Steam generation process is closely related to energy conversion and cleaner energy utilization, and the industrial steam is even regarded as the currency of heat with significant social and economic value. Herein, the possible industrial steam generation paths are analyzed based upon thermodynamics analysis, in which fossil fuel, electric, and heat-pump heating are considered. The large-temperature-lift heat-pump heating for steam generation is proved to be a most reasonable way with high-efficiency and wide adaptability. Furthermore, a possible system sketch is proposed, in which the heat pump is used for preheating for low-temperature evaporation and the generated steam can be thus compressed with a steam compressor to the expected high-temperature (pressure) steam. The key parameters of the heat pump for steam generation are ascertained and the subsequent optimization space is discussed. The results show that the proposed steam-generation path has clear performance advantages and potential for industrial steam generation, which could be a sustainable heating system for industry.

1. Introduction

Steam (water vapor) is an important heat carrier with a latent heat value over 2000 kJ kg⁻¹, which has advantages such as uniform heating, convenient temperature regulation, clean and nontoxic, convenient transportation. Therefore, it has been widely used in industrial production and is mainly provided by industrial boilers.[1] Currently, the research on steam generation mainly focuses on supply energy and efficiency optimization. Fossil fuel, biomass, solar heat, and electric (direct heating) are the input energy source of most boilers.[2]

Fossil-fuel boiler is the most widely used boiler in industrial process. The existing boilers are mainly based on the combustion of coal or natural gas. Recently in China, the coal-fired boiler has been restricted to use due to its severe environmental pollution impacts. In this case, natural gas is regarded as a cleaner energy source; however, it is indicated that the gas-fired boilers will also emit gaseous pollutants such as NOₓ.[3] the high cost of natural gas has increased the cost of production process. Biomass is regarded as carbon-neutral fuel because the CO₂ it produced during the combustion process is equal to the amount which was taken from the atmosphere during the growing stage.[4] However, the biomass will also emit PM₂.⁵ and CO during the combustion process.[5] and its resources are not possible to meet the demand for industry heating.

Solar heating could also serve as an energy source for steam generation, and current research can be mainly divided into solar heating for industrial processes (SHIP)[6] and small-scale solar-driven steam generation.[7] SHIP with solar collector converts solar irradiation hitting on its surface into thermal energy by heating a suitable heat-transport medium. However, the unavailability at night and the large collector area make this technology difficult for stable industrial steam demand and also the high initial cost may deter the customer.

Electricity can be directly used for steam generation in electric boiler, which is quite flexible and easy to use if low-cost electricity is accessible. However, in terms of thermodynamics, electrical boilers have a lower efficiency of its first energy resources (fossil fuel and biomass with efficiency typically between 30% and 50%), whereas electrical-driven heat pump could increase electric heating performance as it absorbs heat from low-temperature heat sources (waste heat, solar heat, or even ambient heat, etc.).[8] There are several heat-pump methods of steam generation appeared to meet the high-temperature industrial steam requirement. Kobe Steel. Ltd.[9] developed two heat-pump-based steam supply systems: the high-efficiency steam-supply system with a steam temperature of 120 °C and the standard steam-supply system with a steam temperature of 165 °C. The heat pump system, which is equipped with a newly developed screw compressor and a high-efficiency motor resistant to high temperatures, uses a refrigerant suitable for high temperature supply. Chamoun et al.[10] proposed a new R718 high-temperature heat-pump system to realize the high temperature output of 145 °C. The water vapor heat pump is investigated to satisfy different temperature demands for industrial use with high performance. Wu et al.[11] studied a water vapor high-temperature heat-pump system for waste heat recovery.
from the 80–90 °C heat source to support high temperature output. They tried to improve the output temperature to 150 °C with a temperature lift of 65 °C.

Traditional boilers supplied by combustion heat are generally with unified structure, which is composed of economizer, water wall, super-heater, and reheater. Also the research mainly focuses on the optimization on the combustion process and the external thermodynamic process. Previous study has taken a theoretical case study on steam generation with different methods and indicates low-pressure water evaporation system for steam generation.[12] We have proposed an air-source heat-pump boiler to absorb energy from ambient air which shows its great potentials in industrial steam-generated heating.[13]

It should be noted that the low-pressure water evaporation and mechanical vapor compression/recompression might bring a new solution on steam generation process.[14] The flash tank[15] and falling film evaporator[16] are usually adopted in this process. Another core component in this process is water vapor compressor, which can maintain a low-pressure environment and deduced the generated water vapor for further compression.

