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Highlights

- Geothermal resources from abandoned oil wells were utilized for power generation.
- An intermediate water cycle was proposed to avoid the blockage problem.
- A specially designed four-stage axial turbine was applied to the system.
- The turbine performed well and could be generalized for geothermal ORC system.
- The proposed system could be a desirable option for geothermal energy applications.
Construction and preliminary test of a geothermal ORC system using
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Abstract: In this study, a low-temperature geothermal organic Rankine cycle (ORC)
system was designed to generate electric power using geothermal resource from
abandoned oil wells in the Huabei oilfield of China. Several main sections were
constructed including the geothermal water cycle, intermediate water cycle, ORC,
cooling water cycle, lubricant oil cycle, bypass and control systems. Using R245fa as
the working fluid, a four-stage axial turbine was specially designed and manufactured,
to drive a grid-connected alternator. An onsite preliminary test was conducted in the
oilfield and part of the experimental data were presented and analyzed. The turbine
isentropic efficiency and ORC efficiency were evaluated. Besides, an intermediate
cycle efficiency was proposed to show the influence of the heat loss of the intermediate
water cycle on the system performance. Experimental results showed that a turbine
efficiency of 78.52% and an ORC efficiency of 5.33% could be obtained. The average
of intermediate cycle efficiency was 77.98%, indicating that the heat loss of the
intermediate water cycle was considerable and should not be ignored. The construction
and preliminary test of this ORC system could provide some meaningful references for
the researchers and engineers to establish desirable geothermal ORC power plants using
abandoned oil wells.

**Key words:** Geothermal resource; Abandoned oil well; Organic Rankine cycle; Multi-stage axial turbine.

**Nomenclature**

**Notation**

- $h$: specific enthalpy (kJ/kg)
- $m$: mass flow rate (kg/s)
- $Q$: heat (kW)
- $W$: power (kW)
- $\eta$: efficiency

**Subscripts**

- $ap$: active power
- $c$: condenser
- $e$: evaporator
- $gen$: generator
- $ie$: intermediate water flowing into the evaporator
- $inter$: intermediate water cycle
- $ip$: intermediate water flowing into the preheater
- $is$: isentropic
- $p$: pump
- $t$: turbine
- $wf$: working fluid
1. Introduction

In the past decades, fossil fuels played the central role to promote the social development. The vast majority of world’s energy consumption came from oil, coal and gas. However, the rate of fossil fuels consumption is much higher than that of exploitation, and the growing energy demand drives humans to explore alternative resources [1]. Meanwhile, the burning of fossil fuels takes main responsibilities for the environment pollution like haze, global warming and air contamination. As the rising energy demand and environment issues become the non-negligible problems demanding prompt solution, the utilization of renewable and clean energy is considered to be a valid way to deal with the energy and environment problems. So, in recent years there has been an increasing interest in the use of renewable and clean energy resources such as solar energy, wind energy, geothermal energy, and tidal energy and so on. Among these energies, geothermal energy finds its advantages of long term stability, higher availability, larger technical potential and favorable cost [2]. Thus, geothermal
power plants have been built for electric power generation in more than 21 countries all over the world [2].

There are four major types of geothermal power plants operating under different thermal conditions: single-flash steam plant, double-flash steam plant, dry-steam plant and binary cycle plant [3, 4]. More than 70% of the geothermal resources are estimated to be medium-low enthalpy type with temperature lower than 150°C [5]. For this low-temperature energy conversion, closed binary Organic Rankine Cycle (ORC) is a state-of-the-art technology showing better adaption and feasibility than other options [4, 6-9]. Thus, much attention has been directed toward the geothermal power generation by ORC systems. Walraven et al. [7] investigated different types of ORC for low-temperature geothermal resources, including subcritical, transcritical, one-pressure and multi-pressure ORC. Each type was further divided into simple ORC, ORC with recuperation and ORC with turbine bleeding. Up to 80 different working fluids were used in these cycles and a comparison with different types of Kalina cycles was also carried out. They found that transcritical and multi-pressure subcritical ORCs showed the best performance and exergy efficiency of more than 50% could be achieved. Besides, the importance of the condenser temperature and pinch point temperature difference were also highlighted. Zare [10] and Imran et al. [11] developed the thermodynamic and exergoeconomic models of basic ORC, recuperated ORC, and regenerative ORC for binary geothermal power plants. Exergoeconomic optimization and performance comparison were conducted to optimize the investment cost and the exergy efficiency. The basic ORC was revealed to have a lowest investment cost while
two other systems showed better exergy efficiency. Guzović et al. [12] pointed out the fact that every geothermal source has distinct thermal conditions of geothermal water and cooling fluid, needing a specific study for each case. Specifically for low-temperature geothermal ORC power plant “Babina Greda” in Croatia, up to 21 refrigerants and hydrocarbons were chosen to find the most desirable working fluids. Wet steam problem, environmental concern, and working pressure of these working fluids were analyzed, showing that R601a and R601 were the most suitable fluids for this ORC power plant. Liu et al. [13] and Kang et al. [14] analyzed the performances of geothermal ORC using zeotropic mixtures as working fluids. System parameters were also optimized to improve the geothermal ORC properties, and the advantages of the mixtures were obtained compared to the pure working fluids. El-Emam and Dincer [15] performed an exergoeconomic analysis for a novel-type geothermal regenerative ORC. Exergy destruction rates of components were studied and influences of system parameters on system thermodynamics and thermoeconomic performances were also revealed.

