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MULTI-OBJECTIVE OPTIMIZATION OF TURBO-EXPANDER IN ORGANIC RANKINE CYCLE SYSTEM

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ABSTRACT
This paper is a mathematical model of low-temperature sub critical Organic Rankine Cycle(ORC) was established in m-file MATLAB, the main contribution of this work was presented by designing the system conjugated with turbo-expander based on the PSO-multi-objective optimization. Under the condition of given cold and heat source parameters, the first function was total heat transfer and the other function was the circulating thermal efficiency required for the power generation. Our work carried out comprehensive parameters of objective functions. The result shows that the reisan optimum pinch point temperature difference in the heat exchangers and an optimum turbo-expander rotational speed that can minimize the combined objective function of ORC system

KEYWORDS: Low-Temperature Waste Heat; Organic Rankine Cycle; Turbo-Expander; PSO-Multi-Objective Optimization

INTRODUCTION
In recent years, with the continuous growth of energy consumption using low-temperature waste heat become the focus of attention of researchers. The organic Rankine cycle (ORC) power generation system has the advantages of high efficiency, pressure suitableand friendly with environment compared with the traditional steam Langken cycle power generation system (Quoilin, S.et. al. 2013).Applications of Low-grade heat are an effectivethermal energy because of different working fluid can recover different range of temperature (Chen et. al. 2010, Tchanche, B. F. et. al. 2011).The expander is the foundation equipment of the ORC system, and the expander has both speed type and volume type (Lemort, V. et. al. 2009). The radial turbo-expander can give a high internal efficiency under the condition of small volume flow (Fiaschi, D. et. al. 2015), but also use different inlet guide vanes to match the geothermal resources at different temperatures so that the system was designedin conditions to maintain a high efficiency(Costall, A. W. et. al. 2015, Zhang, Z. et. al. 2015). The efficiency of the expander is usually set as a fixed value, follow-on the best conditions of the turbo-expander (Bloch et. al. 2001). But, in current study the parameters of ORC system was not consistent with the optimal conditions.

ORC system commercial operation is opposite two major obstacles: (1) necessary design the components to optimize the lack of dynamic operation; (2) the economy limitation(Bernardoni& Christian 2016). Therefore, the study of ORC system in addition to the need to improve thermal efficiency and efficiency, how to reduce investment and operating costs is also an urgent problem to be solved. ORC system, when the temperature difference between the cold and heat source is small, the heat exchanger (including the evaporator and condenser) accounted for the largest investment cost, the proportion of sometimes even as high as 80% to 90% (BaljeO. E. 1981, Konak et. al. 2006, Chen et. al. 2004). In order to improve the system economy, it is necessary to reduce per unit the required heat transfer of power generation (Murata et. al. 1996). In this paper, MATLAB is used as the working fluid, and the total heat transfer and the circulating heat efficiency as the objective function were used to optimize the ORC system. Based on this, the influence of the parameters of the cold and heat source on the system performance is analyzed, In order to obtain parameters achieve the optimal thermal operation of the system.

MATHEMATICAL MODEL OF ORC SYSTEM
Figure 1 shows ORC system's flow chart, the working of fluid pump, evaporator, turboexpander and condenser four main equipment. The liquid organic working fluid is pressurized by the working fluid pump into the evaporator, and then enters the turboexpander to expand the external work. After the work of the organic work in the condenser was condensed into liquid, and then pumped into the evaporator after the pump to complete a closed cycle. The heat process is shown in Fig. 2, which consists of four processes: 1 → 2, adiabatic boost in the pump; 2 → 3, adiabatic endotherm in the evaporator; 3 → 4, adiabatic expansion in the turboexpander, External work; 4 → 1, in the condenser equal pressure exothermic

