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Key issues and novel optimization approaches of industrial waste heat recovery in district heating systems

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Abstract

Large amount of low-grade waste heat is discharged into the environment during industrial processes. This part of waste heat can be collected to serve district heating systems as an important heat source. In most studies of industrial waste heat recovery, the proposed system simulations were unsophisticated in terms of analyzing the real processes. For this reason, the tangency analysis has recently been proposed, and it has been found effective in conducting optimization analysis for direct-heat-exchange systems with multi-heat sources. However, in this study, it has been found that the tangency method has limitations in designing systems with heat pumps, and therefore the disadvantages of tangency analysis are suggested and discussed. Exergy analysis reveals that without considering additional exergy generated by heat pumps, the systems designed by tangency technology tend not be the optimal configuration when heat pumps are employed. In this study, the process optimization principles have been developed from the exergy analysis of heat recovery systems with heat pumps. The optimization principles and mean-heat-transfer-times index are proposed as the key point of process design. Based on the principles, two specific optimization methods in

graphic expression are suggested. In the case studies, energy input decreased by more than 70%, which compares favorably with that of tangency analysis.

Keywords

industrial waste heat recovery, optimization, exergy analysis, district heating, heat pumps

Highlight

- Exergy analysis of multi-heat recovery systems with heat pumps is proposed.
- The mean heat transfer times is proposed as the key point of heat recovery process optimization.
- Two specific optimization methods in graphic expression are suggested, and significant energy reduction can be achieved.

1. Introduction

Large amount of waste heat at low temperature level, mostly between 30℃ and 100℃ [1-3], is discharged into the atmosphere during industrial processes. In China, it was estimated that at least 50% of energy had been wasted in industry sector, mainly in the form of low-grade waste heat [4]. It is also estimated that the loss amount of low-grade waste heat was as high as 7.6 million TJ in northern China each year [5], which is nearly 260 million tons coal equivalent. Waste heat at a high temperature level (i.e. above 120℃) is reused directly for the production processes, while waste heat at an intermediate temperature level (i.e. 60-120℃) is recovered directly by heat exchangers for district heating (DH). Additionally, waste heat featuring a low temperature range (i.e. 30-60℃) is reportedly recovered by heat pumps for district heating. For instance,

several DH networks using in0dustrial waste heat are established in Sweden [6], Germany [7] and Switzerland [8].

In ordinary practices, both middle-grade and low-grade waste heat is employed for district heating. However, the latter one is usually dominant in quantity, which makes heat recovery for district heating very difficult. This point can be evidenced by heat sources from five plants surveyed in this research. Plant I, II and III are oil refineries, while Plant IV is a petrochemical factory. Each plant has a variety of heat sources. To specify features of heat sources, Figure 1 illustrates two heat sources in the Plant I, which refers to two processes in refinery. The refined materials hold some amount of heat, and therefore are capable to serve as heat sources. Due to the control requirements of refinery, the parameters of materials are strictly limited and determined by the industrial process, including inlet and outlet temperatures, flowrate and thereby heat amount. This amount of heat is generally removed by air cooling or circulating cooling water and cooling tower. Therefore, in an industrial waste heat recovery process, the heat sources conditions are set and can be regarded as inputs. The air cooling, etc. is replaced by a heat recovery process, and DH return water serves as cool sources (Figure 1 (a') and (b')). Figure 2 depicts the corresponding T-Q diagrams of the two heat sources in Figure 1. The endpoints of the line on Y-coordinate represent the inlet and outlet temperatures of a heat source, and the projection length of this line on X-coordinate represents heat power of this heat source.

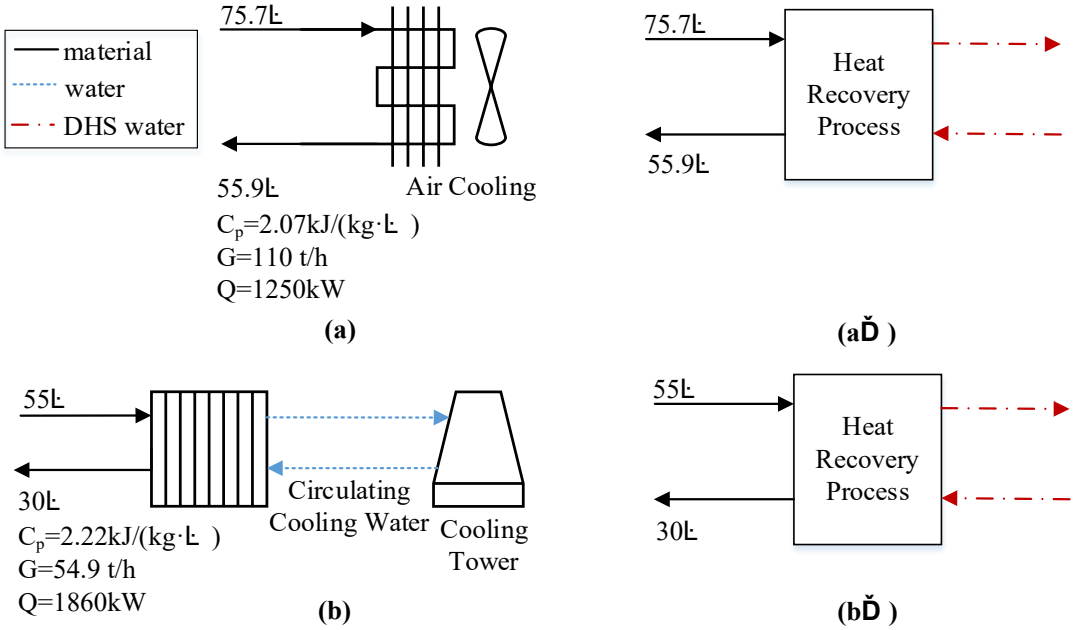
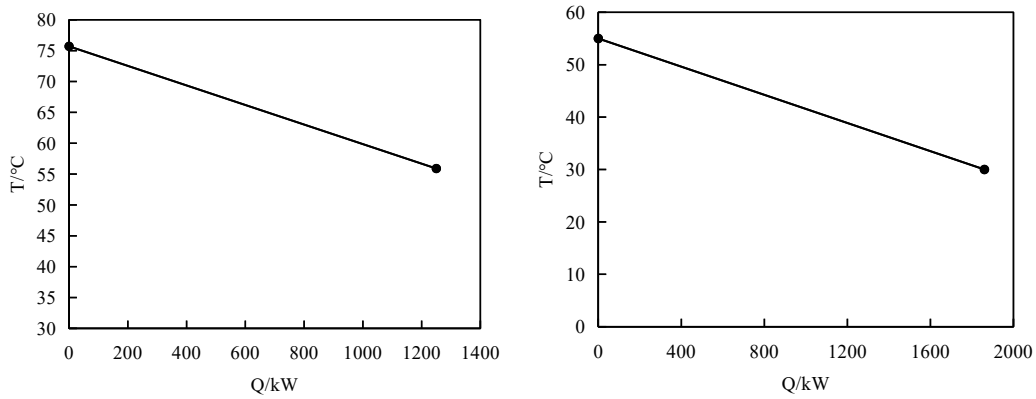


Figure 1. Two heat sources in the Plant I

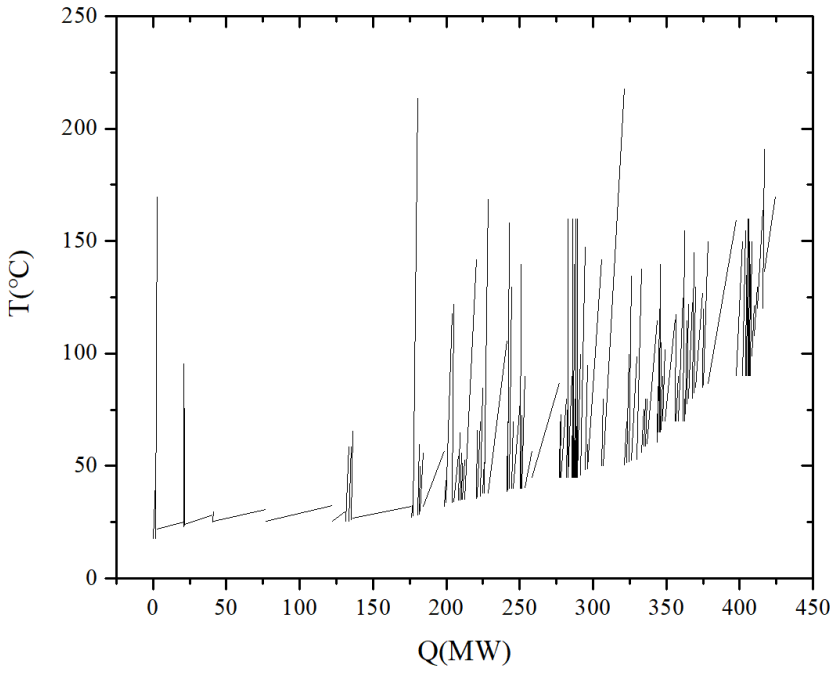


heat source (a) heat source (b)

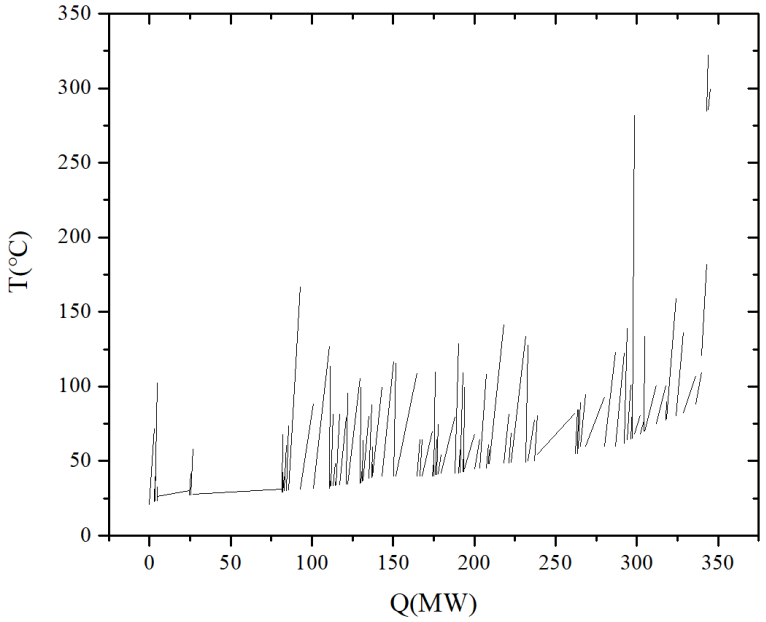
Figure 2. T-Q diagram of the two heat sources in Figure 1

