ORC System Optimization for Multi-Strand Waste Heat Sources in Petroleum Refining Industry

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ABSTRACT

Low-grade waste heat source accounts for a large part of the total industrial waste heat, which cannot be efficiently recovered. The ORC (Organic Rankine Cycle) system has been proved to be a promising solution for the utilization of low-grade heat sources. It is evident that there might be several waste heat sources distributing in different temperature levels in one industry unit, and the entire recovery system will be extremely large and complex if the different heat sources are utilized one by one through several independent ORC subsystems. This paper aims to design and optimize a comprehensive ORC system to recover multi-strand waste heat sources in Shijiazhuang Refining & Chemical Company in China, involving defining suitable working fluids and operating parameters. Thermal performance is a first priority criterion for the system, and system simplicity, technological feasibility and economic factors are considered during optimization. Four schemes of the recovery system are presented in continuous optimization progress. By comparison, the scheme of dual integrated subsystems with R141B as a working fluid is optimal. Further analysis is implemented from the view of economic factors and off-design conditions. The analytical method and optimization progress presented can be widely applied in similar multi-strand waste heat sources recovery.

Keywords: petroleum, ORC, optimization

1. INTRODUCTION

With the development of science and technology, the energy demand is enlarging rapidly, which results in growing consumption of primary fossil fuels and massive discharge of pollutants, especially in China, the biggest developing country in the world. The energy consumption in industry is about 70 % of the total energy consumption in China and more than 50 % of the industrial waste heat could not been effectively recovered [1].

The waste heat sources are generally categorized by temperature levels: low-grade (<230 °C), medium-grade (230-650 °C) and high-grade (>650 °C) [2]. The ORC (Organic Rankine Cycle) system has been proved to be a promising energy conversion technology for the utilization of low-grade heat sources due to its significant advantages such as robust thermal performance under low temperature conditions, system simplicity, and environmental and economic benefits as well. Saleh et al. [3] made comparison of the thermal characteristics among 31 pure working fluids for one heat source, of which the temperature is between 100 ℃ and 30 °C. The fluids are alkanes, fluorinated alkanes, ethers and fluorinated ethers. Thermal efficiencies η_{th} are presented for cycles of different types. In case of subcritical pressure processes one has to distinguish (1) whether the shape of the saturated vapour line in the T, s-diagram is bell-shaped or overhanging, and (2) whether the vapour entering the turbine is saturated or superheated. Moreover, in case that the vapour leaving the turbine is superheated, an internal heat exchanger (IHE) may be used. The highest η_{th}-values are obtained for the high boiling substances with overhanging saturated vapour line in subcritical processes with an IHE, e.g., for n-butane η_{th}=0.130. On the other hand, a pinch analysis for the heat transfer from the heat carrier with maximum temperature of 120 ℃ to the working fluid shows that the largest amount of heat can be transferred to a supercritical fluid and the least to a high-boiling subcritical fluid. Roy et al. [4] studied parametric optimization and performance analysis of a waste heat recovery system based on ORC, using R-12, R-123 and R134a as working fluids, and the results showed that system using R123 had the maximum work output and highest efficiency. Sun et al. [5] studied the influence of controlled variables, including working fluid mass flow rate, air mass flow rate of the condenser. A sizing model of the ORC is proposed, capable of predicting the cycle performance with different working fluids and different components sizes. The working fluids considered are R245fa, R123, n-butane, n-pentane and R1234yf and Solkatherm. Results indicate that, for the same fluid, the objective functions (economics profitability, thermodynamic efficiency) lead to different optimal working conditions in terms of evaporating temperature: the operating point for maximum power doesn't correspond to that of the minimum specific investment cost: The economical optimum is obtained for n-butane with a specific cost of 2136 €/kW, a net output power of 4.2 kW, and an overall efficiency of 4.47 %, while the thermodynamic optimum is obtained for the same fluid with an overall efficiency of 5.22 %. Zhou et al. [6] presented an experimental recovery system, in which R-123 was selected as a working fluid, a scroll expander was used to produce work, and fin tubes heat exchanger was designed as an evaporator. Sharke et al. [7] made a feasibility analysis of the application of ORC as a waste heat recovery system for automotive internal combustion engines, and the results showed that a simple ORC system was able to recover up to 10 % of the engine waste heat.

There are five waste heat sources in a 1.2 million ton-level reforming and extraction unit in Shijiazhuang Refining & Chemical Company, SRCC, in China, which distribute in different temperature levels. Based on the research [8] of single ORC system and the 500 kW ORC test system developed by Tsinghua university and Hangzhou Chinen Steam Turbine Power Co., Ltd., thermodynamic analysis and performance optimization are implemented for the recovery system utilizing these five waste heat sources in this paper. Four schemes are presented in continuous optimization progress, and the simulation results indicate that the scheme of dual integrated subsystems with R141b as a working fluid is optimal and recommended for practical uses in this heat recovery case.