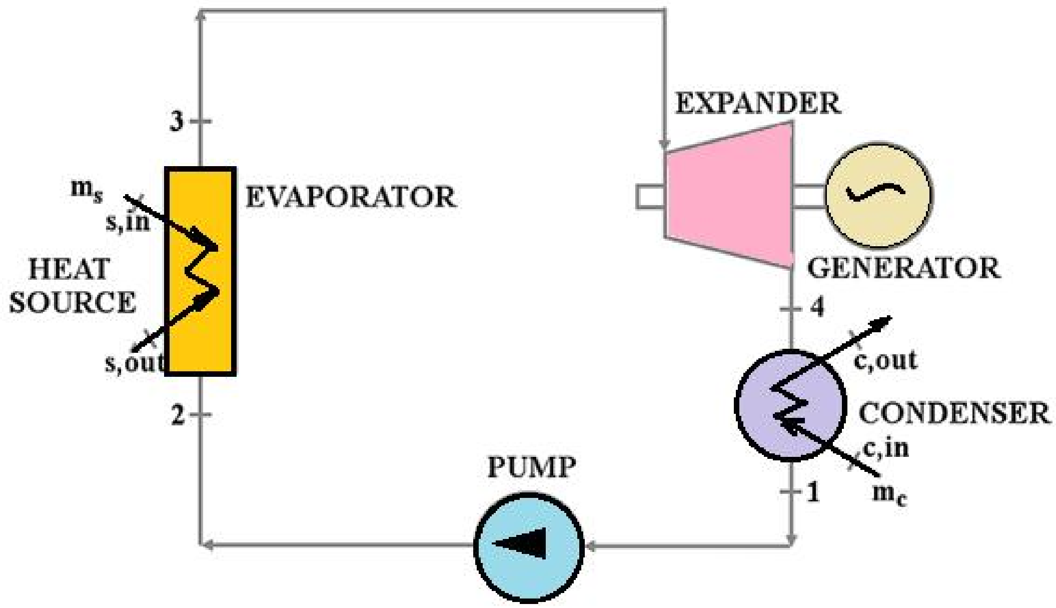

Fig. 1. Scheme of the simple ORC cycle.

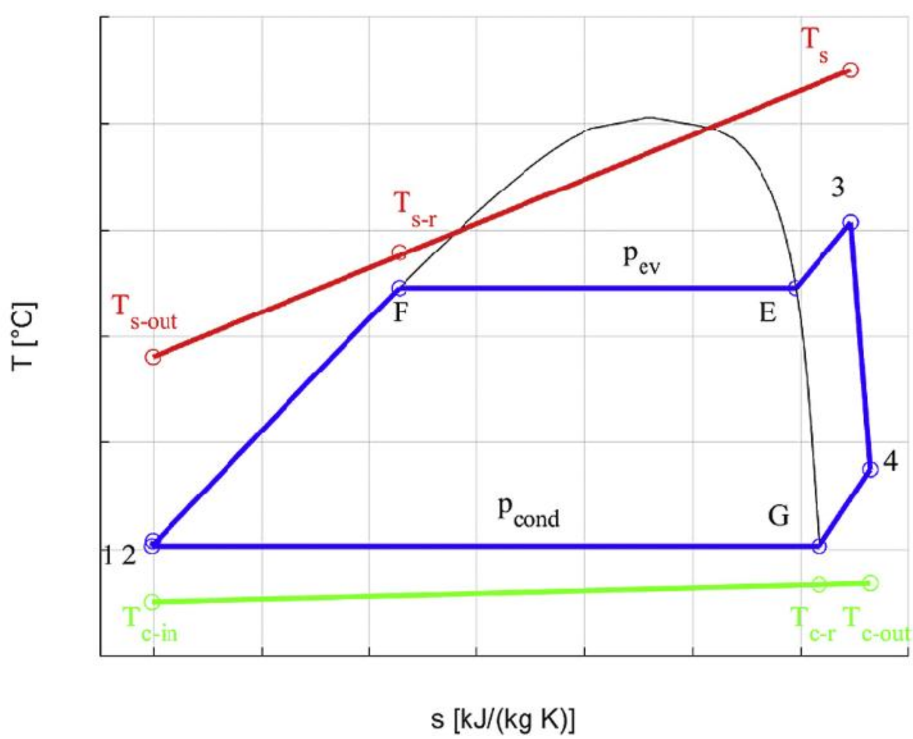

Fig.2 ***T-s* diagram of organic Rankine cycle system.**

To simplify the calculation, the mathematical modeling process assumes the following assumptions:

(1) The system is in a steady flow state.

