Modeling and Off-Design Performance Analysis of a Screw Expander-Based Steam Pressure Energy Recovery System in a Combined Heat and Power Unit

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ABSTRACT: A screw expander-based heating system is proposed based on a 330 MW combined heat and power unit to recover the extraction steam pressure energy. EBSILON Professional software was applied to model the proposed system, and the thermal performance of the traditional system and the new system was compared under different operating conditions. The results show that under the designed heating condition, the standard coal consumption rate of the new system is reduced by 4.74 g/kWh, and the exergy efficiency of the heating process is increased by 17.29%. When the extraction steam pressure changes with the load within a range greater than 0.1 MPa, there is an optimal operating range for the new system, which is related to the built-in pressure ratio of the screw expander and the extraction steam pressure difference, and the distribution of the optimal operating range varies with the supply-water temperature. The heating process exergy efficiency of the two systems shows opposite trends under off-design conditions. Additionally, under the condition of meeting the extraction steam pressure, the method for increasing the output power of the screw expander by adjusting the butterfly valve to increase the extraction steam pressure only increases the standard coal consumption rate of the system. Furthermore, through an application case study, the annual economic benefit of the new system is expected to be ¥ 4.15 million, and the payback period is 5.02 years.

1. INTRODUCTION

With the augmentation of the energy crisis and environmental problems, researchers have been accelerating the development of new energy while also looking for various ways to achieve energy conservation.1 Although renewable energy sources, such as wind energy, solar energy, and biomass, have developed rapidly in recent years, their power consumption is low in practice due to their high cost, unstable power generation, and other issues.2,3 In this case, the energy conservation transformation of the industrial process has become an important way to reduce fuel consumption, lower harmful emissions, and improve production efficiency.4

As an effective method to achieve energy-saving, pressure energy recovery has been widely considered in recent years, with applications including gas pressure reduction stations, air conditioning systems, fuel cells, heat pumps, steam pipes, etc.5−9 Additionally, the research and application of gas pressure energy recovery has been more frequent. Usually, the gas pressure regulating station uses a throttling valve or a pressure control valve to reduce the gas pressure in the transmission pipeline and to convey it to the user. To recover this part of the pressure difference, an expander can be used to replace the original pressure-reducing valve,10 thus realizing the transformation of electrical energy and cold energy.11−15

The steam pressure energy recovery principle is similar to that of gas pressure energy recovery, and there are many application scenarios in industry and commercial processes.16−18 This paper focuses on the pressure energy recovery in the heating pipeline of a combined heat and power (CHP) unit. In regular CHP units, the fifth stage of extraction steam is generally adopted for heating the supply water in the heater for the heating network (HHN), that is, the extraction port is arranged on the connecting pipe between the intermediate-pressure (IP) and low-pressure (LP) cylinders of a steam turbine.19 The exhaust pressure of the IP cylinder generally ranges from 0.2∼0.5 MPa, where the extraction steam is sent to the HHN to heat the return water, and the designed supply-water temperature is approximately 130 °C. Because of climate change and energy-saving...
technology in building construction, the actual supply-water temperature is approximately 90 °C, and the HHN can meet the heat exchange demand when the terminal temperature difference is approximately 10 °C. Thus, the steam pressure needed by the HHN is approximately 0.1 MPa, and the heating pipeline pressure difference reaches 0.1–0.4 MPa. In practice, the high-pressure extraction steam directly enters the HHN for heat exchange, resulting in great exergy loss.

To reduce the exergy loss of the heating process, many options have been proposed and applied to realize the energy cascade utilization. Some researchers proposed to use an absorption heat pump (AHP) for heat supply, which can achieve multiple purposes of waste heat recovery, reducing irreversible heat exchange loss of the heating network, and improving heating load. The thermodynamic and parametric characteristics of the AHP system in CHP units have been investigated by a few authors. Another approach to increase the exergy efficiency of the heating process is the high-back-pressure (HBP) heating. An HBP heating system is constructed of two heating sections in which the supply water can first be heated by the exhaust steam from the LP cylinder and then by the extraction steam in the HHN. By doing this, both the heat capacity and energy utilization efficiency of the CHP units are increased. Although the above approaches can effectively improve the exergy efficiency of the heating process, the system structure is complicated, and its engineering application remains to be verified.

Using an expander, the extraction steam can heat the return water in the HHN after being used for power generation. Some work has been carried on the overall performance of the expander-based heating system. The expander efficiency is normally assumed as constant for system performance estimation, which is plausible for a system operating at a steady point. However, a CHP unit often works under off-design conditions due to the variations of the load. Therefore, it is necessary to combine the off-design performance of the expander with that of the unit to analyze the operating characteristics of the system. In addition, turbines are the most commonly selected expanders; however, for medium- and low-size systems, the volumetric expanders are generally preferred because the turbine component efficiencies are relatively low in these cases. In addition, it is noteworthy that in the application of the pressure energy recovery in the heating pipeline of the CHP unit, the operation conditions of the expander fluctuate with the unit load, and the steam may condense during the expansion process under a low unit load. Turbines have a low efficiency under off-design conditions, and liquid droplets in the steam adversely affect their safe operation.