However, there are still no detailed studies based on basic thermodynamic paths, especially with the low-pressure evaporation. Therefore, it is necessary to explore the thermodynamic paths for steam generation, especially the best way to match steam generation with heat pump, and give a solution on how to construct a high-efficiency sustainable heating system based on the boundary limits.

2. The Paths of Steam Generation

2.1. Steam-Generation Process

Steam generation from water is essentially an energy upgrading process from liquid state to vapor state, from low-temperature and low-pressure state to high-temperature and high-pressure state. In actual applications, steam is of a certain superheat degree to prevent condensation loss along the transmission pipeline. A sketch of steam-generation process is shown in the pressure–enthalpy diagram (Figure 1). It can be found that the enthalpy increment is 2632.59 kJ kg⁻¹ from point A (20 °C, 0.1 MPa) to point B (125 °C, 0.2 MPa), no matter how this process is achieved, and the maximum change in enthalpy occurs in the two-phase region.

2.2. Systems and Devices for Steam Generation

Steam generation is to be achieved through several thermodynamic processes, which can be implemented by different thermodynamic systems and devices. As shown in Figure 2, the thermodynamic system can be classified as closed system (control mass) and open system (control volume). The closed system can be further classified according to fixed and moving boundary conditions. However, there is no mass crossing the boundary regardless of fixed or movable boundary of the closed system. Therefore, the closed system is not suitable for continuous steam supply.

In addition, due to the relatively lower density of the water vapor, for example, 897.8 m³ for 1 ton steam at 125 °C and 0.2 MPa, it is hard to generate water vapor in closed systems due to the excessive vapor storage volume. In contrast, water vapor can be easily generated in open systems. In practical applications, there is usually a steady-flow process in the open system after the start-up period, which means that the properties of the flowing fluid remain constant in any determined control volume section. The following devices are commonly adopted for steady-flow: 1) Heat exchangers are used for the heat exchange between
two moving fluid streams. In general, heat exchangers involve no work interactions \((w = 0)\) and negligible kinetic and potential energy changes for each fluid stream. 2) Compressors and water pump can increase the pressure of fluids and there is usually negligible heat transfer \((Q = 0)\). 3) Nozzles and valves are also commonly used as steady-flow devices. There is usually no heat \((Q = 0)\) and work input \((w = 0)\) in these devices.

In general, the input of work and heat is achieved independently in corresponding devices and it is difficult to construct devices that can provide both work and heat with considerable quantity.

### 2.3. Path of Water-to-Steam in Open System for Steady-Flow

In fact, the most direct steam-generation path from the \(p-h\) diagram is shown as the blue solid line in Figure 3, which is a straight connecting from point A to point B. However, the implementation of this path is extremely difficult in open systems, which often needs to be divided into lots of segments (shown in the dotted line) and brings a lot of difficulties in matching.

Therefore, considering practical processes, the following three steam-generation paths are practically operational, as shown in Figure 4.

The three steam-generation paths can be summarized as follows: 1) Liquid water (atmospheric temperature and pressure) \(\rightarrow\) Liquid water (atmospheric temperature and pressurized pressure) \(\rightarrow\) Saturated water (pressurized pressure) \(\rightarrow\) Saturated vapor (pressurized pressure) \(\rightarrow\) Saturated vapor (atmospheric pressure) \(\rightarrow\) Saturated water (atmospheric pressure) \(\rightarrow\) Superheated steam. 2) Liquid water (atmospheric temperature and pressure) \(\rightarrow\) Saturated water (atmospheric pressure) \(\rightarrow\) Saturated water (sub-atmospheric pressure) \(\rightarrow\) Saturated vapor (sub-atmospheric pressure) \(\rightarrow\) Superheated steam. 3) Liquid water (atmospheric temperature and pressure) \(\rightarrow\) Liquid water (sub-atmospheric pressure) \(\rightarrow\) Saturated water (sub-atmospheric pressure) \(\rightarrow\) Saturated vapor (sub-atmospheric pressure) \(\rightarrow\) Superheated steam.

