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Published in:
Energy

DOI:
10.1016/j.energy.2022.123846

Published: 01/07/2022

Document Version
Peer-reviewed accepted author manuscript, also known as Final accepted manuscript or Post-print

Heat-power decoupling and energy saving of the CHP unit with heat pump based waste heat recovery system

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Abstract

Combined heat and power (CHP) plants are operating under more fluctuating working conditions due to the increasingly inconsistent demands for heat and power and integration of renewable energy. This paper proposes to use an electric heat pump (EHP) to decouple heat and power and save energy by recovering waste heat from the cooling water. The thermodynamic model of the CHP unit under dynamic working conditions is established, and the dynamic EHP model based on an efficiency factor is proposed. The heat-power decoupling and energy-saving potential with different heat and power outputs and the heat pump DH ratio (χ_{HP}) are analyzed for a CHP unit as a case study. Absorption heat pump (AHP) and EHP-based waste heat recovery systems are also compared. The results indicate that the heat-power decoupling potential is bigger when χ_{HP} and the heat demand are increasing. The energy-saving effect is clearer by increasing the coefficient of performance (COP), χ_{HP}, or both. AHP and EHP can help the system obtain a certain level of heat-power decoupling and energy-saving effects, but these effects of the AHP-based system are smaller than that of EHP, especially under the working conditions of high heat demand and low power demand.

Keywords: Electric heat pump (EHP), Absorption heat pump (AHP), Combined heat and power (CHP), Waste heat recovery, heat-power decoupling, Energy saving.

1. Introduction

Combined heat and power (CHP) is the predominant fuel-based energy production technology all over the world. It is characterized by the highest overall energy efficiency due to the cogeneration

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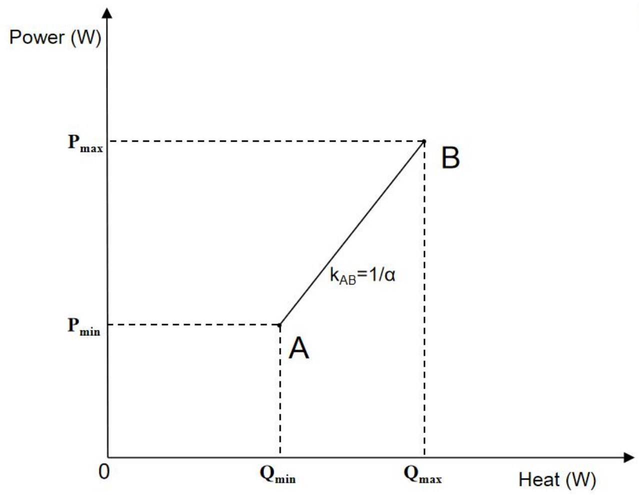
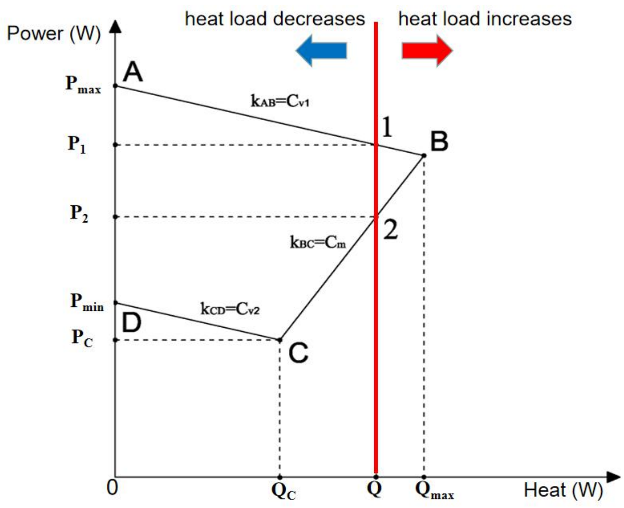
of heat and power. Typically, a CHP unit can operate with an overall efficiency of 65~85%, depending on the scale and technologies used [1], e.g., steam turbine, gas turbine, combined cycle, etc. The installed CHP capacity for power generation is expected to reach 551.7 GW by 2021, with a compound annual growth rate (CAGR) of 3.8% between 2020 and 2025 [2]. Natural gas and coal are the widely used fuels for CHP production. In the near future, CHP is still expected to play an indispensable role in the energy sector. However, the cogeneration of heat and power usually leads to unwanted thermo-electric coupling, which means that the heat and power outputs of a CHP unit are highly coupled with each other, and it is difficult to satisfy both heat and power demands at the same time. In fact, the load profiles of heat and power are with different distributions and sometimes can be contradictory.

Figure 1 shows the typical operating curves and thermo-electric coupling relationships of the backpressure unit and the extraction condensing steam turbine unit. It is clear that the heat and power outputs of the backpressure unit have a linear relationship, and the slope of the AB line segment is equal to the reciprocal of the heat-to-electricity ratio (α). If the backpressure CHP provides 1 unit of heat, it is forced to offer $1/\alpha$ units of power. And if there is no heat demand, the backpressure CHP unit has to be shut down. Therefore, the heat and power production are fully coupled in a backpressure unit, indicating that it doesn't have any peak-shaving capacity. But it has the highest theoretical energy efficiency because of zero condensing loss, and thus it is widely used as the baseload plant.



(a) backpressure unit



(b) extraction condensing steam turbine unit

Figure 1. Typical operating curves and thermo-electric coupling properties.

The thermo-electric coupling relationship for the extraction condensing steam turbine unit is more complicated, as shown in Fig. 1(b). The operating region is not a line segment but an area surrounded by line segments ABCD. AB and DC denote the thermo-electric relationships under the working conditions of the rated and minimum steam intake, respectively. C_{v1} and C_{v2} are the slopes of line segments AB and DC, meaning that if 1 unit of the heat is provided, the power generation will be reduced by C_v units. AD denotes the condensing working conditions without heat output, while CB denotes the backpressure working conditions; and if 1 unit of heat is provided, then it is forced to provide C_m units of the power. Moreover, the feasible operating region of the extraction condensing steam turbine unit includes all working conditions inside the surrounded area. Therefore, this kind of CHP unit has a peak-shaving capability to some extent, depending on the heat output. For example, when the heat output is Q, the peak-shaving capacity for power can be described by [P_2, P_1], which means that the power output can be adjusted in this range. Under design heat load Q', the peak-shaving capacity is degraded to zero, and the peak-shaving capacity is gradually growing while the heat load is decreased until Q=Q_C.

CHP plants are usually operated under the "heat-led" mode, which prioritizes the heat demand, but the power generation and the overall energy efficiency could be reduced due to the thermo-electric coupling, especially with the increasing use of highly fluctuating renewable energy source (RES). Therefore, it is necessary to use some kind of peak-shaving technology to relax the coupling property between heat and power, which can be called thermo-electric decoupling or heat-power decoupling. Currently, the most widely used peak-shaving technology in a CHP plant can be divided into three groups as following.

The first type is called process retrofit: for example, the bypass pressure reducer and attemperator can adjust the heat output to some extent by reducing the steam parameters directly without any power output [3]. Low-pressure turbine (LPT) renovation was also used in some CHP plants, and it was found this could reduce the minimum power output efficiently [4]. Cao et al. [5] reported that the minimum electric power output of traditional CHP units could be decreased with a high pressure-low-pressure bypass under any heating load. Liu et al. [6] proposed to use steam ejectors to achieve certain level of heat-power decoupling, and they also optimized the configuration of the ejectors using thermodynamic models.

The second type is defined as auxiliary heat source (AHS) mode [7,8], including the use of thermal energy storage (TES), an absorption heat pump (AHP) and other heat sources. TES systems have been increasingly designed and operated [9], especially in CHP-based district heating (DH) systems to obtain more operational flexibility and higher renewable energy integration [10–14]. It can be used to match the fluctuating heat demand by charging the excess heat output to the TES tank or discharging the heat to the district heating network (DHN) while maintaining the required power output at high efficiency. AHP is usually used to recover the waste heat from condensing water [15–18], flue gas [19–23], or both [24], in order to increase the heat output and energy efficiency of a CHP plant. The driving source of the AHP is part of the extraction steam intended for heating, and the lower grade heat source is the condensing water after the condenser or the circulating water in the flue gas scrubber [25]. Heat and power can be decoupled to some extent because the consumption of extraction steam in the AHP will lead to more heat output at the cost of less power output. Therefore, the extraction steam used for AHP can be controlled according to the simultaneous fluctuations of the heat and power demands. Ding et al. [26] proposed a solar aided (coal-fired) combined heat and power (SACHP) system, where solar energy could be flexibly used to generate power or to supply heat according to the heating and power demands to provide a certain level of heat-power decoupling capability. They also optimized the operating strategies of the SACHP system according to different heat and power demands [27].

The third type is defined as the power to heat (P2H) mode [8,28], including the use of an electric boiler (EB) [8,29] and an electric heat pump (EHP) [7,30,31]. EB is a popular P2H technology for simultaneous peak shaving of power and heat production. It can also help increase the integration of a renewable energy source [7,8,32] and provide good flexibility for the CHP operation. Although the efficiency of EB can be very high (e.g., 99% [7]), the energy utilization is not reasonable because it just converts high-grade power to low-grade heat with an efficiency of less than 1, not to mention it cannot recover any waste heat that is abundant in the cogeneration process. Some researchers proposed to install small-scale EHPs on the user side, e.g., in heating substations [33] or near buildings [34, 35, 36]. These applications can only change the power-to-heat ratio of the heat demand profile [29] but have little effect on the heat-power decoupling or waste heat recovery in the CHP plant. Fu et al. [37] envisaged a low carbon DH paradigm in the near future in China (2025), and they highlighted the heat-power decoupling using heat pump (HP) and TES together. Liu et al. [38] compared the EB and HP as P2H devices, they found that both of them were capable to enhance the operational flexibility of coal-fired CHP plants and HP is a bit more efficient than EB in reducing the carbon emission and operating cost. From this point of view, large-scale heat pumps are more favorable as they can efficiently reduce the production of excess electricity [34] and provide much more heat with a coefficient of performance (COP) that is much larger than 1 [29].

