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Experimental study of low-temperature organic Rankine cycle with axial flow turbine

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Abstract

We build a low temperature heat recovery ORC experimental system, in which R245fa is used as working fluids, a single-stage axial flow turbine is used to generate work, and the brazed plate heat exchangers are used as evaporator and condenser. The hot water is used to simulate the low-temperature heat source with temperature in the range of 320-370K, and cooling water at ambient temperature used as heat sink. Overall performance of the low-temperature ORC system is investigated. The results show that the axial flow turbine’s isentropic efficiency is near of 0.6 under different evaporating temperature, and the fluctuated characteristics of the turbine appears at evaporating temperature of near 350 K, and causes the stable overall performance of the ORC system at that temperature range. Moreover, overall net power output of the ORC system is over zero when evaporating temperature exceeds 350 K in this research.

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Keywords: Axial flow turbine; low temperature heat source; organic Rankine cycle; power generation.

1 Introduction

About one half of the industrialized waste heat is directly discharged to the ambient air [1]. In the context of today’s trend towards rational use of energy, recovery of the low temperature waste heat is one of the most important ways to increase energy utilization efficiency and reduce pollution emission, and development of conversions low temperature heat into mechanical power and electricity power arouses widely interesting [2-4]. The organic Rankine cycle (ORC) systems are particularly suitable for the low temperature heat recovery due to its use of organic working fluids with low boiling points and high density [5-11].

The small scale ORC systems (several kW) are still need research and develop further considering low temperature heat sources are dispersed distribution and small in quantity. Several theoretical and experimental investigations of small scale ORC systems using different turbines and working fluids have

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also been reported in recent years. Quoilin et al. [7] reviewed a large amount of experimental studies on small scale ORC using scroll turbine, the overall literatures data showed isentropic efficiency of the scroll turbine is of 26%-68%. Pei et al. [9] designed a radial turbine and studied the dynamic performance of a 1 kW scale ORC system with R123 as working fluid, the results showed the maximum turbine efficiency and ORC efficiency are 66% and 6.9%. Although a large amount of small-scale ORC experimental systems with different turbines and working fluids have been studied, seldom information is found on test of the small low-temperature ORC system with partial intake axial flow turbine.

In this paper, the hot water is used as heat source, and cooling water is used as cold source, the R245fa is used as working fluids of the ORC loop. A 5 kW axial flow turbine is designed and applied to the ORC system. The performances of the ORC system are tested under different conditions.

2 Experiment system and facilities

**Figure 1 Design chart of the ORC experimental system.**

Figure 1 shows the ORC experiment system, including three units: the hot water circuit (the heat source), the working fluid circuit and cooling water circuit (the cooling source). In hot water circuit, the water is recycled in a closed circuit and is heated by three diesel stoves to simulate three heat source loads. The water is heated by flue gas of three diesel stoves through the evaporator, and then recycled by hot water pump to the stove evaporator back. The hot water temperature at state point 1 can be controlled by a temperature controller, its maximum temperature is set of 390 K. The power generated by the axial-flow turbine is consumed by different electric-resistance loads (6.6Ω, 7.2Ω, 7.7Ω, 11Ω).

The working fluid circuit consists of evaporator, expander, generator, condenser, working pump, storage tank and other ancillary equipments. The liquid working fluid of R245fa absorbs sensible heat from hot water in the evaporator and becomes saturated vapor at high pressure, and then expands through the expander to generate mechanical power. The expander drives a synchronous generator to generate three-phase electric power, which is consumed by electric resistance (the loads can be changed by selecting different electric resistances). The vapour R245fa discharging the expander flows into the condenser and is cooled by cooling water, and then flows into the storage tank. The liquid R245fa in the storage tank is pressured by a variable frequency pump and send back to the evaporator. A partial intake
axial flow turbine is designed as an expander of the ORC system considering its simple structure. The primary design requirement of the axial-flow turbine is shown in Table 1.

