Process integration of organic Rankine cycle (ORC) and heat pump for low temperature waste heat recovery

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Abstract: Efficient energy utilization can not only increase profit but also alleviate environmental problems. In the process industries, considerable low temperature heat is cooled by cold utility without recovery or reutilization. Organic Rankine cycles (ORC) have been proved to be a mature technology for low temperature waste heat utilization. However, the relative low thermal efficiency and incomplete waste heat recovery limit the power output of ORC system. The reason for the above two problems is that the temperature-enthalpy (T-H) profile of waste heat sources does not match well with the organic working fluid in an ORC system. If the profile of the waste heat matches well with the organic working fluid, higher evaporation temperature can be attained, which leads to higher thermal efficiency of the ORC system. In addition, more waste heat can be recovered, which leads to more heat supplied to the ORC system. Both changes can increase of power output. Heat pumps can lift waste heat to higher temperatures, thus the T-H profile of the waste heat source can be changed. Heat pumps may reinforce the match between waste heat carrier and organic working fluid of the ORC. The isothermal heat released by a heat pump in its condenser can be used to evaporate organic working fluid during isothermal phase change. Thus a better match between waste heat and organic working fluid in the ORC can be obtained. Due to the possibility to improve power output, the integration of an ORC and a heat pump should be investigated. This paper shows the potential profit of the integrating an ORC and heat pump. Proper working fluids in ORCs and heat pumps are investigated simultaneously. The integration between ORCs and heat pumps presents excellent power output increases and deserves to be popularized in practical applications. However waste heat recovery via ORC considering a heat pump upgrade does not always lead to the increase of power output. Two illustrative examples show when and how to integrate heat pump with an ORC. The second example shows the net power output and the amount of waste heat recovered increased by about 9.37 % and 14.73 % respectively.

Key words: organic Rankine cycle, heat pump, integration, low temperature waste heat

1. Introduction

Organic Rankine cycles have received widespread attention since the 1970s. ORCs are a promising way to convert low to medium temperature heat into power [1]. The choice of working fluid, operating conditions

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and the integration of ORC with the background process all exert great influence on the net power output of ORC systems [2]. Organic Rankine cycle modifications, such as recuperation and turbine bleeding [3], can improve the thermal efficiency of the ORC system. Working fluid selection [4, 5] and operating condition determination [2, 6] affect both thermal efficiency and amount of waste heat recovered. Applying optimization methods to process integration can explore the tradeoffs between thermal efficiency, amount of waste heat recovered and economic factors [7, 8]. Yu et al. [9] proposed a new method for simultaneous working fluid selection and operating condition determination. This method is based on a newly defined parameter to locate the pinch point position. Chen et al. [10] proposed a new method based on mathematical programming to simultaneously optimize the ORC and synthesize the heat exchanger network. Yu et al. [11] proposed a superstructure to integrate an ORC into a background process considering turbine bleeding, superheating and regeneration. However, all of these studies aim at improving the power output of a standalone ORC. If another thermodynamic cycle integrated with ORC, it is possible to increase the overall net power output of the whole system.

A heat pump is a reversed Rankine cycle or Brayton cycle to upgrade heat by consuming electricity [12]. According to pinch analysis, heat pumps should be placed across pinch point in a system [13]. This concept has been applied to distillation processes successfully with substantial energy savings [14]. Heat pumps can be classified into closed and open heat pumps respectively according to whether the process stream and the working fluid are the same [15]. From a thermodynamic view, heat pumps can also be classified into reversed Rankine cycles and Brayton cycles [16]. Reversed Brayton cycle heat pumps operate in vapor phase without phase change, thus this kind of heat pump can be also called a sensible heat pump. For a reversed Rankine cycle heat pump, isothermal phase change takes place in the condenser. This isothermal condensation heat can be perfectly matched with the evaporation process of an ORC. Therefore integration of closed reversed Rankine cycle heat pumps with an ORC for low temperature waste heat utilization is considered in this paper.

Fig.1 shows the T-S diagram of an ORC and reversed Rankine cycle heat pump. The condensation heat from heat pump can be used to heat the organic working fluid during phase change. At first sight, it may be unreasonable to upgrade waste heat consuming power via heat pump, and then to generate power via an ORC from the perspective of thermodynamics. However, the involvement of a heat pump may result in higher attainable thermal efficiency of the ORC or better match between the waste heat source and organic working fluid in some cases. As long as the power increase of an ORC overweighs the power consumed by the heat pump, the integration of the heat pump and ORC is profitable. In organic Rankine cycles with pure working fluids, the isothermal phase transition will cause a pinch point between the working fluid and heat source [9]. The pinch point limits the amount of waste heat recovered by an ORC system. The pinch point limitation is caused by the improper ratio of latent to sensible heat of the working fluid, which are dependent on the thermodynamic properties of working fluid, and thus is fixed. Therefore we have to pay attention to changing the profile of waste heat with the help of a heat pump to enhance the match between waste heat source and working fluid.

To the best of our knowledge, no study has been reported to integrate heat pumps with ORCs to increase the net power output of the whole system recovering waste heat. In some specific cases, the integration of a heat pump and an ORC has the potential to increase the net power output of the whole system. The detailed problem statement will be presented in part 2. In this paper, we will investigate when it is beneficial to integrate a heat pump with ORC, and how to optimally integrate heat pump with ORC if it is beneficial.

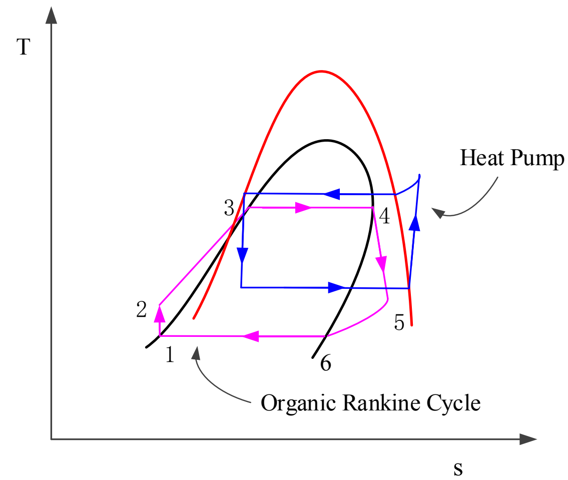

Fig.1 T-S diagram of ORC and heat pump

2. Problem statement

In an organic Rankine cycle, there are generally two ways to increase the net power output, namely increase thermal efficiency and increase the amount of waste heat recovered by organic working fluid. Working fluid selection and ORC modifications aim at increasing the thermal efficiency [17]. The operating conditions exert a great influence on the thermal efficiency as well. The optimal operating condition can be determined with the trade-off between thermal efficiency and the amount of waste heat recovered [18]. However, an important limitation of ORC with pure working fluids is the isothermal boiling, which creates pinch points and leads to bad thermal matching between working fluid and waste heat [19]. The "pinch limitation" problem in organic Rankine cycle is a critical issue to be eliminated or alleviated to improve the net power output. If the evaporating temperature is close to the critical temperature of the working fluid, the evaporation latent heat is very small, which may alleviate the pinch limitation [9]. In the limit as the evaporation temperature approaches the critical temperature, the latent heat approaches 0 and the isothermal phase change of working fluid disappears. Then the working fluid line becomes an oblique line instead of a polygonal line. The waste heat profile can match with the working fluid profile perfectly. The "pinch limitation" is eliminated and waste heat can be recovered totally. On the other hand, some researchers proposed to use supercritical ORCs to avoid the pinch limitation and the results show that supercritical