In addition, waste heat can be seen as clean energy for DH, and waste heat recovery is one of the most efficient energy-saving technologies in a CHP plant [29]. Li and Song [39] studied the economy, energy conservation, and environmental benefits of an AHP-based waster heat recovery system in a 600W supercritical CHP unit. The payback time of the AHP, energy consumption, and CO_2 reduction of the new system were calculated, and the results indicated good techno-economic feasibility of the AHP technology. Huo et al. [40] also analyzed the economic benefits of an AHP waste heat recovery system. They concluded that if the AHP is driven by low parameter extraction steam and used to provide low-grade heat for heating, then it is economically feasible with a good energy-saving capacity. But if the AHP consumes high parameter extraction steam and provides high-temperature water for either the process flow or DH, then this could lead to lower energy efficiency and make it economically not viable. Therefore, parameter-matching and the required supply water temperature after heat recovery are important factors when designing and operating AHP-based waste heat recovery systems. This also applies to the EHP-based waste heat recovery systems, but apparently, EHP is less sensitive to those parameters. It is reported that the COP of EHP can still reach 3.8 when the low-grade water is 50/35°C and supply water is 90/70°C [41]. Moreover, Mateu-Royo et al. [42] proposed a novel high-temperature heat pump with a modified scroll compressor that can achieve a heat sink temperature of 140°C with a COP of 2.23. However, the required supply temperature of the HP-based heat recovery system does not need to be that high for DH because if the recovered heat is insufficient or the supply temperature is lower than the required water temperature, the steam-water heat exchanger can be used to further reheat the water.

However, few researchers have systematically analyzed the heat-power decoupling capability in combination with waste heat recovery or their potential to save energy. Therefore, the novelty of this paper is to propose an improved heat-power decoupling method, which can recover the waste heat from the cooling water using large scale heat pumps, and simultaneously supply more efficient heat for district heating while increase the operation flexibility and energy efficiency of the CHP units. This study mainly focuses on the evaluation of the heat-power decoupling capabilities and energy-saving potentials of the large-scale peak-shaving technologies with AHP and EHP that are integrated with waste heat recovery systems in the CHP plant.

To conclude, AHP- and EHP-based waste heat recovery systems have different characteristics. The driving source of AHP has a lower grade compared to electricity, but EHP has a much higher COP than that of AHP. It is hard to say which one is more thermodynamically efficient. Ni et al. [43] used the energy flow diagram to analyze the primary energy ratio of the heat pump stations driven by different energy sources for waste heat recovery. But few researchers consider peak-shaving capability and waste heat recovery at the same time. Therefore, this paper establishes the thermodynamic system model for the CHP unit under dynamic working conditions. After validation, the model is used to analyze the peak-shaving capabilities and energy-saving potential under dynamic working conditions with EHP and AHP. Finally, the overall comparison of the heat-power decoupling effects and energy-saving capabilities between the AHP- and EHP-based waste heat recovery system is implemented.

2. Integration of waste heat recovery system based on a large scale heat pump in the CHP unit

Taking EHP as an example, the schematic diagram of the CHP unit coupled with the EHP waste heat recovery system studied in this paper is shown in Fig. 2. The EHP consumes the internal power of the CHP. In this way, the heat and power output of the CHP can be adjusted, and simultaneous peak shaving can be achieved. When the heat demand and the required DH supply temperature are not high (e.g., in the beginning or at the end of the heating season), EHP may satisfy all heat demand for DH. Otherwise (e.g., during cold winter days), EHP will provide part of the heat demand, and the rest will be covered by the steam-water heat exchanger (DH heater), where the DH water will be reheated. This process is also an example of cascade energy utilization in the CHP unit, which can help improve the energy efficiency since the COP of the EHP will decrease at higher outlet temperatures. And the main objective of EHP is to recover the low-grade waste heat but not to upgrade it to the required high DH supply temperature. Note that not all DH return water enters the EHP. Instead, the flow in the EHP condenser can be controlled, and the rest of the DH return water will be bypassed and sent to the DH heater. In addition, it is not necessary to recover all the waste heat from the condensing water. Instead, the objective is to achieve the required peak-shaving capability while recovering as much waste heat as possible. This means that the condensing water flow in the evaporator can also be controlled as needed.

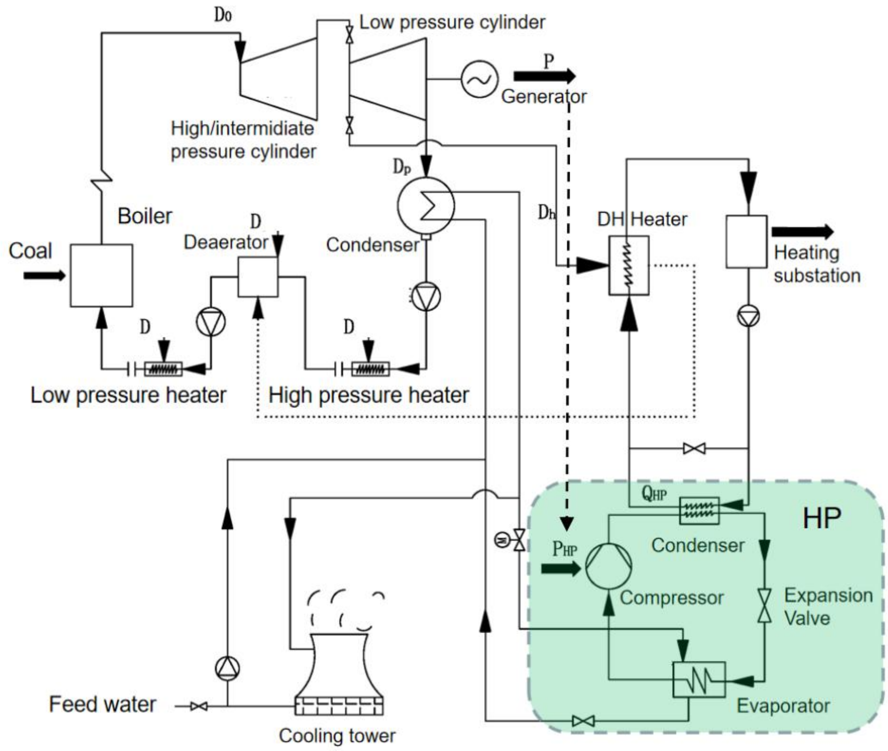


Figure 2. The schematic diagram of the CHP unit coupled with the EHP based waste heat recovery system.

In Fig. 2, the main steam flow D_0 enters the steam turbine, converting the thermal energy into mechanical energy for power generation, and in the last stage of the intermediate pressure cylinder, part of the steam flow D_h is extracted to the DH heater for heating. The exhausted steam flow D_p is condensed in the condenser, and the condensed water will be pumped to the deaerator. The cooling water for the condenser comes from not only the cooling tower but also the evaporator of the HP, which is different from the traditional process. The integration of AHP in the CHP is similar. The difference is that the driving force is not the internal electricity but the extraction steam with a certain level of pressure and temperature.

3. Thermodynamic model for the CHP unit in dynamic working conditions

Peak shaving requires the CHP unit to operate in dynamic or variable working conditions. The heat and power outputs will be adjusted according to the demand profiles of heat and power when peak shaving is needed. Therefore, the characteristics of heat and power production and energy consumption for the HP-based waste heat recovery systems should be studied. For this purpose, the thermodynamic model under dynamic working conditions for the CHP unit should be established first. This is also the basis for evaluating the peak-shaving capacities and the energy-saving potential.

3.1 Establishment of the thermodynamic model

The dynamic working conditions of the CHP unit are essentially the changes of steam flow and thermal parameters, such as extraction pressure, temperature, specific enthalpy, internal efficiency, etc., between different levels of the turbine. The schematic diagram of the typical extraction condensing steam turbine is shown in Fig. 3, where E1~E8 are the steam extraction outlets. The flowrates are small except for the last stage of the intermediate pressure cylinder (E4), where the steam for heating is extracted.

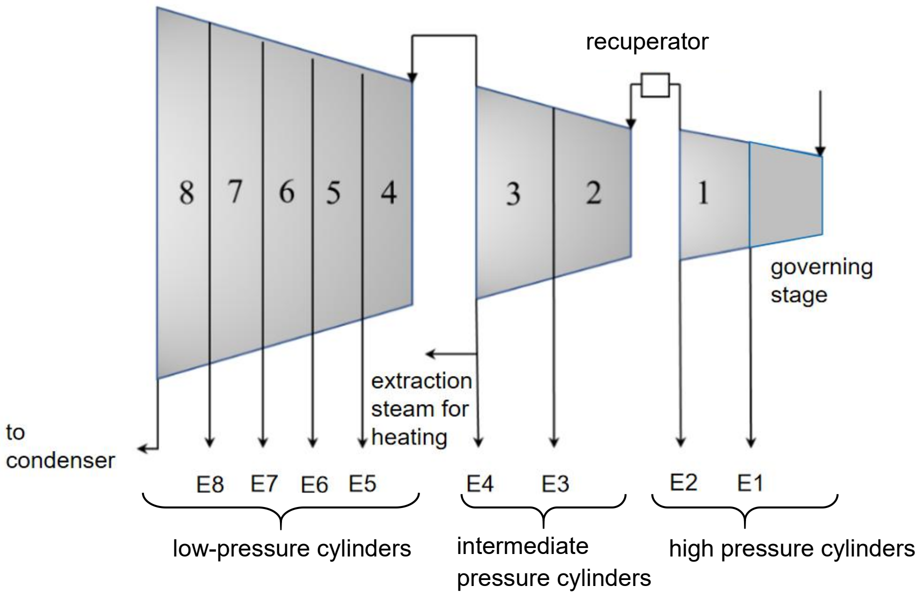


Figure 3. The schematic diagram of a typical extraction condensing steam turbine.

