 | Working fluid selection

Due to the cryogenic characteristics of LNG, the condensation temperature is much lower than the ambient temperature, which results in totally different operating conditions compared with the conventional ORC system. The desired working fluid should have characteristics as below:

1. No ozone depletion potential (ODP) and low global warming potential (GWP)
2. Lower freezing temperature and higher thermal conductivity
3. High critical temperature and chemical stability
4. Non-toxicity, easily available, and low cost

Referring to the literature, R290 is selected as working fluid, which physical parameters are shown in Table 1.

2.3 | Experimental setup

Figure 1 shows a schematic diagram of the experimental with ORC power generation system. Figure 2 shows the picture of the experimental system layout and main facilities of the system. The system consists of three circuits: heat source circuit, R290 circuit, and heat sink circuit. In order to reduce the heat loss in this circuit, all the elements and pipes in the system should be packaged with thermal insulation materials.

2.3.1 | Heat source circuit (red line)

The water pump transfers the heated water into the evaporator and releases the heat to R290. The liquid R290 from

| TABLE 1 | R290 physical parameters
| Items     | Values |
|-----------|--------|
| Critical temperature | 96.7°C |
| Critical pressure | 4.25 Mpa |
| Boiling temperature | −42.2°C |
| Freezing temperature | −188°C |
| ODP | 0 |
| GWP | 3 |

FIGURE 1  Schematic diagram of the experimental system
the WF pump was heated into vapor in the evaporator. The temperature of heating water varies from 20°C to 60°C.

2.3.2 R290 circuit (green line)

This circuit includes main components of the evaporator, vapor-liquid separator, scroll expander, condenser, reservoir, and WF pump. R290 in the reservoir was pressurized by the WF pump and then delivered into the tube side of the evaporator. In order to reduce the pulsation effect, an air chamber was installed on the outlet side of the WF pump. In the evaporator, R290 absorbed the heat and changed into saturated or super-heated vapor. Vapor-liquid separator ensured that R290 state at the inlet of scroll expander was saturated or super-heated.

FIGURE 2 Pictures of the experimental system layout (A), electrical heater (B), working fluid pump (C), and scroll expander (D)
vapor. Then, the high-temperature and pressure R290 vapor expansion in the scroll expander to generate the power and leaves into the condenser. In the condenser, R290 was cooled to liquid by nitrogen, which was collected in the reservoir and recirculated by WF pump. The parameters of the scroll expander, evaporator, separator, condenser, reservoir, and WF pump are presented in Tables 2–7.

Besides the main components, several accessories were used in this circuit. A filter was installed between the evaporator and scroll expander to purify the R290 and prevent impurities to influence the operating performance of the scroll expander. The mass flow meter was located at the outlet of the scroll expander to indicate the mass flow rate of R290. In order to regulate the flow rate of R290, a frequency inverter was used to change the speed of the WF pump. Intelligent electrical parameter measuring instrument is used to obtain the values of the scroll expander current, voltage, frequency, power factor, and power. An electric kettle is used as the load to consume electricity energy generated by the scroll expander.

### 2.3.3 Liquid nitrogen circuit (blue line)

The liquid nitrogen from container came into the condenser and released the cold energy to R290, which was from the outlet of the scroll expander. The liquid nitrogen from the condenser will further raise the temperature to the ambient by the action of the vaporizer and then release into the atmosphere.

### 2.4 Measurement devices

In this experiment, the main measured parameters are the mass flow rate, pressure, temperature, and expander electrical power. The details of the measurement devices related to the above parameters are shown in Table 8.

| Table 2 Parameters of the scroll expander |
| Items | Values |
| --- | --- |
| Model | E15H022A-SH |
| Lubrication | Oil free |
| Nominal output | 1 kW |
| Volume ratio | 3.5 |
| Displacement | 14.5 cm³/Rev |
| Max. speed | 3600 r/min |
| Max. inlet temp. | 175°C |
| Max. inlet pressure | 13.8 bar |
| Ambient temp. range | −20–40°C |

| Table 3 Parameters of the evaporator |
| Items | Values |
| --- | --- |
| Type | Spiral wound heat exchanger |
| Operating pressure of shell side | 0.2 Mpa |
| Operating pressure of tube side | 3.2 Mpa |
| Designed temperature range | −196–180°C |
| Heat transfer area | 3.2 m² |
| Material | Stainless steel |