ORC performs better than subcritical ORC [20]. However, due to operation issues, safety concern and investment cost, supercritical ORCs and near critical state subcritical ORCs have not yet been implemented in practice [21]. Subcritical ORCs still have predominance in practical applications. For a subcritical ORC, the evaporating temperature cannot exceed the waste heat inlet temperature. The evaporating temperature of an ORC is limited by two parameters, namely the critical temperature of working fluid and the waste heat inlet temperature. To guarantee stable operation, the evaporating temperature should not be very close to the critical temperature, therefore the "pinch limitation" problem is a critical problem engineers to face. Fig.2 illustrates the "pinch limitation" on the T-H diagram of an ORC system. In this study, the working fluid in an ORC is cooled by cooling water in the condenser. The condensation temperature of an ORC is assumed to be 45°C. So the minimum waste heat outlet temperature is 55°C assuming the minimum approach temperature as 10°C. The purple line in Fig.2 demonstrates the working fluid evaporating at low temperature. There is no pinch limitation and all of the waste heat can be recovered, but the thermal efficiency is very low and the power output may be not satisfactory. The power output is limited by thermal efficiency in this case. When the evaporation temperature is high, as shown by the green line in Fig.2, the thermal efficiency is high, but the pinch limitation severely limits heat recovery. The power output is restricted by pinch limitation in this case. To tradeoff between thermal efficiency and the amount of waste heat recovered, optimal operating condition can be determined to be the blue line shown in Fig.2. Under optimal operating condition, the thermal efficiency is less than the green line and greater than the purple line, and the amount of waste heat recovered is less than the purple line and greater than the green line, and the pinch limitation is severer than the purple line and lighter than the green line. The thermal efficiency and amount of waste heat recovered are both moderate under the optimal condition, but this often provides maximum power output. Not all of the waste heat is recovered, but the additional thermal efficiency more than makes up for this in the total power produced.

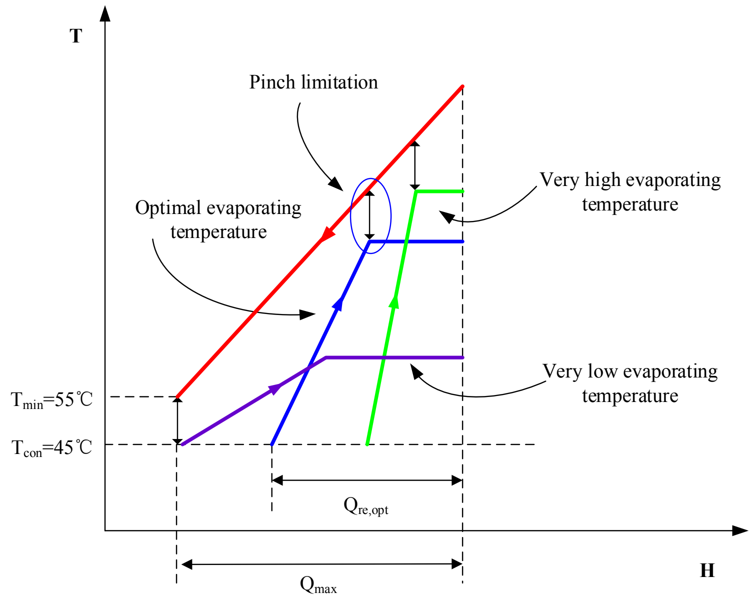

Fig.2 Illustration of pinch limitation for standalone ORC

If a heat pump is introduced, the profile of waste heat can be changed. We will show that this may improve the amount of waste heat recovered and the net power output. From Fig. 2, it is clear that the isothermal phase transition causes the pinch limitation of the system. However, the reverse Rankine cycle heat pump releases heat at a constant temperature neglecting a small portion of superheated gas condensation. This part of latent heat released by the condenser of the heat pump can be used to heat the organic working fluid in an ORC system, with the isothermal phase transitions providing well matched heat exchange profiles. Fig.3 illustrates the T-H diagram of the system integrating heat pumps with an ORC. Waste heat is recovered and utilized by two ways, namely one part is used to heat organic working fluid in ORC system, the other part is upgraded by heat pump first and then heats the organic working fluid at a higher evaporation temperature. Fig.4 shows the flowsheet of the integrated system.

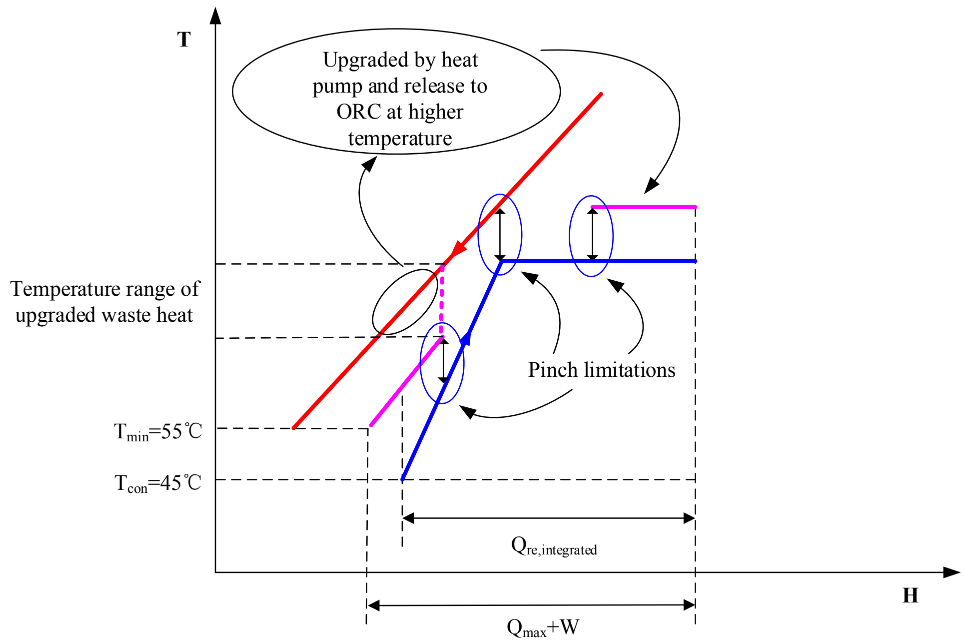

Fig.3 T-H diagram of the integrated system

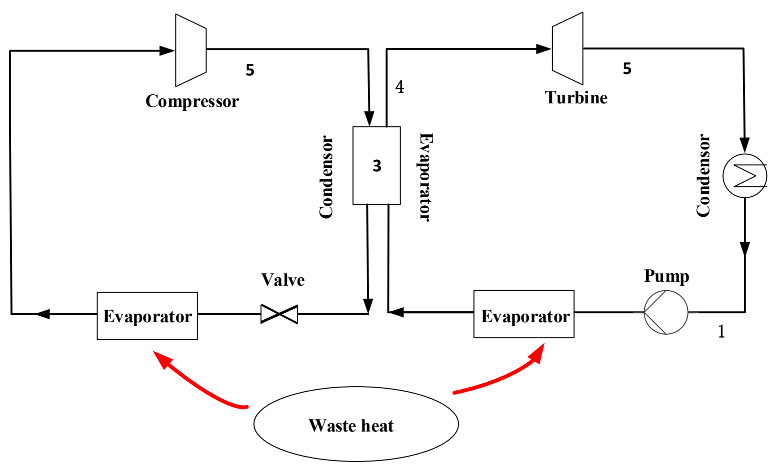

Fig.4 The layout of the integrated system

This study investigates the integration of a heat pump with an ORC for recovering waste heat. Many factors may determine whether or not this configuration is profitable, or offers any advantage over a conventional ORC. The choice of working fluids for both the ORC and heat pump are very important decisions affecting thermal efficiency. The corresponding operating conditions should also be determined appropriately.