Figure 3 depicts all heat resources in the surveyed 4 plants on T-Q diagrams, where heat sources are sorted by the outlet temperature. Similarly, each line represents a heat source in a T-Q diagram. Figure 3 shows that there are 119, 91, 41, and 172 heat sources in plant I, II, III and IV, respectively. It can be seen from Figure 3 that many waste heat sources overlap with each other in terms of temperature range, which makes it difficult to work out the overall temperature and grade level of waste heat

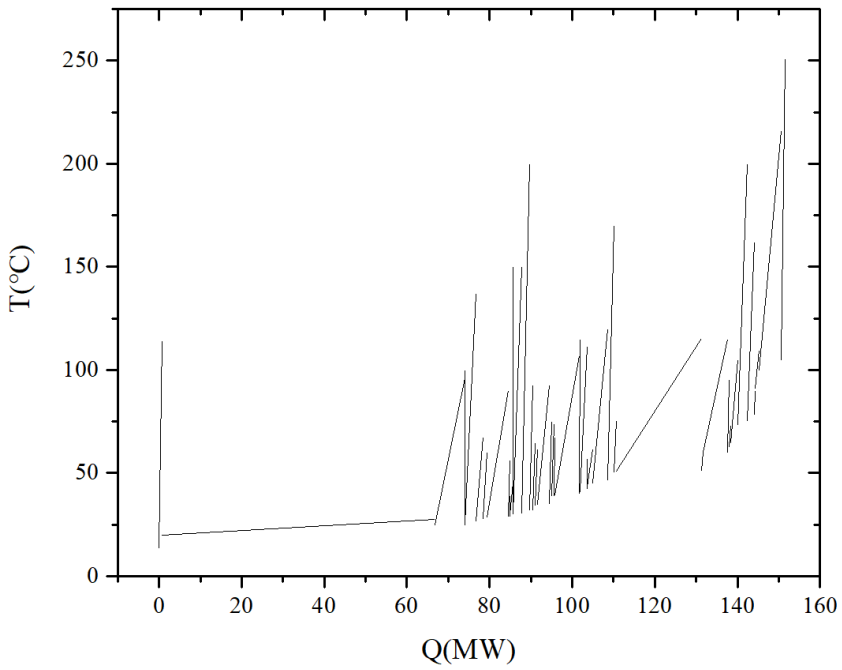
from a plant. Integration of heat sources is therefore needed. For heat sources with overlapped temperature ranges, an integration method was suggested by Fang [9]. Heat sources can be integrated one by one, as shown in Figure 4. The overlapped part on the temperature range can be merged directly by adding heat power ($B_1B+CC_1=B_1C_1$), while the other parts remain the same (AB_1 and C_1D).



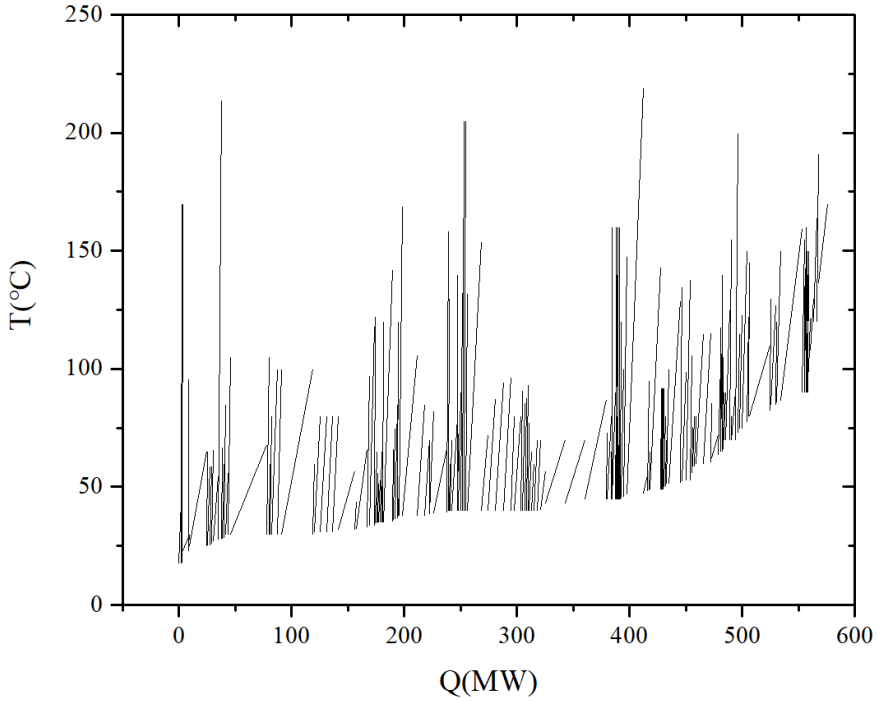
Plant I



Plant II



Plant III



Plant IV

Figure 3. Waste heat from four different plants

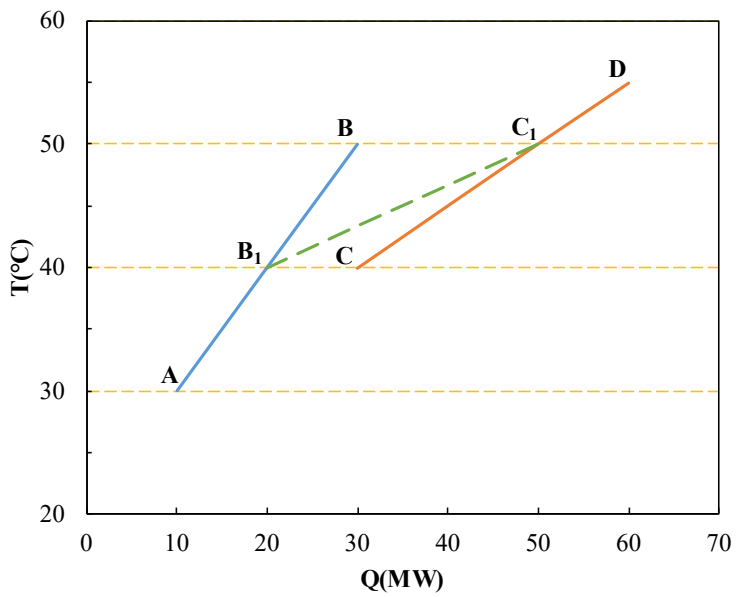


Figure 4. Two Heat sources AB and CD can be integrated into one Heat source AB_1C_1D

All heat sources of the 4 plants are merged into an integrated waste heat curve, as illustrated in Figure 5. According to the heat flux compound curves, it can be seen that about half of waste-heat temperature is under 50℃ in plant A and plant C, and about

half of waste-heat temperature is under 75℃ in plant B and D. The majority of waste-heat temperature is under 100℃. Cool sources at low temperature level below 50℃ is needed for waste heat recovery. However, traditional DH systems possess a relatively high temperature in China, where supply and return water temperature is 100-120℃/45-60℃ in most winter time. Therefore, a large amount of waste heat fails to achieve a proper recovery by traditional DH systems with direct heat exchangers.

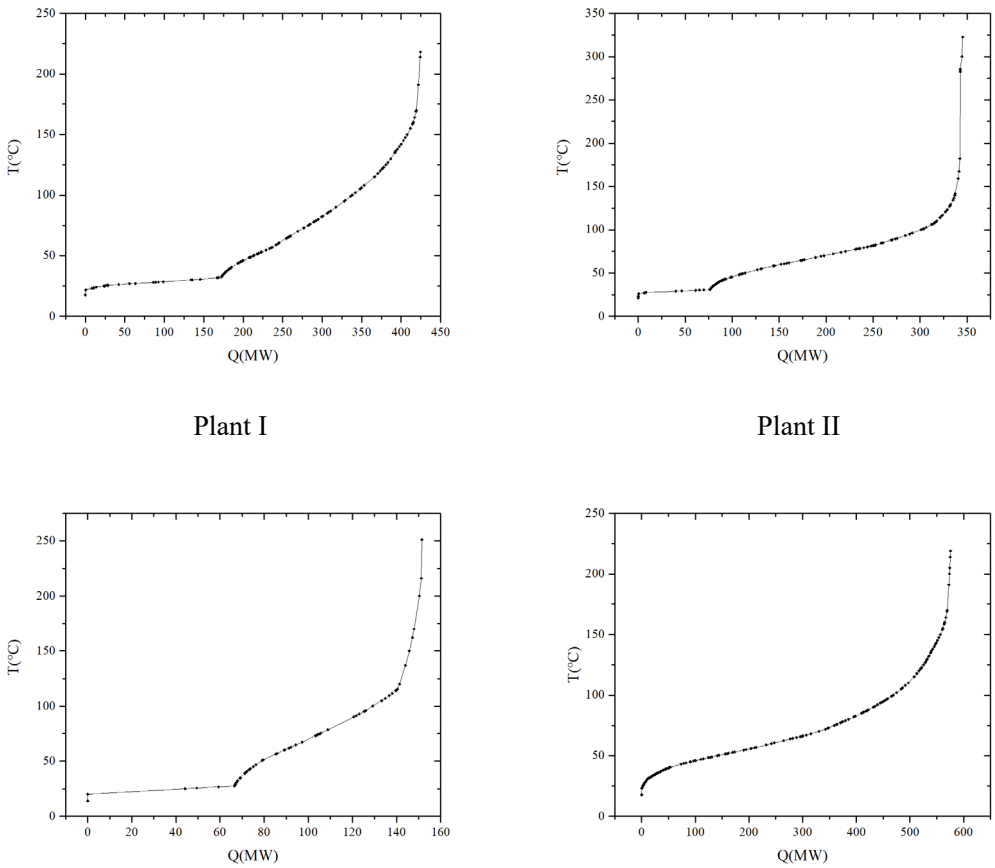


Figure 5. The waste heat flux compound curve of four plants

Industrial waste heat recovery methods has been extensively reported. There is a wide range of heat utilization technologies for the industrial waste heat recovery. Direct heat recovery via heat exchangers and economizers was widely investigated [10-11]. The heat pipe is an effective heat transfer device in waste heat recovery and

fuel consumption saving [12]. The recovery of low-temperature waste heat tends to be difficult and complicated, where the application of inverse refrigeration cycles can play an important role, including electric compression heat pumps [13], absorption heat pumps [14-15], and absorption heat transformers[16-18] for upgrading waste heat. In the presence of absorption heat exchangers, return water temperature of primary network can be reduced to 20℃ under the condition that supply water temperature of primary network is maintained at 120℃. Cascade heating technology can generate a lower return water temperature [19]. Low return water temperatures can facilitate the recovery of low-grade and middle-grade waste heat, which is very important for industry waste heat recovery. Options for heat to power conversion include organic Rankine cycle (ORC) [20] and Kalina cycle [21]. The Kalina cycle is preferably employed to treat relatively high-temperature waste heat, while the organic Rankine cycle can be driven by lower temperature waste heat [22]. The reviewed technologies are summarized in the Table 1.