As one of the most commonly used volumetric expanders, screw expanders (SEs) already have applications in geothermal power generation and waste heat recovery. Compared with turbines, the SE has the following advantages. First, as a full-flow power machine, it accepts not only superheated and dry saturated steam but also wet steam and vapor–liquid mixtures, and the presence of liquid droplets in the machine has little effect on its life. Second, the SE exhibits a rapid start up and shut down and a high efficiency under the off-design condition. Furthermore, the rotation speed is low, and the structure is simple. These advantages enable the SE to be well adapted to the mid–low temperature conditions of the steam Rankine cycle. Iodice et al. proposed a solar thermal power generation system based on the steam Rankine cycle in which SEs and parabolic trough collectors were used. They found that with increasing solar radiation, the optimum evaporation temperature ranges from approximately 204 to 276 °C, and the corresponding maximum solar thermal power efficiency and exergy efficiency are approximately 12.6 and 17.8%, respectively. A similar system was also studied by Li et al. in which tandem SE technology is applied. The results show that when the built-in volume ratio is 3 (high-pressure SE) and 7 (low-pressure SE), the efficiency of the steam Rankine cycle is 18.49%, and the payback period (PP) is 10.48 years for a 1 MW tandem plant. Li et al. studied a solar electricity generation system that uses a cascade steam-organic Rankine cycle and an SE, and by comparing direct and indirect steam generation, they found that the former mode has advantages in thermal performance and economy. Tian et al. used an SE to recover the pressure difference in a steam pipe and studied the operation characteristics of the expander under different working conditions. The fill factor varies from 0.82 to 0.88, and the isentropic efficiency varies from 0.73 to 0.83, which proves the SE to be an effective steam pressure recovery technology.

![System schematic diagram.](https://doi.org/10.1021/acsomega.1c04846)

**Figure 1.** System schematic diagram.

| 1 Boiler | 2 High-pressure (HP) cylinder | 3 Intermediate-pressure (IP) cylinder |
|----------|-------------------------------|--------------------------------------|
| 4 Butterfly valve | 5 Low-pressure (LP) cylinder | 6 Generator one (G1) |
| 7 Condenser | 8 Low-pressure heater | 9 Boiler feed water pump turbine |
| 10 Deaerator | 11 Feed water pump | 12 High-pressure heater |
| 13 SE | 14 HHN | 15 Generator two(G2) |
| 16 Industrial extraction steam | × Stage of extraction steam |
The pressure difference in the heating pipeline of a CHP unit is relatively small, and the extraction steam parameters are not high and fluctuate with the unit load. The high tolerance of the multiphase and excellent part-load behavior of an SE can meet the system characteristics well. Therefore, this study selected the SE to be applied to the CHP unit for recovering the pressure energy of the heating process. The objectives of this work are as follows: (1) build a thermodynamic model to reveal the energy-saving mechanism of the system; (2) compare the performance of the new system (NS) and the traditional system (TS) under off-design conditions; (3) determine and discuss the influence of the extraction steam pressure on the system performance; and (4) analyze the economic benefits of the system through a practical application. A technological foundation for the application of the SE in an extraction steam pressure energy recovery system may be provided in this work.

2. SYSTEM DESCRIPTION

The system schematic diagram is shown in Figure 1, and the main parameters of the 330 MW coal-fired CHP unit adopted in the study are shown in Table 1. The fifth stage of extraction steam is adopted for heating the supply water. The supply-water temperature of the primary heating network is 75 °C. In the TS, the high-pressure extraction steam directly enters the HHN for heat exchange, and the terminal temperature difference of the HHN depends on the extraction steam pressure. In the NS, the extraction steam first enters an SE for power generation and then enters the HHN for heat exchange, the terminal temperature difference of the HHN is kept at 11 °C and the steam pressure in the HHN is 0.06 MPa. EBSILON Professional software38 was used to conduct the simulations of the studied power system. EBSILON Professional is an expert in the design, analysis, and optimization of thermodynamic cycles, which takes advantage of a matrix solution method and requires the linearization of all dependencies. Table 2 shows the model details of the selected main components in this study. Based on mass and energy balance, the simulation model of the system was developed, as shown in Figure A1, Appendix A.

3. MODEL BUILDING

3.1. Twin-Screw Expander Modeling. The ideal working process of an SE is adiabatic and isentropic, but the actual working process is affected by the flow loss and leakage loss, which increases the entropy value after the actual expansion. Ideally, the working fluid undergoes three processes of constant pressure admission, isentropic expansion, and constant pressure discharge in the SE,35,36 as shown in process 2→3→4→1 in Figure 2. In this process, 3→4 is isentropic expansion along the isentropic index k. Because of the irreversibility of the thermal process and pressure mismatch in the actual operation process, losses will occur;19,40 specifically, 2→3′ denotes the inlet process with admission losses, 3′→4′ or 3→4″ is the expansion process with irreversibility losses, blowdown (4′→5′) or blowback (4→5″) effects may be caused by a pressure mismatch, and 5′→1 or 5″→1 refers to the discharge process with discharge loss. In addition, there are mechanical losses due to mechanical friction.