Without considering capital investment, the integration of a heat pump is beneficial as long as the increase of power output by the ORC overweighs the power consumed by the heat pump.

While integrating heat pump with ORC, the following challenges should be addressed: (1) When it is worthwhile to integrate heat pump with ORC? For the purpose of this work, this means when can the additional power generated by the ORC offset the power consumed by the heat pump? (2) If a heat pump is to be included, which part of waste heat should be upgraded via heat pump? (3) What working fluids should be selected for ORC and heat pump respectively? (4) What are the optimal operating conditions of ORC and heat pump? This paper aims at solving the above problems to integrate heat pump with ORC.

3. Methodology

To integrate a heat pump into the system, one should first check whether there is potential in an ORC system to increase the power output. In an ORC can recover nearly all of the waste heat and the evaporation temperature is high enough, the potential to further increase the power output through using a heat pump is narrow. On the contrary, if a large portion of waste heat is not recovered, as shown in the example in Figs. 2 and 3, the integration of a heat pump may be able to increase the net power output.

As the choice of working fluid exerts great influence on the performance of heat pump and organic Rankine cycle, working fluid for both cycles should be determined. If there are very good working fluids, the integration may be more profitable, and vice versa. However, the criteria on working fluid selection for ORC and heat pump is different. For ORC, the ratio of latent to sensible heat is the working fluid selection criteria. For heat pump, coefficient of performance (COP) is the working fluid selection criteria. The reason will be analyzed in the next part in detail. After working fluids are determined for both ORC and heat pump, the optimal operating condition should be determined via the mathematical model discussed in part 3.3.

3.1 Working fluid selection for organic Rankine cycle (ORC)

Many researchers have addressed the issue of working fluid selection for ORCs. Working fluid selection relies on the temperature of waste heat, waste heat type (latent, sensible or combined waste heat) and practical considerations, such as operability, safety and environmental concerns [22]. For a sensible waste heat source, a working fluid whose critical temperature is slightly lower than the inlet temperature of the waste heat source performs well due to a less serious pinch limitation [23]. Without considering integration of a heat pump, the optimum working fluid and corresponding operating conditions can be determined using the methods in the following papers [24-26]. Thermal efficiency of the working fluid is related to the critical temperature, but the difference for all the working is tiny under the same evaporation temperature as shown in Fig.5. In general for sensible waste heat sources, the working fluid with the maximum net power output is one that can recover more waste heat. If heat pump integration is considered, the optimum working fluid in an ORC may change. The purpose of integrating heat pump into this system is to increase the part of waste heat heating the working fluid in isothermal phase changing process to get a better match between waste heat source and working fluid. The smaller ratio of latent to sensible heat of the working fluid in ORC, the less heat need to be upgraded by a heat pump to get better thermal match of the system. The optimal working fluid for the integrated ORC system will be the working fluid with the minimum ratio of latent to sensible heat. The ratio of latent to sensible heat of working fluid depends on not only the working fluid species, but also the operating conditions of ORC. For the same working fluid, the ratio of latent to sensible heat is different under different evaporation temperature. The higher the evaporation temperature, the smaller ratio of latent to sensible heat. So the ratio of latent to sensible heat is a performance indicator considering both the working fluid property and operating conditions. The integration of a heat pump is worthwhile when the ratio of latent to sensible heat of the working fluid is small. The working fluid with the minimum ratio of latent to sensible heat is the most promising candidate for the optimal working fluid in an ORC. It should be note that the ratio of latent to sensible heat of working fluid is a function of condensation temperature and evaporation temperature of an ORC. In this study, the condensation temperature of ORC is assumed as 45°C, then the ratio of latent to sensible heat relies on evaporation temperature.

Table 1 Preselected working fluids for ORC

Working fluids	Chemical formula	T_c (°C)	P_c（bar）	ODP	Type
R227ea	C_3HF_7	101.68	29.12	0.0	Dry
R236fa	$C_3H_2F_6$-D1	124.92	32.19	0.0	Dry
R600a	C_4H_{10}-2	134.65	36.40	0.0	Dry
R236ea	$C_3H_2F_6$	139.23	34.12	0.0	Dry
R600	C_4H_{10}-1	151.97	37.96	0.0	Dry
R245fa	$C_3H_3F_5$-D1	154.05	36.40	0.0	Isentropic
R601a	C_5H_{12}-2	187.25	33.80	0.0	Dry
R601	C_5H_{12}-1	196.55	33.70	0.0	Dry

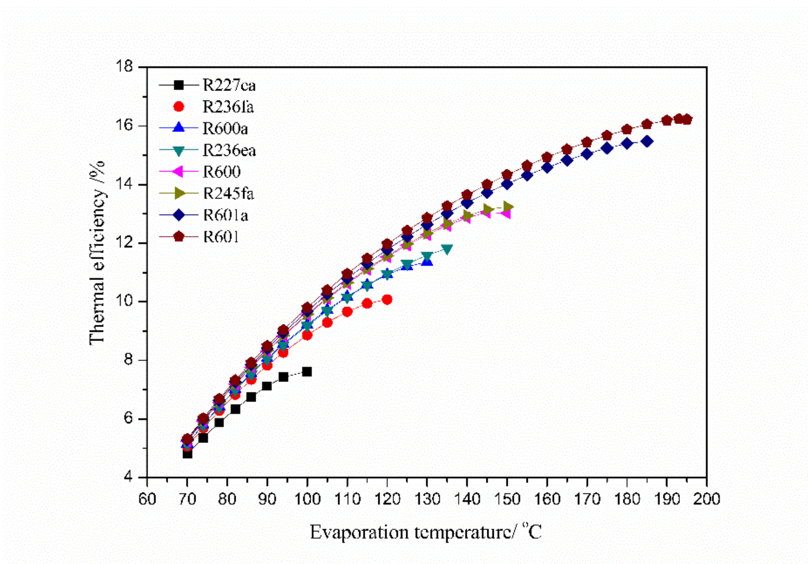

Fig.5 Thermal efficiency vs evaporation temperature for all the preselected working fluids with condensation temperature 45°C

3.2 Working fluid selection for heat pump

As for the working fluid selection for heat pump, there are two main concerns, namely the operability of the system and coefficient of performance (COP) of the working fluid. Operability refers to that the condensation temperature of heat pump should be higher than the evaporation temperature of ORC as shown in Fig.1. Several candidate working fluids for heat pumps whose critical temperatures are higher than 130°C are listed in Table 2. To guarantee the condensation heat from the heat pump evaporates the working fluid in ORC, the condensation temperature of a heat pump must be higher than the evaporation temperature of an ORC. Therefore the critical temperature of working fluid in heat pump cannot be lower than the evaporation temperature of ORC to ensure the operability of the system. The second concern is the COP. Unlike ORCs, the evaporating process of the heat pump is a purely isothermal process, thus no “pinch limitation” takes place in the evaporator. For working fluid selection in ORC, both thermal efficiency and pinch limitation need to be considered. On the contrary, only COP should be considered while selecting working fluid for heat pump. For the integrated system proposed in this study, COP is extremely important in determining whether it is worthwhile to integrate a heat pump with an ORC.