Table 1 Typical technology applications in the recent literature

Heat recovery technologies	Research topic	Effect	Literature
heat exchanger	optimization of heat exchange area and configuration	fuel consumption saving	10,11
heat pipe	optimization of parameters and structure of heat pipe	energy efficiency improvement	12
electric heat pumps	appropriate heat pump integration strategy	operation cost saving	13
absorption heat pumps	absorption cycles used in waste heat recovery systems	high COP , return water elevation, suitable for low temperature heat sources	14,15
absorption heat	effect of absorption heat	COP and heat transfer	16-19

transformers	transformers in industrial waste heat recovery	efficiency increment,	
organic Rankine cycle	ORC integration and optimization	shaft-work output improvement, suitable for low temperature heat sources	20,22
Kalina cycle	Kalina cycle integration	efficient promote, suitable for higher temperature	21

Most previous studies focused on simple heat recovery processes with only a single waste heat source or several waste heat sources in parallel. Researchers proposed system configurations to recollect industrial waste heat, while there is still limited studies for global optimization. Fang [9] analyzed complex multi-heat source systems with multiple low-grade and middle-grade waste heat sources, which was based on a lower DH water return temperature. The application of heat pumps in terminals and thermal-substations can generate low DH water return temperature. In Fang's research, the tangency technology was introduced to figure out the optimal method to recollect heat from multiple waste heat sources. The first step was to draw a heat flux compound curve with integrated method on a T-Q diagram. The second step was to obtain the tangency point, which is the optimal case, by rotating the DH return water line around the inlet temperature point on the T-Q diagram. Tangency technology is similar to the pinch method, which was proposed as an effective method for optimizing the internal heat integration processes inside industrial factories [23, 24]. These methods analyzed the energy flow and optimized the heat exchange network in a straightforward graphic way.

However, these methods still have significant defects and limitations in

application. They are only effective in direct heat exchange processes, and invalid for processes involving work input and output. Secondly, the methods can only be applied to systems without heat conversion, where heat pumps are not employed [25]. Nevertheless, in order to effectively recover low-temperature waste heat for district heating, heat pumps must be used. In such cases, the tangency technology is not suitable. It is therefore necessary to develop an effective optimization method for the analysis of a complex multi-heat source system with heat pumps. The method should be convenient for use and can be expressed with diagrams taking into account the heat-work and heat-heat conversion systems.

The purpose of this paper is to discuss the key issues related to the optimization of the industrial waste heat recovery systems with the employment of heat pumps in the presence of multi-heat sources. Based on the investigation of tangency analysis limitations, the optimization principle is discussed. The optimization solutions in graphical expression are proposed to analyze complex multi-heat source systems with heat pumps along heat sources. Finally, corresponding case studies are introduced to shed light on the achievements of energy efficiency improvement.

2. Tangency analysis shortage discussion

2.1. Tangency analysis in cases with heat pumps

To clarify tangency analysis shortage, Figure 6 demonstrates a tangency analysis for multi-source process with direct heat exchanger and heat pumps along heat sources. The red dashed line demonstrates heat sources. Considering a 5℃ temperature difference in heat exchangers, the heat source line is moved down in the figure to get

the red solid line. The temperature difference of the red dashed and solid line is 5°C. In this case, DH return water temperature is fixed at 20°C, which is generally determined by the DH systems. The black solid line and black dashed lines demonstrate DH return water temperature in different heat exchange processes, where the DH return water lines are rotated pivoting at inlet temperature 20°C. Using tangency analysis, the optimal case is deduced shown as the black solid line, under the condition that is only with direct heat exchangers. The maximum temperature of supply water in this case is 53.3°C.

If DH return water temperature is further lifted in Figure 6, depicted as the blue solid line in the figure, there will be a triangle (the shaded area) consisting of heat source line and DH water line. Such case requires heat pumps to lift heat source temperature above DH return water. Different lines of DH return water mean different system configurations. Therefore, when heat pumps are introduced into a system, tangency analysis fails optimization. Tangency analysis cannot figure out an optimal solution in such cases.

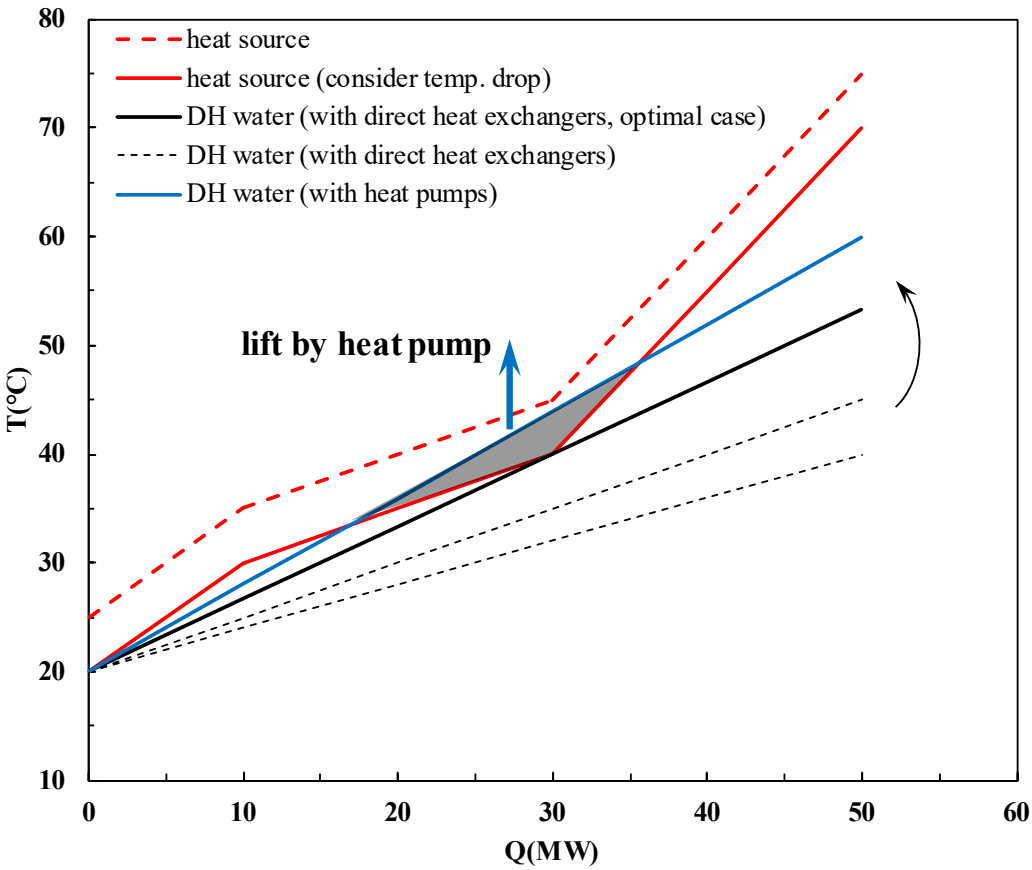


Figure 6. Tangency analysis

An example is introduced to further clarify this point and it will serve as a comparison between tangency analysis and optimization methods proposed in following sections. The application of the tangency analysis in a case with heat pumps is illustrated as follows, and its shortage in practice is discussed in Section 2.2. Four heat sources are presented for explanation in this case in Table 2, and case calculation conditions are listed in Table 3.

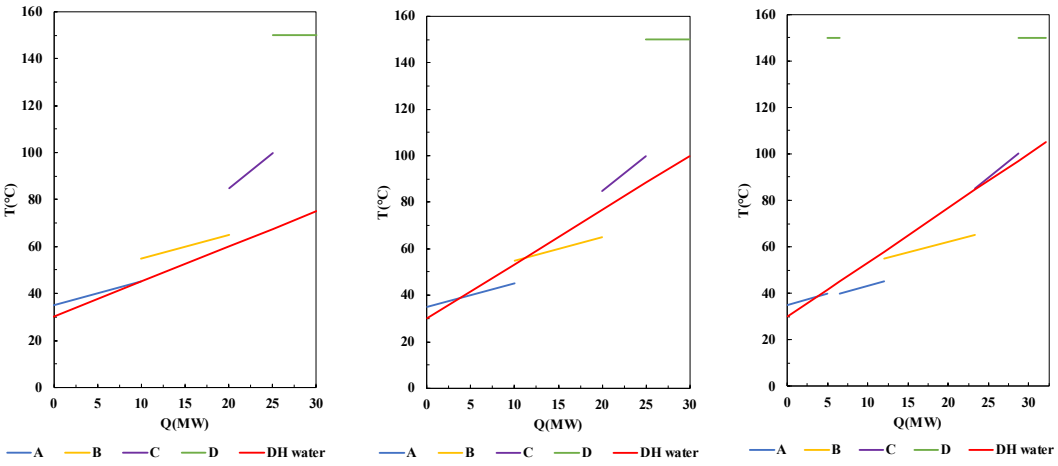
Table 2 Heat source list of the example using heat pumps

heat source	inlet temperature ℃	outlet temperature ℃	quantity MW
A	35	45	10
B	55	65	10
C	85	100	5
D (stream)	150	150	5

Table 3 Calculation conditions

Parameters	Values
Terminal temperature difference of heat exchangers	5℃
Return water temperature of DH network	30℃
Presupposed supply water temperature	100℃
AHP COP	0.7
EHP internal efficiency	0.8
Flowrate of return/supply water of DH network	367 t/h

Four heat sources are arranged and sorted by temperature level, which is the basic step of tangency analysis. If there are heat sources with overlapped temperature ranges, the integration is needed. Figure 7 depicts T-Q diagram of heat sources and DH water. Figure 7(a) shows the maximum case achieved by tangency analysis, where the supply water temperature is 74.0℃ lower than presupposed temperature 100℃. Figure 7(b) shows the case achieving presupposed supply water temperature. Supply water line lifts upward and part of heat source A and almost all of heat source B need the help of heat pumps. The temperature of heat source C is slightly higher than DH water. The temperature of heat source D is much higher than DH water, so it can drive an AHP to recover waste heat from lower temperature heat sources such as heat source A and B. Because heat source D capacity is not enough, electric heat pumps (EHPs) are used in the system.



