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Multi-criteria design optimization and screening of heat exchangers for a subcritical ORC

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Abstract

Evaporator and condenser are two main components of ORC and they are predominant in exergy destruction, cost and material consumption. The optimization and screening of evaporator and condenser is important to achieve the optimal performance of ORC. In this paper, the popular used shell and tube heat exchangers and plate heat exchangers are selected as candidate heat exchangers for ORC. The detailed parametric and structural models of heat exchangers are formulated. The candidate heat exchangers are optimized and screened under the objective of maximum thermal efficiency (THE), minimum specific cost (SIC) and minimum heat exchanger area per unit power output (APR). Genetic algorithm (GA) is used to solve the model. Then two candidates of ORC schemes with different heat exchangers are screened by fuzzy multi-criteria decision making process. Single objective and fuzzy multi-criteria design optimization and screening cases are demonstrated to testify the proposed methodology.

Keywords: ORC, Shell-and-tube heat exchanger, Plate heat exchanger, GA, system optimization;

1. Introduction

Energy shortage and environmental deterioration have become serious issues. A large amount of waste heat is released into the environment from industrial plants. This condition results in serious energy loss and environmental pollution. Organic Rankine cycle (ORC)-based power generation is a promising technology to recover waste energy and/or effectively utilize renewable energy with low enthalpy [1]. The main equipment of ORC includes heat exchangers, an expander, and a pump. Heat exchangers, such as the evaporator and condenser, are the most important components in ORC. Research has shown that the exergy losses of the condenser and evaporator account for 70% to 90% of the total exergy loss in ORC [2,3], and the capital investment cost of heat exchangers accounts for 40% to 90% of the total ORC investment cost [4,5]. ORC driven by low-temperature heat source are more affected by pressure drop and condenser area of heat exchanger than those driven by traditional high-temperature heat source. Increasing the heat transfer coefficient and/or reducing the pressure drop are significant in improving the performance of ORC. Therefore, the screening, design, and optimization of the heat exchangers are important to increase the comprehensive performance of ORC.

Many studies have focused on the parametric and structural optimization of ORC involving different variables and considering different objectives [3,4]. In these studies, the types of heat exchangers are predetermined and the differences between ORCs containing different heat exchangers are not compared. Although Walraven et al. [5] optimized and compared shell-and-tube and plate heat exchangers in ORC; their comparison was limited on the single objective evaluation of exergetic efficiency.

As is well known, the heat exchanger of different type is different in configuration, cost, and compactness. One heat exchanger is superior to another at certain operation condition under different evaluation criteria. Therefore, finding

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optimal heat exchangers as well as their optimal operating parameters under multiple criteria are important for the design of ORC. In this paper, a fuzzy multi-criteria design optimization and screening methodology is presented. The popular used shell-and-tube and plate heat exchangers are selected as candidate heat exchangers for evaporator and condenser. The detailed models of shell-and-tube and plate heat exchangers are formulated. The design objectives are maximum exergy efficiency (THE), minimum specific cost (SIC) and minimum heat exchanger area per unit power output (APR). A GA is applied to simultaneously achieve the optimal configuration of heat exchangers and optimal system parameters. The optimal schemes of ORC with two types of heat exchangers obtained under the three objectives, respectively, are evaluated by the fuzzy multi-criteria design optimization. A case study of an industry waste heat driven ORC is elaborated to testify the proposed methodology.