On the other hand, the expensive drilling cost has become a restriction of geothermal exploitation and development [16-18]. At the same time, millions of oil wells have been abandoned and their leaking problem is a threat to the local environment [19, 20]. As these oil wells can produce plenty of heat energy, changing the oil wells to geothermal wells not only could save a large amount of cost, but also could solve the environment pollution. Using the mature or abandoned oil or gas well to generate geothermal electricity is also called coproduction, a technology combining
the power generation and possible oil recovery, which is regarded as a new and potential way to utilize geothermal energy [21]. Many researchers have investigated the geothermal power generation using abandoned oil wells. Cheng et al. [22] established a modeling including fluid momentum, heat transfer and energy equation to simulate the geothermal ORC power generation. Different organic working fluids were chosen for comparison to find the most suitable choice for geothermal power generation from abandoned wells. Later, they set up a novel heat transfer model coupling two-dimensional thermal reservoirs with one-dimensional wellbore to investigate the influences of thermal reservoirs on the performances of geothermal power generation using abandoned wells [19]. Wight and Bennett [23] discussed the feasibility of the geothermal power generation from abandoned wells and proposed a novel method to utilize the abandoned wells as a heat exchanger of a binary ORC power plant. Apart from the theoretical studies and simulations of geothermal ORC using abandoned oil wells above, several coproduction power plants have also been established all over the world [24-26]. However, very little experimental data have been published in the literatures due to the commercial confidentiality.

Another important part of the geothermal ORC is the expander, which determines the system efficiency directly. Commonly used expansion devices can be divided into two types: one is the volumetric type, such as scroll, screw, reciprocating piston and rotary vane expanders, while the other is the velocity type including radial and axial turbines [27-29]. Recently, Rahbar et al. [27] reviewed the applications of expansion devices in published research and presented the comparison of various ORC expanders.
according to their corresponding nominal power ranges, rotational speeds, advantages and disadvantages. They also summarized the experimental studies of ORC with various expanders. For high power capacities (more than 250kW) and large mass flow rates, it is more suitable to choose an axial turbine as the expansion device. Several investigators conducted the experimental studies using single-stage axial turbines [30-33]. Fu et.al. [30] designed and built a 250-kW ORC system using R245fa as the working fluid. A single-stage axial turbine was specially designed for this system and an isentropic efficiency of 63.7% could be achieved. Klonowicz et.al. [33] presented the design process, numerical simulation and preliminary test of a small scale single-stage axial turbine, which was the key part of a low temperature, sub-critical ORC system. The theoretical turbine efficiency showed a good agreement with the measured result. However, as concluded by Rahbar et al [27], there is still lack of experimental study of the multi-stage expanders in the ORC system.

On the basis of the literature review above, the present study introduces the construction and experiment test of a 500 kW geothermal ORC system using geothermal resource from abandoned oil wells in Huabei oilfield. R245fa is chosen as the working fluid because of its appropriate pressure, nonflammable, noncorrosive and satisfactory thermal stability [34]. A multi-stage axial turbine is specially designed for the ORC system using R245fa. Particularly, the blockage caused by the oil is taken into consideration during the design and construction of the geothermal ORC system. The corresponding solution of this problem is displayed. Part of experimental data are presented to show the feasibility and system performance of the geothermal ORC. The
system efficiency, turbine isentropic efficiency and intermediate cycle efficiency are also evaluated. The projected value of this work is to provide some useful references for large scale applications of geothermal ORC power plants using abandoned oil wells in oilfields.

2. System design and construction

Fig. 1 depicts the schematic diagram of the geothermal ORC system, which consists of six main sections: geothermal water cycle, intermediate water cycle, ORC, cooling water cycle, lubricant oil cycle, bypass and control systems. The lubricant oil cycle and control system are not given in Fig. 1. The flow arrows of geothermal water, intermediate water, R245fa and cooling water are presented, denoting the flow directions of these fluid.