Furthermore, as shown in Figure 4b by T–h diagram, the steam evaporation temperature corresponding to the three steam-generation paths decrease progressively. In essence, the second steam-generation path is equivalent to the third one; both of which are pressurized on the steam side while the first one is pressurized on the water side.

### 3. External Matching and Optimization

As mentioned earlier, the steam-generation path in the open system with steady-flow can be ultimately divided into two types as follows: 1) Pressurization on the liquid side. 2) Pressurization on the vapor side.

For a determined path, a temperature–heat load diagram can be used for analyzing and matching the steam-generation process with different internal processes\,[17]\) so does the T–h diagram which can reflect the corresponding process of unit mass as shown in Figure 4b. Heat input and work input are the most common methods to regulate the thermodynamic properties of the fluid in the open system with a steady flow, and the following details will focus on matching processes: 1) Common heat reservoir mainly includes constant-temperature heat reservoir (CTHR) and varying-temperature heat reservoir (VTHR). CTHRs usually utilize the condensation process of working fluids with large latent-heat value, such as refrigerant or phase-change material (PCM). VTHRs are commonly based on the cooling process in pure working fluid such as hot water, hot oil, and CO\(_2\) at supercritical state, which are of large temperature glide. The condensation process in zeotropic mixture refrigerant can be also served as the heat carrier of VTHR. 2) Compression with work input can also increase the enthalpy of the working fluid, but this method is usually applicable to compressible fluid, such as water vapor and other gaseous fluids.
In contrast, water is incompressible and it is difficult to increase its enthalpy significantly by compressing process.

Therefore, when matching external energy sources for the first steam-generation path, the optimal matching scheme is shown in Figure 5a as the red-solid line with the minimum heat-transfer temperature difference (herein set as 5 °C uniformly).

The red lines represents external heat reservoirs, which kept a temperature difference of 5 °C with the internal fluid. In this case, two VTHRs and one CTHR should be matched for heating in the liquid region, the vapor region, and the two-phase region, respectively (As the pressure of superheated steam and saturated steam is equal, the compression method is inappropriate). The average heat-transfer temperature difference can be easily obtained by the following formula and the \( \Delta H \) represent \( m \cdot h \) (herein \( m \) is set as 1 kg s\(^{-1}\)). In Figure 5a, the average heat-transfer temperature difference is 5 °C.

\[
\Delta T = \frac{\int_{T_{\text{in}}}^{T_{\text{out}}} \left( T_{\text{hot}} - T_{\text{cold}} \right) \, d\delta H}{\Delta H} \tag{1}
\]

In practice, it is difficult and uneconomical to match a thermodynamic process with multiple heat reservoirs, and the matching schemes with single heat reservoir are more common. Figure 5b shows the corresponding matching scheme with a CTHR, which has a constant supply temperature of 130 °C. In that case, the average heat-transfer temperature is as high as 17.96 °C.

To further reduce the heat-transfer temperature difference, a VTHR could be adopted as shown in Figure 5c. Compared with the constant temperature heat source, the VTHR can further reduce the heat-transfer temperature difference by 2.92 to 15.04 °C, and the minimum temperature difference occurs in the beginning of vaporization section as marked.

However, the temperature glide with VTHR is just 5.95 °C, and the heat-transfer temperature difference is over 15 °C. This is mainly because most of the enthalpy changes occur in the two-phase region with a constant temperature. In general, the saturated steam will be introduced into the superheater for separate heating in steam boilers. On this condition, the heat-transfer temperature difference can be further reduced to 12.99 °C as shown in Figure 5d.

Similarly, for the second path shown in Figure 6, a CTHR could be adopted for the saturated steam generation in the front-end of layout (FEOL). Due to the temperature difference during the heat-transfer process, there exists irreversible loss which could be reflected by the entransy dissipation rate as shown in Figure 7 by the temperature–heat load (\( T–Q \)) diagram. In \( T–Q \) diagram, the area between the external heating curve and internal heated curve is the entransy dissipation rate during the heat-transfer process. Therefore, in Figure 7, the area I represents the entransy dissipation rate shared by the two steam-generation paths. The area II and area III represents the entransy dissipation rate of the first and second steam-generation paths, respectively. It is clear from the picture than the second steam-generation path has a relatively smaller entransy dissipation.