2. Organic Rankine cycle

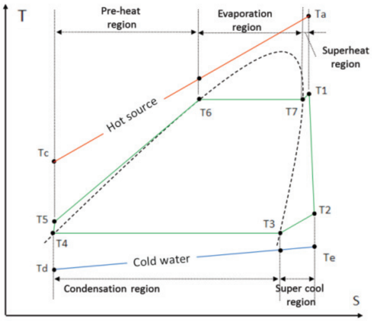

Fig. 1. T-S diagram of an ORC system

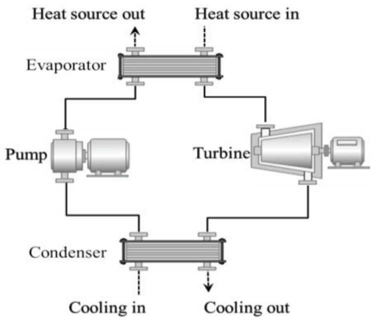

Fig. 2. Schematic diagram of an ORC system

Fig.1 gives the T-S diagram of the cycle. The liquid organic working fluid are compressed to a high pressure (4-5), and then fed to the evaporator where it is preheated (5-6), evaporated (6-7), and superheated (7-1) to high-temperature vapor. Then the superheated vapor expand in the turbine (1-2) to a low pressure to generate electricity. Afterwards, the turbine exhaust is cooled (2-3), condensed (3-4) in the condenser to liquid by cooling water or cooling air.

In this paper, a sub-critical organic Rankine cycle using R134a as working fluid is investigated. Fig.2 shows the schematic diagram of this system. The heat source is industry exhaust oil with temperature of 120℃ and mass flow rate of 0.5kg/s. The cold source is cooling water with inlet temperature of 20 ^{0}C. In all configurations, it is assumed that the state of working fluid at the inlet of turbine and pump are saturated vapor and saturated liquid, respectively. The isentropic efficiencies of the pump and turbine are assumed to be 0.80 and 0.85, respectively.

3. Modelling of heat exchanger

3.1 Plate heat exchanger

Plate heat exchanger is one of the most widely used heat exchangers in ORC system due to its high efficiency and compact structure. In this paper, a well know herringbone corrugation plate heat exchangers is explored. Fig.3. gives the structure and passage of a countercurrent single-pass flow plate heat exchanger. The geometry dimension of the plate surface are characterized by plate width W, plate length L, plate thickness δ, channel spacing b, and chevron angle β. A set of assumptions are made as follows in order to simplify the theoretical models: 1) the PHE working under steady state conditions; 2) heat losses that reject to the environment is negligible; 3) the flow in channels is fully developed.

The heat transfer equation for the heat transfer is given by Eq. (1), where, Q is the heat load, U is the overall heat transfer coefficient; A is the heat transfer surface area; Δt_m is the logarithm mean temperature difference (LTMD).

$$Q=AU\Delta t_m \quad (1)$$

The overall heat transfer coefficient is given by Eq.(2), where, h_h and h_c are the convection heat transfer coefficients for the hot side and the cold side respectively; λ_m is the thermal conductive of the plate; h_{hd} and h_{cd} are the dirt coefficients for the hot side and the cold side respectively.

$$\frac{1}{U}=\frac{1}{h_h}+\frac{1}{h_{hd}}+\frac{\delta}{\lambda_m}+\frac{1}{h_{cd}}+\frac{1}{h_c} \quad (2)$$

Single-phase flow heat transfer correlations

The convection heat transfer coefficient h can be calculated by Eq. (3) [6], Where, D_h is the hydraulic diameter of flow channel [7, 8].

$$h = 0.724\frac{\lambda}{D_h}\left(\frac{6\beta}{\pi}\right)^{0.646} Re^{0.583}Pr^{1/3} \quad (3)$$

Two-phase flow heat transfer correlations

In two-phase flow process, especially in evaporator and condenser, a modified LMTD method is employed to achieve more accurate results. The two-phase heat transfer process are divided into N equal differences, each section are thought to be small enough that the property can be assumed to be constant. So the overall enthalpy change is also divided into N equal differences, and the corresponding properties of each selection can be determined. The convection heat transfer coefficient on hot side of condenser for each section is expressed as [9]:

$$h_{h,i} = 4.118\frac{\lambda_i}{D_h}Re_{eq,i}^{0.4}Pr_i^{1/3} \quad (4)$$

The convection heat transfer coefficient on cold side for each section in evaporator is expressed as [10], Where, Pr_l is the Prandtl number of saturation liquid; $Re_{eq,i}$ and $Bo_{eq,i}$ are equivalent Reynolds and Boiling numbers in each section; $\chi_{m,i}$ is the mean vapor quality in each section; G is the mass velocity of the working fluid.