Fig. 1 Schematic diagram of the geothermal ORC system

1) Geothermal water cycle

The geothermal water is extracted from the oil wells in Huabei oilfield. Huabei
oilfield is one of the largest oilfields in Northern China, and has provided a large amount of petroleum and gas for the industrial manufacture and social activities. Lots of oil wells produced oil through water flooding. Due to years of production, the water contents of the liquid from these old oil wells are more than 90%. Thus, many oil wells are abandoned because of the lack of petroleum production. As the abandoned oil wells store a great deal of geothermal water, they could be easily changed into geothermal wells without expensive drilling cost. It is worthy of taking advantage of the heat energy to generate electric energy and mechanical energy for the oil exploration. In this study, the extraction from the oil wells has the water content of 98%, temperature of 110°C and mass flow rate of 200 t/h, which can be used as the heat source directly. As sometimes little gas is extracted with the geothermal water, a separator is applied to separate the gas and liquid. After separation, a part of the hot water is pumped to the industrial heat users for industrial production, such as oil tube cleaning, heat supply and other oil production process. The rest of geothermal water flows into two plate heat exchangers (HE 1 and HE 2) in parallel and heats the intermediate water. A part of the water then flows into the HE 3 if preheating is necessary. After heating the intermediate water, the geothermal water is sent to the settling tank, in which the oil is separated and stored in an oil tank, and the water is pumped back to underground. Corresponding state points $g_1$, $g_2$, $g_3$ and $g_4$ are marked with blocks around the numbers, representing the geothermal water flowing into and out of the HEs (1, 2 and 3), respectively.

2) Intermediate water cycle

The intermediate water cycle in Fig. 1 is the orange cycle between the geothermal
water cycle (red) and R245fa cycle (blue). 6 key state points are marked in the intermediate water cycle. The state points $i_1$ and $i_4$ denote the outlets of the pump 1 and 2, respectively. $i_2$ and $i_3$ represent the inlet and outlet of the evaporator while $i_5$ and $i_6$ denote the inlet and outlet of the preheater. The intermediate water absorbs the heat energy from the geothermal water in three HEs, and then flows into preheater and evaporator to preheat and vaporize the R245fa. From the perspective of thermodynamics, it is not wise to install such an intermediate cycle between the heat source and working fluid, because the intermediate heat exchanger will lower the system thermal efficiency. However, the geothermal ORC in this study utilizes the actual geothermal water from the oil wells as the heat source. The oil in the geothermal water would jam the heat exchangers and pipes after a certain time, resulting in the deterioration of heat transfer and the decrease of system efficiency. What’s worse, the whole power plant must be stopped to clean the relevant equipment at set intervals. The blockage caused by the oil must be solved in the practical engineering for long-term stable operation. Thus, the addition of the intermediate water cycle would be a valid solution for the blockage problem due to its two obvious advantages:

i) The intermediate water cycle could decrease the cost of equipment cleaning. Without the intermediate water cycle, the oil would jam the evaporator and preheater after a certain time, and cleanout of these equipment will be necessary. However, the cleanout of the evaporator and preheater would cause a certain loss of working fluid. The supplement of the working fluid would be necessary after several times of cleanout, which is a considerable cost because the working fluid of ORC is not cheap. The
intermediate water cycle shifts the oil blockage from the evaporator and preheater to the HE 1 and 2. So, only the HEs need to be cleaned. And the loss of working fluid caused by the cleanout of evaporator and preheater is avoided. Then, the cost of the supplement of working fluid could be reduced.

ii) The intermediate cycle could realize the online cleanout of the heat exchangers and there is no need to shut down the power plant during the cleanout process. The HE 1 and 2 could be cleaned separately and there is always one HE working to maintain the power plant running, which makes it possible to achieve the long-term stable operation.

Besides, the formation mechanism of the oil slick and the anti-scaling measures were also investigated. Corresponding technologies including surface hydrophilization, increasing flow rate and adding surfactant were carried out to prevent the equipment blockage.

3) ORC

The R245fa is pumped from the state 6 to state 1, then preheated in the preheater to the state 2 and evaporated to the state 3. After flowing through the main steam valve and the turbine flow control valve, the R245fa vapor (state 4) expands in a four-stage axial turbine to drive a generator, which is connected to the power grid. Then, the vapor exhaust (state 5) cools down in the condenser to the state 6 and is pumped back to the preheater and evaporator by two pumps in parallel.

4) Cooling water cycle

The cooling water in the cooling tower is pumped to the condensers (state c1) to
absorb the heat energy and condense the R245fa. Then the heated water (state c2) is
sent back to the cooling tower to release the heat.

5) Lubricant oil cycle

The lubrication is of vital importance to the turbine steady operation. Hence, a
lubricant oil cycle is applied to the bearings using the chosen turbine oil. The turbine
oil is pumped by the oil pump to the bearing housing to support the rotor and then flows
back to the lubricant oil box. The temperature of the lubricant oil back should be lower
than 70°C.