### Table 1 Design parameters of the axial flow turbine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotation speed $n$, RPM</td>
<td>0-30,000</td>
</tr>
<tr>
<td>Rotor blade velocity factor $\psi$</td>
<td>0.84</td>
</tr>
<tr>
<td>Nozzle velocity factor $\phi$</td>
<td>0.89</td>
</tr>
<tr>
<td>Nozzle outlet angle $\alpha_1$</td>
<td>11</td>
</tr>
<tr>
<td>Average diameter $d$, mm</td>
<td>73</td>
</tr>
<tr>
<td>Partial intake degree $\beta$</td>
<td>0.75</td>
</tr>
<tr>
<td>Static blade height $l_1$, mm</td>
<td>0.0056</td>
</tr>
<tr>
<td>Rotary blade height $l_2$, mm</td>
<td>0.007</td>
</tr>
<tr>
<td>Number of nozzle</td>
<td>9</td>
</tr>
<tr>
<td>Number of rotor blade</td>
<td>27</td>
</tr>
</tbody>
</table>

In cooling water circuit, a 12 t/h cooling tower and a recycle water pump of 1.5 kW were used. The actual operation condensed temperature of R245fa is determined to be 3-5 K higher than atmospheric temperature in different seasons.

![Figure 2 General schematic of axial flow turbine. (a) Velocity triangles at static blade outlet and rotor blade outlet; (b) Enthalpy-entropy representation of the expansion process.](image)

Figure 2 shows general schematic of axial flow turbine (a) Velocity triangles at static blade outlet and rotor blade outlet; (b) Enthalpy-entropy representation of the expansion process. Table 2 gives the design parameters of the axial flow turbine. The detailed discussion of the most significant equations of the turbine model is presented in Appendix A.

Isentropic efficiency of turbine is given by

$$\eta_s = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad (1)$$

Power output of the axial flow turbine is calculated by

$$P_1 = m_t \cdot \eta_s \cdot (h_3 - h_{4s}) \quad (2)$$

Thermal efficiency of the ORC system is calculated by
\[ \eta_{\text{ORC}} = \frac{P_1}{Q_0} \times 100\% \] (3)

Where \( Q_0 \) is the overall heat recovered by the evaporator, kW.

3 Experiment data and analysis

The experimental results are obtained with three diesel ovens and 7.7\( \Omega \) electric resistances.

Figure 3 Relation between evaporation temperature, condensing temperature and time.

Figure 4 shows the relation between isentropic efficiency of organic turbine and evaporator temperature.

Figure 3 shows the relationship between evaporating temperature, condensing temperature and time. It is seen that the condensing temperature increases as the evaporating temperature is increased, and there exists a temperature fluctuation at evaporating temperature of near 350 K.

Figure 4 shows the relation between isentropic efficiency of organic turbine and evaporator temperature. It illustrates that the isentropic efficiency of axial-flow turbine is near 0.6 under different evaporating temperature. Since the turbine rotational speed increases as the evaporating temperature is increased, thus it is known from Figure 3 that turbine isentropic efficiency keeps stable and near 0.6 under different rotation speed.

Figure 5 Relation between thermal efficiency of ORC and evaporating temperature.

Figure 6 Relation between net power output of ORC system and evaporating temperature.
Figure 5 shows the relation between thermal efficiency of the ORC and evaporating temperature. It is seen that the thermal efficiency of the ORC system increases as the evaporating temperature increases due to the high enthalpy drop at high evaporating temperature. Figure 6 shows the overall net power output of the ORC system, it illustrates that overall net power increases as evaporating temperature is increased, and the net power output is overall zero when evaporating temperature exceeds 350 K. Combined the results shown in Figure 3 to Figure 6, it is observed that the ORC overall performance is fluctuated at evaporating temperature of 350 K due to the unstable characteristics of the axial-flow turbine at near critical rotational speed.

4 Conclusion

In this research, based on a 5kW axial-flow turbine, a low-temperature ORC system is experimental investigated. A partial intake axial flow turbine matches with a synchronous electricity generator is designed based on the thermodynamic properties of the R245fa. The turbine experimental results show isentropic efficiency of the designed turbine is of 60%, and it always works stable at high speed under different evaporation temperature and mass flow rate. The developed ORC system is very suitable for the small dispersed distributed low-temperature thermal recovery.