The overall isentropic efficiency of an SE can be defined as:

\[ \eta_{SE} = \frac{\eta_{th} \cdot \eta_{t} \cdot \eta_{lm} \cdot \eta_{m}}{\eta_{th} \cdot \eta_{t} \cdot \eta_{l} \cdot \eta_{m}} \]  

(1)

where

\[ \eta_{th} = \frac{\eta_{th}}{\eta_{lm}} \]  

(2)

The built-in pressure ratio \( r_p \) and the built-in volume ratio \( r_v \) are shown in eqs 3 and 4, respectively:41

\[ r_p = \frac{P_3}{P_4} \]  

(3)

\[ r_v = \frac{v_4}{v_3} \]  

(4)

\[ r_p = r_v^k \]  

(5)

The theoretical efficiency \( \eta_{th} \) can be calculated as shown in eq 6:35,41
The work generated by an SE is defined as:

$$w_{SE} = \frac{G}{3600}(h_{in} - h_{out}) = \frac{G}{3600}(h_{in} - h_{out,ic})\eta_{SE}$$

where $\eta_{SE}$, $\eta_{boiler}$, $\eta_{ff}$, $\eta_{te}, \eta_{th,D}, \eta_{SE,P}$, and $\eta_{th,p}$ are the overall isentropic efficiency, theoretical efficiency, thermodynamic efficiency, leakage efficiency (due to admission and discharge losses), mechanical efficiency, diagram efficiency, peak overall isentropic efficiency, and peak theoretical efficiency of the SE, %, respectively. $P_1$ and $P_2$ are the inlet pressure and built-in expansion pressure, MPa, respectively; $r_{in}, r_{out}$, and $r_{p}$ are the built-in pressure ratio, built-in volume ratio, and operation pressure ratio, respectively; $v$ is the specific volume, m$^3$/kg; $k$ is the entropy index; $h_{in}$ and $h_{out}$ are the enthalpies of working fluid at the SE inlet and the SE outlet,kJ/kg, respectively; and $G$ is the steam flow rate, t/h.

The parameters of the SE used in the model are shown in Table 3.

### 3.2. CHP Unit Modeling

The operation state of a steam turbine under the off-design condition is calculated by the Fligiel formula:

$$\eta_{th} = \frac{(1 - r_{in}^{1-k}) + (k - 1)(1 - \frac{v}{v_r})}{k(1 - r_{out}^{1-k})} \times 100\%$$

(6)

The peak overall isentropic efficiency $\eta_{SE,P}$ attained at the peak theoretical efficiency $\eta_{th,p}$ can be defined as:

$$\eta_{SE,P} = \eta_{th,p} \cdot \eta_{D} \cdot \eta_{in}$$

(7)

$$\eta_{SE} = \eta_{SE,P} \cdot \eta_{th} = \eta_{SE,P} \cdot \frac{(1 - r_{in}^{1-k}) + (k - 1)(1 - \frac{v}{v_r})}{k(1 - r_{out}^{1-k})}$$

(8)

For CHP units, the total heat consumption $Q$ (MW) is expressed as:

$$Q = \frac{G_b(h_{in} - h_{wb}) + G_f(h_{in} - h_{fw})}{3600\eta_b \eta_{pp}}$$

(11)

where $G_b$, $G_f$, $h_{in}$, $h_{wb}$, and $h_{fw}$ are the live steam and reheat steam flow rates, respectively, t/h; $h_{in}$, $h_{fw}$, $h_{wb}$, and $h_{rh,c}$ are the enthalpies of the live steam, feed water, reheated steam, and cold reheat steam, respectively, kJ/kg; $\eta_b$ and $\eta_{pp}$ represent the boiler efficiency and the pipeline thermal efficiency, respectively.

The heat consumption for heating (MW) is given by:

$$Q_h = \frac{G_{ex}(h_{ex} - h_{cd})}{3600\eta_b \eta_{pp}}$$

(12)

where $G_{ex}$ is the flow rate of the extraction steam for heating, t/h; $h_{ex}$ and $h_{cd}$ denote the enthalpies of the exhaust steam and the condensate in the HHN, respectively, kJ/kg.

The heat consumption for generation (MW) can be expressed as:

$$Q_g = Q - Q_h$$

(13)

The electricity generation efficiency is given as:

$$\eta_g = \frac{w_g}{Q_g} \times 100\%$$

(14)

where $w_g$ is the net electricity output, MW.

To verify the accuracy of the model, taking VWO (valve wide open) as the design working condition, the model under 100% THA (turbine heat acceptance), 75% THA, 50% THA, and 40% THA conditions are calculated. The results are compared with the design data given by the manufacturer, as shown in Table 4, and the maximum relative error is 0.43%, which proves the simulation available and credible.
where $G_{\text{fl}}$ is the flow rate of the LP cylinder exhaust steam, t/h; $h_{\text{exh}}$ and $h_{\text{cond}}$ are the enthalpies of the LP cylinder exhaust steam and the condensate in the condenser, kJ/kg.

### 4.2. Exergy Analysis

Exergy represents the maximum capacity of a system to perform useful work when it proceeds to a given final state in equilibrium with the environment. It is a more scientific index that is based on the second law of thermodynamics, which can identify the locations, magnitudes, and sources of the thermodynamic inefficiencies in a thermal system.\(^{40,43}\)

The exergy of steam (MW) is calculated by:

$$
EX = \frac{G \times [(h - h_{\text{en}}) - T_{\text{en}} \times (s - s_{\text{en}})]}{3600}
$$

(17)

where $h$ and $h_{\text{en}}$ are the specific enthalpies of the steam at the current state and the environmental state, respectively, kJ/kg; $s$ and $s_{\text{en}}$ are the specific entropies of the steam at the current state and the environmental state, respectively, kJ/kgK; and $T_{\text{en}}$ is the environmental temperature, K.