The calculation of the turbine under dynamic working conditions is based on Freuger's formula [44],

$$\frac{D_i}{\overline{D}_i} = \frac{\sqrt{p_i^2 - p_o^2}}{\sqrt{\overline{p}_i^2 - \overline{p}_o^2}} \sqrt{\frac{\overline{T}_i}{T_i}} \tag{1}$$

where D_i and T_i are the steam flow (kg/s) and temperature (K) before a certain stage under dynamic working conditions and p_i and p_o are the pressures at the stage inlet and outlet, Pa. Variables with upper whiskers mean the corresponding parameters before the working condition changes.

This formula does not apply to all stages of the turbine, because the extraction steam flow decreases after E4 due to the steam extraction for heating. Therefore, the thermodynamic model of the turbine should be divided into two sections: 1) from the governing stage to the extraction outlet for heating and 2) from the extraction outlet to the last stage. Then the following assumptions are made for the model development.

1) The steam temperature before each stage does not change under dynamic working conditions, i.e., $\sqrt{\overline{T}_i / T_i} = 1$.

2) The pressure ratio before and after each stage is the same under dynamic working conditions except in the stage with an extraction outlet for heating.

3) In the dynamic working conditions, the exhaust pressure of the last stage is the same as the design condition, i.e., the working pressure in the condenser is stable.

4) The pressure of extraction steam for heating is stable.

5) The terminal temperature differences and pressure losses of high/low-pressure heaters and deaerator are stable, and the losses of steam and water of shaft seals are negligible.

6) The time delay due to the mass flow in the pipelines connecting the different stages of the turbine and other accessories is neglected.

Then the pressure of the governing stage can be determined by

$$p_{1,o} = \overline{p}_{1,o} \frac{D_{1,o}}{\overline{D}_{1,o}} \tag{2}$$

The pressure change between the governing stage and the extraction outlet for heating cannot be neglected. The extraction pressure of stage j (here $2 \le j \le 4$) can be calculated by

$$p_j = \sqrt{{p_h}^2 + (\overline{p}_j^2 - \overline{p}_h^2) \frac{D_j^{\,2}}{\overline{D}_j^{\,2}}} \tag{3}$$

After the extraction outlet for heating, the extraction pressure of each stage is

$$p_j = \overline{p}_j \frac{D_j}{\overline{D}_j} \tag{4}$$

where $p_{1,o}$ and $D_{1,o}$ are the outlet pressure (Pa) and steam flow (kg/s) of the governing stage after the change in working conditions; D_j is the steam flow entering stage j after the change in working conditions, kg/s; and p_h is the extraction pressure for heating under dynamic working conditions, which is assumed to be stable, Pa.

The specific enthalpy at the extraction outlet of the jth stage can also be determined by the thermal equilibrium relationship:

$$h_{j,i} = h_{j,o} - \Delta H_{j,S} \eta_j \tag{5}$$

where $h_{j,i}$ and $h_{j,o}$ are the specific enthalpy of steam at the inlet and outlet of the jth stage under dynamic working conditions, J/kg; $\Delta H_{j,S}$ is the isentropic enthalpy drop of steam in the jth stage determined by the steam pressure ratio before and after the stage; and J/kg; η_j is the internal efficiency of the jth stage.

The internal efficiency of the stages under the reference condition before the change in working condition is:

$$\overline{\eta}_j = \frac{\overline{h}_{j,i} - \overline{h}_{j,o}}{\Delta \overline{H}_{j,S}} \tag{6}$$

In the study, it is assumed that the internal efficiencies of the intermediate pressure cylinders are the same as the reference condition even operating under dynamic working conditions. But the internal efficiencies of high pressure and low-pressure cylinders will be corrected according to the data provided by the power plant in the case study [36].

The extraction steam from different stages to the low-pressure and high-pressure heater has different properties. The steam temperatures before entering the heaters and the condensing water temperatures and the specific enthalpy after the heaters can be determined by the thermodynamic properties of the steam,

$$T_{j,i} = T_{j,o} = f(p_j - \Delta p_{j,loss}) \tag{7}$$

$$h_{j,o} = f\left((p_j - \Delta p_{j,loss}), T_{j,o}\right) \tag{8}$$

where $T_{j,i}$ and $T_{j,o}$ are the temperatures of the extraction steam from the jth stage entering the high/low-pressure heater, and the condensing water temperature after the heater, K; $\Delta p_{j,loss}$ is the pressure loss of extraction steam, Pa; and $h_{j,o}$ is the specific enthalpy of the condensing water after the high/low-pressure heater, J/kg.

The inlet and outlet temperatures at the water side of the high/low-pressure heaters are

$$T_{j,w,o} = T_{j,i} - \Delta T_{j,w} \tag{9}$$

$$T_{j,w,i} = T_{j-1,w,o} \tag{10}$$

where $T_{j,w,i}$ and $T_{j,w,o}$ are the inlet and outlet water temperature of the high/low-pressure heater corresponding to the jth stage, K; $T_{j-1,w,o}$ is the outlet water temperature of the high/low-pressure heater corresponding to the (j-1)th stage; and $\Delta T_{j,w}$ is the heat transfer temperature difference of the high/low-pressure heater corresponding to the jth stage, K.

Similarly, the outlet specific enthalpy at the water side of the high/low-pressure heater can be determined by

$$h_{j,w,o} = f(p_{j,w,o}, T_{j,w,o}) \tag{11}$$

where $p_{j,w,o}$ is the outlet water pressure of the high/low-pressure heater corresponding to the jth stage, Pa. It can be seen as equal to the inlet water pressure of the heater.

3.2 Solution and validation of the thermodynamic model

3.2.1 Solution of the model

This study first selects a known working condition as the reference and obtains all thermodynamic parameters under this condition using the thermodynamic model and the monitored data. Once the working condition changes, new heat and power outputs can be monitored or be determined according to the operating curves of the unit. Then the initial steam flow after the change of working condition can be assumed, and the thermodynamic model is used to calculate corresponding parameters in order to approach the power output. This process will usually need some iterations before the stop criteria are satisfied, and then all parameters can be determined with the thermodynamic model. The above-mentioned process is shown in Fig. 4.

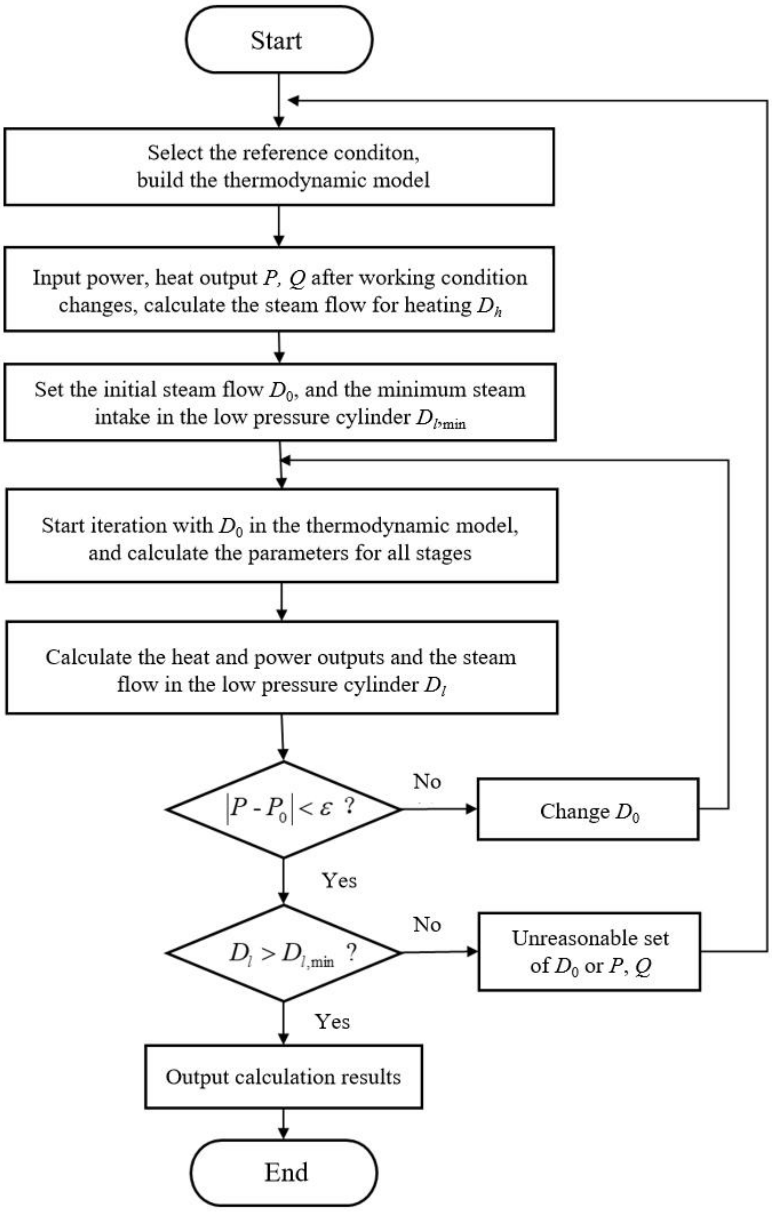


Figure 4. The calculation flow chart for the off-design model of the CHP unit.

Specifically, the process in Fig. 4 can be divided into six steps, as follows:

Step 1: A typical working condition is chosen as the reference, and the thermodynamic model for this reference condition will be established. On this basis, the thermodynamic parameters and internal efficiencies of each stage are calculated according to the mass conservation law and the developed model.