| Table 4 Parameters of the separator |
| Items | Values |
| --- | --- |
| Operating pressure | 1.5 Mpa |
| Designed temperature range | −196–160°C |
| Volume | 0.11 m³ |
| Material | Stainless steel |

| Table 5 Parameters of the condenser |
| Items | Values |
| --- | --- |
| Type | Spiral wound heat exchanger |
| Operating pressure of shell side | 0.43 Mpa |
| Operating pressure of tube side | 1.5 Mpa |
| Designed temperature range | −196–50°C |
| Heat transfer area | 6.6 m² |
| Material | Stainless steel |

| Table 6 Parameters of the reservoir |
| Items | Values |
| --- | --- |
| Operating pressure | 0.37 Mpa |
| Designed temperature range | −196–50°C |
| Volume | 0.043 m³ |
| Material | Stainless steel |

| Table 7 Parameters of the WF pump |
| Items | Values |
| --- | --- |
| Model | SJB150-450/30 |
| Rotation speed | 90–280 r/min |
| Flow rate | 150–450 L/h |
| Max. outlet pressure | 30.0 bar |
| Inlet pressure | 0.2–8.0 bar |
| Effective stroke of piston | 40 mm |
| Diameter of cylinder | 32 mm |
Based on the measured temperatures and pressures, the corresponding enthalpies and entropies for each state point will be obtained by the NIST REFPROP 9.1 software.

### 2.5 Experimental procedures

The flowchart of experimental operating procedures is shown in Figure 3. In step 1, the whole system should be flushed sufficiently and tested sealing performance. In step 2, precooling the system with liquid nitrogen and charging the system with liquid R290 (80 liter). Turn on the power of the electrical heating water and WF pump to produce vapor R290 through the scroll expander. In step 3, the electrical heating power, liquid nitrogen mass flow rate, and frequency of WF pump are manually adjusted until the system parameters are at a stable level. The experimental data were recorded every 2 seconds. In step 4, the electrical heating power, WF pump, and outlet valve of the liquid nitrogen container were stopped until the system completes shutdown.

### 3 THERMODYNAMIC ANALYSIS

In this section, the equations for the analysis of experimental results were presented. Figure 4 presents the T-s plot of the thermodynamic cycle. The involved variables include expander gross output power, expander isentropic
efficiency, thermal efficiency, and cold energy utilization efficiency.

### 3.1 | Expander gross output power

The expander gross output power was denoted by \( W_{\text{exp}} \) and usually calculated in the literature as:

\[
W_{\text{exp}} = m_{R290} (h_1 - h_2)
\]

where \( h_1 \) and \( h_2 \) are the enthalpies of the expander inlet and outlet thermodynamic state, respectively; \( m_{R290} \) is the mass flow rate of vapor R290 through the expander.

### 3.2 | Expander isentropic efficiency

The ideal isentropic expansion process in expander is from 1 to 2s, while the actual process is from 1 to 2. Isentropic efficiency \( \eta_{\text{is,exp}} \) can be expressed by:

\[
\eta_{\text{is,exp}} = \frac{h_1 - h_2}{h_1 - h_{2s}}
\]

where \( h_{2s} \) is the outlet enthalpy through the ideal isentropic expansion process, which can be deduced from the expander inlet entropy and outlet pressure.

### 3.3 | WF pump power consumption

The power consumption of the WF pump can be calculated by:

\[
W_{\text{pump}} = m_{R290,l} (h_6 - h_5)
\]

where \( h_5 \) and \( h_6 \) are the enthalpies of liquid R290 at the WF pump inlet and outlet, respectively; \( m_{R290,l} \) is the mass flow rate of liquid R290 through the WF pump.

### 3.4 | Thermal efficiency

The thermal efficiency \( \eta_t \) of the experimental system can be calculated by:

\[
\eta_t = \frac{W_{\text{net}}}{Q_w} = \frac{W_{\text{exp}} - W_{\text{pump}}}{c_p m_w (T_{w,\text{in}} - T_{w,\text{out}})}
\]

where \( T_{w,\text{in}} \) and \( T_{w,\text{out}} \) are the inlet and outlet temperature of heating water; \( m_w \) is the mass flow rate of heating water.