The dynamic performance of the ORC system was also investigated experimentally, and the factors which influence the performance are tested and analyzed. The maximum average cycle efficiency, turbine efficiency and electricity power is of 7.22%, 56.4% and 4.5kW, respectively. Exclude the power consumption of hot water pump, cold water pump and organic pump and electricity control system, net electric power output of the ORC system is over zero when evaporation temperature exceeds 350 K.

Acknowledgements

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Reference


Appendix A. Equations
In this section, the most important equations of the model are discussed, referring to the same notation proposed in. The expansion process in the axial flow turbine is illustrated in Figure 2, the ideal exit velocity of the nozzle is calculated by

\[ c_1 = \varphi \cdot c_{1s} = \varphi \cdot \sqrt{2000 \cdot \Delta h_s + c_0^2} \]  
(A1)

Where \( \Delta h_s \) is the enthalpy drop during the idea isentropic expanding process of 3-4s, \( \Delta h_s \) is expressed as

\[ \Delta h_s = h_3 - h_{4s} \]  
(A2)

The relative speed of \( w_1 \) is calculated by

\[ w_1 = \sqrt{c_1^2 + u^2 - 2 \cdot c_1 \cdot u \cdot \cos(a_1)} \]  
(A3)

Where \( a_1 \) is the exit gas flow angle of nozzle, \( u \) is the peripheral velocity, m/s. The peripheral velocity of \( u \) is calculated by

\[ u = \frac{d \cdot \pi \cdot n}{60} \]  
(A4)

The actual relative speed of \( w_2 \) is calculated by

\[ w_2 = w_{2s} \cdot \psi \]  
(A5)

Peripheral efficiency of \( \eta_u \) is expressed as

\[ \eta_u = \frac{\Delta h_s - h_n - h_b - h_{c2}}{\Delta h_s} \]  
(A6)

Where \( h_n \) is the nozzle flow loss, \( h_b \) is the rotary blade loss, \( h_{c2} \) is the leaving-velocity loss. \( h_n \) is calculated by

\[ h_n = \frac{c_{1s}^2 - c_1^2}{2000} \]  
(A7)

\( h_b \) is calculated by

\[ h_b = \frac{(w_{2s}^2 - w_2^2)}{2,000} \]  
(A8)

\( h_{c2} \) is calculated by

\[ h_{c2} = \frac{c_2^2}{2,000} \]  
(A9)

Isentropic efficiency of the axial flow turbine is calculated by

\[ \eta_i = \eta_u - \zeta_f - \zeta_v - \zeta_c \]  
(A10)

Where \( \zeta_f \) is the friction loss, \( \zeta_v \) is the blast loss, \( \zeta_c \) is the arc end loss. \( \zeta_c \) is calculated by
\[ \xi_t = \frac{0.001 \cdot d \cdot X_n^3}{e \cdot l_1 \cdot \sin(a_t) \cdot \mu_t} \]  
(A11)

Where \( X_n \) is the idea velocity ratio
\( \xi_v \) is calculated by
\[ \xi_v = \frac{0.065 \cdot l_2 \cdot X_n^3}{1 - e \cdot l_1 \cdot \sin(a_t) \cdot \mu_t} \]  
(A12)

\( \xi_e \) is calculated by
\[ \xi_e = \frac{0.135 \cdot l_2 \cdot B_2 \cdot \eta_v \cdot X_n}{e \cdot l_1 \cdot \sin(a_t) \cdot \mu_t \cdot d} \]  
(A13)

Where \( B_2 \) is the width of the rotary blade, \( \mu_t \) is the nozzle discharge coefficient.
\( \mu_t \) is calculated by
\[ \mu_t = \frac{V_1}{c_{ts} \cdot A_1} \]  
(A14)

Where \( V_1 \) is the exit volume flow rate of nozzle, \( A_1 \) is the exit area of nozzle.

**Biography**

My research focuses on the thermal and economic performance study of middle-low temperature organic Rankine cycle technologies.