Step 2: The heat and power outputs of the CHP unit are determined according to the demand profiles of heat and power. And the extraction steam flow for heating can be calculated by:

$$D_h = Q / \eta_{hx}(h - h_w) \tag{12}$$

where Q is the heat output, W; D_h is the extraction steam flow for heating, kg/s; h is the specific enthalpy of the extraction steam, J/kg; h_w is the specific enthalpy of the condensing water, J/kg; and η_{hx} is the efficiency of the steam-water heat exchanger. It is assumed stable in the study, since the dominant influencing factor on η_{hx} is the latent heat of vapor, which is changing slightly in the range of DH supply temperatures. Other parameters e.g. coefficient of heat conductivity of the condensing water and Nusselt number in the tube side are not the main factors; the flow rates in the tube side also can affect η_{hx}, but with much smaller impact compared to the latent heat of vapor [45].

Step 3: The initial main steam flow is assumed to be D_0, according to the design parameters or the operating curves of the CHP unit. And the minimum steam intake $D_{l,\min}$ in the low pressure cylinder will be determined at the same time.

Step 4: The developed thermodynamic model is used to calculate the thermodynamic parameters, including the steam flow, specific enthalpy, pressure, temperature, etc., for all stages of the turbine.

Step 5: According to the obtained thermodynamic parameters in Step 4, the corresponding steam and water parameters can be determined, the inlet steam flow of the low-pressure cylinder is the steam flow after the extraction stage, and the power output of the unit can be calculated by:

$$P = (D_0 h_0 + D_{rh} q_{rh} - \sum_{1}^{z} D_k h_k - D_p h_p)\eta_{m1}\eta_{m2} \tag{13}$$

where P is the generating capacity of the unit, W; D_0 is the inlet steam flow of the turbine, kg/s; D_{rh} is the amount of reheat steam of the turbine, kg/s; D_k is the extraction steam of the turbine at the kth stage, kg/s; D_p is the condensing steam flow of the turbine, kg/s; h_0 is the specific enthalpy of the main steam, J/kg; h_p is the specific enthalpy of the exhaust steam of the last stage, J/kg; q_{rh} is the enthalpy increase of the reheated steam; h_k is the extraction specific enthalpy of the kth stage; and z is the number of the stage. η_{m1} and η_{m2} are the mechanical efficiency and generator efficiency, they are deemed stable during the dynamic working conditions. Eq. (13) is also valid for stationary working conditions when steam parameters are stable.

The calculated power output will be compared with the set value. If the error exceeds the limit, then change the value of D_0 to start the iterative computation loop until the error is smaller than the set value and the steam intake is larger than the minimum in the low pressure cylinder.

Step 6: The real inlet steam flow, heat, and power output of the CHP unit can be obtained using the above-mentioned iterations. Besides, we can also calculate the waste heat of exhausted steam in the condenser under dynamic working conditions and the energy consumption for heating.

The waste heat of exhaust steam in the condenser is

$$Q_{condens} = D_p (h_p - h_{c,w}) \tag{14}$$

Heat consumption of the unit reflecting the thermal economic indicator can be described as

$$Q_{thermal} = D_0 (h_0 - h_{feed}) + D_{rh} q_{rh} \tag{15}$$

where $Q_{thermal}$ is the heat consumption of CHP unit, J; $h_{c,w}$ is the specific enthalpy of the condensate water in the condenser, J/kg; and h_{feed} is the specific enthalpy of the boiler feed water, J/kg.

3.2.2 Validation of the model

The above-mentioned thermodynamic model under dynamic working conditions was solved by MATLAB in this study. An extraction condensing steam turbine unit, C280/N350-16.7/537/537, was chosen to validate the developed model. It is a sub-critical unit with one reheat cycle. The rated power outputs are 280MW and 350MW at steam extraction condition and pure condensing working condition, respectively. The main steam pressure is 16.7MPa, and the temperatures of the main steam and reheat steam are 537°C. Other parameters of the unit are shown in Table 1.

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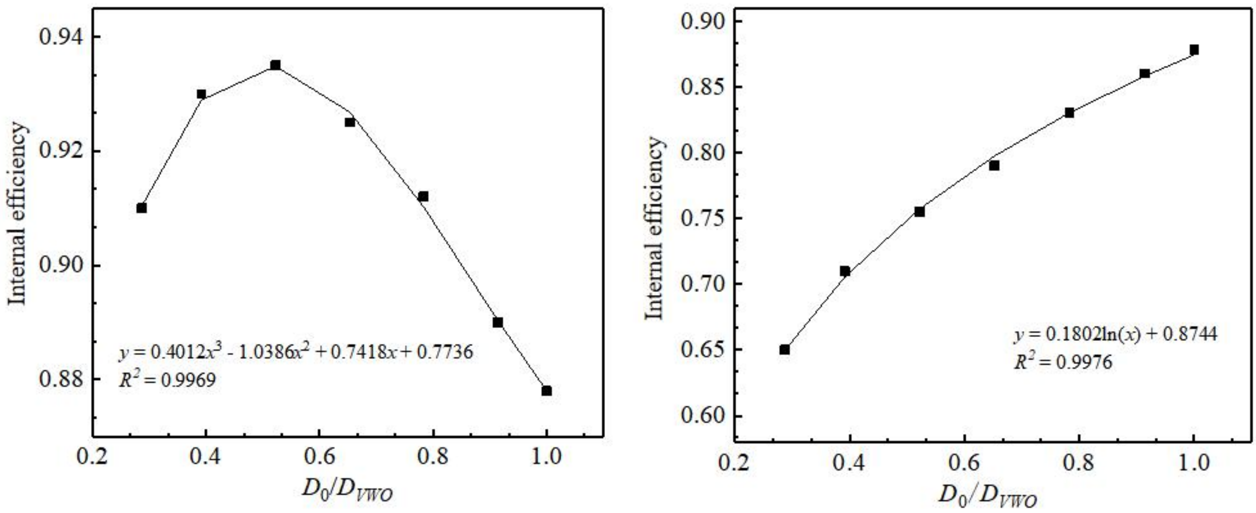
Table 1. Main parameters of C280/N350-16.7/537/537 CHP turbine unit

Parameters	Value
Maximum steam inlet	1165 t/h
Rated main steam flow	1045.29 t/h
Rated reheat steam pressure	3.22 MPa
Rated exhaust steam pressure	4.9 KPa
Boiler feed-water temperature	272.2°C
Maximum heating extraction capacity	680 t/h
Rated heating extraction capacity	480 t/h
Extraction pressure	0.49 MPa
Minimum steam inlet of low-pressure cylinder	144.4 t/h

The valve wide open (VWO) condition was chosen as the reference condition when establishing the thermodynamic model for all stages and the reheat cycle. This study determined the internal efficiencies for high and low-pressure cylinders under dynamic working conditions using the data fit based on the real data provided by the CHP plant under several scattered working conditions with different steam flows. The fitted internal efficiencies agree quite well with the real data, and the R-square indicators are more than 0.99. Therefore, the obtained fitting curve shown in Fig. 5 was used in the developed model.



(a) low-pressure cylinders (b) high-pressure cylinders

Figure 5. The fitted internal efficiency curve of high pressure cylinder and low-pressure cylinder

In addition to the VWO working condition, we also select four turbine heat acceptance (THA) conditions under the design back pressure with different main steam flows and two typical heating conditions to carry out the thermodynamic simulation of this turbine. The simulation results of main steam flow D_0 and heat consumption rate $q_{thermal}$ were compared to the corresponding parameters of $\overline{D}_0$, $\overline{q}_{thermal}$, based on the calibrated thermal equilibrium diagram provided by the turbine manufacturer [46]. The comparison results are shown in Table 2. It can be seen that the errors under those working conditions are smaller than 2%, indicating that the accuracy of the developed model is very good and suitable for the simulation under dynamic working conditions.

Table 2. The accuracy verification of the off-design model of the CHP unit

Condition	D_0 (t/h)	$\overline{D}_0$ (t/h)	Err(D_0) (%)	$q_{thermal}$ (kJ/kWh)	$\overline{q}_{thermal}$ (kJ/kWh)	Err ($Q_{thermal}$) (%)
VWO	1165.52	1165.00	0.04	7856.40	7862.90	0.08
100%THA	1049.72	1045.29	0.42	7880.37	7865.10	0.19
75%THA	757.51	753.70	0.51	7986.02	7949.40	0.46
50%THA	495.97	501.43	1.09	8248.09	8270.70	0.27
40%THA	405.59	413.84	1.99	8498.65	8570.10	0.83
Rated heating condition	1100.82	1107.00	0.56	6191.24	6195.50	0.07
Maximum heating condition	1156.72	1165.00	0.71	5202.78	5210.10	0.14

4. EHP model based on the efficiency factor in dynamic working conditions

The efficiency factor method [47] was adopted to build the EHP model for the analysis of the waste heat recovery system under dynamic working conditions.

The heat supply of the EHP for heating the DH water is

$$Q_{EHP}=G_{DHN,i}c_w(T_{con,o}-T_{con,i}) \tag{16}$$

The recovered heat of the heat pump is

$$Q_{re}=G_{cooling,i}c_w(T_{eva,i}-T_{eva,o}) \tag{17}$$

The energy balance of the EHP can be written as

$$Q_{re}+P_{EHP}=Q_{EHP} \tag{18}$$

The COP can be written as

$$\mathrm{COP}_{EHP}=\frac{Q_{EHP}}{P_{EHP}} \tag{18}$$

where $G_{DHN,i}$ and $G_{cooling,i}$ are the water flow into the heat pump system at the DHN side and the circulating cooling water flow entering the heat pump, respectively, kg/s; $T_{con,i}$ and $T_{con,o}$ are the inlet and outlet temperatures of DHN water at the condensing side of the heat pump, K; $T_{eva,i}$ and $T_{eva,o}$ are the inlet and outlet temperatures of cooling water at the evaporating side of the heat pump, K; and P_{HP} is the driving power of the EHP, W.

COP affects the heating effect of EHP and thus the operation of the whole system. The COP of a given EHP is influenced by the water temperature at the evaporating side and the condensing side in real dynamic operation conditions.