The heating process of the system is shown in Figure 3. For the NS, the exergy loss during the heating process includes the exergy loss of the SE and the exergy loss of the HHN.

$$
\Delta EX_{\text{NS, loss}} = \Delta EX_{\text{SE, loss}} + \Delta EX_{\text{HHN, loss}}
$$

(18)

$$
= (EX_{1} - EX_{2} - w_{SE}) + (EX_{2} + EX_{4} - EX_{3} - EX_{\text{c}})
$$

(19)

For the TS, the exergy loss during the heating process is only the exergy loss of the HHN.

$$
\Delta EX_{\text{TS, loss}} = \Delta EX_{\text{HHN, loss}}
$$

(19)

The exergy efficiency of the heating process is given as:

$$
n_{\text{ex}} = \frac{w_{SE} + EX_{3} - EX_{4}}{EX_{1} - EX_{3}} \times 100\%
$$

(20)

where $w_{SE} = 0$ MW and $EX_{1} = EX_{2}$ in the TS.

### 5. RESULTS AND DISCUSSION

#### 5.1. Overall Analysis

Under certain conditions of the live steam parameters, the extraction steam parameters change with the heating load. By adjusting the butterfly valve on the connecting pipe of the IP and LP cylinders, the extraction steam pressure can be maintained above a certain value. To study the performance of the system when the extraction steam is under a sliding pressure on the premise of meeting the pressure demand of the HHN, the minimum pressure at the extraction port is set as 0.1 MPa. When the pressure is higher than 0.1 MPa, the butterfly valve remains fully open. When the pressure is lower than this value, the butterfly valve begins to turn down so that the extraction steam pressure is maintained at 0.1 MPa.

Table 5 shows the performance indicator comparison under the designed heating condition (all the difference calculations in this paper are equal to the value of the NS minus that of the TS). The live steam flow rate remains at 1054.3 t/h, and the heating load remains at 100 MW. Because of the energy consumed by the SE for power generation, the extraction steam flow rate of the NS is increased by 9.92 t/h, and the power from generator one (G1) is reduced by 1.15 MW. However, the total power output of the NS is increased by 5.74 MW with a 6.89 MW output of the SE, and the corresponding standard coal consumption rate is reduced by 4.74 g/kWh. The detailed key data of the unit at a 100 MW heating load at different live steam flow rates are provided in Table A1, Appendix B.

#### 5.2. Energy-Saving Mechanism

Figure 4 shows the detailed energy balance diagram of the system under the designed heating condition. The pressure difference on the extraction pipe is recovered in the NS, which has little influence on the cylinders front of the steam extraction port, so the energy flows of this part for the two systems are basically the same (the input energy from the coal is calculated by eq 10, the slight difference in input energy between the two systems is due to the difference of the reheat energy and the difference of the feed water enthalpy caused by the extraction steam flow rate difference). With the addition of the SE, the extraction steam flow rate in the NS is increased, and its energy for heating is increased from 108.73 MW of the TS to 116.37 MW. For this part of the energy, 7.01 MW is used by the SE for power generation. In the TS, the extraction steam with an energy of 7.64 MW is conserved and enters the condenser after driving the rotor in the LP cylinder, resulting in 1.15 MW more electricity generated by G1 and 5.67 MW more loss of the condenser. According to the energy balances of the system, after adding the SE, the total power generation is increased by 5.74 MW, which is nearly equal to the energy loss reduction (5.67 MW).

Figure 5 shows the energy flow differences in the NS. Although the extraction steam heat of the TS is smaller than that of the NS, most of the reduced heat (about 80%) is dissipated in the condenser after driving the rotor in the LP cylinder. It is obvious from the figure that the increased total power generation and the decreased heat loss in the condenser of the NS are basically the same. Combined with Figure 4, it can be concluded that the increased total power generation is mainly determined by the reduction of heat loss in the condenser.

The graphical exergy analysis\(^{38}\) of the HHN was carried out under the designed heating condition as shown in Figure 6. While the heating load is maintained constant, the exergy loss of the HHN in the NS is reduced by 7.3 MW compared with that in the TS. The total exergy loss during the heating process is reduced by 4.92 MW, and the exergy efficiency of the heating process is increased by 17.29%, which indicates that the NS improves the overall performance of the heating process by recovering the pressure difference with more sufficient energy cascade utilization.

#### 5.3. Performance Indicator Analysis

When the extraction steam is under a sliding pressure above 0.1 MPa, the
operation performance of the two systems is compared under different conditions. Figure 7 shows the power generation of the two systems under a heating load of 100 MW. The total power generation of the NS is larger than that of the TS as mentioned above. With the increase of the live steam flow rate, the power output of the SE increases due to the increase of the extraction steam pressure. The total power increment also augments with the increase of the live steam flow rate.