(a) (b) (c)

Figure 7. T-Q diagram of heat source and DH water based on tangency analysis

Figure 7(c) shows the actual case involving heat pumps. Because of the heat unbalance of absorber and evaporator, part of heat source D is inserted between heat source A and B. Since COP of the AHP is set as 0.7, supply water receives about 0.7 Q_D in the AHP condenser and Q_D in the AHP absorber, where Q_D denotes the heat power of heat source D. Waste heat recovery system with detail calculation results are showed in Figure 8 based on Figure 7(c). Figure 8 is the flow chart of the waste heat recovery process of Figure 7(c). However, the usage of EHP consumes electricity and brings the additional energy input, so the supply water temperature will exceed presupposed supply water temperature 100°C rising to 105.2°C. The total recovered waste heat is 30MW and consumes 2.2MW electricity.

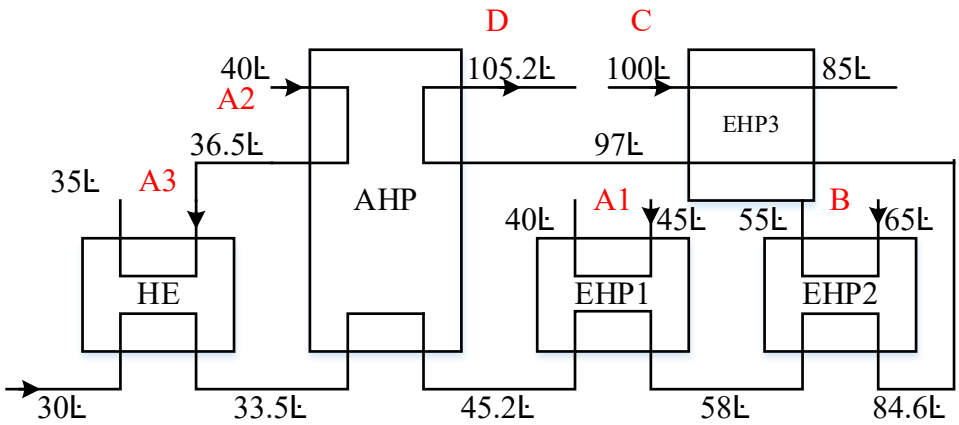


Figure 8. Flow chart of waste heat recovery based on tangency analysis

In an AHP, the generator and condenser can be regarded as a pair of heat-transfer group, while the absorber and evaporator the other pair of heat-transfer group. Therefore, AHPs can be regarded as two heat-transfer groups depicted in Figure 9 and the flow chart in Figure 8 can be simplified into a more straightforward one (Figure 10). It helps us to better understand the heat transfer process.

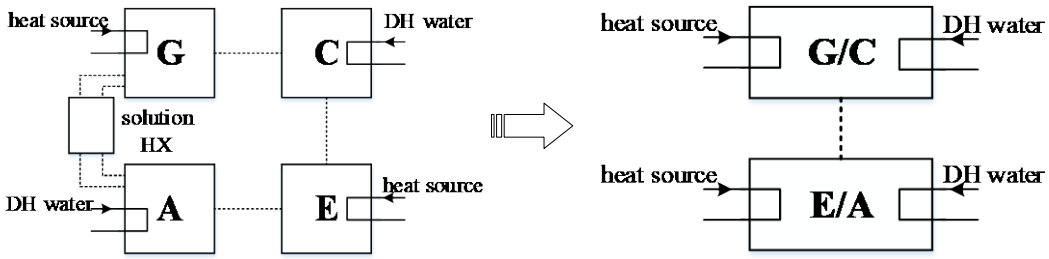


Figure 9. Flow chart of an AHP and its simplified representation
(G: generator; C: condenser; A: absorber; E: evaporator)

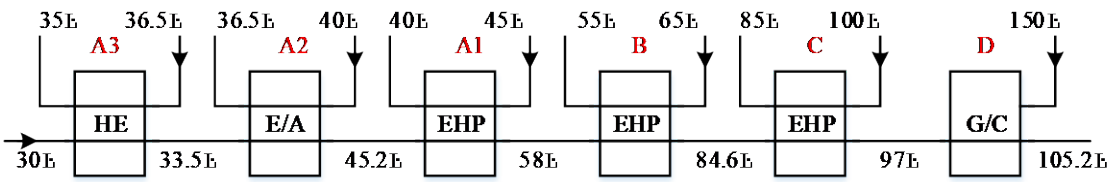


Figure 10. Simplified flow chart of waste heat recovery based on tangency analysis

2.2. Limitations of tangency analysis

In Section 2.1, we can see that if heat source temperature is at low level, EHPs will

be used along the heat sources to supply additional exergy to a system based on tangency analysis. For EHPs, additional exergy supply means additional power consumption. However, there will be extra exergy supply than system demand if EHPs are involved in a system.

As shown in Figure 11, heat source B can be recovered directly while heat source A needs EHP assistance. The theoretical exergy demand of heat source A is showed in Figure 11(a). The shaded area represents the exergy demand. It contains the temperature gap between heat source and DH water and the temperature drop in the heat exchanger. When an EHP is used in the system, there will be additional heat transfer between heat source and evaporator occurs, and this heat transfer needs additional exergy supply. The shaded area in Figure 11(b) shows the minimum additional exergy supply due to temperature difference of heat exchange, and in this illustrative example the terminal temperature difference in the evaporator is set as value in Table 3. In actual cases, because phase-change heat transfer occurs in the condenser and evaporator, the temperature of the refrigerant in the condenser and evaporator is constant. This actual exergy supply is shown as the shaded area in Figure 11(c), which is far more than the theoretical exergy demand (Figure 11(a)). Besides, the shaded area in Figure 11(c) can be divided into the shaded area shown in Figure 11(b) and two additional triangles from condenser side and evaporator side, respectively. These two triangles result from the constant temperature levels in the condenser and evaporator. These additional triangles can be reduced by setting some EHPs in series (Figure 11(d)). If the number of the EHPs in series tends to be infinity, the two additional triangles will

disappear.

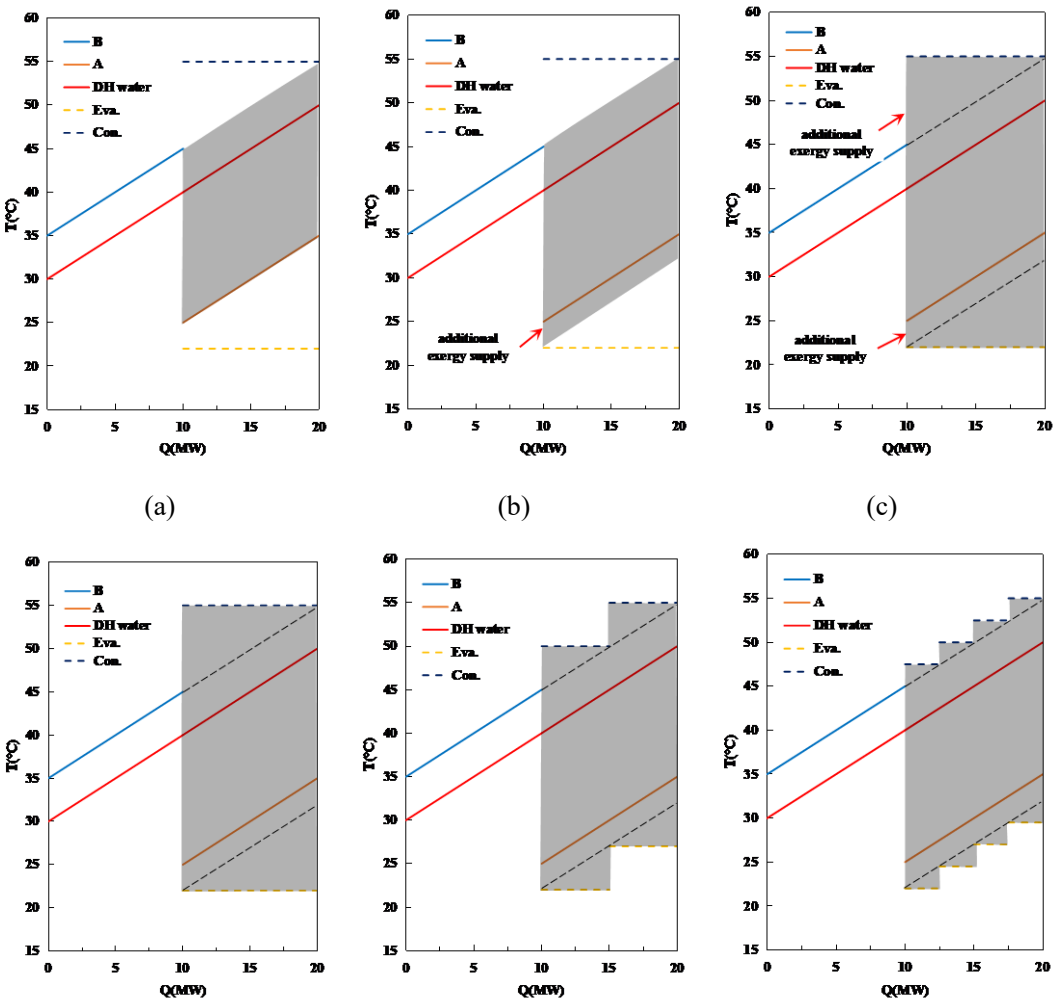


Figure 11. Exergy demand and supply in systems

(a) theoretical exergy demand

(b) exergy supply (heat transfer temperature difference is considered)

(c) exergy supply (detailed thermal property of a EHP is considered)

(d) additional triangles can be reduced by setting some EHPs in series

The exergy supplied by EHPs is more than theoretical exergy demanded in the system in application. There will be additional exergy generation in heat transfer process when heat pumps are integrated into districted heating systems installed in factories, thereby consuming more energy. However, tangency analysis fails to take this point into account. The tangency analysis is developed from the perspective of resources integration. The configuration designed according to tangency analysis tends

to require more exergy supply, and is not the optimal case with lower energy consumption. The optimized method discussed in the following sections is developed with consideration of additional exergy supply during the heat exchange process with heat pumps.