(2) The total heat transfer coefficient of the condenser and the evaporator is regarded constant

(3) Neglects heat losses of the system; losses of working fluid resistance; Equivalent entropy efficiency of the pump

Heat exchanger heat transfer model

2.1.1 Condenser model of the system uses water cooling method to cool the working fluid, organic refrigerant in the condenser heat dissipation:

$$Q_{con} = m_{cf} c_{p,cf} \left(T_{cf,out} - T_{cf,in} \right) = m_{wf} \left(h_4 - h_1 \right) \quad (1)$$

Where Q is heat, kW; m is the mass flow rate, kg / s; cp is the specific enthalpy, kJ / kg. The subscripts "1" and "4" correspond to the state points in Fig. 1. According to the heat transfer equation, the heat transfer area of the condenser is:

$$A_{con} = \frac{Q_{con}}{U_{con} \Delta T_{ln,con}} \quad (2)$$

Where A is the heat transfer area (m^2), U is the heat transfer coefficient W/(m^2·K) ;

ΔT_{In} is the difference of temperatures K.

Evaporator model.

The amount of heat absorbed by the organic working fluid in the evaporator is:

$$Q_{eva} = m_{hf} c_{p,hf} \left(T_{hf,in} - T_{hf,out} \right) = m_{wf} \left(h_3 - h_2 \right) \quad (3)$$

Where, the subscript "eva" and "hf" denote the evaporator and hot water, respectively;

The subscripts "2" and "3" correspond to the state points in Fig 1.

According to the heat transfer equation, the heat transfer area of the evaporator is:

$$A_{eva} = \frac{Q_{eva}}{U_{eva} \Delta T_{ln,eva}} \quad (4)$$

The logarithmic mean temperature difference between the heat exchanger (evaporator and condenser) is calculated by the segmentation method (liquid phase, gas-liquid two-phase and gas phase). The formula is as follows:

$$\Delta T_{ln} = \frac{Q}{\sum_{i=1}^{3} \frac{\Delta Q_i}{\Delta T_i}} \quad (5)$$

$$\Delta Q_i = m_{wf} \Delta h_i \quad (6)$$

$$\Delta T_i = \frac{\Delta T_{max} - \Delta T_{min}}{\ln \frac{\Delta T_{max}}{\Delta T_{min}}} \quad (7)$$

Where i is the order of the liquid phase, the gas-liquid two-phase and the gas phase of the working fluid, i = 1, 2, 3.

Expansion machine model

From the evaporator out of the organic working fluid to superheated steam form into the expansion machine work, steady state under the expansion of the output of the machine:

$$W_{exp} = m_{wf} \left(h_3 - h_4 \right) = m_{wf} \left(h_3 - h_{4,s} \right) \varepsilon_{exp} \quad (8)$$

The turboexpander efficiency is calculated by empirical formula (BaljeOE. 1981) as follows:

$$\varepsilon_{exp} = 0.87 - 1.07 \left(n_s - 0.55 \right)^2 - 0.5 \left(n_s - 0.55 \right)^3 \quad (9)$$

ns is the specific speed, the formula is as follows:

$$n_s = \frac{2\pi N m_{wf}^{0.5}}{60 \rho_{wf}^{0.5} \Delta h_T^{0.75}} \quad (10)$$

The expansion ratio of the turboexpander is defined as the ratio of the pressure at the inlet and outlet:

$$\delta = P_3 / P_4 \quad (11)$$

Where W is the power in (Kw), N is the speed in (rpm), ε is the Equivalent entropy efficiency, ρ is the density in (Kg/m^3), "exp" is the expander, "s" is the isentropic process. P is the presser in (Pa)

2.3 working fluid pump (pup) model

ORC operation process, the working fluid power consumption:

$$W_{pup,wf} = m_{wf} \left(h_2 - h_1 \right) = m_{wf} \left(h_{2,s} - h_1 \right) / \varepsilon_{pup,wf} \quad (12)$$

2.4 Pump model

Circulating water pumps include hot water pumps and cooling water pumps, the power consumption were:

$$\eta_{net} = \frac{W_{net}}{Q_{eva}} \quad (15)$$

Total heat transfer area of ORC system

$$A_{tot} = A_{evp} + A_{con} \quad (16)$$

Net power generation of ROC System

$$W_{net} = W_{exp} \varepsilon_{gen} - W_{pup,wf} - W_{pup,hf} - W_{pup,cf} \quad (17)$$

Where the subscript "gen" indicates the generator.