2. ORGANIC RANKINE CYCLE SYSTEM

2.1. Thermodynamic model of the ORC system

Fig. 1 illustrates the system of a basic Organic Rankine Cycle system for waste heat recovery. The liquid organic working fluid from the condenser is pumped to the preheater, heated and then transported to the evaporator. In the evaporator, the liquid working fluid is heated and turns into saturated or superheated vapor. Hence, in the turbine, the organic vapor expands to a low pressure state, driving the turbine to produce power, which is turned into electricity by a generator connected to the shaft with the turbine. Afterwards, the exhaust organic gas from the turbine outlet is condensed to liquid again in the condenser by cooling water.

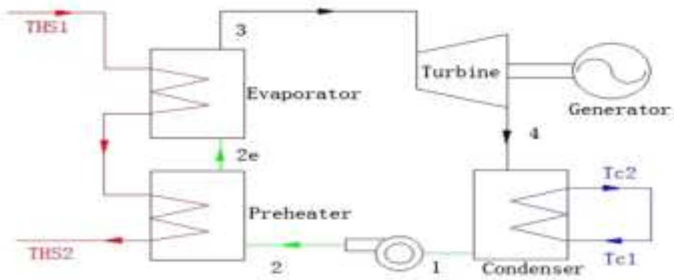

Fig. 1. Schematic diagram of an Organic Rankine Cycle system for waste heat source recovery. [18]

The process of Organic Rankine Cycle is shown in the T-s diagram in Fig. 2, which can also be described as follows:

Mass flow rate of the organic working fluid

$$G_W = \frac{Q_{total} \bullet (1 - \varepsilon_Q)}{h_3 - h_2} \qquad (1)$$

Where Q_{total} is the heat load of waste heat source, and ε_Q is the heat loss of the heat exchanger. As the beginning and ending temperatures of the waste heat source are fixed in this research, the heat load absorbed by ORC system can be calculated as

$$Q_{total} = G_S \bullet C_{P \cdot S} \bullet (T_{HS1} - T_{HS2}) \qquad (2)$$

where $C_{P.S}$ is the average specific heat capacity of fluid or gas of the waste heat source, and T_{HS1} and T_{HS2} are defined as the beginning and ending temperature.

Process 1 to 2 (Pump)

$$W_P = \frac{G_W \cdot (h_{2rs} - h_1)}{\eta_{pump}} \quad (3)$$

where h_{2rs} is the isentropic enthalpy of the working fluid after compressed in the pump , and η_{pump} is the efficiency of the pump.

Process 2 to 2e (Preheater)

$$Q_{prh} = \frac{G_W * (h_{2e} - h_2)}{1 - \varepsilon_Q} \quad (4)$$

Process 2e to 3 (Evaporator)

$$Q_{eva} = \frac{G_W * (h_3 - h_{2e})}{1 - \varepsilon_Q} \quad (5)$$

Process 3 to 4 (Turbine)

$$W_T = G_W * (h_3 - h_{4S}) * \eta_T \quad (6)$$

where h_{4s} is the isentropic enthalpy of the exhaust gas in the turbine outlet , and η_T is the efficiency of the turbine.

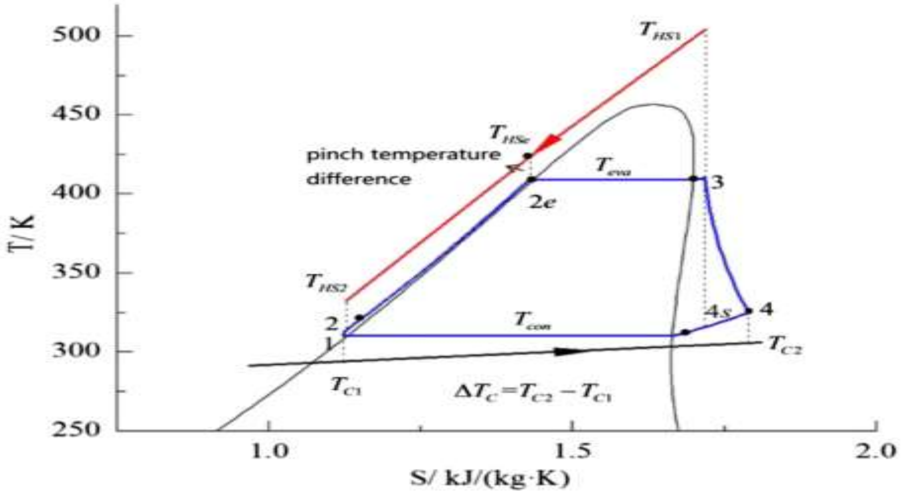

Fig. 2. T-s diagram of Organic Rankine Cycle. [18]

Process 4 to 1 (Condenser)

$$G_C = \frac{G_W * (h_4 - h_1)}{(1 - \varepsilon_Q) * C_{P.C} * (T_{C2} - T_{C1})} \quad (7)$$

Where $C_{P.S}$ is the average specific heat capacity of cooling water. The net power output of the system is

$$W_{net} = W_T - W_P \quad (8)$$

Here the power required by auxiliaries is neglected since its amount is considerably small and the thermal efficiency is defined as

$$\eta_e = \frac{W_{net}}{Q_{total}} = \frac{W_T - W_P}{Q_{total}} \quad (9)$$

Based on the experiments conducted on the 500 kW ORC test system [8], some design parameters are given in Table 1.