The ideal value of the COP can take form

$$\mathrm{COP}_{ideal}=\frac{T_{CON}}{T_{CON}-T_{EVA}} \tag{20}$$

And the real COP at dynamic working conditions can be written as [47]

$$\mathrm{COP}=\beta_{EHP}\mathrm{COP}_{ideal} \tag{21}$$

$$\beta_{EHP}=M+\frac{N}{\mathrm{COP}_{ideal}} \tag{22}$$

where β_{EHP} is the efficiency factor of the EHP, reflecting the influence of internal heat transfer loss on the actual dynamic operation of the system; and T_{CON} and T_{EVA} are the EHP’s condenser temperature and evaporator temperature, respectively, K—they are calculated according to Eqs. (23) and (24). M and N are the coefficients, depending on the working fluid and working temperature ranges of the heat pump.

The temperatures of DHN water and condensing water are all changing in the dynamic working conditions. T_{CON} and T_{EVA} can be calculated using the logarithmic average of inlet and outlet water temperatures of the condenser and evaporator [47].

$$T_{CON} = \frac{T_{con,o} - T_{con,i}}{\ln\left(\frac{T_{con,o}}{T_{con,i}}\right)} \tag{23}$$

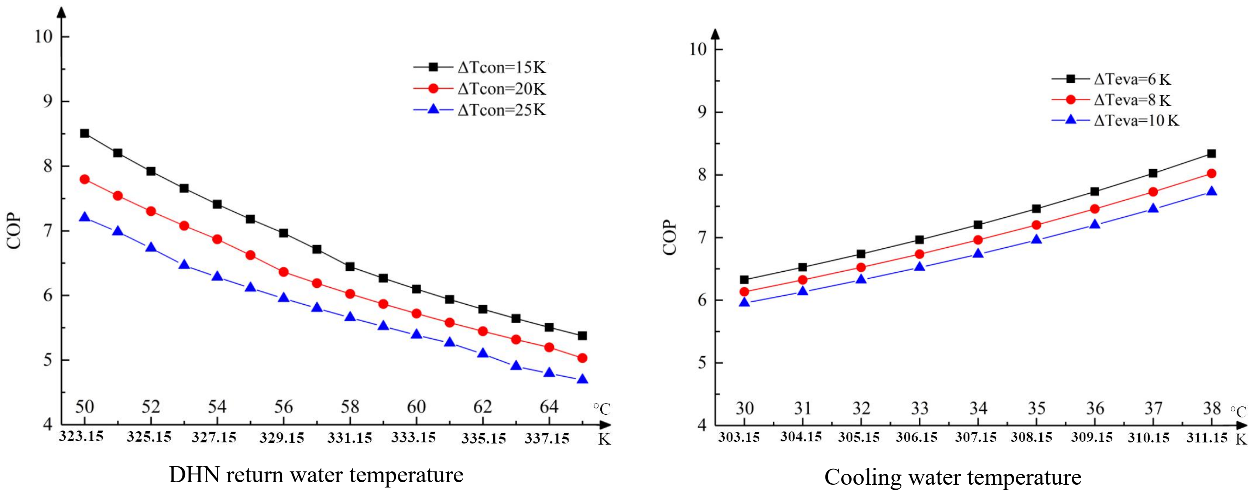
$$T_{EVA} = \frac{T_{eva,i} - T_{eva,o}}{\ln\left(\frac{T_{eva,i}}{T_{eva,o}}\right)} \tag{24}$$

In this study, R600 (*n*-butane) is selected as the working fluid, and the coefficients M and N corresponding to different evaporation and condensation temperatures are shown in Table 3 [47]. Under the design condition, the circulating water temperature on the evaporating side is reduced from 38℃ to 30℃, while the DHN return water temperature is increased from 65℃ to 90℃. Accordingly, COP_{ideal} of the EHP is 8.08, but the actual COP is calculated to be 5.06 using the efficiency factor method. Sometimes the real DHN water temperature can be lower than this scope, and thus the COP can be higher.

Table 3. The values of coefficients M and N corresponding to different evaporation and condensation temperatures of the water source heat pump with R600 as the working fluid [47].

T_{CON}/K	T_{EVA}/K	M	N
[318.15, 328.15)	[283.15, 313.15]	0.7319	−0.5154
[328.15, 338.15)	[283.15, 323.15]	0.7259	−0.5602
[338.15, 348.15)	[283.15, 333.15]	0.7181	−0.6077
[348.15, 358.15)	[288.15, 343.15]	0.7081	−0.6582
[358.15, 368.15)	[293.15, 353.15]	0.6781	−0.7639
[358.15, 378.15)	[298.15, 363.15]	0.6217	−0.8149
[378.15, 388.15)	[303.15, 373.15]	0.5586	−0.7767

Figure 6 shows the variation of COP with different DHN return water temperatures and the circulating cooling water temperature. It can be seen that the COP increases when the DHN return water temperature decreases and/or the circulating cooling water temperature increases. In addition, the lower the temperature differences are at the condenser or the evaporator sides, the higher the COP will be. The COP in any dynamic working conditions of the EHP can be obtained according to this model.



(a) COP variation with DHN return temperature (b) COP variation with cooling water temperature

Figure 6. The effects of DHN return water temperature and the circulating water temperature on the COP of EHP.

5. heat-power decoupling capability of the CHP unit with EHP-based waste heat recovery system

It is important to determine the operating region of the heat and power outputs for a CHP unit with an HP-based waste heat recovery system. They are essential for calculating the CHP unit parameters under dynamic working conditions.

The CHP unit described in Section 3.2 is used as a case study. Based on the developed thermodynamic model under dynamic working conditions, when the steam extraction and heat outputs are determined, the maximum and minimum power outputs can be calculated by solving D_0 iteratively according to Fig. 4. Therefore, the theoretical operating region can be obtained. The maximum and minimum power outputs are 382.93MW and 149.7MW, respectively, under two extreme condensing working conditions, and the maximum heat output is 447.48MW while the power output is 267.11MW. But the minimum power output is assumed to be no less than 175MW, considering the safety of the turbine when assessing the decoupling effect of the heat and power.

The heat and power balances under dynamic working conditions for the CHP unit with the HP-based waste heat recovery systems can be written as:

$$Q_{EHP} + Q_{CHP} = Q_D \tag{25}$$

$$P_{CHP} = P_{EHP} + P_D \tag{26}$$

where Q_{CHP} and P_{CHP} are the heat and power outputs of the CHP unit, W; and Q_D and P_D are the heat and power demands, MW.

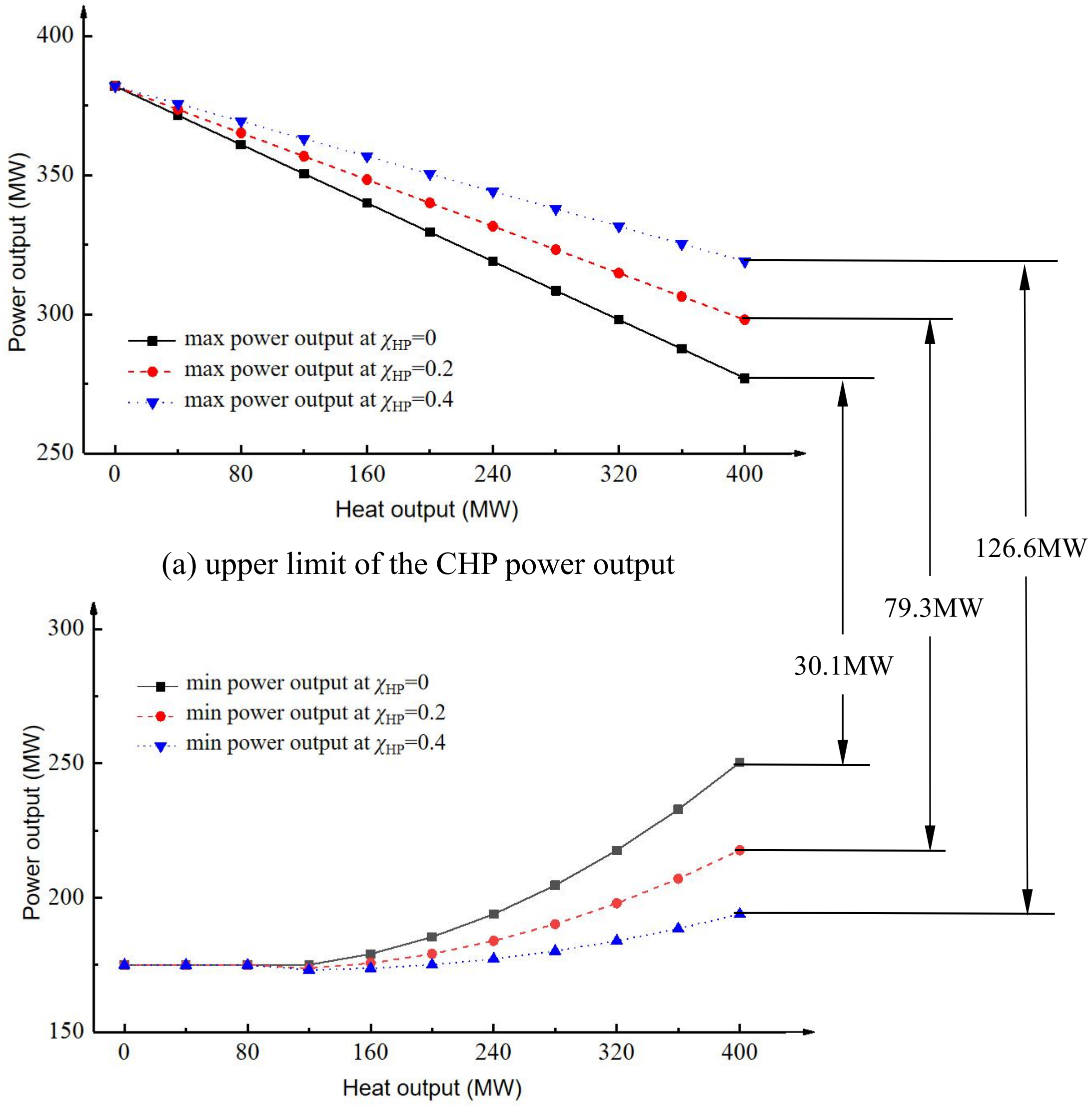
The power output is different for the CHP unit with the EHP-based waste recovery system compared to the traditional system because the EHP can produce heat with an efficiency of COP at the cost of some generated electricity. This contributes to the decoupling of heat and power, and the decoupling effect depends on the share of the heat supply from the HP-based waste heat recovery system. Therefore, the ratio between the heat supply of the HP and the total heat demand is called the heat pump DH ratio, which is defined as:

$$\chi_{HP} = \frac{Q_{HP}}{Q_D} \tag{27}$$

The peak-shaving capability of the power corresponding to a certain heat output is defined as the difference between the CHP unit's maximum and minimum power output.

$$\Delta P = P_{\max}(Q_{CHP}) - P_{\min}(Q_{CHP}) \tag{28}$$

where $P_{\max}(Q_{CHP})$ and $P_{\min}(Q_{CHP})$ are the maximum and minimum power output of the CHP unit when the heat output is Q_{CHP}, MW.