In this part, the performance of working fluids in Table 2 is investigated. A heat pump operated under various evaporating temperature with constant condensation temperature (130°C) is simulated in Aspen Plus [27]. Fig.6 illustrates the variation of COP with the evaporation temperature. It is clear that N-hexane has the maximum COP among all the preselected working fluids. Even though the results is just under fixed condensation temperature, the trend is the same under other operating conditions. Therefore, N-hexane is chosen as the working fluid used in heat pump in this study.

Table 2 The preselected working fluid for heat pump

Working fluids	Chemical formula	T_c (°C)	P_c (bar)	ODP	Type
R600a	C_4H_{10}-2	134.65	36.40	0.0	Dry
R236ea	$C_3H_2F_6$	139.23	34.12	0.0	Dry
R600	C_4H_{10}-1	151.97	37.96	0.0	Dry
R245fa	$C_3H_3F_5$-D1	154.05	36.40	0.0	Isentropic
R123	$C_2HCL_2F_3$-D1	183.79	36.76	0.02	Dry
R601	C_5H_{12}-1	196.55	33.70	0.0	Dry
N-hexane	C_6H_{14}-1	234.45	30.25	0.0	-

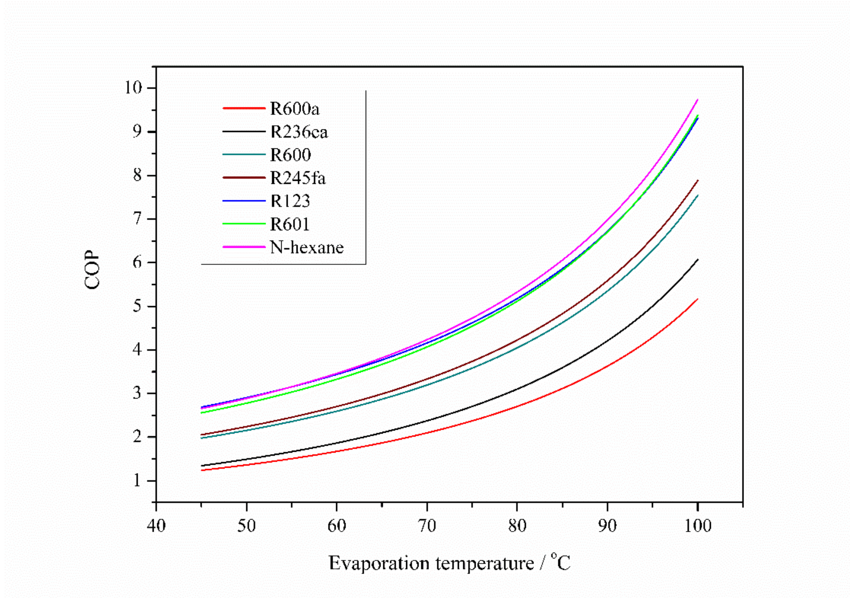

Fig.6 COP of various working fluid with fixed condensation temperature

3.3 The determination of operating conditions of the integrated system

In this section we consider the operating conditions of the combined ORC and heat pump system. First we should determine the optimal evaporating temperature of organic Rankine cycle. The evaporating temperature may be constrained by the waste heat inlet temperature or the critical temperature of working fluid for subcritical ORC. The reason is that the evaporating temperature can never be greater than these two temperatures for a subcritical ORC. As long as the evaporation temperature is determined, the corresponding heat pump condensation temperature can be determined. It is clear that the condensation temperature of heat pump should be ΔT_{min} degrees higher than the evaporating temperature of the ORC in order to maximize thermal efficiency. The higher the evaporation temperature, the smaller the latent heat of working fluid in ORC, the smaller the ratio of latent to sensible heat, and the more chances of profitable integration of heat pump. The evaporation temperature of the ORC should be as high as possible.

Under a specific ORC evaporation temperature, the most difficult problem is to determine the evaporating temperature of heat pump. The evaporating temperature of heat pump not only affects the COP, but also determines which part of waste heat should is upgraded as shown in Fig.6. The waste heat absorbed by the ORC is also related to the evaporating temperature of heat pump. When the evaporation temperature of the heat pump is 45°C (the condensation temperature of the ORC), waste heat above 55°C can be upgraded. The waste heat can be recovered totally by the ORC working fluid if the heat pump load is large enough. This is not worthwhile in most cases since the evaporation temperature is too low to justify the integration of a heat pump. Fig.7 shows the detailed notation of the parameters in this system, which will be used in the mathematical equations later. When the evaporating temperature is higher than the condensation temperature of the ORC, the waste heat above the evaporation temperature plus minimum temperature approach ($T_{eva,hp} + \Delta T_{min}$) can be upgraded by the heat pump. Then a "cliff" is formed in the waste heat curve as shown in Fig.7. As part of the waste heat is upgraded by the heat pump, the new cliff will become a new potential pinch limitation. Then how much waste heat should be upgraded? When three pinches develop simultaneously as shown in Fig.7, the amount of waste heat to be upgraded under the specified evaporation temperature reach optimum. The reason is that if we upgrade more waste heat, pinch point I will not exist, in essence more waste heat is upgraded than necessary. Pinch II limits the amount of waste heat recovered at this time. When the amount of waste heat upgraded is very small, pinch II will not develop. And Pinch I limits the amount of waste heat recovered. As the main purpose of integration with a heat pump is to increase the total amount of waste heat recovered, the function of increasing the amount of waste heat recovered is limited under this circumstance. As long as the integration of the heat pump can increase both the waste heat recovered and net power output, we should increase the amount of waste heat upgraded until three pinch points develop simultaneously as shown in Fig.7. Under this circumstance, the potential of the power output increase reaches to the limitation. So under specified evaporation temperature, the optimal amount of waste heat to be upgraded should be calculated, namely determining T_{cliff} as shown in Fig.7. Then the corresponding operating condition can be determined.

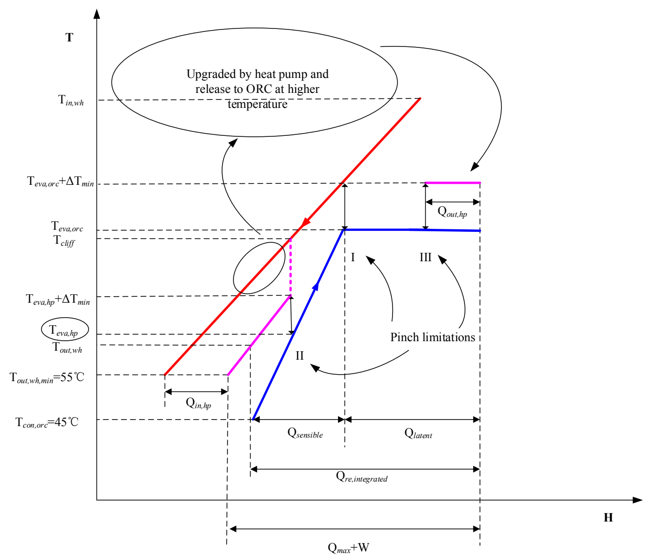

Fig.7 T-H diagram of the integrated system with detailed notation

The above discussion only guarantees the optimal amount of waste heat to be upgraded under specific evaporating temperature. To get the final optimal operating conditions, heat pump at various evaporation temperature should be integrated. Then the optimal evaporation temperature of heat pump can be determined. In theory, we should try other ORC evaporation temperatures and determine the corresponding optimal evaporating temperature of heat pump and the optimal amount of waste heat upgraded using the above method. However, as shown in the case study part, only when the evaporating temperature exhibit the minimum ratio of latent to sensible heat, the integration of a heat pump is profitable. Through the above method, the optimal design of the integrated system can be obtained. The mathematical model is presented as follows;