The extraction steam flow rates of the two systems under a 100 MW heating load are depicted in Figure 8. With the decrease of the live steam flow rate, the extraction steam flow rate of the NS decreases while that of the TS increases. For the TS, since the extraction steam pressure decreases with the live steam flow rate, the terminal temperature difference of the HHN decreases and the extraction flow rate increases. When the live steam flow rate is lower than 750 t/h, the extraction steam pressure is lower than 0.1 MPa, the butterfly valve begins to turn down so that the extraction steam pressure is maintained at 0.1 MPa. Since the inlet steam of the IP cylinder is the reheated steam, the temperature remains unchanged. When the live steam flow rate continues to decrease, the inlet pressure of the IP cylinder decreases, while the outlet pressure remains unchanged, so the outlet temperature increases. The extraction steam flow rate of the TS decreases because of the rise in its temperature. For the NS, the terminal temperature difference of the HHN is maintained as 11 °C, the heating load remains unchanged and the power output of the SE decreases with the live steam flow rate as shown in Figure 7. Therefore, the extraction steam flow rate decreases with the decrease of the live steam flow rate. In addition, when the live steam flow rate is small, the power generation of the SE is also low, and the extraction steam flow rate difference between the two systems is small. With the increase of the live steam flow rate, the power generation of the SE increases, and the extraction steam flow rate difference also increases.

Figure 9 shows the standard coal consumption rate and power generation thermal efficiency of the system under a 100 MW heating load. As the live steam flow rate increases, the thermal efficiencies of the two systems declines and the thermal efficiency of the TS is lower than that of the NS.
efficiency improvement of the NS gets larger, resulting in that more fuel can be saved.

When the heating load remains unchanged, the live steam flow rate regulation is mainly to adjust the generation load of the unit, while the heating load of the unit, which is mainly adjusted by the extraction flow rate regulation also changes during operation. Both loads will affect the performance of the unit.

Figure 10 presents the extraction steam pressure (at the extraction port) difference between the two systems at different heating loads and flow rates of the live steam. It is apparent that the pressure difference increases with the live steam flow rate at a large heating load. This increase is because the extraction steam pressure increases with the live steam flow rate, so the output power of the SE also increases, and the extraction steam flow rate difference between the two systems increases, resulting in an increase in the extraction pressure difference. When the heating load is low, the extraction steam flow rate is small. Even if the live steam flow rate is increased, the output power of the SE is also small, and the extraction steam flow rate difference between the two systems is small, so the pressure difference is basically unchanged.

Under a certain live steam flow rate, the extraction steam pressure difference between the two systems first increases and then decreases with an increase in the heating load. This change is because when the heating load is small, the extraction steam flow rate is small and the power output of the SE is correspondingly low, so the extraction steam pressure difference between the two systems is also small. With an increase in the heating load, the output power of the SE increases correspondingly, and the pressure difference between the two systems also increases. However, a further increase in the heating load causes the extraction steam pressure to decrease. Even if the extraction steam flow rate is large, the efficiency of the SE

Figure 5. Energy flow difference in the NS.

Figure 6. Graphical exergy analysis of the HHN. (a) Traditional system. (b) New system.

Figure 7. Power generation of the two systems under a 100 MW heating load.

Figure 8. Extraction flow rate at different live steam flow rates.
decreases, and its output power is limited, so the pressure difference between the two systems is also reduced. The increase in the live steam flow rate provides support for the maintenance of the extraction steam pressure, so the region with a larger pressure difference extends in the direction of the high heating load with an increase in the live steam flow rate.

The standard coal consumption rate difference between the two systems is shown in Figure 11. In order to explore the performance of the NS under different supply-water temperature requirements, the pressure in the HHN is set as 0.04 MPa (supply-water temperature is 65 °C), 0.06 MPa (supply-water temperature is 75 °C), and 0.09 MPa (supply-water temperature is 85 °C), corresponding to Figure 11a−c, respectively. It is apparent that the distribution characteristics of Figure 11 are consistent with the extraction steam pressure difference between the two systems. As with the above analysis, it is mainly related to the output power of the SE. Except for some conditions with a high heating load and a low live steam flow rate or low heating load such that the NS is consistent with the standard coal consumption rate of the TS (under these conditions, the output of the SE is small because of a low-pressure ratio, low flow rate, or the unit does not work due to the low steam intake of the LP.

Figure 11. Coal consumption rate difference at different heating loads and live steam flow rates. (a) Pressure of the steam in the HHN is 0.04 MPa. (b) Pressure of the steam in the HHN is 0.06 MPa. (c) Pressure of the steam in the HHN is 0.09 MPa.
cylinder), the standard coal consumption rate of the NS under all other conditions is lower than that of the TS.

It is apparent from Figure 11 that there is an optimal operating range for the NS in which the standard coal consumption rate can be reduced more (greater than or equal to three-fourths of the maximum) than that of the TS. Comparing Figure 11a–c, when the pressure of the steam in the HHN increases from 0.04 to 0.09 MPa, the optimal operation range of the system moves downward. This trend is because the pressure of the steam in the HHN, i.e., the exhaust pressure of the SE, increases, and under a certain live steam flow rate, the pressure difference between the extraction port and the HHN can match the built-in pressure ratio of the SE at a lower heating load. Therefore, when the SE is applied to a system, it should be selected according to the daily operation load law of the system to make the built-in pressure ratio match the extraction steam pressure difference so that the optimum operation range can be within the normal load range of the system and the energy-saving benefits can be maximized.