6) Bypass and control systems

The bypass and control systems are designed and constructed for the purpose of
safe operation. The bypass system connects the evaporator and condenser directly to
protect the turbine and generator during the start and shutdown, especially the
emergency shutdown. The control system mainly contains following functions:

i) Rotational speed control. As the generator is connected to the power grid, it is
necessary to maintain the rotational speed at the operational speed, 1500 rev/min, by
the turbine governor.

ii) Liquid level control in the condenser. The height of liquid level in the condenser
is maintained at a set value by adjusting the frequencies of two ORC pumps.

iii) Overspeed protection. The control system would give an alarm, close the
turbine flow control valve and stop the ORC pumps if the turbine is overspeed.

iv) Liquid level protection. The control system would give an alarm, close the
turbine flow control valve and stop the ORC pumps if the liquid level in the condenser
is higher (or lower) than the upper (or lower) limit.

v) Back pressure protection. If the back pressure is higher than the upper limit, there might be a lack of cooling water or the ORC system is overpressure. The control system would give an alarm, close the turbine flow control valve and stop the ORC pumps.

3. Equipment apparatus

The experiment device consists of the heat exchangers (HE 1, 2 and 3), intermediate water pumps (1 and 2), evaporator, preheater, turbine, generator, ORC pumps, condenser, cooling tower, cooling water pump, lubricant device, monitoring device, control panel, pipelines, valves and sensors. Fig. 2 presents the front and back views of the experimental layout in the Huabei oilfield, Cangzhou City.

The evaporator shown in Fig. 2(a) is a shell and tube heat exchanger, having a length of 7356 mm, a diameter of 1000 mm and a weight of 7500 kg. The working fluid in the tube side is intermediate water while that in the shell side is R245fa, because the average temperature of the R245fa is lower than that of the intermediate water and the heat loss of the evaporator can be reduced. The designed inlet and outlet temperatures of the intermediate water are 105 °C and 80.7 °C while those of the R245fa are 50 °C and 85.6 °C. The designed inlet and outlet pressures of the intermediate water are 200 kPa and 150 kPa while those of the R245fa are 818.8 kPa and 700 kPa.

A main steam valve is employed to control the mass flow rate of the working fluid flowing into the turbine. The main steam valve is a manually operated valve, which can not be adjusted by the control system. Thus, in order to achieve the accurate rotational speed control, an electromagnetic valve is set up before the turbine inlet, also called the
turbine flow control valve. The valve opening can be adjusted by the actuator to maintain the rotational speed at 1500 rev/min and can be closed automatically to prevent the turbine overspeed. Meanwhile, a bypass valve is set up in the bypass system to protect the turbine and generator during the starting and closing processes. At the bottom of Fig. 2(a), the lubricant oil system containing an oil pump, pipelines and a blue box (oil box) provides lubricant for the bearings.
Fig. 2 Experimental layout of the geothermal ORC system, (a) front view, (b) back view

The axial turbine is specially designed and developed for the geothermal ORC system because the axial turbine design offers advantages such as easy to achieve multistage arrangements, a lower rotating speed, a higher power capacity, partial admission for off-design and a lower price due to the mature production technology [27]. The turbine is supplied by the Qingdao Jieneng Turbine Co. Ltd. Fig. 3 shows the design drawing of the four-stage axial turbine. The numbers and heights of four stage stator blades are 76, 76, 76, 76 and 31.6, 43.2, 58.2, 77.6 mm, respectively, and those of rotor blades are 143, 143, 143, 143 and 36.4, 50.25, 67.6, 89 mm, respectively. The average diameter of the turbine is 650 mm. Some design working parameters of the turbine are listed in the Table 1. The turbine is coupled with an alternator by a shaft coupling and the electricity is sent to the power grid. As the rotational speed is designed as 1500 rev/min, in order to be consistent with the grid frequency, 50 Hz, the alternator
contains two pair of magnetic poles. The lower rotating speed of 1500 rpm could avoid some tough problems such as the strength and vibration of the blades, the choice of the high speed bearing and rotordynamic problem. Besides, the multistage arrangement is more suitable for high power turbines and this turbine design thought could be easily generalized to the large-scale geothermal ORC power plants for oilfields.

![Diagram of a four-stage axial turbine]

**Fig. 3 Four-stage axial turbine**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Working fluid</td>
<td>-</td>
<td>R245fa</td>
</tr>
<tr>
<td>Turbine inlet temperature</td>
<td>°C</td>
<td>84</td>
</tr>
<tr>
<td>Turbine inlet pressure</td>
<td>kPa</td>
<td>700</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>kg/s</td>
<td>33.04</td>
</tr>
<tr>
<td>Outlet pressure</td>
<td>kPa</td>
<td>187.142</td>
</tr>
<tr>
<td>---------------</td>
<td>-------</td>
<td>---------</td>
</tr>
<tr>
<td>Rotating speed</td>
<td>rev/min</td>
<td>1500</td>
</tr>
<tr>
<td>Power output</td>
<td>kW</td>
<td>656.86</td>
</tr>
</tbody>
</table>

All three heat exchangers in the intermediate water cycle are plate heat exchangers. The HE 1 and 2 are in parallel and totally same with the width of 708 mm and the height of 2103 mm. Their designed inlet and outlet temperatures of geothermal water are 110 °C and 85.7 °C, while those of intermediate water are 80.7 °C and 105 °C. The HE 3 generally works at high power demand with the width of 732 mm and the height of 1519 mm. The designed inlet and outlet temperatures of geothermal water and intermediate water are 85.7 °C, 70.6 °C and 50 °C, 55 °C, respectively.