3. Optimization methods of process with heat pumps

3.1 Optimization principle

When heat pumps are used in the system, there will be additional exergy supply. In order to reduce power consumption of the system, designers should take efforts to reduce the additional exergy supply. If EHPs in series are considered in the system, the main additional exergy supply is caused by the extra heat transfer. Therefore, reducing extra heat transfer is the key point in the additional exergy supply reduction, and reducing additional exergy supply can reduce total energy consumption. Considering these, mean heat transfer times N is introduced in the analysis, which is described by Equation (1). Smaller N means less heat transferred and less exergy loss, and means less additional exergy supply.

$$N = \frac{\sum Q_i}{Q_{total}} \tag{1}$$

In which, Q_i is the heat power transferred in each heat transfer process; Q_{total} is the total heat received by the heat sink in the system.

N is heat exchange times for heat transferring from source to sink. For plate heat exchangers, heat directly transfer from source to sink by heat exchangers, N=1. For EHPs, the heat transfer process is source-refrigerant-sink, N=2. Similarly for AHP, the

heat transfer process is source-LiBr-sink, N=2. At present, most of waste heat recovery systems are composed by heat exchangers, EHPs and AHPs. According to the analysis above, mean heat transfer times of waste heat recovery systems can be described as Equation (2)

$$N = \frac{\sum Q_x + 2\sum Q_e + 2\sum Q_a}{Q_{total}} = 1 + \frac{\sum Q_e + \sum Q_a}{Q_{total}} \tag{2}$$

In which, Q_x is the heat transferred by heat exchangers; Q_e is the heat transferred by EHPs; Q_a is the heat transferred by AHPs; Q_{total} is the total heat received by the heat sink in the system.

According to Equation (2), the weight of the heat transferred by heat pumps is higher, so if we want to reduce mean heat transfer times and to reduce additional exergy input, Q_e and Q_a should be reduced. On the other hand, $\sum Q_e + \sum Q_a$ is the total cooling capacity of the system. For a specific project, if the heat source and the temperature of supply/return water are given, then the exergy demand is fixed. If we want to reduce the cooling capacity of the system, EHPs should offer the same exergy with less cooling capacity. Thus, the less cooling capacity means the bigger temperature difference lifted by heat pumps, and it means the COP of EHPs will be lower. From another angle, COP can be described as the ratio of cooling capacity and the exergy offered by EHPs, so less cooling capacity means lower COP. It is a quite different conclusion from existing designs. Designers should take effort to reduce the cooling capacity, not to improve the COP of EHPs. As a result, in industrial waste heat recovery, lower cooling capacity, bigger temperature difference and lower COP means lower power consumption.

Therefore, to reduce the mean heat transfer times N is the key point of heat recovery process design, and it is the basic principle of process optimization. However, the existing waste heat recovery process design mentioned above does not consider from this perspective, and it cannot reduce cooling capacity effectively. In the following Section 3.2, two novel methods of reducing cooling capacity and illustrative examples will be introduced, which are based on the basic principle mentioned above and different from the traditional tangency technology.

3.2 Optimization method I

According to the tangency analysis steps above, heat sources are sorted by inlet temperature (Figure 6). The order of heat sources is based on the integration method, and it can present heat sources condition clearly. However, if heat pumps are used in the system, it may not be the best order, so that the order should be adjusted. As shown in Figure 12, it is based on the case shown in Figure 6. If the heat source A1 is moved to the left (A2), this part of heat can be recovered directly by heat exchangers. Therefore, in a T-Q diagram, if the heat source is at a lower temperature level, it should be moved to the left at first (Figure 12), instead of being moved upward by heat pumps (Figure 6).

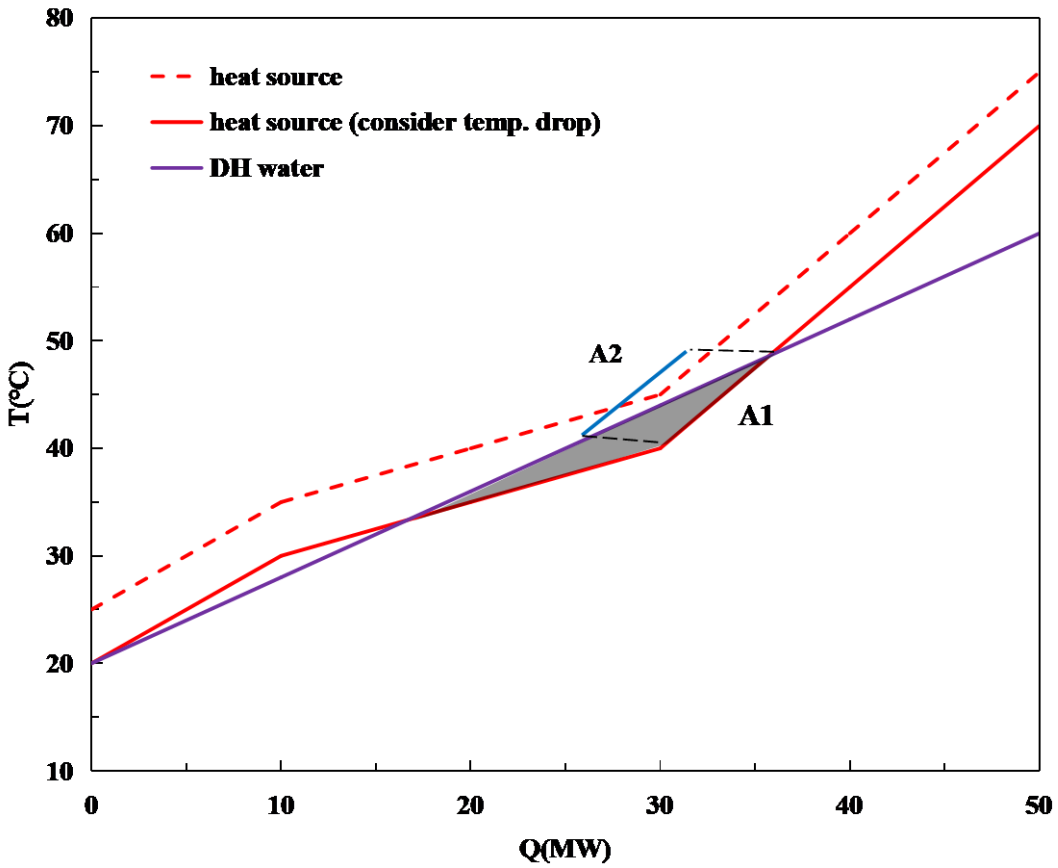
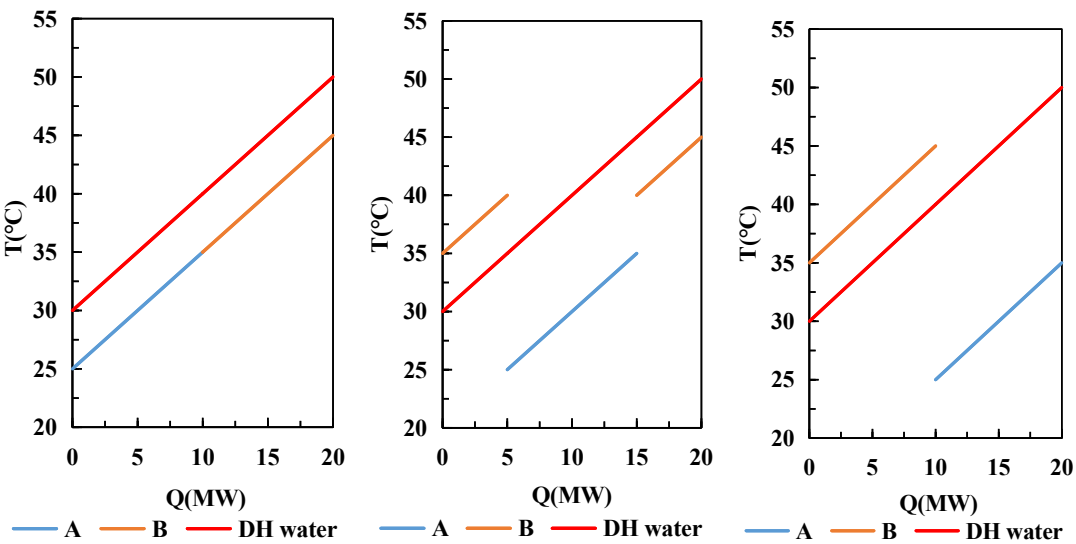


Figure 12. Order adjustment of heat sources

A basic model is analyzed to introduce the thought 'move to the left' by a graphic expression. As shown in Figure 13(a), there are two heat sources in the system. Inlet temperature of heat source A is 35℃ and outlet temperature is 25℃. Inlet temperature of heat source B is 45℃ and outlet temperature is 35℃. Inlet temperature of DH water is 30℃ and outlet temperature is 50℃. The minimal terminal temperature difference of heat exchanger is 5℃. At first, both heat source A and B need heat pumps to shift temperature. Therefore, the cooling capacity is 20MW (Figure 13(a)). Then part of heat from heat source B is moved to the left, and this part of heat can be recovered directly by heat exchangers. Therefore, with the increase of the moved heat power, the cooling capacity reduces (Figure 13(b)). When all of heat source B are moved to the left, the cooling capacity reaches the minimum 10MW

(Figure 13(c)). Cooling capacity decreases with moved heat power. The corresponding additional exergy can be calculated.