$$h_{c,i} = 1.926\frac{\lambda_i}{D_h}Re_{eq,i}^{0.5}Pr_l^{1/3}Bo_{eq}^{0.3}\left[1-\chi_{m,i}+\chi_{m,i}\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right] \quad (5)$$

$$Re_{eq,i} = \frac{GD_h}{\eta_1}\left[1-\chi_{m,i}+\chi_{m,i}\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right] \quad (6)$$

pressure drop correlations

The plate pressure drop can be divided into two parts, the friction pressure loss ΔP_c in channel and pressure drop due to the contraction and expansion losses through the ports ΔP_p [12], Where, L_p is the path lengthl N_p is the number of passesl u_c and u_p are the velocity through channel and port, respectively.

$$\Delta P_c = 4.8Re^{-0.3}(L_p/D_h)\frac{\rho u_c^2}{2} \quad (7)$$

$$\Delta P_p = 1.3\frac{\rho u_p^2}{2}N_p \quad (8)$$

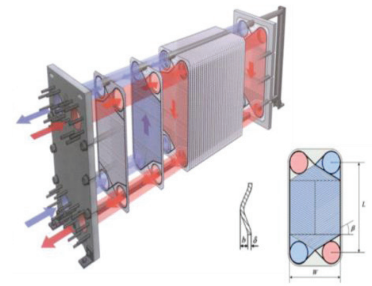

Fig.3. Countercurrent single-pass flow in plate heat exchanger

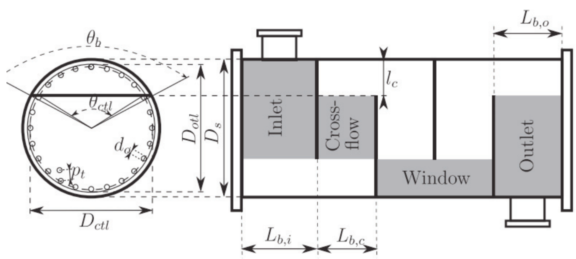

Fig.4. Shell-and-tube heat exchanger geometry and tube pattern

3.2 Shell-and-tube heat exchanger

In this paper, the basic TEMA-E [6, 7] type of shell-and-tube heat exchanger is applied. It has a single shell pass with the inlet and outlet at the opposite ends of the shell. Fig.4. gives the geometrical characteristic. The geometrical parameters optimized in this paper are: the outside diameter of the shell D_s, the outer diameters of the tubes d_o, the tube length l_t, pitch between the tube centers pt, and the baffle spacing l_b. All the configuration and geometry calculating expression can be found in the literature [6, 7, 11,12].

$$\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_{sd}} + \frac{d_o}{2\lambda_m}\ln\left(\frac{d_o}{d_i}\right) + \frac{d_o}{d_i}\frac{1}{h_{td}} + \frac{d_o}{d_i}\frac{1}{h_t} \quad (9)$$

The overall heat transfer coefficient equation is given by Eq.(9), where, h_s and h_c are the convection heat transfer coefficients for the shell side and tube side respectively; λ_m is the thermal conductive of the plate; h_{sd} and h_{td} are the dirt coefficients for the shell side and tube side respectively. The main heat transfer correlations are discussed below.

tube side

The heat source and cold source always flow in the tube without phase change, so a single phase model in tube is needed. While the flow state in the tube will have a relatively small impact on this model, it is assumed to be laminar flow, and the film heat transfer coefficient and pressure drop in tube are expressed as[12] as Eq. (10)-(11), Where, μ and μ_w are the fluid viscosity at bulk flow and the wall temperature, respectively; $j_{f,t}$ is the friction factor [12].

$$h_t = 1.86\frac{\lambda}{d_e}(RePr)^{0.33}\left(\frac{d_e}{L}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10)$$

$$\Delta P_t = \left[8j_{f,t}\left(\frac{L}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{0.25} + 2.5\right]\frac{\rho u_t^2}{2} \quad (11)$$

shell side

The condensation transfer coefficient $h_{s,con}$ and evaporation transfer coefficient $h_{s,eva}$ are expressed as Eq.(12) and (13)[12], where, λ is the condensate conductivity; ρ_l and ρ_g are the liquid and vapor density; μ_l is the condensate viscosity; L is tube length, N_t is the total number of tubes; W_c is condensate flow rate; r is the latent heat; μ_v is the vapor viscosity; T_w and T_s are the wall and saturation temperature, respectively.

