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Modelling and optimization of organic Rankine cycle driven by industrial waste heat and solar energy

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ABSTRACT

In this study, an organic Rankine cycle (ORC) driven by solar energy and industrial waste heat is proposed to achieve a collaborative optimization from the heat source side and the working fluid side. An integrated heat exchanger network-organic Rankine cycle (HEN-ORC) system model is established in the GAMs environment. The parameters of physical properties for the working fluid are calculating by the PR equation of state. The optimal ORC integrated system driven by hybrid heat sources at different inlet temperatures of the waste heat streams are obtained through the developed optimization method. Comparative analysis on the ORC driven by waste heat, ORC driven by solar energy, and the hybrid heat source-driven ORC are conducted. In addition, the influences of working fluids on the system performance is also analyzed.

Keywords: Heat exchanger network, Organic Rankine cycle, Hybrid heat source, Integration, Mix integer non-linear programming

NONMENCLATURE

Symbols	
A_c	collecting area, m^2
dtorci	temperature difference at hot end, K
dtorco	temperature difference at cold end, K
FCSep	split ratio of heat capacity flow rate of cold stream, kW/K
FCT	heat capacity flow rate of cold stream, kW/K
FHSep	split ratio of heat capacity flow rate of hot stream, kW/K
FHT	heat capacity flow rate of hot stream, kW/K
G_b	solar beam radiation, kW/m^2
h	enthalpy, kJ/mol
I	Index of hot stream
J	Index of working fluid stream
m_{col}	heat capacity flow rate of the thermal oil, kW/K
MF	mole flow rate, mol/s
q	heat exchanging quantity, kW
Q_u	heat input in the ORC from the sun, kW
QC_{orc}	heat released by the working fluid in the condenser, kW
QE_{orc}	heat absorbed by the working fluid in the HRS, kW
QMAX	maximum heat exchanging quantity, kW
QMIN	minimum heat exchanging quantity, kW
STG	Index of stages in matching of hot and working fluid stream
T	temperature, K
Ti	outlet temperature of non-isothermal split cold stream after exchanging heat with hot stream, K
To	outlet temperature of non-isothermal split hot stream after exchanging heat with cold stream, K
TDMAX	maximum temperature approach, K
TMAPP	minimum temperature approach, K
WP_{orc}	organic fluid pump power, kW
WT_{orc}	work output from the expander, kW
Z	binary variables denote the existence of heat exchange units, -

η	efficiency, -
Subscripts	
am	ambient
col	collector
in	inlet
out	outlet
i	hot stream
j	cold stream
stg	stage

1. INTRODUCTION

With the rapid development of economy and society, the world is facing more and more serious problems of environment and energy. The large amount of medium and low temperature thermal energy discharged in the industrial production process should not be ignored. In addition, the world is abundant in clean energy such as solar energy and geothermal energy. Organic Rankine cycle (ORC) is considered one of the promising technologies utilizing medium and low temperature heat to generate power.

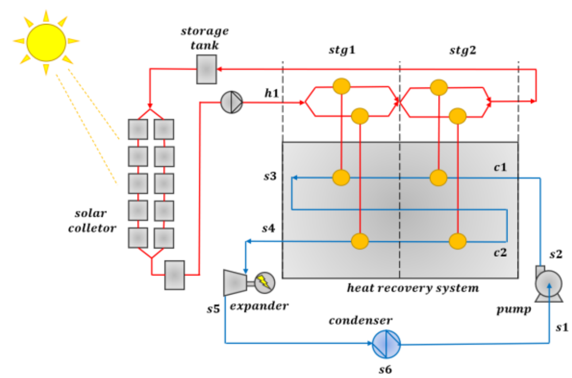
Large number of previous researches have been focused on the ORC components design and optimization, working fluid selection for cycle structure. The simulation and optimization for interior design of the ORC have been well studied. Recently, the exterior integration design of heat source driving the ORC to improve the utilizing efficiency of energy become more and more important. Huang et al. [1] proposed an integration system to recover the low-temperature surplus heat from process hot stream through the ORC. The ORC is used to replace the traditional external cold utility for cooling process hot stream. Thus, more thermal energy was recovered and the total annual cost of entire integration system was reduced. Bellos et al. [2] presented an integration concept hybrid heat source of waste heat stream and solar energy drive the ORC. Four ORC systems with different working fluids were studied by changing the inlet temperature of waste heat stream. However, the utilization of solar energy was limited to the two-phase region.