Figure 14 demonstrates the additional exergy needed varying with cooling capacity corresponding to Figure 13. In Figure 14, the blue line means the exergy difference of heat source and DH water in heat process, i.e. the exergy demand of system. The orange line takes temperature drop in evaporators into consideration (Figure 11(b)), and the grey line takes two additional triangles exergy losses into consideration, which implies actual case (Figure 11(c)). If only temperature drop in evaporators is taken into consideration, exergy supply will increase gradually with the cooling capacity. However, when all factors are included, exergy supply will increase significantly with the cooling capacity, representing by grey line.



(a) (b) (c)

Figure 13. The process of moving heat source B to the left
(a) before moving; (b) half moved; (c) all moved

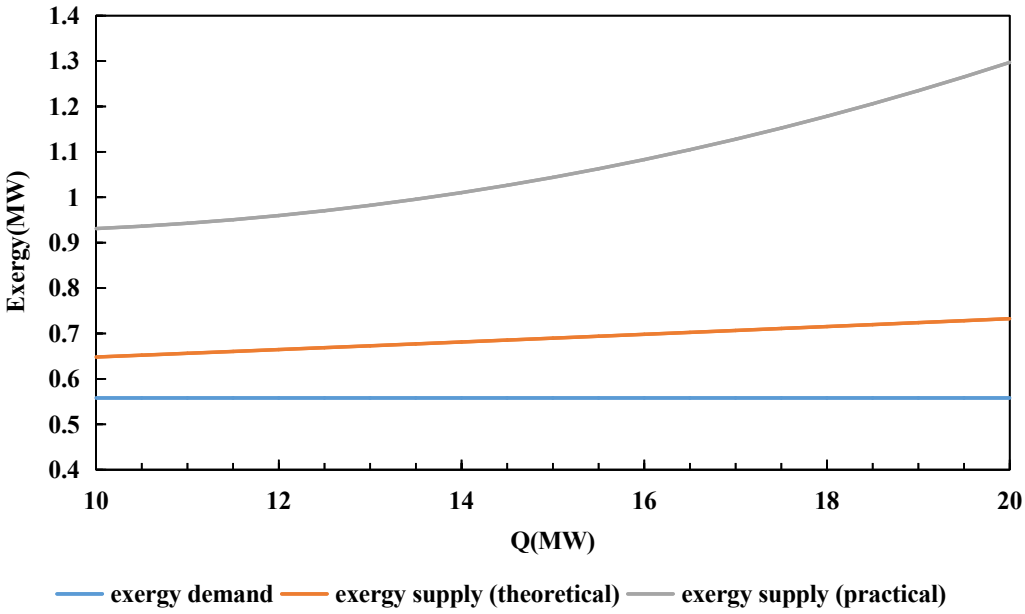


Figure 14. Additional exergy varies with cooling capacity

Based on the analysis above, take case in Section 2 as an example. The T-Q diagram can be redrawn as Figure 15. In this new system design, heat source A is divided into 4 parts, and heat source B is divided into 3 parts. The order of these parts are adjusted base on DH water temperature seriation. The calculation conditions are same as Table 3. Flow chart is shown in Figure 16.

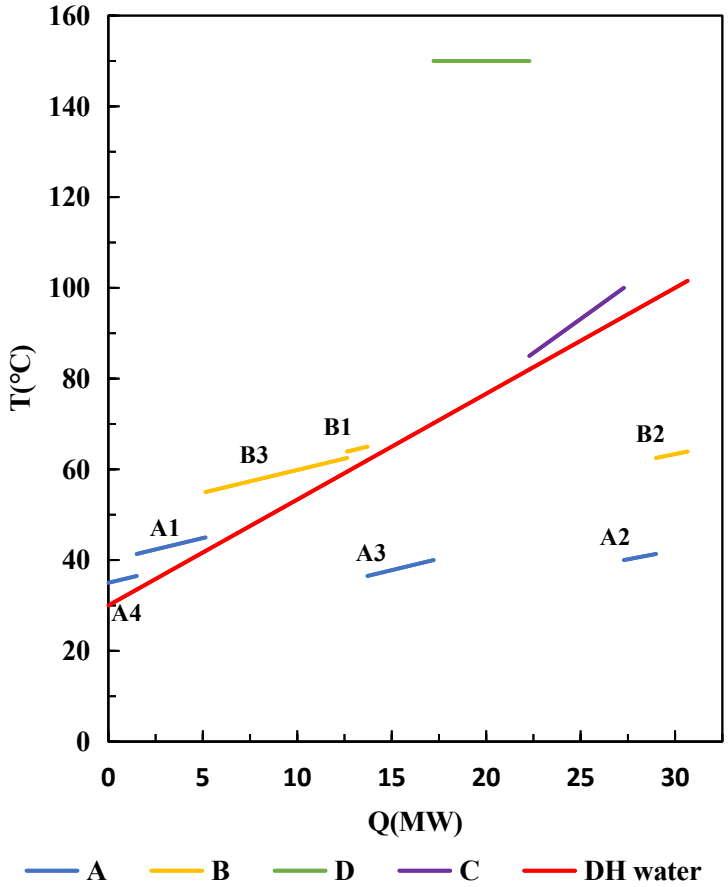


Figure 15. T-Q diagram based on the method I

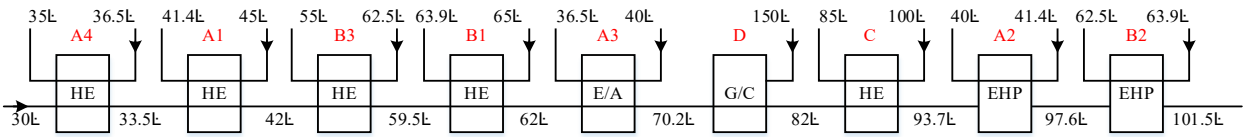


Figure 16. Simplified flow chart based on method I

This new system configuration recover as much waste as the one in Section 2, i.e. 30MW. The cooling capacities of this new and former system are 2.8MW and 20MW. The mean COPs (total cooling capacity/total electric consumption) of this new and former system are 4.7 and 9.3. The electric power consumed is 0.59MW, far less than that of former, which is 2.1MW. It is a decrease of 72%. These results totally align with the analysis.

3.3 Optimization method II

Different from the method I, another method can also reduce cooling capacity of heat recovery systems (Figure 17). The integration method of heat sources is introduced in Figure 4 and vice versa, one heat source can be divided into two sources. In Figure 17, heat source A is divided into A1 and A2. A1 can be recovered by heat exchangers and only A2 is recovered by heat pumps. Using this method, cooling capacity can be reduced.

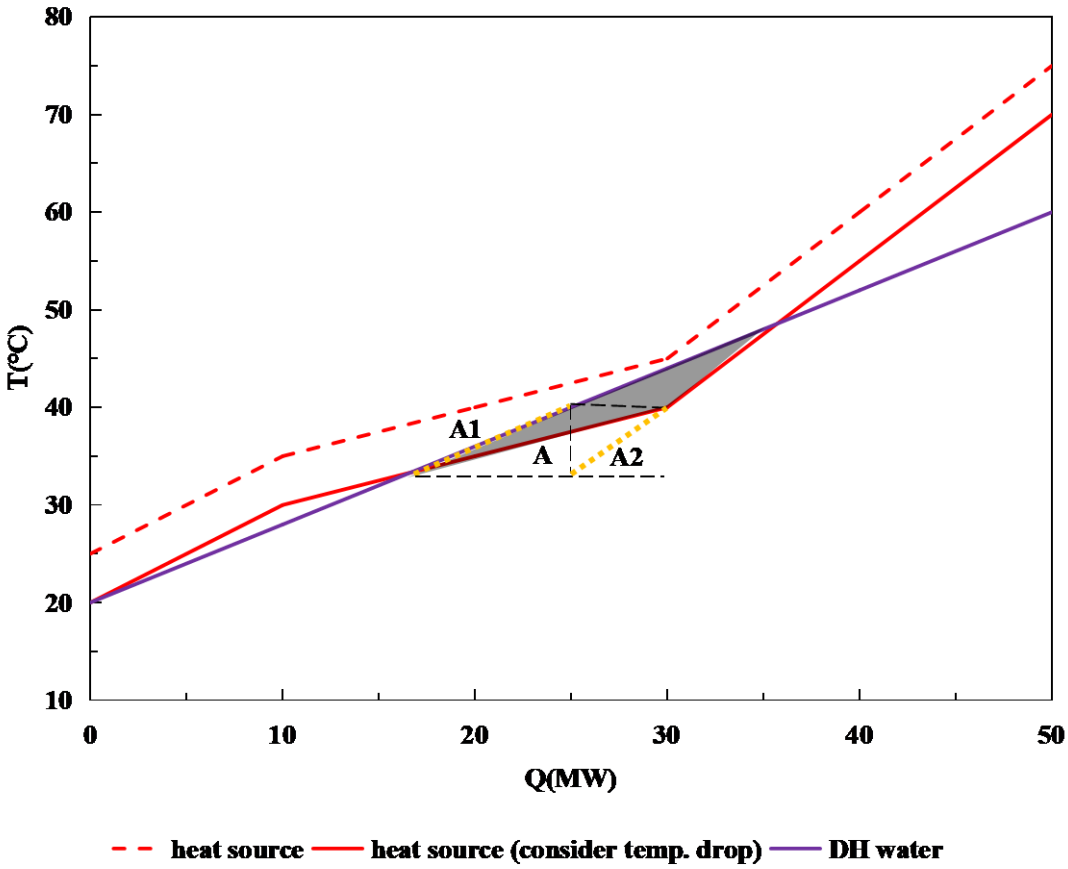


Figure 17. Disintegration of heat sources

A typical example is analyzed to introduce the thought of disintegration. As shown in Figure 18 and Figure 19(a), inlet temperature of heat source A is 55℃ and outlet temperature is 40℃. Inlet temperature of DH water is 30℃ and outlet temperature is 70℃. The minimal terminal temperature difference of heat exchanger

is 5℃.