The exergy efficiency of the heating process is shown in Figure 12. For the TS, it decreases with an increasing live steam flow rate. This result is mainly because under a certain heating load, the extraction pressure increases with the live steam flow rate, so the terminal temperature difference of the HHN in the TS increases, and the energy transfer irreversibility of the HHN increases. Of course, the extraction pressure remains at 0.1 MPa at a high heating load and a low live steam flow rate as mentioned above, so the exergy efficiency remains unchanged under these conditions. For the NS, the exergy efficiency of the heating process is mainly related to the SE operation efficiency. When the heating load is low, the pressure of the extraction steam is high, and an increase in the live steam flow rate will continue to increase the extraction steam pressure, which makes the operating pressure ratio first approach and then deviate from the built-in pressure ratio of the SE, so the exergy efficiency first increases and then decreases. When the heating load is high, the pressure of the extraction steam is low. With an increase in the live steam flow rate, the operating pressure ratio gradually approaches the built-in pressure ratio of the SE, so the efficiency of the SE is improved, and the corresponding exergy efficiency of the heating process is also improved. The maximum exergy efficiency of the heating process of the TS is 70.17% when the heating load is 150 MW and the live steam flow rate is 850 t/h. This efficiency decreases slightly with a decrease in the live steam flow rate, and the reason can be provided with reference to Figure 8. For the NS, the lowest exergy efficiency of the heating process is 72.18%, when the live steam flow rate is 750 t/h and the heating load is 150 or 125 MW.

The difference between the exergy efficiencies of the heating processes of the two systems is shown in Figure 13. The exergy efficiency of the heating process of the NS is higher than that of the TS. Additionally, the exergy efficiency increment increases from the upper left corner to the lower right corner of Figure 13, which implies that the larger the available extraction steam

![Figure 13. Exergy efficiency difference of the heating process between the two systems.](image1)

![Figure 14. Overall isentropic efficiency of the SE at different heating loads and extraction steam pressure limits. (a) Live steam flow rate is 1054.3 t/h. (b) Live steam flow rate is 750 t/h.](image2)

![Figure 15. Coal consumption difference at different extraction steam pressure limits.](image3)
pressure difference, the more advantageous the NS. Of course, when the heating load is high, the exergy efficiency difference first decreases and then increases with a decrease in the live steam flow rate because of the slight decrease in exergy efficiency of the heating processes of the TS, as mentioned above. The exergy efficiency difference reaches the maximum value of 20.06% under the condition of a 1054.3 t/h live steam flow rate at a 25 MW heating load.

5.4. Effect of the Pressure Limits. The adjustment of the butterfly valve has a great influence on the whole system. To study the regulation, the minimum pressure of the extraction steam is set to 0.5, 0.4, 0.3, 0.2, and 0.1 MPa. When the extraction steam pressure is lower than the set value, the butterfly valve will start to turn down to maintain the extraction steam pressure at the set value.

Figure 16. IP cylinder output of the NS at different heating loads and different extraction steam pressure limits. (a) Live steam flow rate is 1054.3 t/h. (b) Live steam flow rate is 750 t/h.

Figure 17. SE output at different heating loads and different extraction steam pressure limits. (a) Live steam flow rate is 1054.3 t/h. (b) Live steam flow rate is 750 t/h.

Figure 18. Coal consumption rate of the NS at different heating loads and pressure limits. (a) Live steam flow rate is 1054.3 t/h. (b) Live steam flow rate is 750 t/h.
The built-in pressure ratio of the SE is 4.17. Therefore, the $\eta_{SE}$ is maintained at a high level when the extraction steam pressure limit is close to 0.25 MPa. When the live steam flow rate is 1054.3 t/h and the pressure limits are 0.3, 0.4, and 0.5 MPa, respectively, the extraction steam pressure is lower than the pressure limit. Therefore, with the increase of the heating load, the extraction steam pressure is maintained at the pressure limit, and the $\eta_{SE}$ remains unchanged because of the constant operating pressure ratio (it is similar when the live steam flow rate is 750 t/h and the pressure limits are 0.2, 0.3, 0.4, and 0.5 MPa, respectively). When the pressure limits are 0.1 and 0.2 MPa, the extraction steam pressure is higher than 0.25 MPa. An increase in the heating load continues to decrease the extraction steam pressure, so the $\eta_{SE}$ first increases and then decreases as mentioned in the explanation of Figure 12. When the live steam flow rate is 750 t/h and the pressure limit is 0.1 MPa, the extraction steam pressure is lower than 0.25 MPa. With an increase in the heating load, the extraction steam pressure decreases and the operating pressure ratio deviates from the built-in pressure ratio of the SE, so the $\eta_{SE}$ decreases.

Figure 15 shows the difference between the standard coal consumption rates of the two systems. It is apparent that when the pressure limit is low, the standard coal consumption rate difference increases with the live steam flow rate, as described in Figure 11. When the pressure limit is high, the difference in the standard coal consumption rates decreases with an increasing live steam flow rate. This change is because the pressure limit is high, and under a certain heating load, the extraction steam parameters remain basically unchanged with an increasing live steam flow rate. Therefore, the output power of the SE is basically unchanged, and its proportion in the total power generation is reduced, thus reducing the standard coal consumption rate difference. From the entire figure, the standard coal consumption rate difference increases in the direction of the increasing pressure limit, the decreasing live steam flow rate, and the increasing heating load. However, this result does not mean that a higher-pressure limit is better.

Figures 16 and 17 show the IP cylinder output and the SE output.