MULTI-OBJECTIVE OPTIMIZATION MODEL AND ITS SOLUTION

OPTIMIZATION MODEL

Optimize target

In order to take account into the technical and economic nature of the ORC system, this paper uses the total heat transfer area and the cyclic thermal efficiency required for the unit net power generation as the optimization target construction function as follows:

$$\min f_1(X) = A_{tot} / W_{net} \tag{18}$$

$$\min f_2(X) = 1/\eta_{net} \tag{19}$$

The above objective function requires a minimum in a certain range, where f1 is the total heat transfer area required for the net power generation unit of ORC system, and f2 is the reciprocal of the thermal efficiency of the ORC system. In this paper, we use the linear weighted sum method to construct two objective functions into a comprehensive objective function. The expression is:

$$\gamma = \omega_1 f_1(X) + \omega_2 f_2(X) \tag{20}$$

Where ω1 and ω2 are the weighting factors and can be calculated according to the following formula:

$$\omega_1 = \frac{f_{1,\min}^2 - f_{2,\min}}{\left(f_{2,\min}^1 - f_{1,\min}\right) + \left(f_{1,\min}^2 - f_{2,\min}\right)} \tag{21}$$

$$\omega_2 = \frac{f_{2,\min}^1 - f_{1,\min}}{\left(f_{2,\min}^1 - f_{1,\min}\right) + \left(f_{1,\min}^2 - f_{2,\min}\right)} \tag{22}$$

Constraint function in the optimization model to solve

1) consider the range of the turboexpander speed is set as: 5000(rpm) <N <150000(rpm);

2) Temperature difference is the minimum heat transfer temperature difference in the heat exchanger, the simulation of the pinch temperature difference is usually taken from 3 to 20 K, so this paper set Tpp ≥ 3 K;

3) according to the system heat transfer process, Heat can be carried out the hot water inlet temperature should be greater than the evaporation temperature, the evaporation temperature is greater than the condensation temperature, the condensation temperature is greater than the cooling water inlet temperature, ie

$$T_{hf,in} > T_{eva} > T_{con} > T_{cf,in}$$

OPTIMIZATION ALGORITHM

Multi-objective Particle Swarm Optimization

PSO algorithm has been successfully applied in a wide of variety of optimization tasks in which it has shown a high convergence rate this algorithm explain briefly in chapter 2. However, Multi-objective optimization is quite different from single-objective optimization (Konak et. al. 2006), the latter has one objective function so it is easy to calculate the global best position, then the basic PSO version is not effective to solve multi objective problem directly, because there are many objectives and incommensurable the global best particle among all these objectives, it is impossible to create all objective functions reach their minimum value at the same time (Chen et. al. 2004). The effective method to solve the desired solution thought MPSO algorithm by converting the Multi - objective problem into a Single - objective problem using Weighted Sum Approach (WSA) (Murata et. al. 1996), which represented in an equation (23)

$$\min F(x) = \sum_{j=1}^{m} w_j \cdot f_j(k) \tag{23}$$

Where: $w_m \in [0,1]$, $\sum_{j=1}^{m} w_m = 1$

$\boldsymbol{f_j}$ **(k)** representing the best optimal finesses value of the j-th objective function in the k-th generation.

Hopefully by expression (1) we can get the Pareto solutions along 23 points against 23 Global best particles presented by the w_1 =[0.04 0.08 0.12 ……… 0.96] and w_2 =[0.96 0.92 0.88……… 0.04] making the current solutions to move toward the direction of the minimum distance from current position to each objective's best optimal value.

CALCULATION RESULTS AND ANALYSIS

This paper assumes that the heat source is 373.15 K hot water; the cooling water inlet temperature and the ambient temperature are set at 293.15 K. The mass flow rate of the heat source fluid is set to 1.0 kg / s (Cavazzini, G. et. al. 2017).

In the parameters given on the ORC system from the results of the optimization are shown under given operating conditions, the condenser folder temperature difference, the evaporator pinch temperature difference and vortex

Under the optimal design conditions, the total heat transfer area of the ORC system is 23.18 m^2, the net power generation is 5.75 kW, the thermal efficiency,the rotational speed of the wheel expander is the best47704 r / min., which is 6.94 K, 13.82 K, respectively. The rate is 7.55% and the overall objective function is 2.04.