Heat source A is disintegrated into A1 and A2. According to the optimization principle, the cool capacity of heat pumps should be minimized. In this system, when a small quantity of the heat is allocated to A2 (Figure 19(b)), EHP1 still needs to run. The quantity of heat transferred by heat exchangers is increased, which means the cooling capacity is decreased. When more heat is allocated to A2 and the terminal temperature difference of A1 and DH water reaches minimum of 5℃ EHP1 is not needed. In this situation, the heat transferred by heat exchangers reaches the maximum and this is the optimal process (Figure 19(c)). If more heat continue to be allocated to A2 afterwards, the heat transferred by heat exchangers decreases and cool capacity of heat pumps increases (Figure 19(d)).

Figure 20 demonstrates the theoretical and practical exergy supply of the example. Theoretical exergy demand almost remains the same as heat allocation changes, while practical exergy supply reaches the minimum at around 20MW, where cooling capacity reaches the minimum at 20MW. Detailed exergy curves of A1 and A2 are shown in Figure 21.

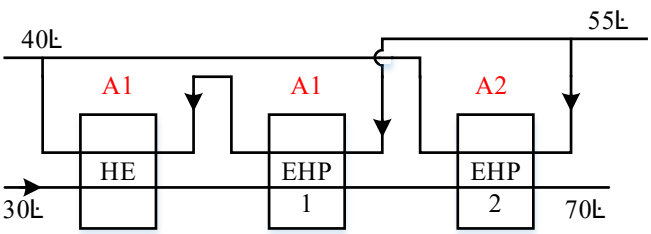


Figure 18. A simple example of method II

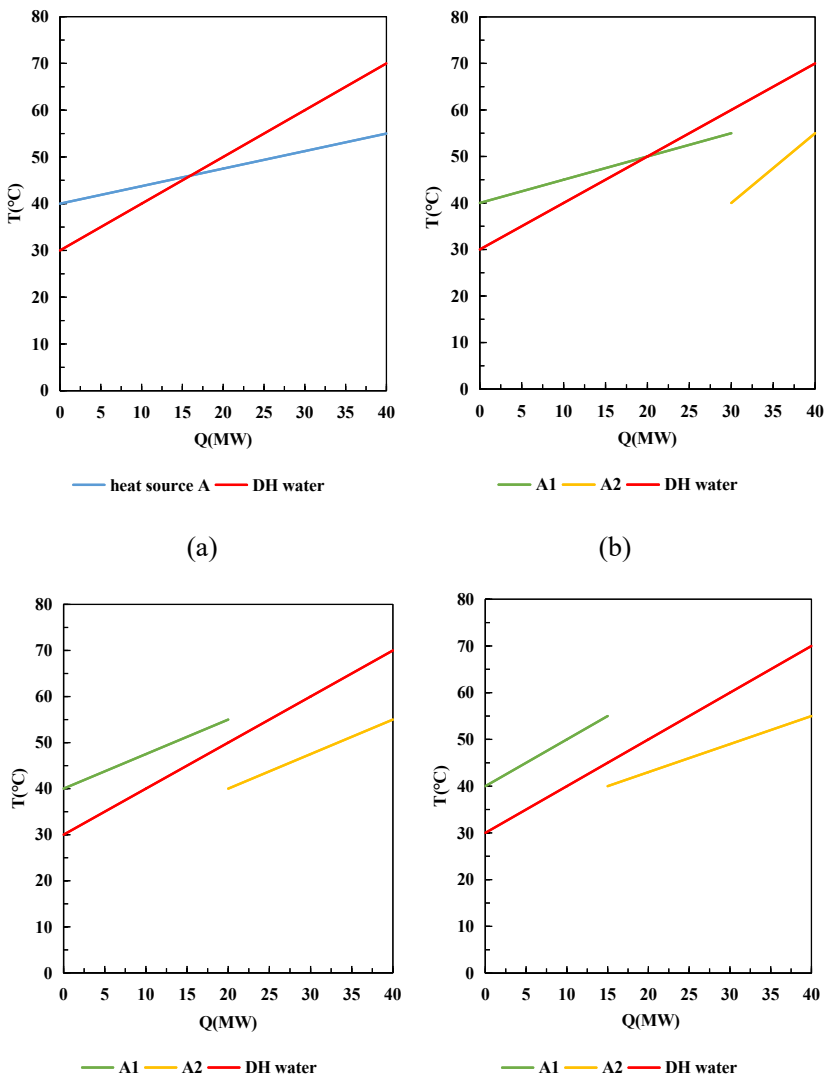


Figure 19. T-Q diagram of allocated heat to A2

(a) no heat is allocated to A2; (b) 10MW is allocated to A2;

(c) A1 can be recovered by HEs directly; (d) A1 is parallel to DH water in the figure

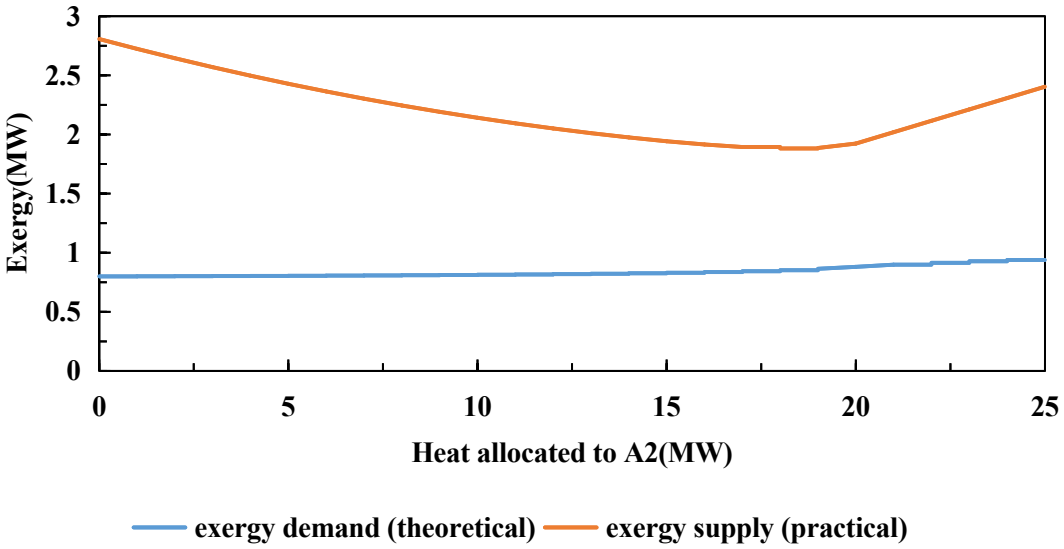
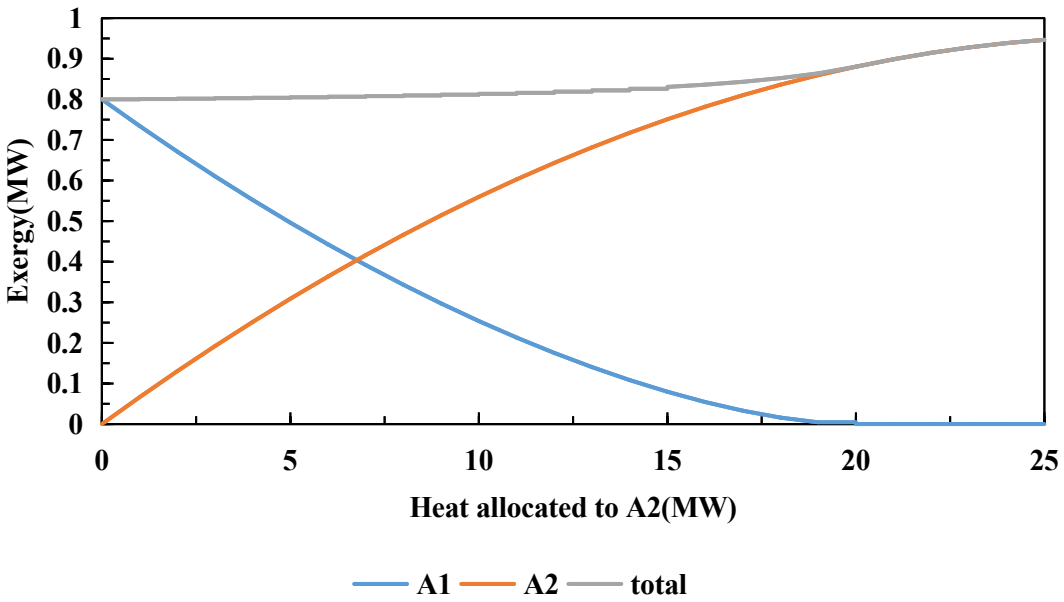
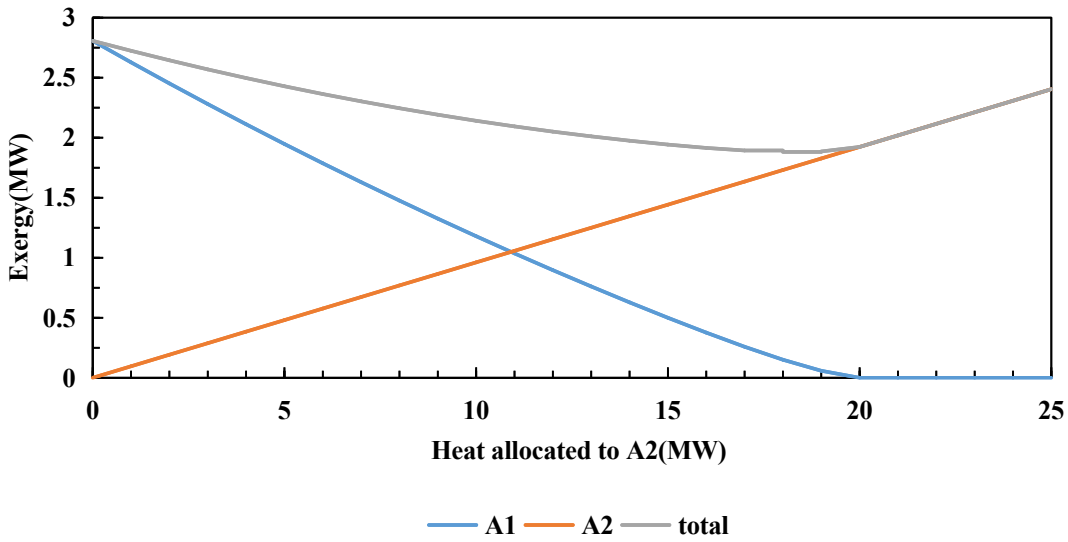


Figure 20. Theoretical and practical exergy supply of the example



(a)



(b)

Figure 21. Detail exergy demand and supply analysis

(a) theoretical exergy demand; (b) practical exergy supply;

The advantages of method II can be illustrated in the following application case. There are three heat sources in the case (Table 4). The calculation conditions are listed in Table 5.