$$h_{s,con} = 0.95\lambda\left[\frac{\rho_l(\rho_l-\rho_v)g}{\mu_l}\frac{LN_t}{W_c}\right]^{1/3} \tag{12}$$

$$h_{s,eva} = 0.62\left[\frac{\lambda^3(\rho_l-\rho_v)\rho_v g r}{\mu_v d_o(T_w-T_s)}\right]^{1/4} \tag{13}$$

The shell-side pressure drop can be calculated by following correlation [12], where $j_{f,t}$ is the friction factor and L is the tube length.

$$\Delta P_t = 8j_{f,s}\left(\frac{D_s}{d_e}\right)\left(\frac{L}{l_b}\right)\frac{\rho {u_s}^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{14}$$

4. ORC system optimization model

4.1 *Objective functions*

Three important indicators, thermal efficiency (THE), specific investment cost (SIC) and heat exchanger area per unit power output (APR) are selected as the objective functions. The thermal efficiency is defined by Eq. (15), where, $\mathrm{W_p}$ is the power consumption of the pump; $\mathrm{W_{tur}}$ is the power output from the turbine; $\mathrm{Q_{eva}}$ is the heat transferred from the heat source to the working fluid.

$$\mathrm{THE} = (\mathrm{Wtur} - \mathrm{Wp})/\mathrm{Qeva} \tag{15}$$

The total investment cost that indicates the economic performance of the installation is composed of the purchased-equipment cost of heat exchangers ($\mathrm{PEC_{ht}}$), pump ($\mathrm{PEC_p}$), turbine ($\mathrm{PEC_{tur}}$) and generator ($\mathrm{PEC_{gen}}$). PECs are expressed by Eqs. (16-20) [14], where A is the heat exchanger area; η_m is the mechanical efficiency of the pump; η_{el} is the electric efficiency of the generator.

$$\mathrm{PEC_{ht}} = 10{,}000 + 324\mathrm{A}^{0.91} \tag{16}$$

$$\mathrm{PEC_p} = 422\mathrm{W_p^{0.71}}\left[1.41 + 1.41\left(\frac{1-0.8}{1-\eta_m}\right)\right] \tag{17}$$

$$\mathrm{PEC_{tur}} = 6000\mathrm{W_{tur}^{0.7}} \tag{18}$$

$$\mathrm{PEC_{gen}} = 60(\eta_{el}\mathrm{W_{tur}})^{0.95} \tag{19}$$

$$\mathrm{SIC} = (\mathrm{PEC_{ht}} + \mathrm{PEC_p} + \mathrm{PEC_{tur}} + \mathrm{PEC_{gen}})/(W\mathrm{tur} - W\mathrm{p}) \tag{20}$$

Heat exchanger area per unit power output is defined by the ratio of total heat transfer area to the net power output and is expressed by Eq. (13), where $\mathrm{A_{tot}}$ is the total heat transfer area.

$$\mathrm{APR} = \mathrm{A_{tot}}/(W\mathrm{tur} - W\mathrm{p}) \tag{13}$$

4.2 *Optimization algorithm*

4.2.1 *Genetic algorithm*

The candidate scheme of ORC with different heat exchangers under different design criteria are predetermined and listed in Fig.5, where ORC with shell-and-tube heat exchanger is named ORC-A while ORC with plate heat exchanger is named ORC-B. Genetic algorithm (GA) [15] is used to perform multivariable optimization for different ORC scheme under different single objective function. The flow diagram to produce alternative schemes is shown in Fig.5. The two ORCs (ORC-A and ORC-B) are optimized by GA with three different objection functions (THE, SIC and APR), so that six optimal solutions can be achieved. Each solution (A1, A2, A3, B1, B2 and B3) corresponds to a special ORC working condition and the types of heat exchangers.