In this study, a method to integrate heat exchange network (HEN) in heat recovery system (HRS) is proposed to achieve a collaborative optimization from the heat source side and the working fluid side for the ORC system. Solar energy stream is able to exchange heat with working fluid in entire heating section and not only limited in two-phase heating region through calculation. The comparison between hybrid heat source-driven ORC system and single heat source-driven ORC operating in parallel mode is conducted to verify the effectiveness of optimization for heat exchange matching from heat source side. In addition, four working fluids are selected to study the performance for the proposed ORC system, illustrating the availability for improving the ORC system from working fluid side.

2. PROBLEM DESCRIPTION AND MODEL FORMULATION

2.1 System description

Fig. 1 shows the schematics of three integration systems, namely, the solar-driven ORC (SORC), the ORC system driven by waste heat stream (WORC), and the ORC driven by hybrid heat source of multiple waste heat streams and solar energy (SWORC). As shown in Fig.1, the ORC system is composed of a pump, a HRS, a expander and a condenser. The heat recovery system act as an evaporator. Working fluid and hot stream exchange heat through the HEN in the HRS and thus, the heat of waste heat and solar energy are utilized. Liquid working fluid is pumped into the HRS by the pump and heated into saturated vapor in the HRS. Then, the vapor expands in the expander and condensed into saturate liquid in the condenser.



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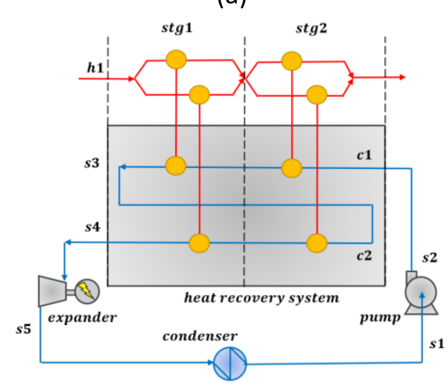

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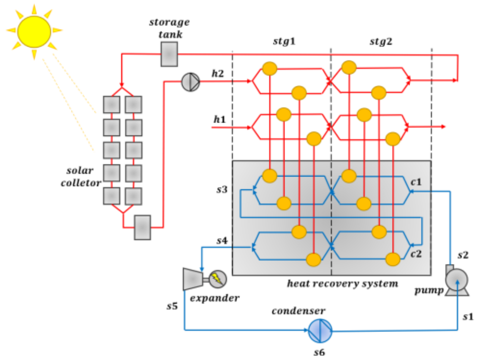


(c)

Fig. 1 Representation of three integration system: (a) SORC, (b) WORC, (c) SWORC

2.2 Model formulation

The model of the presented ORC system consists of solar energy collection model, ORC model, EOS model, HEN model and objective function.

2.2.1 Solar energy collection model

In the present study, the commercial parabolic trough collector (PTC) Eurotrough II is selected as solar energy collector. Eq. (1) shows the solar radiation energy Q_{solar} which is related to the collector area and solar radiation. Eq. (2) indicates the total heat of collected solar energy entering the ORC system. Eqs. (3)-(4) give the thermal efficiency of solar energy collector.

$$Q_{solar} = A_c * G_b \quad (1)$$

$$Q_u = m_{col} * (T_{col,out} - T_{col,in}) \quad (2)$$

$$\eta_{col} = Q_u / Q_{solar} \quad (3)$$

$$\eta_{col} = 0.7408 - 0.0432 * \frac{T_{col,in} - T_{am}}{G_b} - 0.000503 * G_b * \left(\frac{T_{col,in} - T_{am}}{G_b}\right)^2 \quad (4)$$

2.2.2 ORC model

The ORC model includes the mass and energy balances of a pump, a HRS, a expander and a condenser. Eqs. (5)-(6) show the energy balance of the HRS. The heat that the HRS absorbs from hot stream is equal to the heat that the working fluid absorbs from the HRS. Eqs. (7)-(9) give the energy balance for the condenser, expander, pump respectively.

$$QE_{orc} = \sum_{h\in I}\sum_{j\in J}\sum_{stg\in STG} q_{i,j,stg} \quad (5)$$

$$QE_{orc} = MF_j(h_4 - h_2) \quad (6)$$

$$QC_{orc} = MF_j(h_5 - h_1) \quad (7)$$

$$WT_{orc} = MF_j(h_4 - h_5) \quad (8)$$

$$WP_{orc} = MF_j(h_2 - h_1) \quad (9)$$

2.2.3 EOS model

In the present study, the EOS model consists of the PR-EOS model, Antoine equation model and thermodynamic properties calculation model. Compared with the calculation data of Aspen Plus V9, the accuracy of the PR-EOS is high enough as the compression factor's relative error is less than 0.1%.