Table 4. Heat sources list of the example

heat source	inlet temperature ℃	outlet temperature ℃	quantity MW
A	31.9	56.6	14.38637
B	85	127.02	4.591101
C	136.39	170	7.5

Table 5. Calculation conditions

Parameters	Values
Terminal temperature difference of heat exchangers	5℃
Return water temperature of DH network	20℃
Presupposed supply water temperature	100℃
AHP COP	0.7
Flowrate of return/supply water of DH network	283 t/h

Figure 22(a) depicts heat sources and DH supply water on T-Q diagram. The temperature of heat source C is much higher than DH water, so it can drive AHPs to lift

lower temperature heat sources A. Therefore, part of heat source A is lifted by AHPs, and another part is lifted by EHPs. Figure 22(b) shows the tangency analysis. In this system, heat source A is divided into three parts. The first part can be recovered directly by heat exchangers, the second part is recovered by AHPs driven by heat source C, and the last part is recovered by EHPs. Simplified flow chart is shown in Figure 23. The calculation conditions are listed in Table 5. The analytical results are that 26.9MW waste heat is recovered, cooling capacity is 7.0MW and electric power is 0.41MW.

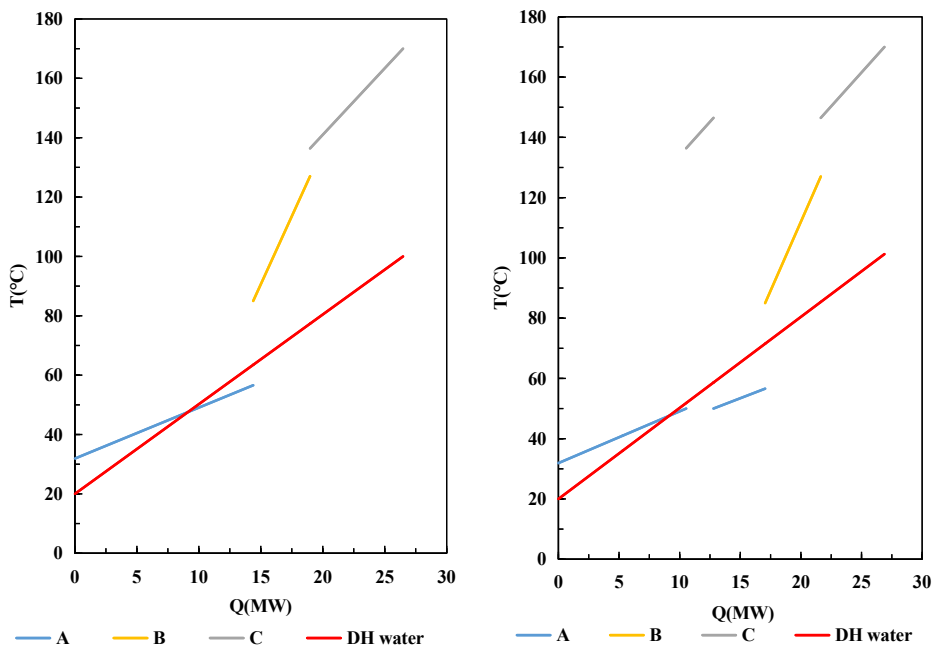


Figure 22. T-Q diagram of application example with traditional design

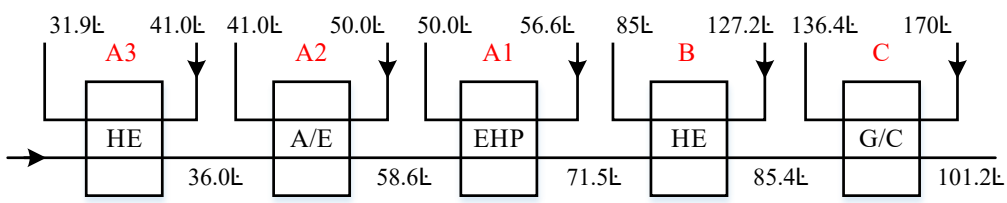


Figure 23. Simplified flow chart of the application example with traditional design

With method II, part of heat source A can be divided into two parallel parts (Figure 24). In this system, heat source A is divided into three parts. The first two parts can be recovered directly by heat exchangers, and the last part is recovered by AHPs. The second part is parallel to the DH water, in this situation, the heat capacity of heat exchangers reaches the maximum. Simplified flow chart is shown in Figure 25. The calculation conditions are the same as Table 5. The amount of recovered waste heat power is the same as 26.9MW. It is estimated that cooling capacity is 3.9MW and there is no need for EHPs. Those are a 100% decrease of electricity and a 44% decrease of cooling capacity.

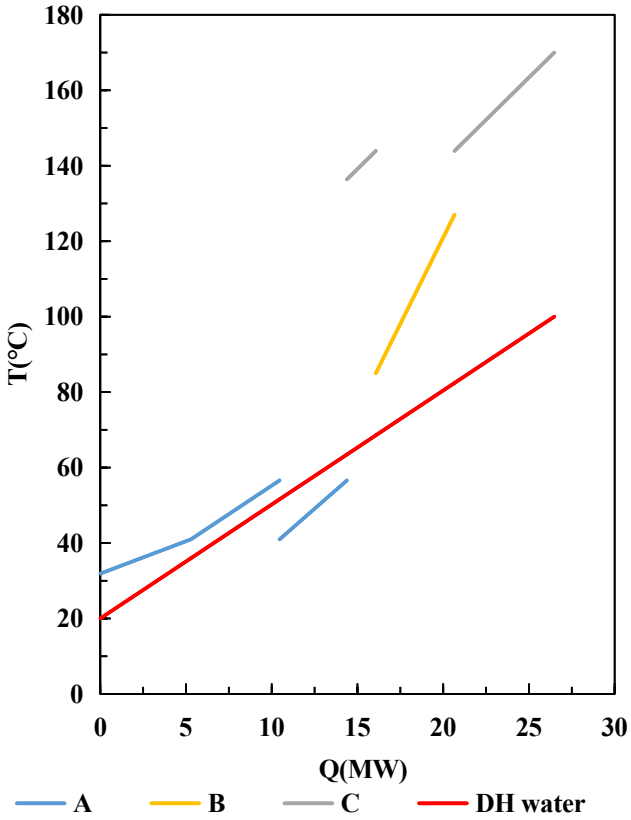


Figure 24. T-Q diagram of the application example with method II

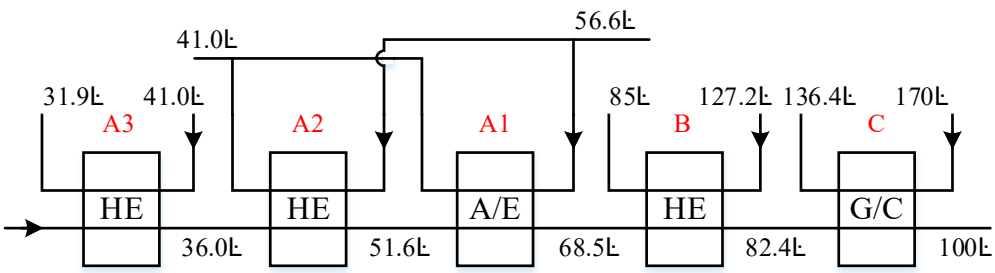


Figure 25. Simplified flow chart of the application example with method II

4. Conclusions

This study introduces optimization methods for complex multi-heat source systems with heat pumps along heat sources. The methods can be used to effectively recover low-temperature and middle-temperature industrial waste heat, and to remarkably elevate the energy efficiency inside industrial and district heating sectors. Based on the analysis presented above, the following conclusions can be drawn as follows:

- Tangency analysis was developed from the mere perspective of resources integration, and the additional exergy generation by using heat pumps is not involved for consideration. As a result, the tangency technology is effective only for direct heat exchange systems in multi-heat source systems. It is not ideal for systems in the presence of heat pumps.
- When industrial waste heat sources struggle to provide enough exergy, heat pumps will be needed in the system. For a system, because of the exergy loss during heat transfer, the exergy provided by heat pumps is greater than the systematic theoretical exergy demand. In order to evaluate the exergy loss, mean heat transfer times N is proposed and served as an optimization index.
- From the application of the mean heat transfer times N in the analysis, it is found

that the lower the cooling capacity of heat pumps, the lower the exergy loss, thereby the less electrical energy consumption. The optimization principle of complex multi-heat source systems with heat pumps is to decrease cooling capacities of heat pumps.

- Two novel methods to reduce cooling capacity are proposed for process optimization in this article. These methods can be presented in a diagrammatic way. Corresponding case studies for these two methods showed the effectiveness in reducing exergy loss and energy consumption in the system. Compared to the system using tangency analysis, the new system using method I significantly reduced the electricity input by up to 72%. In the other case, it is shown that EHPs can be avoided, and electricity declined by 100% in the new system using method II.

This study figures out the optimization solutions for complex multi-heat source systems. However, there are apparently other factors requiring consideration for real practices, such as economic and investment factors. Thus, a multi-objective optimization taking into account the economic aspect is necessary after the technical and exergy optimization process.

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