The condenser is installed at the top of the testbed to prevent the possible cavitation of the R245fa pump. As same as the evaporator, it is a shell and tube heat exchanger with a length of 7086 mm, diameter of 1100 mm and weight of 7000 kg. The designed inlet and outlet conditions of the R245fa are 54.2 °C/187.1 kPa and 30.0 °C/177.0 kPa. While the designed inlet and outlet conditions of the cooling water are 20.0 °C/120.0 kPa and 28.5 °C/114.0 kPa. Also, the working fluid flows through the shell side while the cooling water is in the tube side.

The preheater is a brazed heat exchanger with the width of 304 mm and the height of 694 mm. The designed inlet and outlet conditions of the R245fa are 30.4 °C/861.8 kPa and 50.0 °C/818.7 kPa, while those of the intermediate water are 55.0 °C/150.0 kPa and 50.0 °C/100.0 kPa, respectively.

The ORC pumps are chosen considering the cavitation resistance and
leakproofness. The cavitation would threaten the safe operation of the system and also cause the fluctuation of the mass flow rate of the working fluid. Thus, the sliding vane pumps are chosen as the ORC pumps because of the effective anti-cavitation performance. Meanwhile, the ORC pumps are installed at the bottom of the testbed to prevent the cavitation. On the other hand, the sliding vane pumps shows a good performance of leakproofness, which could decrease the loss of the expensive ORC working fluid.

Two intermediate water pumps 1 and 2 are installed in the intermediate cycle to maintain the water circulation. As the pressure differences between the inlets and outlets of the pumps are only about 50 kPa, the power consumed by the pumps are very limited and could be ignored compared to the amount of heat transfer in the intermediate cycle. Thus, in the thermodynamic analysis, the power consumption of the intermediate water pumps are not considered.

The control cabinet of the pumps is depicted in the Fig. 2(b), providing the power for the ORC pumps and intermediate water pumps. Two operation modes of the ORC pumps could be switched on the control cabinet: manual operation and automatic operation. The frequencies of the ORC pumps could be adjusted manually in the manual operation mode. Also, in the automatic operation mode, the frequencies could be regulated automatically by the control system to maintain the liquid level in the condenser.

A number of sensors are installed on the key state points to monitor the operating state and protect the whole trial project. Fig. 1 displays the measurement points in the
geothermal ORC testbed. The platinum resistance thermometers, PT100, are used to measure the temperatures with tolerance ±0.5 °C. The pressures in the ORC are measured by the pressure sensors with tolerance of ±0.5% full scale, while the mechanical pressure gauges are applied to other measurement points. The liquid levels in the evaporator and condenser are measured by using the float type level gauges. All measured data are recorded and stored by a computer.

4. Thermodynamics analysis

![Temperature-entropy chart of the ORC](image)

The temperature-entropy chart of the ORC is presented in Fig. 4. The state points 1, 2, 3, 5, 6 denote the outlets of the ORC pumps, preheat, evaporator, turbine and condenser, respectively. The state point 4 represents the turbine inlet. In the designed conditions, the main steam valve and turbine flow control valve are fully opened; i.e., the parameters of the working fluid at evaporator outlet are same as those at the turbine inlet. Thus, the state point 3 is same as 4, as shown in Fig. 4. In the actual onsite working
conditions, the valves might be opened partly and there would be an isenthalpic process between points 3 and 4, as explained in the next section. The state points $1_s$ and $5_s$ are the isentropic enthalpy points corresponding to points 1 and 5. The thermodynamics analysis is evaluated and several key parameters could be determined according to following equations.

The heat absorbed by the intermediate water from the geothermal water could be written as

$$Q_{\text{inter}} = m_{ie} (h_{12} - h_1) + m_{ip} (h_{5} - h_{4})$$  \hspace{1cm} (1)

where $m_{ie}$ is the mass flow rate of the intermediate water flowing into the evaporator, while $m_{ip}$ is the mass flow rate of the intermediate water flowing into the preheater.

The heat absorbed by the R245fa from the intermediate water in the evaporator and preheater is

$$Q_e = m_{wf} (h_3 - h_2) + m_{wf} (h_2 - h_1)$$  \hspace{1cm} (2)

where $m_{wf}$ is the mass flow rate of the R245fa.

The output power of the turbine is calculated by

$$W_t = m_{w_f} (h_4 - h_5)$$  \hspace{1cm} (3)

Then, the turbine isentropic efficiency can be written as

$$\eta_{is} = \frac{W_t}{m_{w_f} (h_4 - h_{5_s})}$$  \hspace{1cm} (4)

where the $h_{5_s}$ is the enthalpy of the turbine outlet through the ideal isentropic process.