2.2. Working fluid selection

Generally, organic working fluids can be divided into three categories, dry, isentropic and wet, according to the saturation vapor curve slope, as shown in Fig. 3. Hung et al. [9] proved that the major physical properties affecting the system performance included latent heat, specific heat of liquid and vapor. Chen et al. [10] suggested that fluids with high latent heat and low liquid specific heat were preferable. The thermodynamic and physical properties, stability, environmental impacts, safety and compatibility, and availability and cost are among the important considerations when selecting a working fluid. The paper discusses the types of working fluids, influence of latent heat, density and specific heat, and the effectiveness of superheating. A discussion of the 35 screened working fluids is also presented. However, Yamamoto et al. [11] suggested low latent heat fluid was better.

In this paper, the organic working fluid candidates are selected according to following principles:

- Proper thermal properties for the recovery system
- Chemical stability under the operating condition
- Slighter impact on the environment (low GWP (Global Warming Potential) and low ODP (Ozone Depletion Potential));
- Security during the cycle (non-flammable, non-explosive and non-toxic); and Availability and low cost

3. BACKGROUND OF THE WASTE HEAT SOURCES

Petroleum industry is one of the key industries in China, which relies heavily on the primary energy resources. Liu et al. [12] presented models and analysis of the energy-savings potential for refining and conversion processes in the context of technological change. The results indicated that upgrading process heaters had been a priority during.

Table 1
Design parameters of the ORC system. [18]

Property	Symbol	Value
Pump efficiency	η_{pump}	0.8
Heat loss of heat exchanger	ε_Q	0.01
Preheater pressure loss	ζ_{Qex}	0.08
Minimum temperature difference of evaporator	$\Delta t_{eva}/K$	6.0
Turbine inlet pipe pressure loss	ζ_{P1}	0.02
Turbine efficiency	η_T	0.8
Turbine outlet pipe pressure loss	ζ_{P2}	0.015
Cooling water initial temperature	T_{C1}/K	293.15
Cooling water temperature rise	$\Delta T_C/K$	8.0
Minimum temperature difference of condenser	$\Delta t_{con}/K$	6.0

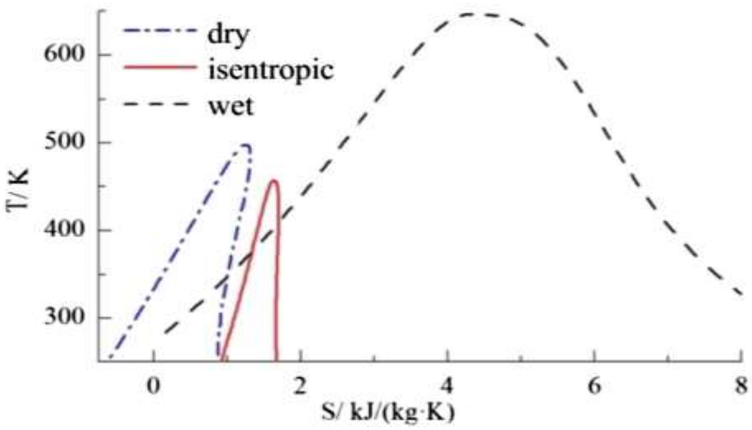

Fig. 3. T-s diagram of dry, isentropic and wet organic fluids. [18]

Varga et al. [13] summarized the results of an engineering study of partial substitution of an air cooler, which was situated in one of the main process units of the refinery and cooled the process stream from 140 °C to 45 °C meantime dissipated heat of 32 MW into environment, with an organic Rankine cycle. Huang [14] made analysis of an ORC system using R601A as a working fluid, in order to utilize the low-grade waste heat in refining units. Meacher [15] designed a low temperature organic Rankine cycle system using R-113 as a working fluid for recovery and conversion of process waste heat for typical application in oil refineries and chemical plants.

This paper focuses on the recovery of waste heat sources distributing in various temperature levels from the catalytic reforming unit in Shijiazhuang Refining & Chemical Company, SRCC, in Hebei Province, China, where the water resource is considerably lacking. The renovation of the unit is mainly conducted by Sinopec Engineering Incorporation, SEI, and it is currently in the process of implementation. According to the temperature conditions of the waste heat sources, the Organic Rankine Cycle technology is selected with the consideration of cold water saving. The system design and performance optimization is conducted by Tsinghua University and SEI.

Table 2
Five waste heat sources to be recovered. [18]

Term	Source	Temperature level (°C)	Heat load (MW)	Proportion
A	Bottom oil of xylene	215–80	2.20	6.75%
B	Xylene products	205–80	3.79	11.63%
C	Evaporation tower	147–80	8.59	26.37%
D	Reforming tower	104–80	11.50	35.30%
E	Benzene tower	98–80	6.50	19.95%
Total			32.58	100%

Table 3
Simulation results of the five independent subsystems. [18]