The COP of heat pump is the ratio of heat released in the condenser to the power consumed as shown by Eq. (1).

$$COP = \frac{Q_{out,hp}}{W} \tag{1}$$

From the first law of thermodynamics, the sum of waste heat upgraded and the power consumed by the heat pump equals to the total heat released by the heat pump condenser.

$$Q_{in,hp} + W = Q_{out,hp} \tag{2}$$

Then the heat pump released heat and the absorbed heat satisfy Eq. (3).

$$Q_{in,hp} = Q_{out,hp}(1-\frac{1}{COP}) \tag{3}$$

The ratio of sensible heat to latent heat of working fluid is a constant under specified evaporating temperature and condensation temperature.

$$r_{l/s} = \frac{R_l}{R_s} = \frac{Q_l}{Q_s} = Constant \tag{4}$$

When all the three pinches develop, pinch II and pinch I can be used to calculate the maximum organic working fluid flowrate. Referring to the notations in Fig.6, the sensible heat of working fluid from $R_{s,cliff}$ can be calculated by Eq. (5).

$$\frac{R_{s,cliff}}{R_s} = \frac{T_{eva,orc} - T_{eva,hp}}{T_{eva,orc} - T_{con,orc}} \tag{5}$$

When pinch I and pinch II occur simultaneously, the optimal T_{cliff} should satisfy Eq. (6)

$$\frac{R_{s,cliff}}{R_l} = \frac{FCp_{wh}(T_{eva,orc} + \Delta T_{min} - T_{cliff})}{FCp_{wh}(T_{in,wh} - T_{eva,orc} - \Delta T_{min}) + Q_{out,hp}} \tag{6}$$

From Eqs. (1)-(5), and eliminating $R_{s,cliff}$, the waste heat to be upgraded can be calculated by Eq. (7).

$$Q_{in,hp} = \frac{FCp_{wh}\left[(T_{eva,orc} - T_{con,orc})R_{l/s} - (T_{in,wh} - T_{eva,orc} - \Delta T_{min})\right]}{R_{l/s}(T_{eva,orc} - T_{con,orc})/(T_{eva,orc} - T_{eva,hp}) + [COP/(COP-1)]} \tag{7}$$

Then the power consumed by heat pump can be calculated by Eq. (8).

$$W_{hp} = (COP-1)Q_{in,hp} \tag{8}$$

The “cliff” temperature T_{cliff} can be obtained by Eq. (9).

$$T_{cliff} = T_{eva,hp} + \Delta T_{min} + Q_{in,hp} / FCp_{wh} \tag{9}$$

The heat capacity flowrate of organic working fluid is determined by the Pinch I and II, which can be calculated by Eq. (10).

$$FCp_{orc} = \frac{FCp_{wh}(T_{eva,orc} + \Delta T_{min} - T_{cliff})}{T_{eva,orc} - T_{eva,hp}} \tag{10}$$

The heat capacity flowrate of organic working fluid must be less than that of waste heat as indicated by Eq. (11), which is consistent with Fig.6.

$$\frac{FCp_{orc}}{FCp_{wh}} = \frac{(T_{eva,orc} + \Delta T_{min} - T_{cliff})}{T_{eva,orc} - T_{eva,hp}} = \frac{(T_{eva,orc} - T_{eva,hp} - Q_{in,hp} / FCp_{wh})}{T_{eva,orc} - T_{eva,hp}} < 1 \tag{11}$$

Then the total sensible heat of organic working fluid can be calculated by Eq. (12).

$$Q_{orc,s} = FCp_{orc}(T_{eva,orc} - T_{con,orc}) \tag{12}$$

The waste heat cannot be recovered is the maximum waste heat load below $T_{eva,orc} + \Delta T_{min}$ minus the sensible heat recovered by organic working fluid, which can be calculated by Eq. (13).

$$Q_{wh,unre} = FCp_{wh}(T_{eva,orc} + \Delta T_{min} - T_{cliff} + T_{eva,hp} + \Delta T_{min} - T_{out,wh,min}) - Q_{orc,s} \tag{13}$$

Then the outlet temperature of waste heat can be calculated by Eq. (14).

$$T_{out,wh} = T_{out,wh,min} + Q_{wh,unre} / FCp_{wh} \tag{14}$$

The total recovered waste heat load is the sum of maximum recoverable waste heat and power consumed by the heat pump minus the unrecovered waste heat load as shown in the following equation:

$$Q_{wh,re} = FCp_{wh}(T_{in,wh} - T_{out,wh,min}) + W - Q_{wh,unre} \tag{15}$$

Then the power output of the ORC can be calculated by

$$P_{orc} = \eta_{orc} Q_{wh,re} \tag{16}$$

The net power output of the system is

$$P_{net} = P_{orc} - W_{hp} \tag{17}$$

All the important parameters of the integrated system can be determined by the above equations with given evaporation temperature of heat pump and ORC. These equations guarantee that three pinches occur, which results in the maximum net power output. The above equations only guarantee the optimal condition under fixed heat pump evaporation temperature. The evaporation temperature of heat pump is a free variable. To find the optimal evaporation temperature of heat pump, Eqs. (1)-(17) should be solved under various evaporating temperatures. With the results under various evaporation temperatures of heat pump, the optimal operating condition of the integrated system can be determined.

4. Case study

Two example considering different situations are presented to illustrate when and how to optimally integrate heat pump into the system. These two cases is taken from a paper [23], in which the effect of the waste heat source characteristic on the working fluid selection is investigated. The waste heat sources are classified into sensible, latent, and combined waste heat sources. Illustrative example 1 is the case where it is not necessary to integrate the heat pump into the system. Illustrative example 2 demonstrates how to optimally integrate heat pump into the system when it is necessary. The effect of the ratio of latent to sensible heat of working fluid in ORC on the integrated system is also investigated.

The assumptions and parameters used in the original paper are as follows:

- The target temperature of waste heat can be as low as ambient temperature
- The condensation temperature of ORC is assumed as 45°C
- The heat capacity flowrate of sensible waste heat is assumed as 100 kW/°C
- Minimum heat transfer approach temperature is assumed as 10°C

4.1 Illustrative example 1

In this case, a combined waste heat source is investigated. The inlet temperature of waste heat is 120°C and the latent heat load is 6500 kW. The maximum sensible heat source is 100*(120-55) = 6500 kW. The latent heat load is 6500 kW. Then the total heat load of the waste heat source is 13000 kW. The T-H diagram of the waste heat is the red line shown in Fig.8. Fig.9 shows the variation of the net power output and the amount of waste heat recovered with evaporation temperature. It is apparent that R600a gives the maximum power output. Fig.8 illustrates the T-H diagram of waste heat source and organic working fluid.