The heat released by the R245fa to the cooling water in the condenser is

$$Q_c = m_{w_f} (h_5 - h_6)$$  \hspace{1cm} (5)

The power consumed by the ORC pumps is
\[ W_p = m_w (h_1 - h_0) \] (6)

Up to now, the net power and the ORC efficiency can be calculated as follows

\[ W = W_f \eta_{gen} - W_p \] (7)

\[ \eta_{ORC} = \frac{W}{Q_e} = \frac{W \eta_{gen} - W_p}{Q_e} \] (8)

where the \( \eta_{gen} \) is the generator efficiency.

Actually, in the experiment, the turbine output is not calculated according to the Eq. (3). Instead, the active power of the generator, \( W_{ap} \), is measured to reflect the turbine output. Hence, in this study, the turbine isentropic efficiency and the ORC efficiency are defined as

\[ \eta_{ap}' = \frac{W_{ap}}{m(h_1 - h_{ss})} \] (9)

\[ \eta_{ORC}' = \frac{W_{ap} - W_p}{Q_e} \] (10)

The heat loss of the intermediate cycle to the environment should also be paid attention to in this study, to determine whether this loss could be ignored or not. Thus, an intermediate cycle efficiency is defined as follows

\[ \eta_{inter} = \frac{Q_e}{Q_{inter}} = \frac{m_w (h_1 - h_i)}{m_w (h_2 - h_1) + m_w (h_3 - h_4)} \] (11)

The denominator of the \( \eta_{inter} \) is the heat absorbed by the intermediate water from the geothermal water, while the numerator is the heat absorbed by the R245fa from the intermediate water. In the theoretical study, the heat loss from the intermediate cycle to the environment could be ignored, and thus \( Q_e \) is equal to \( Q_{inter} \). However, in the experiment, this heat loss should be considered and evaluated to show the influence of
the intermediate cycle on the overall system efficiency. The higher $\eta_{\text{inter}}$ means the lower heat loss in the intermediate cycle, namely the slighter influence to the system efficiency, and vice versa. Hence, $\eta_{\text{inter}}$ is regarded as an important parameter to judge the system performance.

In the above equations, the $h$ denotes the enthalpy of the state point, and the subscripts represent corresponding state points in the system.

5. Experiment results

The preliminary test was conducted after the construction and test run of the geothermal ORC system. The cooling water cycle was started firstly to provide sufficient cold source and prevent the possible overheating. Meanwhile, the intermediate water cycle was also put into operation. Then, two ORC pumps were operated at manual operation mode, to provide required working fluid for the evaporator and turbine. During the starting procedure of the system, the mass flow rate of the R245fa was increased manually by increasing the frequencies of the ORC pumps. Next, the geothermal water valve was opened gradually, providing heat source for the system and heating the working fluid. After the working fluid was heated to the superheated state, the bypass valve was closed gradually and the turbine flow control valve was opened so that the working fluid flowed into the turbine to rotate the turbine and generator. Considering about the thermal stress during the starting, the mass flow rate of the geothermal water was increased slowly and the rotational speed increased step by step to 1500 rev/min. After that, the generator was connected to the electricity grid and the rotational speed was maintained at 1500 rev/min due to the rotational speed
control system. Then, one of the ORC pump was switched to the automatic operation mode to control the liquid level in the condenser.

The experimental test started at about 14:30 on December 23rd 2015 and closed at about 08:27 on December 27th 2015. In consideration of the stability of the experimental data, only part of the experimental results on December 26th from 7:50 to 23:04 are presented here. The experimental investigation was conducted under the actual onsite working conditions. The mass flow rate and temperature of the geothermal water were about 60-120 t/h and 105-107 °C, respectively. The evaporating pressure and temperature of the working fluid were about 0.44-0.68 MPa and 59.6-80.4 °C, respectively. The output power of the turbine was about 60-160 kW. The rotational speed was maintained at 1500 rev/min. As the power demand was lower than the designed value, the HE 3 was closed and the preheater was not employed in the preliminary test.

Fig. 5 presents the inlet temperature and mass flow rate of the geothermal water as they varied with time. The mass flow rate before 11:00 maintained at around 63 t/h with very limited undulation but varied obviously afterwards. Several peaks occurred after 11:00. The possible reason of this phenomenon was that part of extracted geothermal water was used for other industrial productions and their water consumption always changed with time. Besides, several sharp peaks could be observed at about 11:30, 16:37 and 20:32, respectively. This was due to the sudden manual operation of the valve in the geothermal water cycle. After 17:00, as the industrial productions stopped, the mass flow rate of the geothermal water was increased by adjusting the valve opening
manually. The maximum mass flow rate was 128.53 t/h at 22:58. The inlet temperature of the geothermal water was very smooth without fluctuation and the average temperature was 106.12 °C.