Heat sources	Working fluid	Evaporation temperature, K	Mass flow rate, kg/s	Net power output, kW	Thermal efficiency, %
A	R123	429.7	9.63	350.08	15.93
	R601	428.2	4.02	337.26	15.35
	R601a	433.2	4.20	335.97	15.29
B	R123	412.2	16.93	558.81	14.74
	R601a	422.7	7.39	557.20	14.70
	R245ca	420.2	13.86	556.28	14.68
C	R600a	381.7	21.43	938.27	10.92
	R236ea	381.5	43.42	930.75	10.83
	R245fa	375.0	37.04	923.43	10.75
D	R141b	352.2	45.21	961.30	8.36
	Heptane	352.9	25.97	952.42	8.29
	R601	353.4	26.35	948.50	8.25
E	R141b	350.9	25.63	530.43	8.16
	2-Butene	351.4	14.90	520.22	8.01
	R601a	351.7	14.99	519.49	7.99
Total	Maximum net power output and highest thermal efficiency			3338.89	10.25
	Hydrocarbons as working fluids			3300.73	10.13

The beginning temperatures of the waste heat sources are 215 °C, 205 °C, 147 °C, 104 °C and 98 °C, and the ending temperature is 80 °C as required by the technological process. The corresponding heat loads are 2.20 MW, 3.79 MW, 8.59 MW, 11.50 MW and 6.50 MW. The data of the five waste heat sources are listed in Table 2.

4. THERMODYNAMIC ANALYSIS AND PERFORMANCE OPTIMIZATION

In this section, four schemes of the comprehensive ORC system for the five waste heat sources are presented in continuous optimization progress and comparison is made from the view of thermodynamics, system simplicity and technological feasibility.

4.1. Five independent ORC subsystems

It is easy to come up with the design of five independent ORC subsystems of different parameters to recover the five waste heat sources one by one. According to the principle of working fluid selection mentioned above, several organic fluids are selected as candidates. The properties of the working fluids are calculated by REFROP 9.0 [16] developed by the National Institute of Standards and Technology of the United States. The system simulation is carried out by a computer program written by authors under FORTRAN environment.

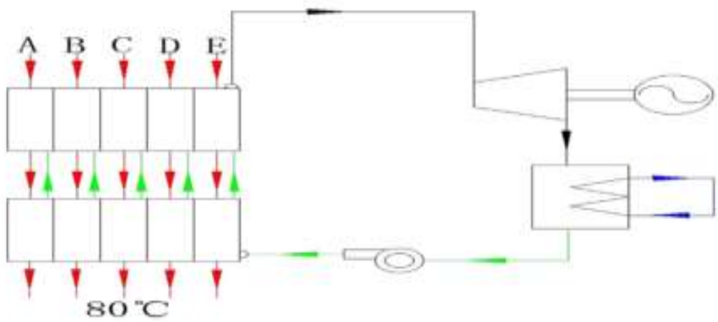

Fig.4. Schematic diagram of single ORC system. [18]

For each waste heat source and each kind of working fluid, the simulation program arrives to a proper evaporation temperature through continuous iteration, the heat load above which is utilized to convert the working fluid liquid into vapor in the evaporator and the heat load below which is used in the preheater. The computation results are listed in Table 3.

As shown in Table 3, R123 is a working fluid suitable for waste heat sources A and B, because its critical temperature is relatively high, which is a little lower than the beginning temperatures of heat sources A and B, at 215 °C and 205 °C. fluid suitable for heat sources C for the same reason. The temperature of heat source D and E is low and the temperature drop is relatively small. Systems with R141b as a working fluid have a good performance for heat source D and E though the critical temperature of R141b is fairly high. The main reason is that the latent heat is in a larger amount than the sensible heat at a low evaporating temperature for the organic fluid whose critical temperature is high. Thus systems with these kinds of working fluids give out comparatively better thermal performance.

The scheme above is theoretically simple. But it is evident that the entire system including five independent subsystems is complex in structure, and requires high capital cost and large space occupation.

Table 4
Simulation results of the single ORC system. [18]

Working fluid	Evaporation temperature, K	Mass flow rate, kg/s	Net power output, kW	Thermal efficiency, %
R141b	354.15	126.96	2813.02	8.63
R123	355.35	164.43	2806.48	8.61
Heptane	355.25	72.58	2795.04	8.58
R245fa	357.15	146.34	2793.94	8.58
R601	355.85	73.66	2784.73	8.55

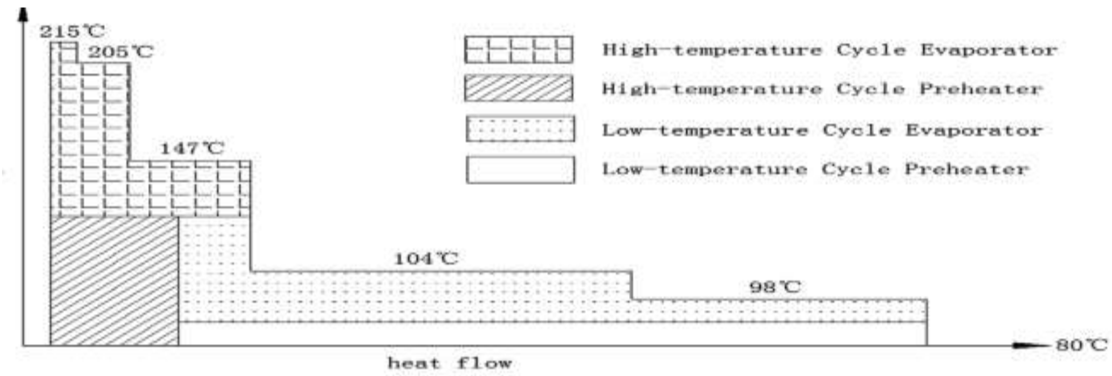

Fig. 5. Heat load distribution of dual independent ORC subsystems with different working fluids. [18]