(b) lower limit of the CHP power output

Figure 7. The variation of the upper and lower limits of the CHP unit with the heat pump DH ratio χ_{HP}=0, 0.2 and 0.4.

For the studied case, as the heat output increases from 0MW to 400MW, the variation in the upper and lower power outputs when χ_{HP} is 0, 0.2, and 0.4 is shown in Figure 7. It can be seen that the upper power output increases while the lower power output decreases as the heat output increases. This leads to a wider scope of the power output in the operating region of the CHP unit with EHP-based waste heat recovery system, and thus the heat and power are decoupled to some extent.

When the CHP unit is running under the maximum steam inlet condition, the upper power output of the CHP unit increases due to the smaller extraction steam flow for DH, but part of the generated electricity should be used for driving the EHP. When the CHP unit is running under the minimum steam inlet condition, the safe operation of the turbine and the smallest steam flow in the low pressure cylinder should be satisfied. In this case, the use of EHP reduces the steam extraction for DH and the steam flow in the CHP unit, leading to a lower power output compared to the normal CHP unit. To conclude, the power output scope is wider or the heat-power decoupling extent is larger when χ_{HP} and the heat load increase compared to the traditional CHP unit. For example, When χ_{HP}=0, all the heat demand should be supplied by the CHP unit, and the power output scope is only 30.1MW when the heat output is 400MW. But when χ_{HP}=0.2 and 0.4, the heat supply of the CHP unit will be reduced to 320MW and 240MW, and the power output scope will be increased to 79.3MW and 126.6MW, respectively, as can be seen from Figure 7. This means that the power peak-shaving capacity becomes 2.6 and 4.2 times larger than with the traditional system. This will also bring income for the CHP plant for providing the peak-shaving services to the power grid. In theory, the EHP-based waste heat recovery system can afford the total heat demand when χ_{HP}=1, which means complete heat-power decoupling. But this is not necessary since CHP unit should also satisfy the power demand at the same time. And the main purpose of the HP-based waste heat recovery system is to increase the flexibility of the power and heat generation while satisfying the same heat and power demands.

6. Energy-saving potential of the CHP unit with EHP-based waste heat recovery system

6.1 Energy consumption characteristic

The use of EHP to recover waste heat reduces the heat output of the CHP unit, but it consumes the generated electricity of the plant and converts it to heat at the efficiency of COP, which is larger than 1. However, it is not clear whether the system can save primary energy and to what extent. Therefore, the coal consumption characteristics of the system should be analyzed.

The coal consumption of the CHP unit combined with EHP for heating can be calculated by

$$B = \frac{Q_{thermal}}{q_{net}\eta_b} \tag{29}$$

where $Q_{thermal}$ is the heat consumption of the CHP unit, W; q_{net} is the lower heating value of the standard coal equivalent, 29,307,600J/kg; and η_b is the boiler efficiency.

The coal consumption for heating can be written as

$$B_h = \frac{Q_h}{q_{net}\eta_b\eta_p} \tag{30}$$

where Q_h is the heat consumption for heating; η_p is the efficiency of heating pipelines.

Thus, the coal consumption for power generation is:

$$B_e = B - B_h \tag{31}$$

Then, the system energy consumption characteristics of the CHP unit with EHP-based waste heat recovery system can be analyzed. We take the working condition when the heat output is 350MW and the power output is 250MW, as an example. In the case study, η_b =0.9, η_p =0.98, and the COP of the EHP under the design operating condition is 5.06. Figure 9 shows the relationship of the coal consumption in the standard coal equivalent for heating, for power generation, and their sum when the heat pump DH ratio χ_{HP} is from 0 to 0.5 since the heat supply of EHP does not need to be very high but should be effective to make the system more flexible and energy-efficient.

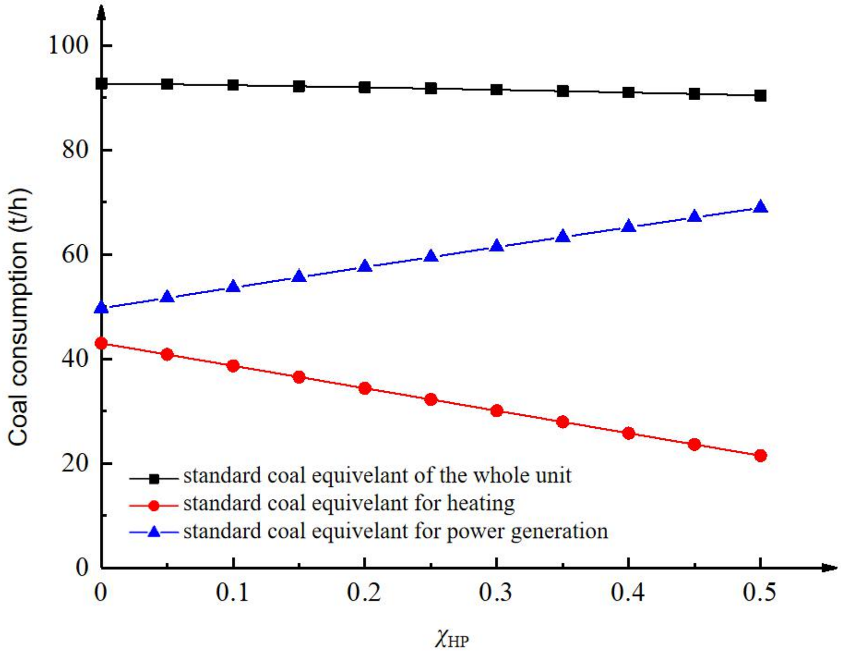


Figure 8. The influence of heat pump DH ratio (χ_{HP}) on energy consumption characteristics of the CHP unit with EHP waste heat recovery system.

Figure 8 indicates that the coal consumption for power generation increases while the coal consumption for heating decreases when χ_{HP} is increasing from 0 to 0.5. This is because the use of EHP dramatically reduces the heat output of the CHP unit at the cost of increasing the power output in order to drive the heat pump and satisfy the power demand at the same time, and this makes the coal consumption for power generation climb. But for the system, the total coal consumption of the whole CHP unit gradually decreases, and the larger χ_{HP} is, the higher the energy-saving potential will be. It can be calculated that the total coal consumption of the CHP unit is the 94.13t/h standard coal equivalent when the heat and power outputs are 350MW and 250MW, respectively. And the total coal consumption is reduced to 90.4 t/h at χ_{HP}=0.5, while the EHP affords 175MW of the DH heat demand with 140.4MW waste heat recovered from the circulating water. The standard coal equivalent of 3.73t/h can be saved at χ_{HP}=0.5 compared to the traditional system at χ_{HP}=0, leading to about 4% energy savings.

6.2 Energy-saving potential in different working conditions

In addition to the heat pump DH ratio χ_{HP}, the energy-saving potential of the system is also influenced by the COP of the heat pump. Apparently, the higher the COP is, the larger the energy-saving potential will be. However, it is very important to determine the minimum threshold of the COP, under which the whole system will not save energy. It is also important to indicate the energy-saving potential in different working conditions with different combinations of the heat and power outputs.

In this section, the energy-saving potential described as the saved standard coal equivalent can be defined as:

$$\Delta B = B_0 - B \tag{32}$$

where B_0 is the standard coal consumption of the CHP unit without the EHP-based waste heat recovery system, t/h.

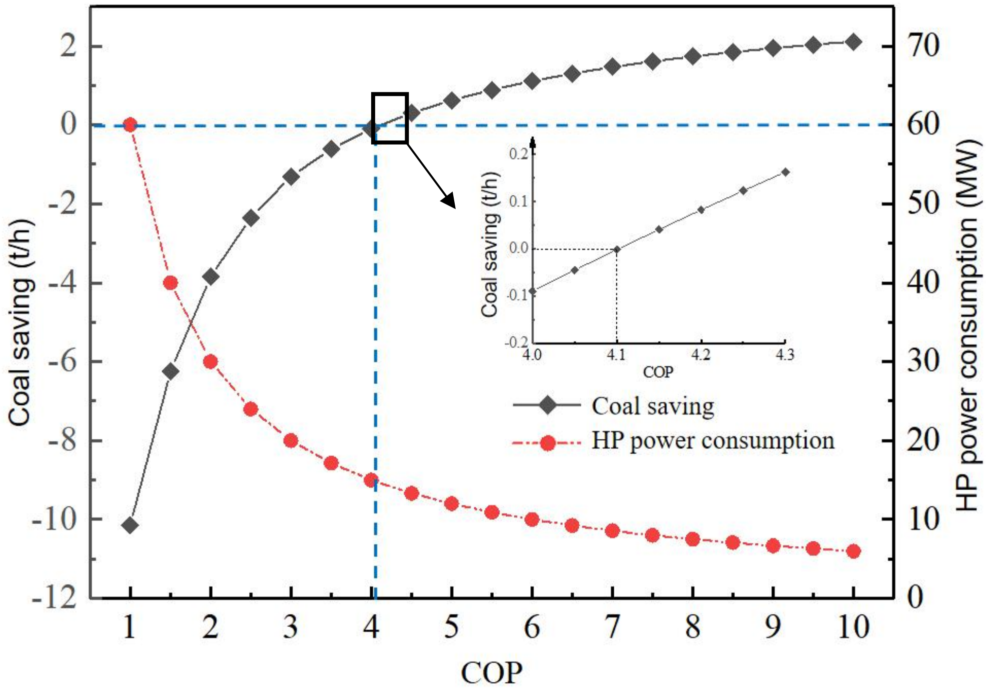


Figure 9. The influence of COP on the driving power consumption for the EHP and the energy-saving potential of the whole system, Q_{EHP}=60MW, χ_{HP}=0.18, Q_D=350MW, P_D=250MW.