The GA parameters are determined as follows: the population size is 40, the generation size is 100, the crossover fraction is 0.7 and the migration fraction is 0.2. The optimization processes are programmed on software Matlab2013a, and the thermal properties of working fluid are referred to REFPROP 9[15].The design variables and their lower/upper bounds are shown in table 1.

4.2.2 Non-structural fuzzy decision making process

After six alternative optimal schemes under different single criteria are achieved, the screening of heat exchangers under multiple criteria is followed. In this paper, a non-structural fuzzy decision making method (NSFDMM) is used. NSFDMM has the advantages in making a choice among several alternatives and providing a comparison of the considered options [16]. It is so robust to reduce the inherent uncertainly an imprecision in the pair-wise comparison [17] and has been used in the evaluation of compact heat exchangers [18].

The NSFDMM applies three basic rules. Firstly, the problem is decomposed into several elements. Secondly, comparative judgment with pairwise comparisons is performed to form the corresponding matrix. Finally, synthetically argument of the priorities is conducted. At different level, the decision solution with the maximum relative weight is the optimum scheme with best comprehensive performance.

The procedure can be summarized to the following eight steps : 1) information collection (Criteria and its value); 2) pair-wise comparisons; 3) modification of consistency and priority ordering; 4) assignment of semantic score and normalization; 5) calculation of weight; 6) multi-Level fuzzy evaluation; 7) final priority orders; (8) output data and priority discussion.

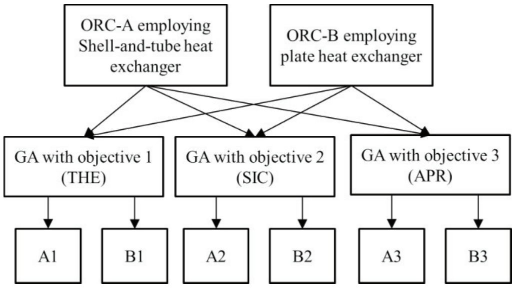

Fig.5. Flow diagram to produce alternative schemes

Table 1 Lower and upper bounds of the genetic algorithm variables used in this article

Items	Variable	Lower bound	Upper bound
ORC system	Evaporating pressure	1.5MPa	3.0MPa
	Condensation pressure	0.7MPa	1.0MPa
Shell-and-tube	Outside diameter of tube d_o	20mm	80m
Heat exchangers	Ratio of tube length and shell outside diameter L/D_s	5	10
	Relative baffle pitch p_b/D_s	0.2	1.0
	Relative baffle cut l_c/D_s	0.15	0.45
Plate heat	Plate length L	0.3m	1.2m
exchangers	Plate width W	0.2m	0.5m
	Channel space b	3 mm	5 mm
	Plate thicknessσ	0.3 mm	0.5 mm

5. Results

At the given condition of heat resource, six alternative schemes of ORC system are obtained using GA with three different objective functions. The results of each candidate are shown in table 2. Schemes A1, A2, A3 give the optimal configuration of ORC with shell-and-tube heat exchangers of different objective functions while scheme B1, B2, B3 give the optimal results of ORC with plate heat exchangers of different objective functions. In comparison of scheme As and Bs, the ORCs with plate heat exchanger has better performance in general. For plate heat exchangers, the heat transfer coefficients of condenser are higher than shell and tube heat exchangers, while the pressure drops are a little bit higher than that of shell and tube heat exchangers. Considering the specific investment cost and area of per unit net-power out, the value of schemes B1,B2 and B3 are all lower than those of A1, A2 and A3. In terms of the system indicators, the six optimal schemes perform well and get the near values with the range from 9.36% to 12.38% of thermal efficiency, and 43.7% to 46.8% of exergy efficiency. The results show that best heat exchanger and their combination scheme can be achieved based on different objective functions.