4.2. Single ORC system

The single ORC system for all five heat sources is the simplest in structure. As shown in Table 2, heat sources D and E account for more than half of the total heat loads while the high-grade parts, A and B, only make up 18 %. If the evaporation temperature is set too high, the mass flow rate of working fluid will be low. Thus heat load needed for preheating is much smaller than the remaining part after evaporation, and the ending temperature of waste heat source will be higher than 80 °C. The schematic diagram of this scheme is shown in Fig. 4 and the simulation results of the single ORC system are shown in Table 4, which shows that the thermal efficiency is quite low due to the low evaporation temperature. The maximum net power output of this scheme is just 2813.02 KW, nearly 16 % lower than that of the scheme of five independent subsystems. Therefore, the single ORC system, in spite of the simplicity, is not efficient enough and neglected in further discussion.

4.3. Dual independent ORC subsystems with different working

Poor simulation results of the single ORC system indicates the necessity of two ORC subsystems, which is presented, corresponding to a high-temperature cycle and a low-temperature cycle.

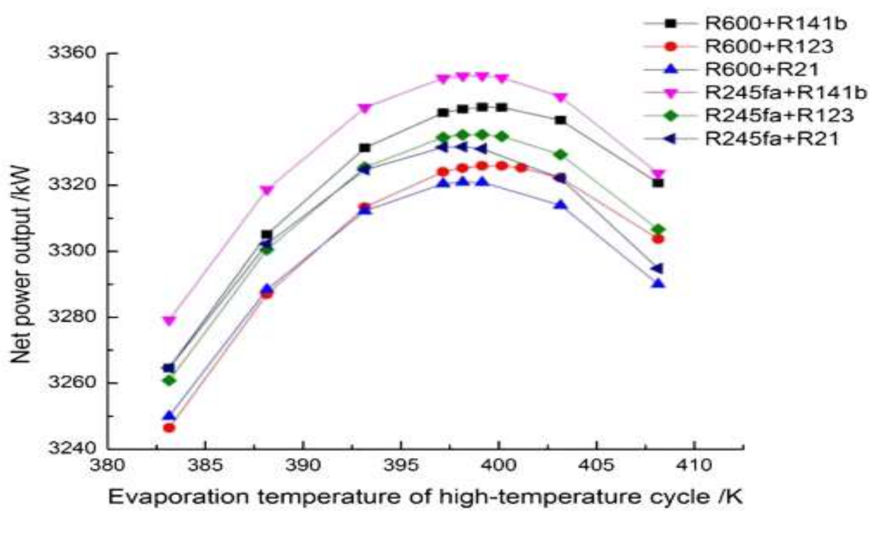

Fig. 6. Simulation results of dual independent ORC subsystems with different working fluids. [18]

Table 5
Thermodynamic parameters of dual independent subsystem with different working fluids. [18]

	Working fluid	Evaporation temperature, K	Mass flow rate, kg/s	Net power output, kW
High-temperature cycle	R245fa	399.15	50.7	1588.8
Low-temperature cycle	R141b	354.85	78.5	1764.5

The evaporation temperature of the high-temperature cycle is firstly set, and by calculating the ratio of latent heat to sensible heat of the working fluid, the total heat load for this cycle is determined. Then for the low - temperature cycle, a proper evaporation temperature can be determined through interaction by the simulation program as the total heat load remaining is fixed. The heat load distribution for two ORC subsystems is shown in Fig. 5.

Two different working fluids are selected for the high-temperature cycle and the low-temperature cycle, in order to achieve better performance of both subsystems. The simulation results are demonstrated in Fig. 6, which indicates that system with R245fa for the high-temperature cycle and R141b for the low-temperature cycle gives out the maximum net power output, 3353 kW, the highest among discussed schemes. Table 5 shows the optimal set of thermodynamic parameters for both cycles.

Compared to the five independent subsystems, this scheme cuts the number of the subsystems from five to two. The entire system is relatively simplified and the efficiency is increased.

4.4. Dual integrated ORC subsystems with one working fluid

Improved on the basis of the schemes above, scheme of dual integrated ORC subsystems with one working fluid is presented. The condensing temperatures of both cycles are set the same thus only one condenser and one working fluid pump are needed. The heat load distribution for the two cycles is shown in Fig. 7, and simulation results of different working fluids are shown Fig. 8, which indicate that system with R141b as a working fluid gives out the best performance among the selected working fluids, with maximum net power output at 3325 kW, which is close to that of the scheme of dual independent subsystems and the difference is ignorable. By comparison, the scheme of dual integrated subsystems is more compact and easier to operate, thus it is regarded as the optimal scheme for this waste heat recovery case. Table 6 shows the optimal set of thermodynamic parameters for both cycles.