### Table 6. Main Parameters of the SE

| item                        | value |
|-----------------------------|-------|
| rated power (MW)            | 2.46  |
| rated inlet steam flow (t/h)| 50    |
| rated inlet steam pressure (MPa) | 0.25 |
| rated inlet steam temperature (°C) | 240 |
| rated exhaust steam pressure (MPa) | 0.07 |
| rated speed (rpm)           | 3000  |

### Table 7. First Cost of the Project

| system                      | subitem                  | value (thousand ¥) |
|-----------------------------|--------------------------|--------------------|
| thermodynamic system        | equipment cost           | 8612.30            |
|                            | installation cost        | 4053.60            |
|                            | construction cost        | 952.10             |
| electrical system           | equipment cost           | 274.20             |
|                            | installation cost        | 426.20             |
| thermal control system      | equipment cost           | 454.20             |
|                            | installation cost        | 623.10             |
| others                      | other cost               | 1081.40            |
| the whole system            | total cost               | 16,477.10          |

“Remarks: Other costs include construction site demolition and cleanup fees, project management fees, technical service fees for project construction, and trial operation fees.”

Figure A1. Simulation model of the system.
of the NS when the live steam flow rate is 1054.3 and 750 t/h, respectively. As the limit value of the extraction steam pressure, i.e., the exhaust pressure of the IP cylinder, increases, the IP cylinder output decreases. When the pressure limit is increased by 0.1 MPa, the IP cylinder output decreases about 10 MW (when the heating load is low and the pressure limit is low; the extraction steam pressure is higher than the pressure limit, so the power drop of the IP cylinder is smaller). The SE output increases with the pressure limit because of the increase of the pressure ratio and the increase of the P(H) as described in Figure 14. However, the increment of the SE output is lower than the reduction of the IP cylinder output.

Figure 18 shows the standard coal consumption rate of the NS when the live steam flow rates are 1054.3 and 750 t/h. It is apparent from this figure that under different heating loads, the standard coal consumption rate increases with the pressure limit. This is mainly due to the certain throttler loss of the butterfly valve and the decreased output of the IP cylinder. Although the output power of the SE is increased, the total generating power of the unit is reduced. Therefore, for extraction steam pressure difference recovery, it is not advisable to increase the extraction steam pressure through the butterfly valve to increase the output power of the SE under the condition of meeting the extraction steam pressure. It will only increase the standard coal consumption rate of the system, and the result is not worth the loss.

6. CASE STUDY

6.1. Description. The scheme proposed in this paper was used to reconstruct a CHP unit located in northern China, and the unit is identical to that studied above. At present, the extraction steam flow rate of a single unit is 100 t/h. During the actual operation of the unit, the butterfly valve on the connecting pipe of the IP and LP cylinders is fully opened, and the pressure of extraction steam into the HHN is adjusted by a throttle valve. According to the statistical average value of the data from the heating seasons in previous years, the extraction steam pressure is basically maintained at 0.25 MPa with a temperature of 245 °C in winter, and the parameters fluctuate slightly with the heating load. The retrofit was carried out on the No. 2 unit. Due to the limitation in the SE manufacturing size, the rated power of a single SE is small. Therefore, two SEs were used in parallel, and the output of the SE was used for auxiliary power. The main parameters of the SE are shown in Table 6, and Figure 19 shows the SE at the project site.

6.2. Economic Analysis. The PP can be calculated by:

$$PP = \ln \left( \frac{w_{net}C_{e} - C_{om}}{w_{net}C_{e} - C_{om} - jC_{tot}} \right) / \ln(1 + j)$$ (21)

where \( w_{net} \) is the net power generation, kWh; \( j \) is the interest rate with a value of 4.8%; \( C_{om}, C_{tot}, \) and \( C_{e} \) are the cost of operation and maintenance, initial capital cost, and electricity price, respectively, ¥, ¥/kWh. The value of \( C_{om} \) is 4% of the equipment cost.

The first cost of the project is listed in Table 7. Under the design condition, the generating power of the two SEs is 4.92 MW. The heating season of this case lasts 120 days, and the output of the two SEs can be 14169.6 MWh per year. Of course, under this working condition, after the SEs are put into operation, when the live steam flow rate and heating load remain unchanged, the power generation of G1 is reduced by 0.92 MW, so the net increased power generation in each heating season is 11,520 MWh. In 2017, the power plant’s annual power generation of the No. 2 unit was 1650 million kWh, the annual auxiliary power consumption of the No. 2 unit was 116 million kWh, and the auxiliary power consumption rate was 7.03%. After the SEs are put into operation, the auxiliary power consumption rate will be 6.33%, a decrease of 0.7%. The price of electricity is 0.36 ¥/kWh, the increased power generation profit will be ¥ 4.15 million per heating season, and the PP is 5.02 years.