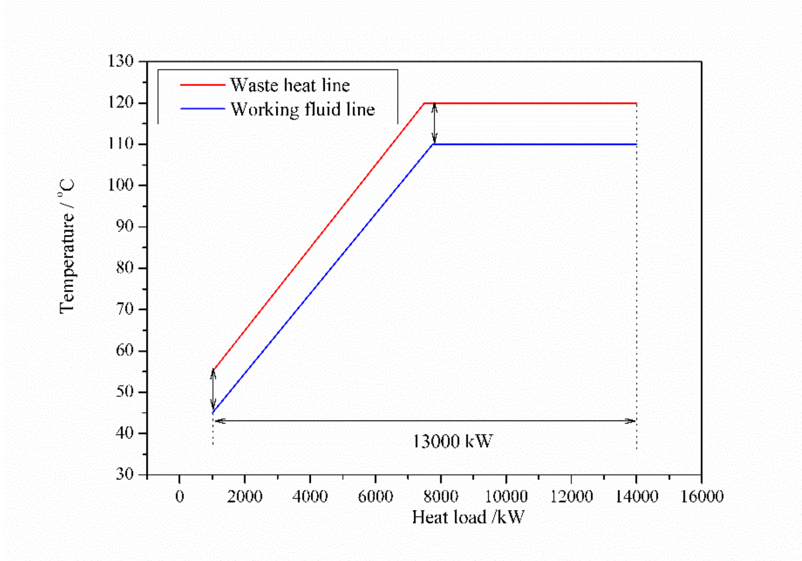

Fig.8 T-H diagram of waste heat source and R600a under optimal operating conditions

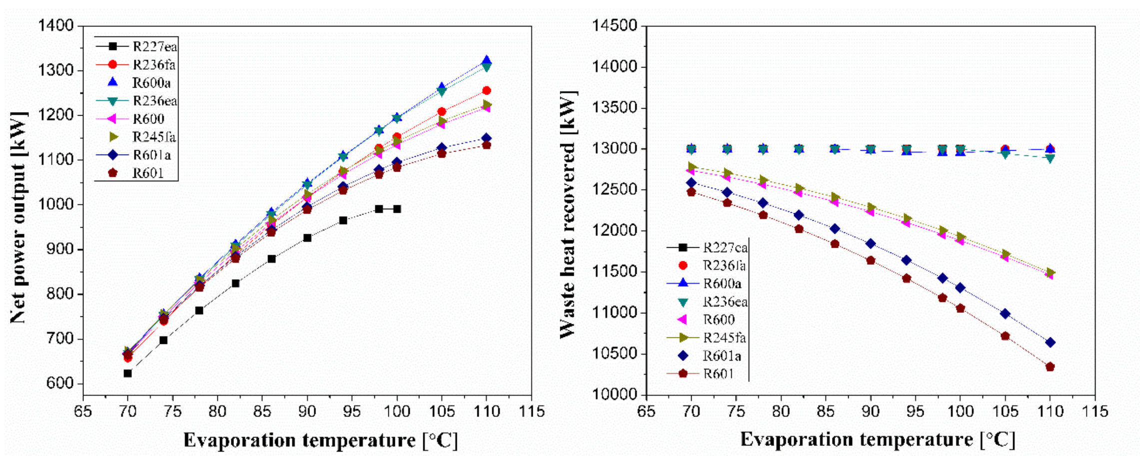

Fig.9 Net power output and waste heat recovered of standalone ORC for example 1

Based on Fig.8, it is clear that the waste heat is recovered totally, and the evaporation temperature has reach the maximum. There is no space to increase the amount of waste heat recovered or the thermal efficiency. Waste heat source match very well with the organic working fluid. For this case, heat pump cannot improve neither the amount of waste heat recovered nor the thermal efficiency of ORC. It is not profitable to integrate a heat pump in this case.

4.2 Illustrative example 2

In this case, a sensible heat source is investigated. The inlet temperature of waste heat source is 150°C. Then the maximum available waste heat is 100*(150-55) =9500 kW. Fig.10 shows the variation of net power output and the amount of waste heat recovered with evaporation temperature for a stand-alone ORC. It is apparent that R236fa gives the maximum power output 805 kW. The optimum working fluid is R236fa and the optimal evaporating temperature is 120°C.

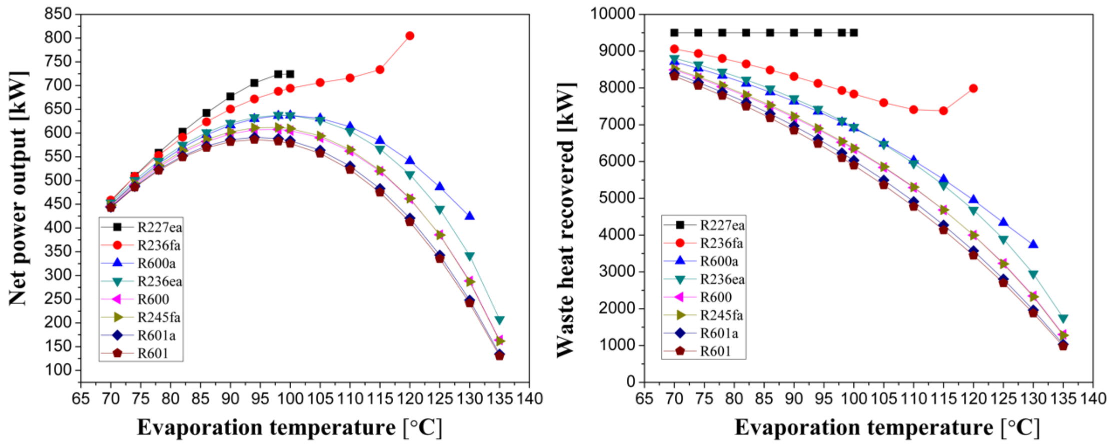

Fig.10 Net power output and waste heat recovered of standalone ORC for example 2

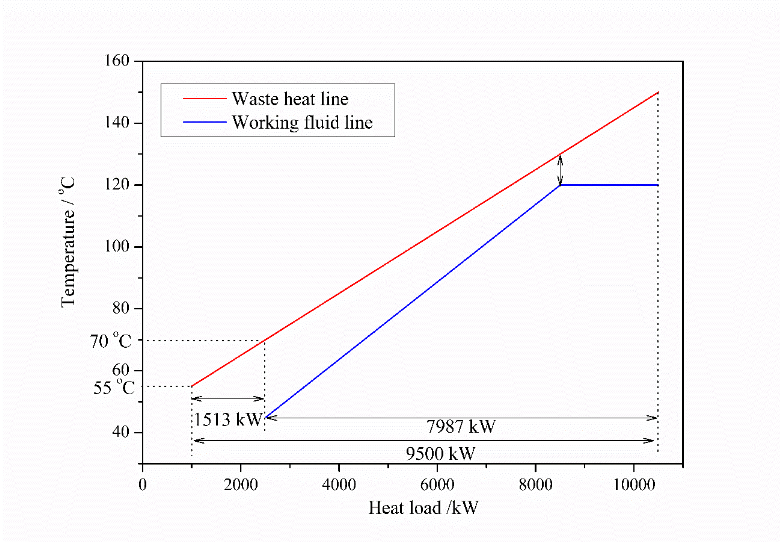

Fig.11 T-H diagram of standalone ORC under optimal operating conditions

Fig.11 illustrates the T-H diagram for waste heat source and organic working fluid R236fa under optimal operating conditions. It should be note that the amount of waste heat recovered is about 7987 kW, which is less than the maximum available waste heat 9500 kW. Only 84% waste heat is recovered. In addition, the optimal evaporating temperature is only 120°C, which is limited by the critical temperature of R236fa. Due to the waste heat inlet temperature of 150°C, the evaporating temperature may be increased, so there is space to increase the amount of the waste recovered or the thermal efficiency of ORC. It may be profitable to integrate a heat pump into the system. Based on the results shown in Fig.6, N-pentane is selected as the working fluid in the heat pump. As discussed in part 3.1, the working fluid in the ORC should be the one with the minimum ratio of latent to sensible. Table 3 lists the ratio of latent heat to sensible heat of several working fluids under specific evaporating temperatures. It can be seen that R236fa evaporating at 120°C exhibits the minimum ratio of latent to sensible heat. So R236fa should be investigated firstly. R600a evaporating at 130°C exhibits the second minimum ratio. R600a is also investigated to explore the effect of the ratio of latent to sensible heat on the integrated system.