2.2.4 HEN model

In the HRS, liquid working fluid is heated into saturated gas. The hot stream release heat to working fluid through the HEN. Eqs. (10)-(13) give the energy balance for hot stream and working fluid in the HEN. Eqs. (13)-(17) give the mass balance for hot stream and working fluid. Eqs. (18)-(21) show the temperature constrains of each stream. Eqs. (22)-(27) indicate the temperature and energy constrains for each heat exchangers.

$$FHT_i(T_{i,in} - T_{i,out}) = \sum_{j\in J}\sum_{stg\in STG} q_{i,j,stg}, \quad \forall i \in I \quad (10)$$

$$FCT_j(T_{j,out} - T_{j,in}) = \sum_{i\in I}\sum_{stg\in STG} q_{i,j,stg}, \quad \forall j \in J \quad (11)$$

$$q_{i,j,stg} = FHSep_{i,j,k}(T_{i,stg} - To_{i,j,stg+1}) \quad \forall i \in I, stg \in STG \quad (12)$$

$$q_{i,j,stg} = FCSep_{i,j,stg}(Ti_{i,j,stg} - T_{j,stg+1}) \quad \forall j \in J, stg \in STG \quad (13)$$

$$FHT_i T_{i,stg+1} = \sum_{j\in J} FHSep_{i,j,stg} To_{i,j,stg+1} \quad \forall i \in I, stg \in STG \quad (14)$$

$$FCT_j\, T_{j,stg} = \sum_{i\in I} FCSep_{i,j,stg} Ti_{i,j,stg} \quad \forall j \in J, stg \in STG \quad (15)$$

$$FHT_i = \sum_{j\in J} FHSep_{i,j,stg} \quad \forall i \in I, stg \in STG \quad (16)$$

$$FCT_j = \sum_{i\in I} FCSep_{i,j,stg} \quad \forall j \in J, stg \in STG \quad (17)$$

$$T_{i,stg+1} \le T_{i,stg} \quad \forall i \in I, stg \in STG \quad (18)$$

$$T_{i,out} \le T_{i,stg=CARD(STG)+1} \quad \forall i \in I, stg \in STG \quad (19)$$

$$To_{i,j,stg+1} \le T_{i,stg} \quad \forall i \in I, j \in J, stg \in STG \quad (20)$$

$$T_{i,out} \le To_{i,j,stg=CARD(STG)+1} \quad \forall i \in I, stg \in STG \quad (21)$$

$$dtorci_i \ge TMAPP \quad \forall i \in I \quad (22)$$

$$dtorci_i \le T_{i,stg} - Ti_{i,j,stg} + TDMAX(1 - Z_{i,j,stg}) \quad \forall i \in I, j \in J, stg \in STG \quad (23)$$

$$dtorco_i \ge TMAPP \quad \forall i \in I \quad (24)$$

$$dtorco_i \le To_{i,j,stg+1} - T_{j,stg+1} + TDMAX(1 - Z_{i,j,stg}) \quad \forall i \in HI, j \in J, stg \in STG \quad (25)$$

$$Q_{i,j,stg} \le QMAX_{i,j} Z_{i,j,stg} \quad \forall i \in I, j \in J, stg \in STG \quad (26)$$

$$Q_{i,j,stg} \ge QMIN_{i,j} Z_{i,j,stg} \quad \forall i \in I, j \in J, stg \in STG \quad (27)$$

2.2.5 Objective function

The objective function is the maximization of the net power output (NPO) of the ORC system as formulated by Eq. (28).