The results of the optimization are obtained under the condition of given cold and heat source conditions. However, in the practical engineering application, the temperature difference between the hot and cold source temperature and the hot and cold source is often fluctuating. After the optimization (the evaporator, the condenser temperature difference and the turboexpander speed remain unchanged) ORC system for cold and heat source temperature and cold and heat source temperature difference between the sensitivity of the import and export changes.

Temperature difference between the mouth temperature and the temperature of the evaporator, the heat absorbed by the working fluid in the evaporator remains unchanged, and the logarithmic mean temperature difference of the evaporator heat transfer increases from 21.36 K to 22.23 K due to the heat transfer coefficient, Resulting in reduced heat transfer area from 14.93 m^2 to 14.42 m^2. At the same time, the increase of hot water temperature will lead to the increase of the evaporation temperature of the working fluid, which leads to the increase of the temperature of the refrigerant in the expander. With the increase of the inlet temperature, the temperature and the temperature of the expander of the expander are gradually increased, the expansion efficiency increases first and then decreases, and the expansion ratio of the expander increases gradually, which leads to the increase of enthalpy in the expander. The output of the expansion of the more power. So the net output power and thermal efficiency of the system gradually increase, while the energy conservation is easy to know, the condenser heat load decreases. Fixed cooling water parameters and the temperature difference between the condenser grips, due to the same heat transfer coefficient, heat transfer equation from the condenser to reduce the heat transfer area, the size reduced from 8.65 m^2 to 8.29 m^2. So as the temperature of the heat source increases, A_{tot} Decreases, W_{net} rises, η_{net} rises, so that γ decreases gradually as shown in Fig 3.

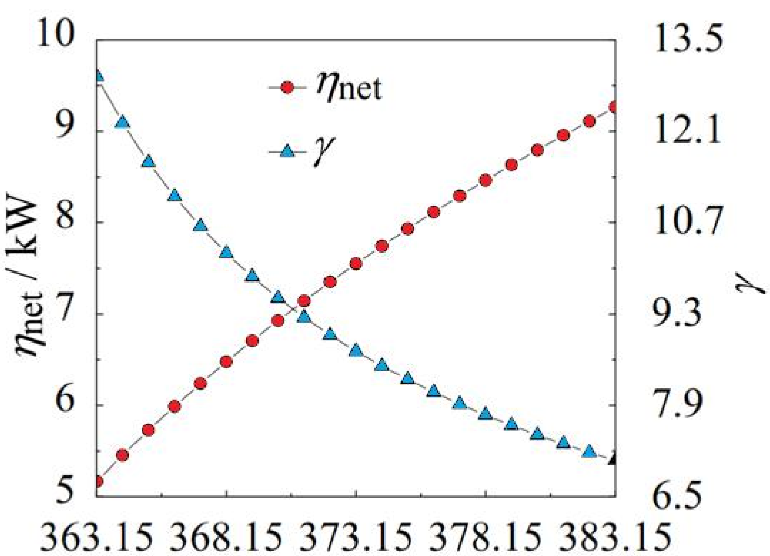

Fig.3 Heat efficiency and combined with parto of objective function versus hot water temperature.

CONCLUSION

A thermodynamic optimization of Organic Rankine Cycle (ORCs) for power generation from heat source temperatures in the range between 80 and 150C was carried out. Influence of Variation of Cold and Heat Source was the first function gave an optimal heat exchanger had difference temperature and the turbo-expander speed other function was maximized the rise in the cooling water temperature that mean measured the difference between outlet of hot water as well as cooling water (inlet) A simple sub-critical ORC cycle has been modeled and optimized by means of a PSO based algorithm, which was modified in order to allow the swarm particles to dynamically search for the best working fluid among a list of 37 candidates simultaneously. As a consequence, during the optimization procedure, the search domain of each particle of the swarm was continuously and dynamically modified iteration-by-iteration due to the different vapor saturation lines ofthe chosen working fluids. The assignment of each particle to a working fluid was based on a probability distribution, which was assumed to be initially uniform in order to allow the swarm to uniformly investigate the working fluid candidates. Subsequently, the assignment was based on the roulette-wheel process, which favors the most performing fluids in order to concentrate the swarm on them with a consequent significant reduction in computation time, which was halved in comparison with distinct single-fluid optimizations and showing comparable results in terms of accuracy

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