Table 2. Results of each scheme

	A1	A2	A3	B1	B2	B3
Volume(m^3)	0.262	0.167	0.148	0.017	0.008	0.007
Total area(m^3)	11.268	10.345	8.992	4.684	2.168	1.718
Heat transfer coefficient of evaporator ($W/m^2{\cdot}K$)	1122.744	1467.105	1526.010	1156.497	4008.039	4385.561
Heat transfer coefficient of condenser ($W/m^2 \cdot K$)	1543.229	1750.927	1598.621	1587.277	3675.834	3544.999
Pressure drop on organic fluid side(kPa)	1.866	35.396	63.349	2.005	137.365	137.762
Network output(kW)	8.409	8.476	7.998	6.716	7.255	6.484
Thermal efficiency (%)	0.391	0.364	0.376	0.380	0.318	0.361
Exergy efficiency(%)	0.099	0.093	0.096	0.102	0.085	0.096
Special investment cost(yuan/ kW)	5133.134	5034.471	5208.642	5502.134	5111.222	5484.566
Area of heat exchanger per work output (m^2/kW)	1.340	1.221	1.124	0.698	0.299	0.265

According to the basic rules of NSFDMM and the steps of three-level fuzzy evaluation, the priority of the six candidate schemes of ORC systems are obtained, the results of which are shown in Fig.6. Seeing from Fig.6, the first level fuzzy comprehensive evaluation result is obtained by multiplying the priority and weight matrices of volume, total area, pressure drop, and heat transfer coefficient for condenser and evaporator. It reflects the performance of heat exchangers without the system respects. The second level which is added by the network output, exergy efficient and thermal efficiency, which mainly focus on the system performance indicators. The scores of this level include the result of first level by assigning weight as 0.25. Similarly, the result of the third level under the consideration of special investment cost and area of heat exchanger per work output can be achieved.

According to the basic procedure of the non-structural fuzzy decision making method, fuzzy evaluation of the six ORC systems is conducted. Fig. 6 shows the evaluation results. As is shown in Fig 6a, the plate heat exchangers has better performance than shell-and-tube heat exchangers only considering the heat exchanger design. The highest score is the configuration of B3, which is an ORC employed plate heat exchangers with the minimum APR of 0.265 m^2/kW. In the second level, the priority order has changed when the system indicators are explored. A1 ranks the first, and B1 follows behind. It indicates that the schemes with highest thermal efficiency perform the best in this level. In the third level, scheme B2 and A2, are superior to others. From the comparison at three different levels, we can see that the optimal configuration varies in different level of evaluation. Therefore, these results give a clear guidance to designer or decision maker.

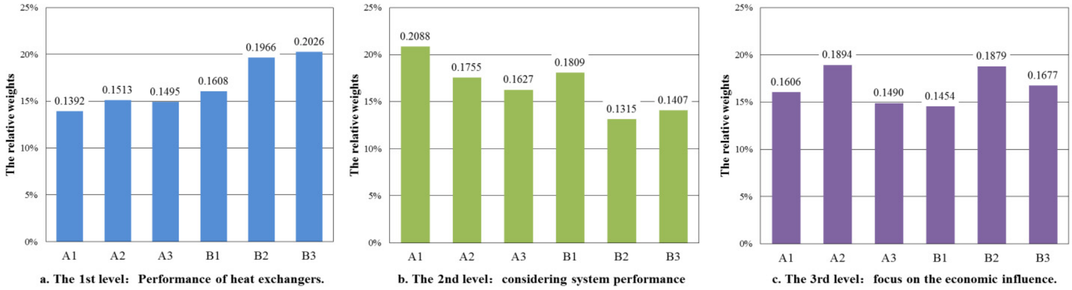

Fig. 6 The relative weight scores allocation diagram of the three levels

6. Conclusions

In this paper, the models of two kinds of ORC systems using different type of heat exchangers are formulated. The genetic algorithm is employed to optimize the two kinds of ORC system. Six basic schemes are obtained with different ORC and heat exchanger configuration. Then the candidates are compared by using a three-level fuzzy evaluation method combined with NSFDMM. The results show that plate heat exchangers may have a better performance than shell and tube heat exchangers when only equipment performance is consideration. When the system technique parameters and economic indicators are taken into account, the optimal results changed. Optimizing the ORC system and the heat exchangers simultaneously can reduce the influence of improper parameter assumptions. The employment of a three-level fuzzy evaluation method can help researchers to obtain a more reasonable solution.

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