$$Max\ NPO = WT_{orc} - WP_{orc} \quad (28)$$

The formulated model is a mixed integer non-linear programming model. The optimization variables as follow: the inlet and outlet temperature, heat capacity flow rate of solar stream, the existence of heat exchanger at each stage or not, ORC working fluid mole flowrate, evaporation pressure and condensing pressure of the ORC system. In the present study, the MINLP model is formulated in GAMS 25.1 on a 3.2GHz Intel (R) Core (TM) 2 PC, and SBB is selected to solve the MINLP model. [3]

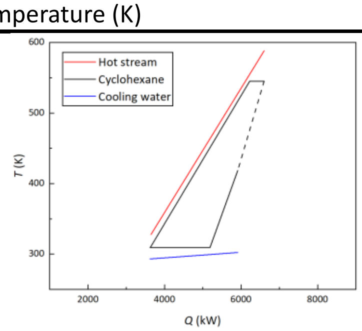
3. RESULTS AND DISCUSSION

The basic parameters are listed in Table 1. Fig.2 (a) gives the *T-Q* diagram of SORC. Fig.2 (b) and Fig.2 (c) show the *T-Q* diagrams of WORC and SWORC where the inlet temperature of waste heat stream is 473.15K.

As shown in Fig.2, irreversible loss is mainly located in the two-phase heating zone for the cyclohexane-WORC, and irreversible loss in the preheating section can't also be ignored. The SORC provide the better matching of the ORC system and heat source in the preheating section. The SWORC provides less irreversible loss in two-phase region and most part of the preheating section. The maximum *NPO* of SWORC increase by 38.5% compared to the total maximum *NPO* of SORC and WORC under the same inlet temperature of waste heat, and the system efficiency of SWORC increase by 3.6%.

Table 1 The basic parameters

Parameter	Value
Heat capacity flow rate of waste heat stream (kW/K)	20
Isentropic efficiency for expander	0.8
Isentropic efficiency of compressor	0.8
ΔT_{min} (K)	10
Ambient temperature (K)	298.15



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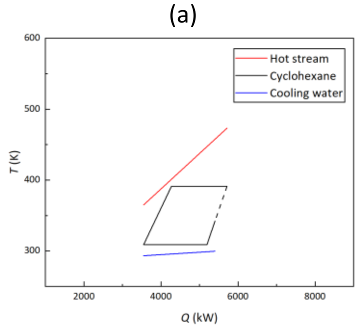

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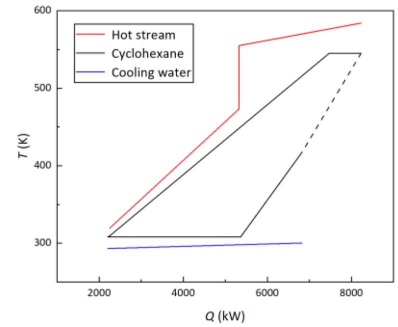

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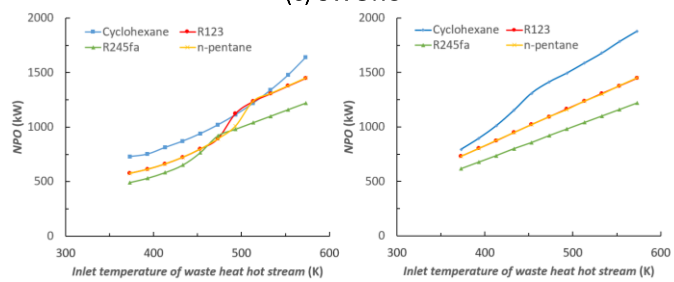
Fig 2 *T-Q* diagram of three ORC system: (a) SORC; (b) WORC; (c) SWORC



(a) (b)

Fig. 3 Maximum *NPO* of different working fluid system: (a) SORC+WORC, (b) SWORC

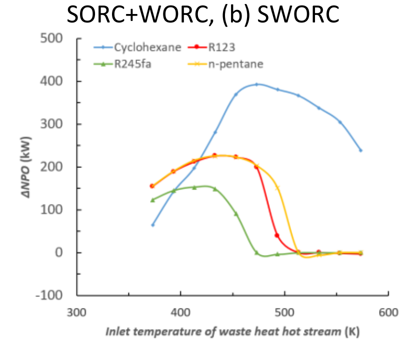
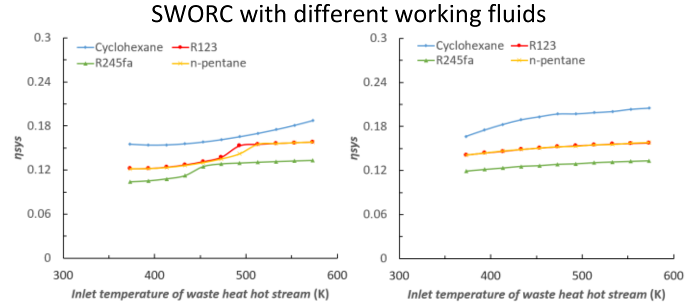