An EHP with 60MW (χ_{HP}=0.18) heat output was used to analyze the energy-saving potential. Figure 9 shows the relationship between COP and the power consumption of the EHP as well as the energy-saving potential when the heat and power demands are 350MW and 250MW, respectively. It can be seen that, in this case, the power for driving the EHP is increasing very quickly while the coal saving is decreasing dramatically when the COP is decreasing. This leads to a smaller energy-saving potential and even will make the whole system consume more coal when COP<4.1 compared to the CHP unit without EHP. Therefore, this is called the critical COP, which is different under different working conditions. The bigger the χ_{HP} is, the smaller the critical COP will be. Here, when COP>4.1, the whole system will recover more waste heat, and thus the driving power of the

EHP will be reduced, leading to lower energy consumption and thus a higher energy-saving potential, but this benefit will increase more slowly with a larger COP.

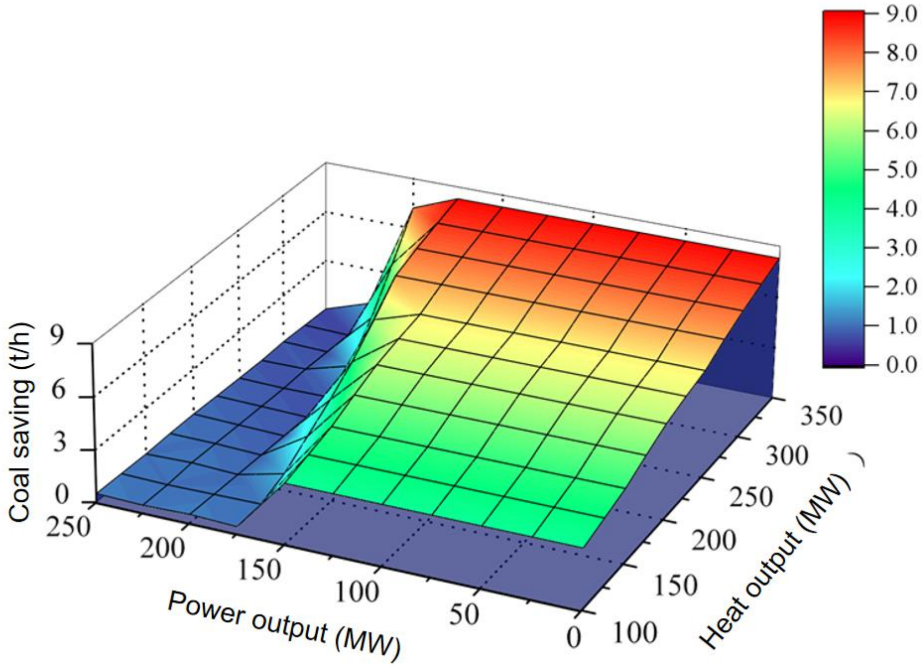


Fig. 10 The energy-saving potential of the CHP system with the EHP waste heat recovery system under different working conditions.

In order to study the influence of heat and power demands under different working conditions on the energy-saving potential, we take the same case study with EHP having a rated heating output of 60MW. The variation scopes of the heat and power demands are [100MW, 350MW] and [0, 250MW]. Figure 10 shows the energy-saving potential in terms of different working conditions. It is clear that the coal saving effect is growing as the heat demand is increasing and the power demand is decreasing. This means that the energy potential is becoming better with larger heat demand and lower power demand because under these working conditions, the use of EHP not only reduces the heat output of the CHP unit but also reduces the forced power generation caused by the thermoelectric coupling effect since the forced power generation is used to drive the EHP and provide more heat. Thus, the system can provide a downward peak-shaving capability for power generation. However, when the heat demand maintains a certain level and the power demand gradually increases, the energy-saving potential will decrease and finally become zero. This is because when the power demand is high, the system will not need any downward peak-shaving capability but has to generate more power to drive the EHP. And the coal saving effect caused by reducing the forced power output when the heat pump replaces the CHP unit for heat supply will disappear. Therefore, the power demand is more sensitive for the energy-saving potential of the system, and it is better to let the whole system operate more extensively under the conditions of high heat demand and low power demand.

7. Comparison of the heat-power decoupling capabilities and energy-saving potentials between AHP and EHP-based waste heat recovery systems

AHP is more often used in the CHP plant for recovering the waste heat from circulating cooling water, and it is reported that this technology is techno-economically feasible [48-50]. The results in this study also show that the EHP-based waste heat recovery system can save energy and help the CHP unit realize heat-power decoupling to some extent, depending on the heat pump DH ratio (χ_{HP}). However, to the best of our knowledge, it is still not clear which one is better in terms of energy-saving and heat-power decoupling. Therefore, the two technologies are compared under different working conditions in this section.

The energy balance of the AHP can be written as

$$W_P + Q_{AHP} = Q_{AHP,re} + Q_g \tag{33}$$

where W_P is the solution pump energy consumption of AHP, W; Q_{AHP} is the heat supply of AHP, W; $Q_{AHP,re}$ is the recovered waste heat from the cooling water, W; and Q_g is the heating energy of driving steam, W.

If the power consumption of the solution pump is ignored, then

$$Q_{AHP} = COP_{AHP} Q_g \tag{34}$$

where COP_{AHP} is the coefficient of performance of AHP.

The steam consumption of AHP can be calculated by

$$D_g = \frac{Q_g}{(h_g - h_g')} \tag{35}$$

where D_g is the driving steam flowrate of AHP, kg/s; h_g is the enthalpy of driving steam, J/kg; and h'_g is the saturated water enthalpy of driving steam, J/kg.

Before the comparison study, the heat and power outputs should be calculated for the CHP unit with AHP-based waste heat recovery system, assuming that the driving steam of the AHP is the same extraction steam for the DH heater. The heat output of the CHP unit is calculated as

$$Q = Q_D - Q_{AHP} + Q_g \tag{36}$$

where Q is the actual heat output of the unit after coupling the AHP waste heat recovery system, MW. Q_D is the heat demand of the DH network, MW.

In this comparison study, the heating capacity of the AHP and EHP is set to be 60MW. Two different working conditions are analyzed with the same heat demand of 350MW, while the power demand is 250MW and 195.86MW, respectively, in order to reflect two typical working conditions of high and low power demands. The COP of AHP and EHP is 1.75 and 5.06 under the design conditions of each heat pump. Then the capabilities for heat-power decoupling as well as the energy saving are demonstrated in Tables 4 and 5.

Table 4. The comparison of the potentials for heat-power decoupling and energy saving between CHP units with AHP and EHP when the power and heat demands are 250MW and 350MW.

Parameter	With no HP	With AHP	With EHP
Power output of the CHP unit (MW)	250	250	261.85
Heat output of the CHP unit (MW)	350	324.28	290
Power demand (MW)	250	250	250
Standard coal consumption for heat supply (t/h)	46.06	42.99	35.62
Standard coal consumption for power supply (t/h)	48.07	49.70	56.44
Total coal consumption (t/h)	94.13	92.69	92.06
Coefficient of the performance	/	1.75	5.06
Energy saving in standard coal equivalent (t/h)	0	1.44	2.07
Waste heat recovery (MW)	0	25.71	48.15

Table 5. Comparison of the potentials for heat-power decoupling and energy saving between CHP units with AHP and EHP when power and heat demands are 195.86MW and 350MW.

Parameter	With no HP	With AHP	With EHP
Power output of the CHP unit (MW)	195.86 + 33.03(need peak-shaving)	195.86 + 23.33(need peak-shaving)	195.86 + 11.85(for driving EHP)
Heat output of the CHP unit (MW)	350	324.28	290
Power demand (MW)	195.86	195.86	195.86
Standard coal consumption for heat supply (t/h)	46.03	42.99	38.69
Standard coal consumption for power supply (t/h)	45.39	44.29	43.21
Total coal consumption (t/h)	91.42	87.28	81.9
Coefficient of the performance	/	1.75	5.06
Energy saving in standard coal equivalent (t/h)	0	4.14	9.52
Waste heat recovery (MW)	0	25.71	48.15

It can be concluded from Tables 4 and 5 that both AHP and EHP can help the system obtain certain heat-power decoupling as well as energy-saving effects. Table 4 indicates that when the power demand is high (P_D=250MW), there is no need for the peak shaving in the CHP unit, but with EHP, the unit has to increase the power output in order to satisfy the power demand and drive the EHP for heating. That is why the power output of the CHP unit with the EHP will be more than 250MW. Under this working condition, it is 261.85MW because the EHP consumes 11.85MW of power and recovers 48.15MW of heat (Q_{EHP}=60MW), which means that the heat output of the CHP unit is 290MW. Because the AHP consumes steam and produces heat, the power output of the CHP unit with the AHP is 250MW, but the heat output is higher than the system with the EHP since extra steam will be used to drive the AHP. Here the heat output is 324.28MW, including the driving steam, and the heat recovery with AHP is 25.71MW, leading to the total heat demand of 350MW. To conclude, the CHP unit with EHP has higher coal consumption for the power supply but lower coal consumption for the heat supply compared to the system with AHP. Under this working condition of (P_D, Q_D)=(250MW, 350MW), the standard coal consumption with EHP for the power supply is

6.74t/h higher, while consumption for the heat supply is 7.37t/h lower than that with the AHP. Therefore, the total coal consumption is reduced by 0.63 t/h with EHP compared to the system with AHP.