Fig. 5 Inlet temperature and mass flow rate of geothermal water

Fig. 6 displays the temperature at \(i_2\) and mass flow rate of the intermediate water cycle. \(i_2\) denotes the inlet of the intermediate water flowing into the evaporator, as shown in Fig. 1. It could be found that the mass flow rate of the intermediate water was about 150 t/h with slight fluctuation. The temperature at \(i_2\) could be adjusted by the mass flow rate of the intermediate water. For the purpose of the stable operation, the mass flow rate of the intermediate water was kept invariant in this preliminary test. As a result, the temperature at \(i_2\) was mainly influenced by the mass flow rate of the geothermal water. Hence, the change trend of temperatures at \(i_2\) was very similar to that of mass rate of geothermal water. The maximum of the temperature at \(i_2\) was 91.92 °C, which was about 15 °C lower than the geothermal water. It meant that, as for the ORC, the average temperature of the heat absorption was decreased due to the addition
of intermediate water cycle. Hence, it can be inferred that the ORC efficiency in this experiment system will be lower than that in the traditional system with direct heat source.

Fig. 6 Temperature at \( i_2 \) and mass flow rate of intermediate water cycle. The temperatures and the mass flow rate of the cooling water cycle are depicted in Fig. 7. The mass flow rate fluctuated between 450 t/h and 500 t/h, and the average was 472.3 t/h, which was closed to the designed flow. The large flow rate was to avoid the possible overheat and overpressure of the ORC system. The temperature at \( c_1 \) varied with time between 12.13 °C and 16.84 °C, which was much higher than the ambient temperature. Actually, the maximum and minimum air temperatures of this day were 5 °C and -7 °C, respectively. The large difference between the temperature at \( c_1 \) and the ambient temperature was due to the insufficient cooling in the cooling tower. Two cooling towers were set up to supply the cold source for system, but in the preliminary test, as the power demand was lower than the designed conditions, only one cooling tower was put into operation. Thus, the temperature at \( c_1 \) was far above the ambient
temperature, which also resulted in the higher temperature at \( c_2 \).

Fig. 7 Temperatures at \( c_1 \) and \( c_2 \) and mass flow rate of cooling water

Temperatures of the key state points (evaporator outlet, turbine inlet, turbine outlet and condenser outlet) are presented in Fig. 8, while corresponding pressures are shown in Fig. 9. The temperature and pressure of the evaporator outlet were not equal to those of the turbine inlet as the working fluid flowed through the main steam valve and the turbine flow control valve. As the valves were not fully opened, during this process, the pressure of the working fluid was decreased but the enthalpy was nearly invariant due to no work output. Thus, this process was regarded as the isenthalpic process. The enthalpies of the evaporator outlet and the turbine inlet were equal. The pressure of the turbine inlet was measured and the temperature was calculated through the isenthalpic process. As shown in the Fig. 9, the differential pressure between the evaporator outlet and the turbine inlet was obvious, which indicated that the pressure drop of two valves was considerable. It could be speculated that the turbine output would be limited due to this pressure drop. At the evaporator outlet and turbine inlet, it was determined from
REFPROP that the temperature measured was higher than the saturation temperature of the measured pressure. Thus, the R245fa vapor was superheated, implying the favorable performance of the evaporator. The temperature of the turbine outlet also varied with time and showed the same change trend as the evaporator outlet and turbine inlet. The temperature of the exhaust at turbine outlet was higher than the saturation temperature, indicating that the exhaust was in the superheated region. There was no need to worry about the liquid erosion and damage of the last stage blade. Thus, R245fa is a favorable working fluid for the geothermal ORC system. Meanwhile, due to the large mass flow rate of the cooling water, the working fluid was subcooled at the condenser outlet, in favor of the stable and safe operation of the ORC pumps without concern about the cavitation erosion.

Fig. 8 Temperatures of the evaporator outlet, turbine inlet, turbine outlet and condenser outlet
Fig. 9 Pressures of the evaporator outlet, turbine inlet, turbine outlet and condenser outlet

The active power of the generator $W_{ap}$ and the heat absorption $Q_{inter}$ and $Q_e$ are presented in Fig. 10 and Fig. 11, respectively. The active power represented the actual power output of the ORC system to the power grid. Obviously, the overall trends of the $W_{ap}$, $Q_{inter}$ and $Q_e$ were related to the mass flow rate of the geothermal water. In the preliminary test, the mass flow rate of the geothermal water was inadequate and the R245fa vapor pressure was decreased due to the valves before the turbine inlet. Thus, the active power was lower than the designed power output, varying from 62 kW to 163 kW. The maximum was 163.44 kW at 23:03. $Q_{inter}$ denoted the heat absorption of the intermediate water from the geothermal water while $Q_e$ denoted the heat absorption of the R245fa from the intermediate water. Thus, the difference between $Q_{inter}$ and $Q_e$ was the heat loss of the intermediate water cycle. As shown in Fig. 11, the average of $Q_{inter}$ was 3294 kW while that of $Q_e$ was 2566 kW. The difference between the average values was 728 kW, meaning that 22.1% heat absorption of the intermediate water was...
released to the environment. Thus, it can be sure that the system efficiency would be limited due to the heat loss of intermediate water cycle. To see the influence of the intermediate water cycle, the intermediate cycle efficiency, $\eta_{\text{inter}}$, is given in Fig. 12.