Table 3 The ratio of latent to sensible heat of working fluid under various conditions

Working fluids	Evaporation temperature(°C)	R_l (kJ/kg)	R_s (kJ/kg)	Ratio (r)
R236fa	120	38.72	115.91	***0.33***
R600a	120	150.20	221.86	0.68
R600a	130	97.18	265.61	***0.37***
R236ea	120	79.40	106.50	0.75
R236ea	130	63.43	123.88	0.51
R600	120	213.34	213.70	1.00
R600	130	184.01	248.08	0.74
R245fa	120	111.76	111.63	1.00
R245fa	130	97.31	128.99	0.75
R601a	120	248.48	195.41	1.27
R601a	130	233.26	224.61	1.04
R601	120	271.13	196.47	1.38
R601	130	256.72	225.57	1.14

4.1.1 R236fa as the working fluid in ORC

As the critical temperature of R236fa is 124.92°C, the evaporation temperature is limited to 120°C for R236fa. In addition, R236fa evaporating at 120°C exhibits the minimum ratio of latent to sensible heat, which makes it the most promising working fluid candidates for ORC in the integrated system. Then the condensation temperature of the heat pump is set as 130°C. With Eqs.(1)-(17), the optimal amount of waste heat to be upgraded under various evaporation temperatures can be determined. Fig.12 shows the variation of upgraded waste heat load, unrecovered waste heat load, heat load absorbed by ORC and waste heat outlet temperature.

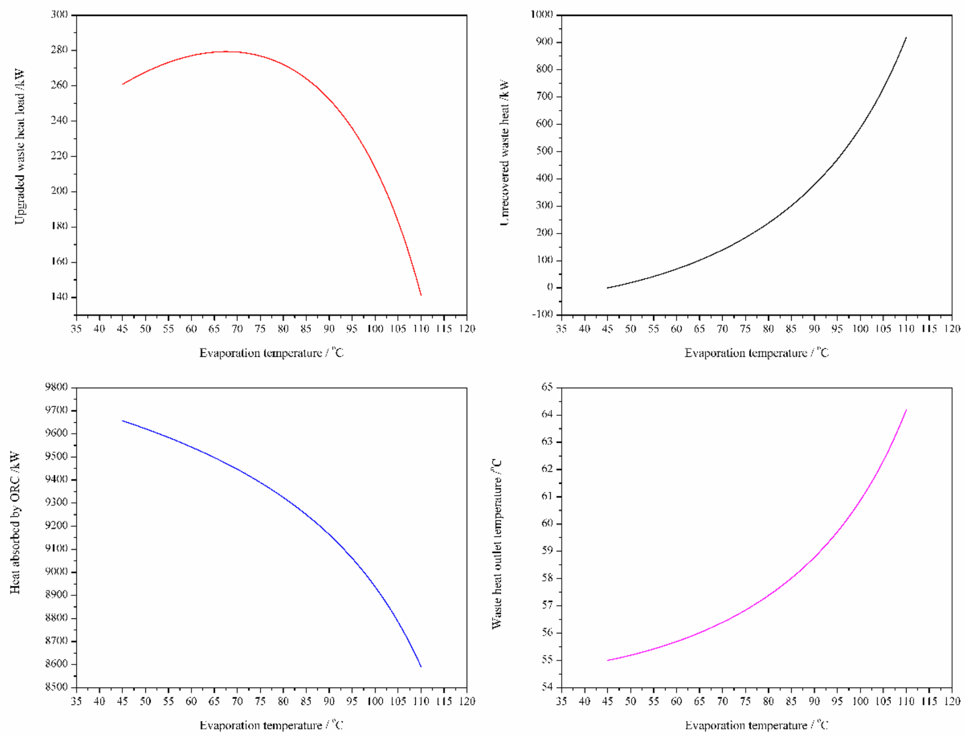

Fig.12 Key parameters with the variation of heat pump evaporation temperature

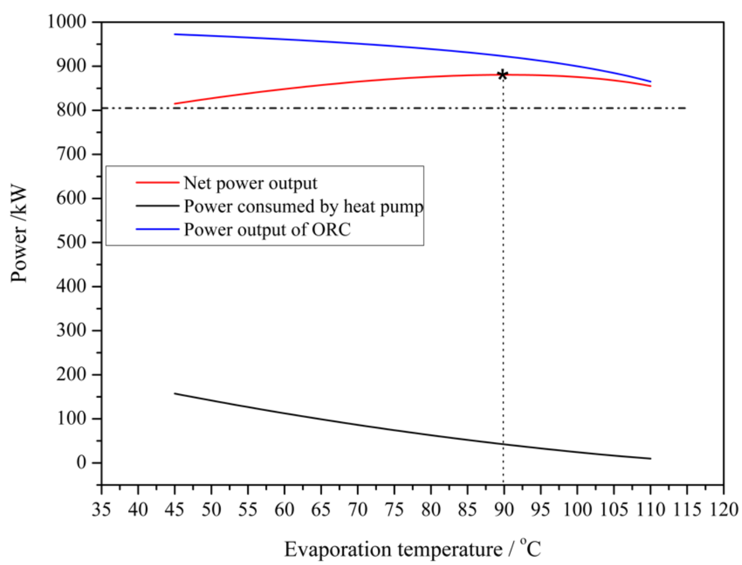

Fig.13 Integrated system performance with the variation of heat pump evaporation temperature with R236fa as working fluid

Fig.13 illustrates variation of net power output, heat pump power consumption and ORC power output with evaporation temperature of heat pump. The horizontal dash line indicates the maximum power output (805 kW) of a stand-alone ORC recovering waste heat. The integration of heat pump improves the net power output of the system. When the evaporation temperature of heat pump is 45°C, the COP of heat pump is 2.65. Based on Eqs. (1)-(3), the amount of waste heat to be upgraded is 280 kW, and the power consumed is 169 kW, and the total waste heat is 9669kW. The thermal efficiency of an ORC with R236fa evaporating at 120°C is about 10.07%, then the net power output is about 805 kW. The net power output of the integrated system is same as that of the original stand-alone ORC. So it is not profitable to integrate the heat pump at the evaporation temperature of 45°C. The reason is that the COP is too low to justify the integration of a heat pump under this evaporation temperature. With the increase of evaporation temperature of heat pump, the COP of the heat pump increases, and the amount of waste heat to be upgraded increases first and then decreases, and both the power consumed by heat pump and the power output by ORC decrease. When the evaporation temperature of heat pump is 90°C, the net power output reach maximum 880.5 kW. Heat pump upgrade the waste heat within 100°C and 102.52°C to 130°C consuming 42.2 kW power. The COP of heat pump is 6.97 at this point. Then the total available heat load is 9542.2 kW and the amount of waste heat recovered is 9164 kW, the final outlet temperature of waste heat source is 58.78 °C. For stand-alone ORC, only 7897 kW waste heat is recovered and the outlet temperature of waste heat is 70 °C. At the expense of 42.2 power consumption, the amount of waste heat is increased by 14.73%, the power output is increased by 9.37%. Fig.14 illustrates the optimal T-H diagram of the integrated system. Compared with the T-H diagram of stand-alone ORC system in Fig.11, waste heat matches much better with the working fluid after heat pump is integrated.