### 3.5 | Cold energy utilization efficiency

The cold energy utilization \( \eta_c \) is an also important indicator for evaluating the system, which can be calculated by:

\[
\eta_c = \frac{W_{\text{net}}}{Q_{N2}} = \frac{W_{\text{exp}} - W_{\text{pump}}}{m_{N2} \left( h_{N2,\text{in}} - h_{N2,\text{out}} \right)}
\]

where \( m_{N2} \) is mass flow rate of liquid nitrogen; \( h_{N2,\text{in}} \) and \( h_{N2,\text{out}} \) are the enthalpies of the inlet and outlet liquid nitrogen.

### 4 | RESULTS AND DISCUSSION

After the experimental system was set up, the system was operated to obtain steady-state conditions. The system operational parameters display an obvious effect on system overall performance. The mass flow rate of heat sink varies in the range of 80 kg/h~120 kg/h through adjustment. The heat source temperature can be changed by an electrical heater (from 20°C to 60°C).

In the steady-state conditions, the temperature, pressure, WF mass flow rate, and other supporting data were recorded. The experimental investigation results were categorized into two situations: different pressure drop and different heat source temperature.

#### 4.1 | Effect of pressure drop (\( P_d \))

The pressure drop can be received by deducting the WF pump inlet pressure from the scroll expander inlet pressure. In some extent, the higher pressure drop illustrates the better power generation system performance. In the experiment, the degree of superheated vapor at the inlet of scroll expander was kept at about 2 ± 1°C.

Figure 5 presents the variation of scroll expander gross output power and isentropic efficiency against changing the pressure drop for different liquid nitrogen mass flow rate conditions. The figure demonstrates that the \( W_{\text{exp}} \) was enhanced by increasing \( P_d \) or \( m_{N2} \). The reason is significantly of increasing WF mass flow rate and enthalpy drop through the expander. The maximum \( W_{\text{exp}} \) was 662.2 W, obtained at \( m_{N2} \) of 120 kg/h and \( P_d \) of 1.05 Mpa. Moreover, with the increasing \( P_d \), the \( \eta_{\text{is,exp}} \) demonstrates a trend of a rapid increase first and then a flat variation. This is due to the enhancement of evaporative capacity, and the WF mass flow rate increases with the rise of the \( T_{w,\text{in}} \). The value of \( \eta_{\text{is,exp}} \) can describe the operational performance and gross losses including leakage, heat, and friction in the expander.
expander. The recorded maximum $\eta_{\text{is,exp}}$ value under $m_{\text{N2}}$ of 120 kg/h was 0.604 at $P_d$ of 1.05 Mpa.

Figure 6 presents the variation of thermal efficiency against changing the pressure drop for different liquid nitrogen mass flow rate conditions. With the increasing $P_d$, the $\eta_t$ demonstrates a trend of increasing first and then decreasing. When the evaporator heat input power and liquid nitrogen cooling capacity were in the optimal matching state, the maximum $\eta_t$ was 0.0586, obtained at $m_{\text{N2}}$ of 120 kg/h and $P_d$ of 1.02 Mpa. Meanwhile, the increment in $P_d$ ensures the deterioration of $\eta_{\text{is,exp}}$, which results in a decrease in $\eta_t$.

Figure 7 presents the variation of cold energy utilization efficiency against changing the pressure drop for different liquid nitrogen mass flow rate conditions. With the increasing $P_d$, the $\eta_c$ demonstrates a trend of increase first and then decrease. The maximum $\eta_c$ was 0.112, obtained at $m_{\text{N2}}$ of 120 kg/h and $P_d$ of 0.96 Mpa.

Combined with Figures 5-7, it can be concluded that the higher pressure drop will benefit the ORC power generation system performance, but a higher pressure drop means higher expander inlet pressure, representing more pressure-bearing requirement for the evaporator and higher investment. From the economic point of view, the optimum $P_d$ was 1.02 Mpa; the corresponding $W_{\text{exp}}, \eta_t$, and $\eta_c$ were 638.2 W, 0.0586, and 0.0976, respectively.
4.2 | Effect of heat source temperature ($T_{hs}$)

The heat source temperature refers to the temperature of heated water entering the shell inside of the evaporator. In the experiment, the WF pump frequency was kept constant at 20 Hz, while the evaporating pressure and condensing pressure are adjusted by changing the temperature of heat source and mass flow rate of liquid nitrogen.