The intermediate cycle efficiency fluctuated with time and the average was 77.98%, showing that the heat loss in the intermediate cycle should not be ignored. The lower intermediate cycle efficiency indicated that much work need to be conducted to improve the system performance. As this test project is a preparation for the actual applications of geothermal ORC power plants, it is necessary to investigate how to improve the intermediate cycle efficiency, which will be our research mission in the future.

Fig. 10 Active power of the turbine-alternator during the preliminary test
Finally, using the Eqs. (9) and (10), the turbine efficiency $\eta_{t_2}'$ and ORC efficiency $\eta_{ORC}'$ are shown in Fig. 13 and Fig. 14, respectively. Some sudden rises and drops could be observed in the turbine efficiency curve. These abnormal values were not reasonable, caused by the sudden change of the active power and the heat absorption of the working fluid. Getting rid of these abnormal values, the average of the turbine efficiency was

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**Fig. 11** Heat absorption of the intermediate water and the working fluid

**Fig. 12** Intermediate cycle efficiency during the preliminary test
78.52%, showing the good performance of the turbine. Hence, the multi-stage axial
turbine was a desirable choice for the geothermal ORC system.

As same as the turbine efficiency, the abnormal values of the ORC efficiency in
Fig. 14 were removed. The average efficiency was 4.46% and the maximum efficiency
was 5.33%. The limitation of the ORC efficiency could be explained by five reasons.
Firstly, the mass flow rate of the geothermal water was lower than the designed value
and the system was operated at highly off-design conditions. Secondly, the preheater
was not applied in this preliminary test and the heat energy of the geothermal water had
not been fully taken advantage of. Thirdly, the heat loss in the intermediate water cycle
led to the lower intermediate cycle efficiency, which decreased the evaporator outlet
temperature of the working fluid. Fourthly, the power output was limited by the lower
turbine inlet pressure, which was caused by the pressure drop in the main steam valve
and the turbine flow control valve. Finally, due to the insufficient cooling in the cooling
tower, the temperature of the cooling water was higher than the environmental
temperature, which resulted in a higher turbine output pressure and also reduced the
power output. Therefore, based on these reasons, further investigations need to be
conducted to improve the system efficiency.
6. Conclusions

This study displayed the long-term work of the design and construction of the geothermal ORC system using abandoned oil wells as heat source. With the consideration of actual engineering problems, the system comprising six sections was more complicated than the ideal ORC system. An intermediate water cycle was
proposed to prevent the possible blockage of key apparatus. Three assistant systems containing the lubricant oil cycle, bypass system and control system maintained the long-term stable and safe operation of the overall system. A specially designed and manufactured four-stage axial turbine was applied to this system. The turbine efficiency, intermediate cycle efficiency and ORC efficiency were evaluated as the system assessment criteria. From the detailed discussions of the results, some important conclusions can be listed as follows.

1. The intermediate water cycle was a reliable and feasible method to avoid the blockage of the important apparatus and pipes. With its help, the online real-time cleaning of equipment will be achieved and the corresponding cost will be decreased. However, as the result of the intermediate heat exchange, the average heat-absorbing temperature of ORC was declined and the ORC efficiency will definitely lower.

2. The heat loss in the intermediate water cycle should not be ignored. The intermediate cycle efficiency of 77.98% implied that the heat-work conversion of the overall system would be deteriorated and more efforts need be made to reduce the heat loss.

3. The four-stage axial turbine performed well even at highly off-design conditions. An average turbine efficiency of 78.52% was obtained. Therefore, the multi-stage axial turbine is suitable and desirable for the middle-large size geothermal ORC system. Moreover, the low rotational speed of the multi-stage turbine could avoid some strength and vibration problems, allowing for the safe and reliable operation
of the system.

4. The average ORC efficiency was about 4.46%, while a maximum of 5.33% could be obtained. Several reasons took the responsibilities for the lower efficiency including the insufficient mass flow rate of geothermal water, offline of the preheat, heat loss of the intermediate water cycle, lower turbine inlet pressure and higher turbine outlet pressure. It is believed that higher system efficiency could be achieved in the normal conditions.

Acknowledgement

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References


Figure captions:

Fig. 1. Schematic diagram of the geothermal ORC system

Fig. 2. Experimental layout of the geothermal ORC system, (a) front view, (b) back view

Fig. 3. Four-stage axial turbine

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Fig. 14 ORC efficiency during the preliminary test
Table captions:

Table 1 Design working parameters of turbine
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