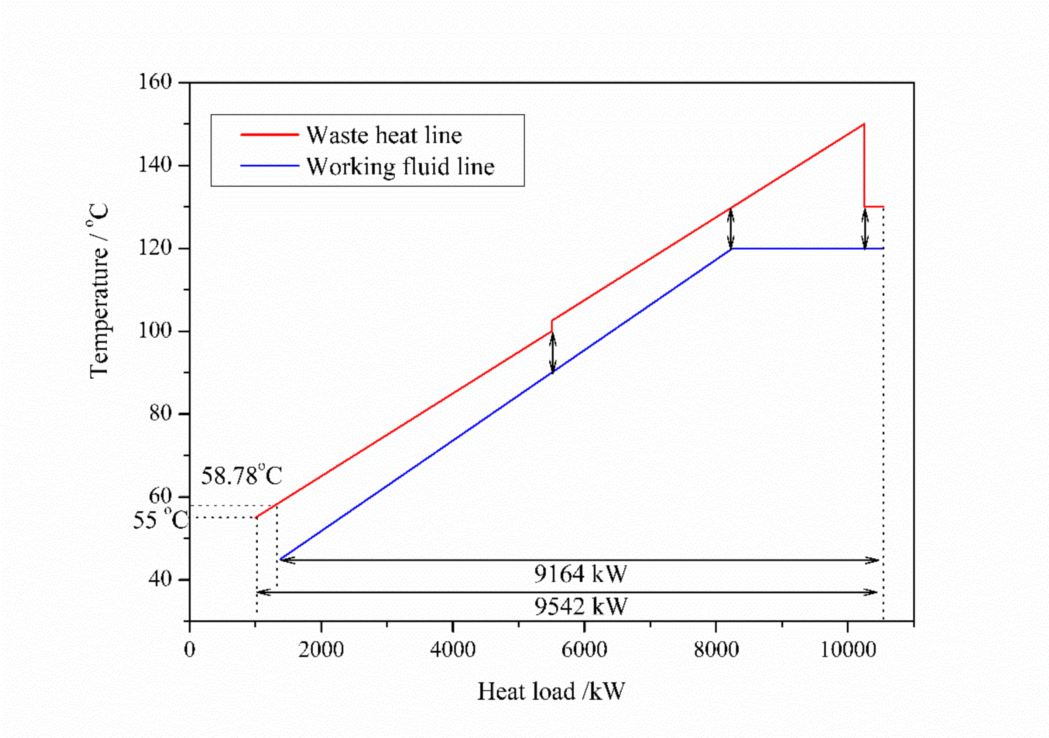

Fig.14 T-H diagram of integrated system under optimal operating conditions

R236fa is selected as the working fluid in ORC for the integrated system since it has the minimum ratio of latent to sensible heat. To show the critical effect of the ratio of latent to sensible heat on the integration problem, R600 is selected as another option. As the critical temperature of R236fa is 124.92°C, evaporation temperature cannot be higher or close to critical temperature. On the contrary, R600 has higher critical temperature. Therefore higher thermal efficiency can be obtained at higher evaporation temperature. As the inlet temperature of waste heat source is 150°C, the evaporation temperature can be higher than 120°C. When working fluid R600 evaporates at 130°C, the condensation temperature of heat pump is 140°C accordingly. It should be note that higher evaporating temperature in ORC indicates higher condensation temperature of ORC, which means lower COP of heat pump and more power consumption. Fig.15 shows the net power output of the integrated system under various evaporation temperature. It can be seen that the net power output is less than the original stand-alone ORC. If the working fluid in the ORC is R600, it is not profitable to integrate heat pump into the system.

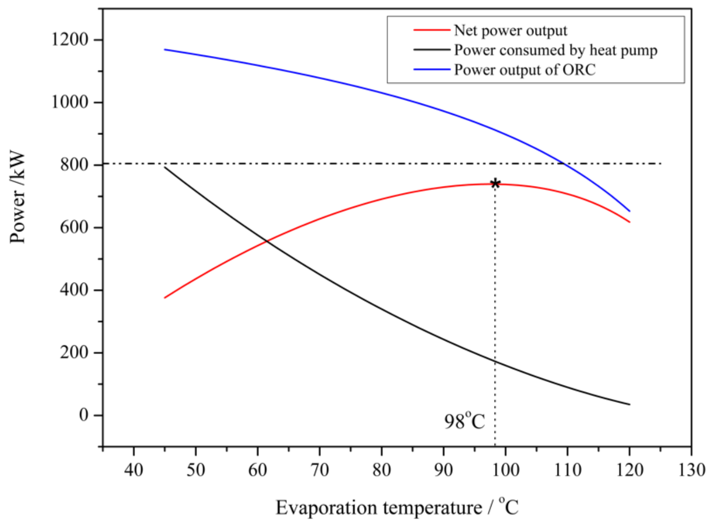

Fig.15 Integrated system performance with the variation of heat pump evaporation temperature with R600 as working fluid

Even though R600 can attain higher thermal efficiency compared with R236fa, it is not profitable to integrate a heat pump with an ORC in this case. It can be seen that the ratio of latent to sensible heat is the dominant factor in the integrated system. It is not always profitable to integrate a heat pump into the system. The minimum ratio of latent to sensible heat is the most important factor to be considered while designing the integrated problem. The ratio depends on the working fluid and evaporation temperature. Further, the evaporation temperature is limited by waste heat inlet temperature and critical temperature of working fluid. Therefore working fluid, evaporation temperature and waste heat inlet temperature determine the performance of the integrated system jointly.

5. Conclusion

This study presents a methodology to integrate a heat pump with an ORC to improve the net power output of the system. The method can, however, indicate whether there is potential to increase the power output and whether it is profitable to integrate a heat pump into this system.

Integrating a heat pump into the system does not always result in the increase of net power output. Whether it is profitable to integrate heat pump relies on the ratio of latent to sensible heat of working fluid and the COP of heat pump. When the critical temperature of working fluid is lower than the inlet temperature of waste heat source, the ratio of latent to sensible heat is small, then the integration is probably profitable. If the COP of heat pump is improved, the chance to integrate heat pump will be higher. A systematic method to determine the working fluid in both ORC and heat pump, and the optimal operating conditions of the integrated system is proposed. The second example shows the net power output and the amount of waste heat recovered increased by about 9.37 % and 14.73 % respectively with the proper selection of working fluid and operating conditions.

In this paper, only simple closed compression heat pump cycles are investigated. Other cycles with modification to increase the COP of heat pump cycle such as closed compression cycles with economizer, cascade configuration, and mechanical vapor recompression are also promising alternatives to enhance the system performance. What is more, equipment cost is not taken into account in this paper. In the future work, other kinds of heat pump cycles can be exploit to enhance the performance of the system and the equipment cost of the system should be considered.

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Nomenclature

COP	coefficient of performance
FCp	heat capacity flowrate

H	enthalpy
ORC	organic Rankine cycle
ODP	ozone depletion potential
P	power output/pressure
Q	heat load
r	ratio
R	specific heat
S	entropy
T	temperature
W	electricity consumption
η	thermal efficiency
Subscript	
c	critical
cliff	cliff formed in the T-H diagram
con	condensation/condenser
eva	evaporation/evaporator
hp	heat pump
in	inlet/ absorbed heat
l	latent
min	minimum
orc	organic rankine cycle
out	outlet/ released heat
re	recovered
s	sensible
unre	unrecovered
wh	waste heat

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