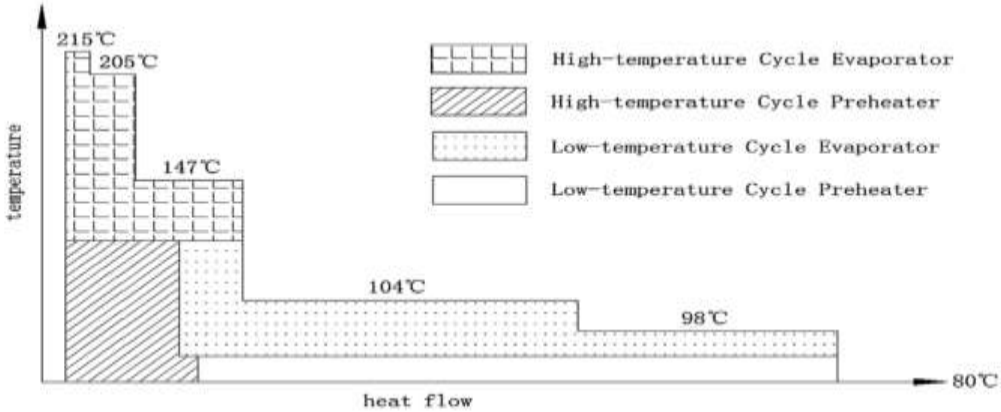

Fig. 7. Heat load distribution for dual integrated ORC subsystems with one working fluid. [18]

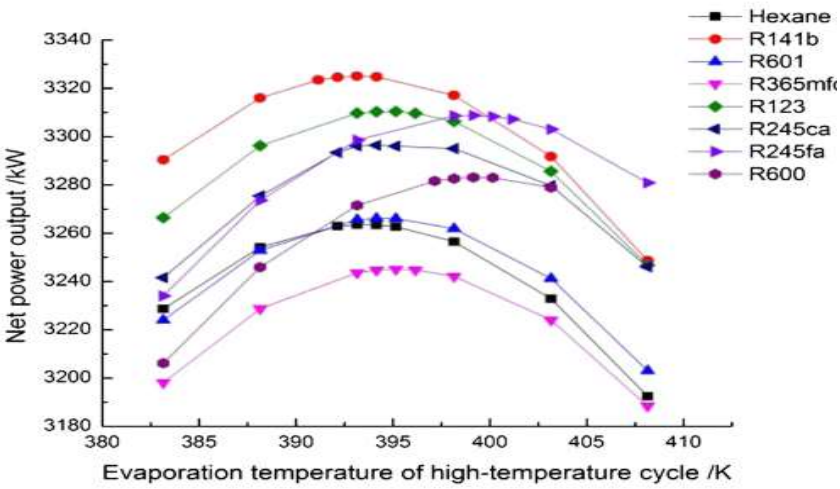

Fig. 8. Simulation results of dual integrated ORC subsystems with one working fluid. [18]

Schematic diagram of the recovery system including two integrated subsystems with R141b as a working fluid is shown in Fig. 9 and the flow procedure of the system is given as below: Heat sources A, B and C firstly go through the evaporator of the high-temperature cycle, and the high-grade heat load transforms the working fluid into vapor. Then A, B and a part of heat source C go through the preheater of the high - temperature cycle, making the working fluid preheated up to the evaporation temperature. Heat sources D, E and the remaining part of C go through the low-temperature subsystem in the same way, as the heat load is divided into two parts according to the ratio of latent heat to sensible heat. The distribution of the heat load of the five heat sources is listed in Table 7. It should be emphasized that all the working fluid goes into the preheater of the low-temperature cycle and then is divided into two parts, one of which goes into the evaporator of the low- temperature cycle and the other goes to the high-temperature cycle. The ratio of mass flow for the high-temperature cycle is nearly 30 % of the total according to the simulation results.

5. FURTHER ANALYSIS

The simulation results and analysis above illustrated that the system including two integrated subsystems is optimal among the schemes. In this section, further analysis is made from the view of economic factors and off-design conditions.

5.1. Economic factors

Suppose this ORC system runs 8000 h per year, and the price of electricity is 0.08 \$/kWh in China, it will produce 2.671×10^7 kWh electricity, worth 2.14 million dollars. Rough estimation for the cost of ORC system would be 2500 \$/kWh, and overall initial investment would be 8.3 million dollars. The investment recovery period would be 3.5-4 years.

Table 6
Thermodynamic parameters of dual integrated subsystems with one working fluid. [18]

	Working fluid	Evaporation temperature, K	Mass flow rate, kg/s	Net power output, kW
High-temperature cycle	R141b	393.15	37.4	1426.3
Low-temperature cycle	R141b	353.85	86	1898.7