Fig. 4 Absolute increase value for the maximum *NPO* of the SWORC with different working fluids

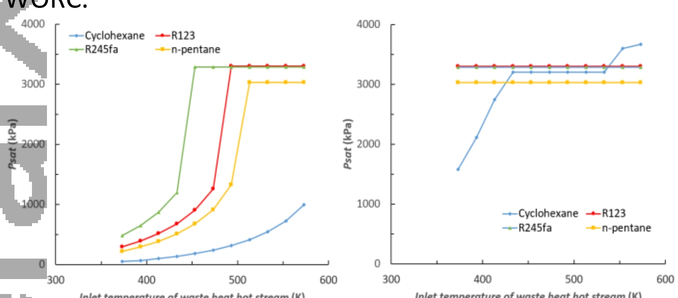


(a) (b)

Fig. 5 System efficiency of different working fluid system: (a) SORC+WORC, (b) SWORC

Fig. 3 show the maximum *NPO* of the ORC system with different working fluid varies with the inlet temperature of waste heat stream. The results indicate that the ORC performances are influenced by the inlet temperature of waste heat stream and working fluid. As shown in Fig.3, The maximum *NPO* of the ORC system increases monotonously with the increase of the inlet temperature of waste heat stream. The maximum *NPO* of the SWORC with cyclohexane at each inlet temperature is higher than that of the SWORC with any other selected working fluid and it varies from 793.92kW to 1879.73kW. The maximum *NPO* of the SWORC with R123 is similar to that of the SWORC with n-pentane. Fig.4 gives the difference between the maximum *NPO* of the SWORC system and the total maximum *NPO* of the SORC and WORC varies as the change of the inlet temperature of waste heat stream. The difference of the maximum *NPO* first increases and then decreases with the increase of inlet temperature of the waste heat stream.

Fig. 5 shows the influence of inlet temperature of waste heat stream for the system efficiency of the ORC system with different working fluid. It indicates that the total system efficiency of two independent ORC system utilizing cyclohexane is greater than that of the corresponding system with other selected working fluids. So does the system efficiency of the SWORC. The system efficiency of the SWORC is higher than that of the total system efficiency of two independent ORC system at the same temperature for all selected working fluids. As shown in Fig. 6 (a), the ORC evaporation pressure increases first, reaching the maximum evaporation pressure and then tend to stabilize with the increase of inlet temperature of waste heat stream in the WORC for R123, R245fa and n-pentane. However, the evaporation pressure of the WORC with cyclohexane increases continually as inlet temperature of the waste heat stream increases. Fig. 6 (b) indicates the evaporation pressure of the SWORC with R123, R245fa or n-pentane reach the maximum from the beginning and then stays constant. The maximum *NPO* and system efficiency are related to the evaporation pressure. If the evaporation pressure of the WORC does not reach the maximum, the maximum *NPO* of the SWORC has improvement at the corresponding inlet temperature of waste heat stream compared with the total maximum *NPO* of the SORC and WORC.



(a) (b)

Fig. 6 Evaporation pressure of different working fluid system: (a) WORC, (b) SWORC

4. CONCLUSION

A simultaneous optimization model for solar field, waste heat stream and the ORC system is formulated. The comparisons for different kinds of the ORC system are conducted through optimization calculation in GAMs environment. Following conclusions are drawn.

The heat source and working fluid have a better heat exchange match in the SWORC in contrast with the SORC and WORC system operating in parallel mode. The cyclohexane-ORC system yields maximum NPO and system efficiency at each inlet temperature of waste heat stream compared with the SWORC system utilizing the other working fluids. The improvement of maximum NPO for the SWORC system is related to maximum evaporation pressure of working fluids. The maximum relative increase of the maximum *NPO* for the SWORC system is 38.5%.

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