Table 5 shows that when the power demand is lower (P_D=195.86MW) and the heat demand is maintaining at 350MW, the CHP unit with no HP will generate surplus electricity (33.03MW) due to the coupling effect of heat and power. Of course, the CHP plant can choose to fulfill the power demand with priority and activate peak heating devices to supply the insufficient heat supply due to the less heat output corresponding to the less power output. But the heat-power decoupling and energy saving potentials should be evaluated based on the same heat and power output for different technologies. Therefore, the downward peak-shaving capability is needed for reducing the extra power output. Although the heat demand is the same, the system with AHP will output more heat than that with EHP in order to drive the AHP, and thus the power output is also higher for the AHP-based system. This means that the heat-power decoupling potential of the CHP unit with AHP is relatively weak compared to the system with the EHP, since 23.33MW of power needs downward peak-shaving. In this case, the forced power outputs of the system with AHP and EHP are reduced by 9.7MW and 21.18MW, respectively compared to the traditional system. Therefore, the power consumption of the EHP will enable a larger capability to accommodate renewable energies, e.g., wind power. The use of an AHP will lead to a higher overall load of the CHP unit to satisfy the same amount of heat and power, and this makes the energy-saving potential of AHP-based system smaller than that with EHP. The lower the power demand is, the bigger differences the energy-saving potential will be. Under this working condition, the standard coal consumption for power and heat supply with EHP-based waste heat recovery system is 1.08t/h and 4.30t/h lower than that with AHP-based system, and the overall standard coal consumption is thus reduced by 5.38t/h.

To conclude, the system with the EHP waste recovery system is better than that with the AHP system, in terms of heat-power decoupling and the advantage is more clearer as the power demand decreases and/or heat demand increases. It is found that COP is a sensitive factor for the EHP based system to be energy efficient. For the studied CHP unit and under the DHN return water temperature at around 65°C and supply temperatures at about 90°C, COP_{EHP} is 5.06, however if for some reasons, e.g. incorrect design or manufacturing defects, the COP_{EHP} is lower than this value, the energy saving merit compared to the AHP based system will be reduced. If the real COP_{EHP} is decreased to 4.6, the energy saving potential of the EHP based system becomes smaller than the AHP based system under some working conditions of high power demand, although the heat-power decoupling potential is still better than the latter. If the real COP_{EHP} is decreased to 4.1, it is even not energy efficient compared to the traditional system with no HP. Therefore it is of vital importance to match the parameters and choosing the working conditions for designing and operating an EHP based waste heat recovery system considering the requirement of heat-power decoupling at the same time.

8. Conclusions

This paper proposes to use an electric heat pump (EHP) in a CHP plant to recover the waste heat from cooling water and simultaneously help the CHP unit obtain the capabilities of heat-power decoupling as well as energy saving. The heat-power decoupling, as well as energy-saving potentials between the EHP and the absorption heat pump (AHP), are also compared. The major conclusions are as follows.

1) The use of large-scale heat pumps (e.g., EHPs and AHPs) to recover waste heat can extend the feasible operating region of the CHP unit because the upper power output increases while the lower power output decreases as the heat output increases. Therefore, the thermo-electric coupling property is relaxed to some extent, and thus the heat-power decoupling effect can be obtained according to different working conditions and the heat pump DH ratio χ_{HP}. When the heat demand is stable, the heat-power decoupling effect will be better with higher χ_{HP}. And when χ_{HP} is determined, the decoupling potential is bigger along with the increasing heat demand. Therefore, the power output scope is wider, and thus the heat-power decoupling effect is bigger when χ_{HP} and/or the heat load are increasing.
2) The integration of an HP-based waste heat recovery system can bring energy-saving benefits to the whole system, provided that the coefficient of performance (COP) is bigger than the critical value under different working conditions. In the case study, when the EHP provides 60MW of heating (χ_{HP}=0.18), the critical COP_{EHP} is 4.1, which means that the whole system can save energy if COP_{EHP} is larger than 4.1. Otherwise, the system will consume more energy than the traditional system (χ_{HP}=0). The larger the χ_{HP} is, the smaller the critical COP_{EHP} will be.
3) Since COP_{AHP} is much lower compared to COP_{EHP}, χ_{HP} needs to be much higher to make the whole system save energy and be economically feasible. But χ_{HP} should not be too large since the CHP unit should also satisfy the power demand at the same time, and the main purpose of the HP-based waste heat recovery system is to increase the flexibility of cogeneration while satisfying the same heat and power demands. Specifically, when χ_{HP}=0.5 and the COP_{EHP} is 5.06, the energy-saving potential of the whole system is about 4% compared to the traditional system.
4) The energy-saving effect can be increased either by increasing the COP or χ_{HP}. In addition, the energy-saving potentials are also influenced by the working conditions, i.e., different combinations of heat and power outputs. The energy-saving effect becomes greater as the heat demand increases and the power demand decreases. And the power demand is more sensitive for the energy-saving potential. It is better to let the whole system operate more extensively under the conditions of high heat demand and low power demand in order to reach a better energy-saving potential.
5) Both the AHP and EHP can help the system obtain a certain level of heat-power decoupling as well as energy-saving effects. The use of the AHP will lead to a higher overall load of the CHP unit to satisfy the same amount of heat and power. This makes the energy-saving potential of the AHP-based system smaller than that of the EHP. And the lower the power demand is, the

bigger differences the energy-saving potential will be. Under the working conditions of (P_D, Q_D)=(250MW, 350MW) and (195.86MW, 350MW), the total standard coal consumption with the EHP-based waste heat recovery system is 0.63t/h and 5.38t/h lower than that with the AHP-based system. And in the latter working condition, the forced power outputs of the system with the AHP and EHP are reduced by 9.7MW and 21.18MW, respectively. To conclude, the system with the EHP waste recovery system is better than that with the AHP system in terms of heat-power decoupling and the advantage is clearer as the power demand decreases. COP_{EHP} is a sensitive factor affecting the energy saving potential of EHP based system. If COP_{EHP} is lower than the model value because of e.g. incorrect design or manufacturing defects, the energy saving merit compared to the AHP based system will be reduced. Therefore it is of vital importance to match the parameters and choosing the working conditions for designing and operating an EHP based waste heat recovery system considering the requirement of heat-power decoupling at the same time. In the future, the economic performances of the two HP based system should also be analyzed considering the peak-shaving services provided for the power grid for making deliberate decision.

Acknowledgments

This work was supported by the China national key research and development program – China-Finland intergovernmental cooperation in science and technology innovation (Funding No. 2021YFE0116200) and academy research fellow funding from Academy of Finland (Funding No. 336268 and 334205).

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Nomenclature

Abbreviation	
AHP	Absorption heat pump
AHS	Auxiliary heat source
CAGR	Compound annual growth rate
CHP	Combined heat and power
COP	Coefficient of performance
DH	District heating
DHN	District heating network
EB	Electric boiler
EHP	Electric heat pump
HP	Heat pump
P2H	Power to heat
RES	Renewable energy source
TES	Thermal energy storage
THA	Turbine heat acceptance
VWO	Valve wide open

Symbols	
B	Total standard coal consumption of the CHP system, t/h
B_h	Standard coal consumption of CHP plant for heating, t/h
B_e	Standard coal consumption of the CHP plant for electricity generation, t/h
B_0	Standard coal consumption of the CHP unit without HP, t/h
c	Specific heat capacity, J/(kg·K)
D_0	Main steam flow entering the steam turbine, kg/s or t/h
D_h	Steam flow for heating, kg/s or t/h
D_p	Exhausted steam flow, kg/s or t/h
$D_{1,o}$	Outlet steam flow of the governing stage, kg/s or t/h
$D_{l,\min}$	Minimum steam intake of the turbine, kg/s or t/h
D_{rh}	Amount of reheat steam of the turbines, kg/s or t/h
D_g	Driving steam flow of AHP, kg/s or t/h
$G_{DHN,i}$	Water flow into the heat pump system at the DHN side, kg/s or t/h
$G_{cooling,i}$	Circulating cooling water flow entering the heat pump, kg/s or t/h
h	Specific enthalpy, J/kg
H_S	The isentropic enthalpy of steam, J/kg
p	Steam pressure, Pa
P	Power output or power consumption, W/MW
Q	Heat output or heat consumption, W
$Q_{thermal}$	Heat consumption of CHP unit, J
$q_{thermal}$	Heat consumption rate of CHP unit, kJ/kWh
q_{rh}	Enthalpy increase of the reheated steam, J/kg
q_{net}	Net calorific value of standard coal, J/kg
$Q_{condens}$	Waste heat of exhaust steam in the condenser, W
Q_g	Heat energy of driving steam, W
t	Temperature, K

z	Number of turbine stages
α	Heat-to-electricity ratio
β	Efficiency factor of the EHP
χ_{HP}	The heating ratio of the HP-based waste heat recovery systems, 1
η	Efficiency, %

Subscript

min	Minimum value
max	Maximum value
AHP	Relating AHP
HP	Relating HP
EHP	Relating EHP
w	Water
$'$	Design value
re	Recovered
D	Demand
i	Inlet
o	Outlet
w	water
h	heating
l	low
rh	reheat
ht	heat transfer
hx	heat exchanger
ex	exhaust steam
re	recovered