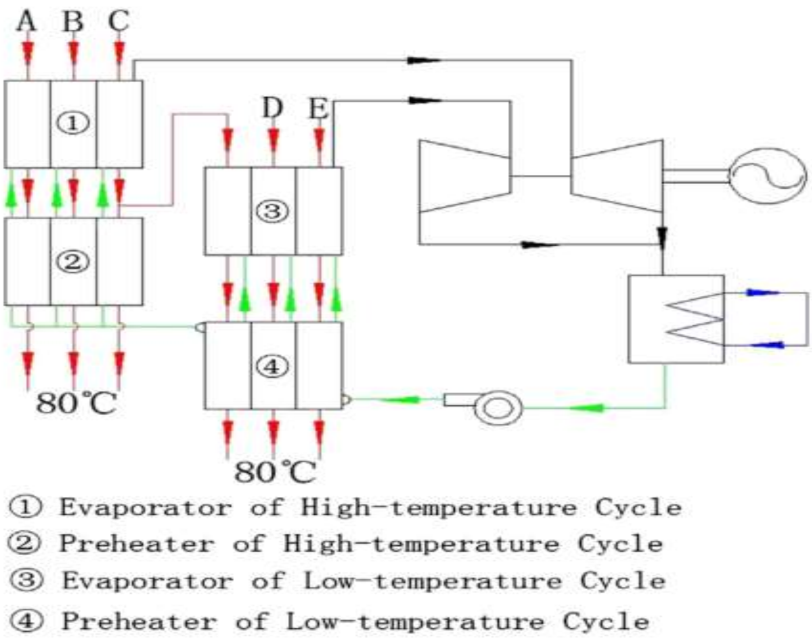

Fig. 9. Schematic diagram of dual integrated subsystems with R141b as a working fluid. [18]

Generally it is difficult to make an accurate estimation of the whole capital cost of an ORC system due to the fluctuation of the component prices. The cost of the pump and the turbine changed slightly under the condition of small power capacity level [17]. Thus it is reasonable that the economic analysis of the ORC system mainly involves the cost of heat exchangers, and the cost of other components can be neglected. A ratio of net power output to total heat transfer area is selected as the performance evaluation criterion for purpose of predicting the system performance from view of both thermodynamics and economics [17].The heat transfer area is determined by :

$$A = \frac{Q}{U \bullet \Delta t_m} \qquad (10)$$

Where U is the overall heat transfer coefficient and Δt_m is the logarithmic mean temperature difference. The thermodynamic properties of the three organic fluids are calculated by REFROP 9.0 [12], and here the flow states are assumed the same.

Fig. 7 indicates that systems with R141b, R123 and R245fa as working fluids give out comparatively higher net power output under optimal operating condition. 7 indicates that systems with R141b, R123 and R245fa as working fluids give out comparatively higher net power output under optimal operating condition. It indicates that system with R141b as a working fluid is the most economical since its ratio of net power output to total heat transfer area, up to 0.9804, is the highest among the three working fluid candidates.

Table 7
Distribution of the heat load of all the waste heat sources. [18]

Heat source	Temperature level	Distribution of the heat load			
		High-temperature cycle		Low-temperature cycle	
		Evaporator	Preheater	Evaporator	Preheater
A	215 °C	65.9%	34.1%		
B	205 °C	63.2%	36.8%		
C	147 °C	31.3%	0.8%	58.0%	9.9%
D	104 °C			72.1%	27.9%
E	98 °C			62.8%	37.2%

Table 8
Ratios of the net power output to total heat transfer area. [18]

Working fluid	Net power output a design point, kW	Total heat transfer area, m^2	Ratio of net power output to total heat transfer area, kW/m^2
R141b	3325.03	3391.61	0.9804
R123	3310.43	3571.32	0.9269
R245fa	3308.80	3403.02	0.9723

There is little to none difference among the total heat transfer areas of the three systems under optimal condition. The reason is that the thermodynamic properties of the three organic fluids are similar and the flow states are assumed the same in the computation program.

In reality, the total heat transfer coefficient is related to many factors and is usually determined through heat transfer experiments. It remains to be a part of the future work to get a more accurate overall heat transfer coefficient for purpose of more precise analysis and further optimization.

5.2. Off-design working conditions

It is evident that the working condition of the reforming and extraction unit changes along with time and environment, thus the hot and cold ending conditions of the waste heat recovery system will vary. It is important to analyze the performance of the system under off-design conditions. The study is implemented for the scheme of dual integrated subsystems with R141bas a working fluid. The mass flow rate of the waste heat sources changes in line with the product output of the reforming and extraction unit. Fig. 10 shows that the net power output of the ORC system rises as the mass flow rate of the waste heat sources increases. The net power output of the system reaches maximum when the evaporation temperature is 393.15 K in each case.The temperature of the environment changes with time and it plays influence on the condensing temperature of the ORC system. Fig. 11 shows the net power output of the system under different condensing temperatures, which proves the point of view that the lower the condensing temperature, the higher the net power output. The maximum net power outputs under different condensing temperatures are shown in Fig. 12.