7. CONCLUSIONS

A mechanism that uses the SE to recycle the pressure energy of the extraction steam in a CHP unit is proposed in this paper. The thermodynamic performance of the TS and the NS under different working conditions is compared based on a typical 330 MW CHP unit. In addition, the economic benefits of the system are analyzed through a practical application case. The following conclusions are derived:

Table A1. Detail Key Data of the Unit under a Heating Load of 100 MW

| item                              | live steam flow rate (t/h) | TS   | NS   | difference |
|-----------------------------------|---------------------------|------|------|------------|
| flow rate of the extraction steam for heating (t/h) | 1054.3 | 136.41 | 146.33 | 9.92       |
|                                   | 1000 | 137.16 | 146.22 | 9.06       |
|                                   | 950  | 137.66 | 145.85 | 8.19       |
|                                   | 900  | 138.27 | 145.38 | 7.11       |
|                                   | 850  | 139.03 | 144.77 | 5.74       |
|                                   | 800  | 139.97 | 143.94 | 3.97       |
|                                   | 750  | 141.21 | 142.75 | 1.54       |
|                                   | 700  | 140.97 | 142.00 | 1.03       |
| power output of the G1 (MW)       | 1054.3 | 281.65 | 280.50 | -1.15      |
|                                   | 1000 | 273.13 | 272.18 | -0.95      |
|                                   | 950  | 259.21 | 258.41 | -0.80      |
|                                   | 900  | 245.22 | 244.58 | -0.64      |
|                                   | 850  | 231.19 | 230.72 | -0.47      |
|                                   | 800  | 217.15 | 216.88 | -0.27      |
|                                   | 750  | 203.19 | 203.10 | -0.09      |
|                                   | 700  | 187.11 | 186.96 | -0.15      |
| power output of the SE (MW)       | 1054.3 | 6.89  | 6.89  |           |
|                                   | 1000 | 6.25  | 6.25  |           |
|                                   | 950  | 5.60  | 5.60  |           |
|                                   | 900  | 4.82  | 4.82  |           |
|                                   | 850  | 3.85  | 3.85  |           |
|                                   | 800  | 2.62  | 2.62  |           |
|                                   | 750  | 0.10  | 0.10  |           |
|                                   | 700  | 0.72  | 0.72  |           |
| total power output of the CHP unit (MW) | 1054.3 | 281.65 | 287.39 | 5.74       |
|                                   | 1000 | 273.13 | 278.43 | 5.30       |
|                                   | 950  | 259.21 | 264.01 | 4.80       |
|                                   | 900  | 245.22 | 249.40 | 4.18       |
|                                   | 850  | 231.19 | 234.57 | 3.38       |
|                                   | 800  | 217.15 | 219.49 | 2.34       |
|                                   | 750  | 203.19 | 204.10 | 0.91       |
|                                   | 700  | 187.11 | 187.69 | 0.58       |
| power generation standard coal consumption rate (g/kWh) | 1054.3 | 241.04 | 236.30 | -4.74      |
|                                   | 1000 | 235.17 | 230.78 | -4.39      |
|                                   | 950  | 231.62 | 227.48 | -4.14      |
|                                   | 900  | 227.53 | 223.79 | -3.74      |
|                                   | 850  | 222.79 | 219.63 | -3.16      |
|                                   | 800  | 217.20 | 214.92 | -2.28      |
|                                   | 750  | 210.52 | 209.60 | -0.92      |
|                                   | 700  | 204.38 | 203.76 | -0.62      |
Under the condition of constant live steam parameters and heating load, although the extraction steam flow rate is increased and the power generation of G1 is reduced by using the SE to recycle the pressure energy, the total power generation of the system still increases. The increased total power generation is mainly determined by the energy loss reduction in the condenser. Under the designed heating condition, the standard coal consumption rate of the NS is reduced by 4.74 g/kWh, and the exergy efficiency of the heating process is increased by 17.29%.

When the extraction steam pressure changes with the load, the NS has an optimal operation range. Therefore, the SE should be selected according to the daily operating load law of the system when it is used for recycling the extraction steam pressure difference so that its built-in pressure ratio is matched with the extraction steam pressure difference. In this way, the optimal operation range can be within the normal load range of the system, and the energy-saving benefits can be maximized.

A comparison of different extraction steam pressure limits shows that when the pressure limit is increased by 0.1 MPa, the IP cylinder output decreases about 10 MW. It is not advisable to increase the extraction steam pressure through the butterfly valve to increase the output of the SE under the condition of meeting the extraction pressure. This approach only increases the standard coal consumption rate of the system, and the result is not worth the loss.

Through a practical application case study, the annual economic benefit is expected to be ¥ 4.15 million and the PP is 5.02 years.

APPENDIX A. SIMULATION MODEL OF THE SYSTEM

APPENDIX B. DETAIL KEY DATA OF THE UNIT UNDER A HEATING LOAD OF 100 MW

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Notes

The authors declare no competing financial interest.

ACKNOWLEDGMENTS

This work was supported by the Natural Science Foundation of Beijing (3172031).

NOMENCLATURE

Abbreviations

AHP absorption heat pump
CHP combined heat and power
G1 generator one
G2 generator two
HHN heater for the heating network
HBP high back-pressure
IP intermediate-pressure
LP low-pressure
NS new system
PP payback period
SE screw expander
SC screw compressor
TS traditional system
THA turbine heat acceptance
VWO valve wide open

Symbols

C cost
b standard coal consumption rate, g/kWh
EX exergy, MW
G flow rate, t/h
h enthalpy, kJ/kg
j annual interest rate
k isentropic index
P pressure, MPa
Q heat consumption, MW
r expansion ratio
s entropy, kJ/kgK
T temperature, K
v specific volume, m³/kg
w power output, MW
Δ difference
η efficiency

Subscripts

1∼5 state points
b boiler
c cold
cd condensate
d diagram
d design condition
e electricity
en environmental state
es extraction steam
fw feed water
g generation
h heat
in inlet
is isentropic
l leakage
ls live steam
m mechanical
net net
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