Figure 8 displays the increase in heat source temperature; the evaporating pressure and condensing pressure are adjusted by changing the temperature of heat source and mass flow rate of liquid nitrogen. The recorded maximum $\eta_{\text{exp}}$ was 0.606, obtained at $m_{N_2}$ of 120 kg/h. But, $W_{\text{exp}}$ shows a trend of increasing first and then decreasing with the increase in the $T_{hs}$. When the $T_{hs}$ increases to 50°C at $m_{N_2}$ of 120 kg/h, the average temperature difference ($\Delta T$) between the liquid nitrogen and heat source reaches the maximum (as shown
in Figure 8); $W_{exp}$ reaches the peak of 674.2 W. As the $T_{hs}$ continues to increase, the average temperature difference ($\Delta T$) begins to decline, which is attributed to the decrease in the enthalpy drop.

Figure 10 presents the variation of thermal efficiency against changing the heat source temperature for different liquid nitrogen flow rate conditions. The trend shows a similar with $W_{exp}$ variation, which is attributed to change in the $W_{exp}$ and the evaporator heat input power increase with the $T_{hs}$ arise. The recorded maximum $\eta_t$ under $m_{N_2}$ of 120 kg/h was 0.594 at $T_{hs}$ of 50 °C.

Figure 11 presents the variation of cold energy utilization efficiency against changing the heat source temperature for different liquid nitrogen flow rate conditions. The trend shows a similar with $W_{exp}$ variation. The decrease in the $\eta_c$ was mainly due to the $T_{hs}$ raise and $W_{exp}$ decline. The maximum $\eta_c$ was 0.104, obtained at $m_{N_2}$ of 120 kg/h and $T_{hs}$ of 40 °C.

Combined with Figures 8–11, to some extent, increasing the $T_{hs}$ has a positive effect on ORC system performance. The maximum $W_{exp}$ corresponds to the optimum matching between heat source and heat sink. From the comprehensive view, the optimum $T_{hs}$ is 50 °C under $m_{N_2}$ of 120 kg/h; the corresponding $W_{exp}$, $\eta_t$, and $\eta_c$ were 674.2 W, 0.0594, and 0.099, respectively.

5 | CONCLUSION

The experimental study on power generation plant of a 1 kW small-scale Organic Rankine Cycle system using a scroll expander and R290 has been investigated. The
effects of $P_d$ and $T_{hs}$ were analyzed. Based on the calculation results, the following conclusion can be drawn.

1. In some extent, the experimental results demonstrated that the $W_{\text{exp}}, \eta_{\text{is,exp}}, \eta_c$, and $\eta_c$ were enhanced by increasing $P_d$ or $T_{hs}$. The experimental results show that the $P_d$ exhibits relatively higher than the $T_{hs}$ about the system overall performance. The recorded maximum $W_{\text{exp}}, \eta_{\text{is,exp}}, \eta_c$, and $\eta_c$ were 674.2 W, 0.606, 0.0594, and 0.112, respectively.

2. A higher $P_d$ will benefit the ORC overall system performance, but it means more pressure-bearing requirements of evaporator and the higher investment. The optimum $P_d$ was 1.02 Mpa under $m_{N_2}$ of 120 kg/h, and the corresponding $W_{\text{exp}}, \eta_{\text{is}}, \eta_c$, and $\eta_c$ were 638.2 W, 0.0586, and 0.0976, respectively.

3. The maximum $W_{\text{exp}}$ corresponds to the optimum matching between heat source and heat sink. The optimum $T_{hs}$ was 50°C under $m_{N_2}$ of 120 kg/h, and the corresponding $W_{\text{exp}}, \eta_c$, and $\eta_c$ were 674.2 W, 0.0594, and 0.099, respectively.

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NOMENCLATURE
symbols
- $c_p$: isobaric heat capacity [kJ/kg°C]
- $h$: enthalpy [kJ/kg]
- $m$: mass flow rate [kg/h, m³/h]

Greek symbol
- $\eta$: efficiency

Subscripts
- $1$: inlet
- $8$: outlet
- $c$: cold energy
- $d$: drop
- $\text{exp}$: expander
- $\text{is}$: isentropic
- $\text{in}$: inlet
- $\text{out}$: outlet
- $\text{t}$: thermal
- $\text{pump}$: working fluid pump
- $\text{w}$: water
- $N_2$: nitrogen
- $\text{net}$: net
- $v$: vapor
- $l$: liquid
- $\text{hs}$: heat source

W  power [W]
Q  thermal energy [kW]
T  temperature [°C]
P  pressure [Mpa]

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