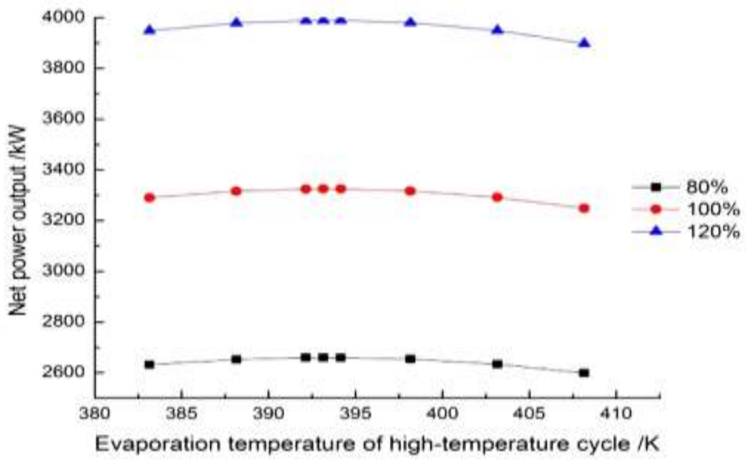

Fig. 10. Net power output under different heat loads. [18]

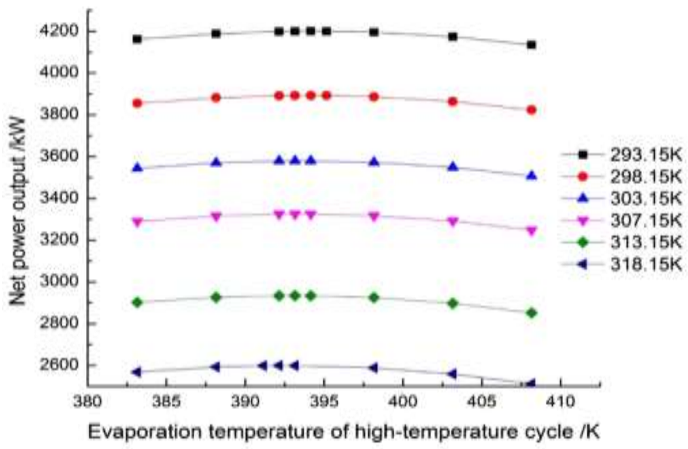

Fig. 11. Net power output under different condensing temperatures. [18]

The off-design condition analysis in this paper only takes the heat load of the waste heat sources and the condensing temperature into account. In other words, all the other parameters of the system are considered the same, i.e. the turbine efficiency, the heat losses and the pressure losses and the minimum temperature differences. In the real case, the parameters mentioned above are related to the system working condition, thus it will be part of the future work to research the off-design condition more exactly.

6. CONCLUSIONS

This paper analyzes and optimizes a comprehensive ORC recovery system utilizing five waste heat sources distributing in different temperature levels, from a 1.2 million ton-level reforming and extraction unit in Shijiazhuang Refining & Chemical Company of China. Four schemes are presented in continuous optimization progress with the aim of simplifying the entire system and achieving both the robust thermal performance and economic benefit. The main conclusions are summarized as follows:

(1) The scheme of five independent subsystems is easy to design but it is difficult to handle in operation as the entire system is extremely large and complex. The maximum net power output is 3338.89 kW and the thermal efficiency is 10.25 %.

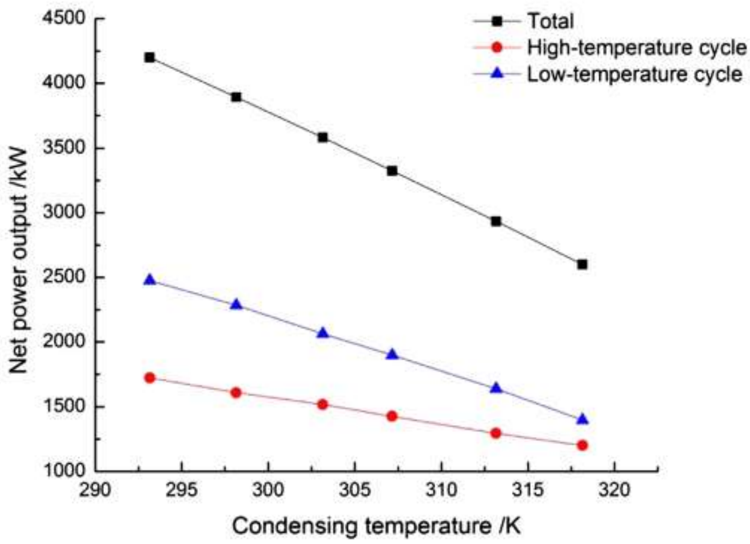

Fig. 12. Maximum net power output under different condensing temperatures. [18]

The scheme of single ORC system is withdrawn due to its considerably low thermal efficiency. The scheme of dual independent subsystems using R245fa and R141b as working fluids gives out the maximum net power output at 3353.3 kW. The scheme of dual integrated subsystems is more compact and the maximum net power output is 3325 kW, which is very similar with those of other schemes, thus it is recommended for this heat recovery case.

(2) A ratio of net power output to total heat transfer area is selected as the performance evaluation criterion from view of both thermodynamics and economics. By comparison, scheme of dual integrated subsystems R141b as a working fluid is the most recommended in this recovery case.

(3) Further analysis is implemented for the system of dual integrated subsystems with R141B as a working fluid, the simulation results indicate that the system performs well under off-design conditions.

(4) Thermal performance is the most important criterion for waste heat recovery system, and the system simplicity and technological feasibility as well as the economic factors still should

be concerned for practical application. The analytical method and optimization progress proposed in this paper can be applied in similar recovery system for multi-strand waste heat sources.

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