For the back-end of layout (BEOL), it is more feasible to adopt a steam compressor for pressurization on the generated steam.

**Figure 5.** The first steam-generation path in \( T–h \) diagram matching with external heating. a) Optimal matching scheme. b) CTHR. c) VTHR. d) Improved CTHR.
with a relatively lower temperature. It should be noted that adopting the steam compressor may bring more work consumption.

4. Optimal Steam-Generation System with Heat Pump

4.1. Discussion on Heat Pump for Heating and for Steam Generation

Regardless of the form of heat supplier, the steam-generation path is essentially similar and can be uniformly summarized as first steam-generation path. As mentioned earlier, heat exchanger can input heat and the fuel combustion; solar energy and electricity can be the heat supplier. As shown in Figure 8a, heat pump can transfer heat from a low-temperature medium to a high-temperature medium usually with free heat input and higher efficiency. Therefore, the inverse Carnot efficiency reflect the ideal ratio between the supply heat and the network input, which is always higher than 100% as the following equation shown.

\[ \varepsilon_{\text{carnot}} = \frac{Q_H}{W_{\text{net,in}}} = \frac{T_H}{T_H - T_L} \]

When taking heat pump as heating device, there are already some heat-pump systems for residential heating and even for steam generation as shown in Figure 8b.

However, there exists a clear blank space enclosed by the black-dotted line as shown in Figure 8b which could represent the working condition with ambient air as the heat source for industrial steam generation. The reason lies in the decrease in the COP value with temperature lift increasing as shown in Figure 9a.

Temperature lift reflect the difference between the high temperature and low temperature as shown in following equation.

\[ \Delta T_{\text{lift}} = T_H - T_L \]

According to the definition of inverse Carnot efficiency, Equation (2) is further sorted out as

\[ \varepsilon_{\text{carnot}} = \frac{T_H}{\Delta T_{\text{lift}}} \]
It is clear that the temperature lift is inversely proportional to the inverse Carnot efficiency when the high temperature is constant. Figure 9a further shows the inverse Carnot efficiency of heat pump under different temperature lift values ($T_{HI} = 125 \, ^\circ C$). With the temperature lift increase from 40 to 130 $^\circ C$, the inverse Carnot efficiency declines sharply from 9.95 to 3.06. Moreover, there is a big difference between the actual heat pump’s coefficient of performance (COP) and inverse Carnot efficiency, and the thermodynamic perfectness is commonly used to reflect this difference and is defined as

$$\eta_{HP} = \frac{\text{COP}}{\varepsilon_{\text{carnot}}}$$ (5)

According to practical experience, the thermodynamic perfectness is hard to reach 50% under a large temperature lift. Under these circumstances, when the temperature lift is above 100 $^\circ C$, the COP value is less than 2. As shown in Figure 9b, there is few research in the blue circle region with large temperature lift. Paradoxically, the larger the temperature lift value is, more heat sources can be used.[18] If the temperature lift value is above 100 $^\circ C$, the low-temperature medium can be the ambient air or water or those from the industry cooling water tower, which is usually easy to obtain or even free.

Therefore, it is of great significance to explore novel steam-generation path based on heat pump to reach the target region as shown in Figure 9b.

### 4.2. Discussion on Large-Temperature-Lift Air-Source Heat Pump for Steam Generation

To further improve the temperature lift range of heat pump, many scholars have optimized the cycle form of heat pump. At present, multi-stage compression and cascaded cycle are two typical heat-pump cycles to increase the temperature lift of heat pump. Two-stage compression can also be regarded as a kind of multi-stage compression, and the corresponding sketch is shown in Figure 10a. For multi-stage compression, due to the

Figure 9. Heat pump’s sketch and the inverse Carnot efficiency. a) Inverse Carnot efficiency and COP. b) COP and temperature lift of the current heat-pump systems.

Figure 10. Heat pump’s sketches a) Two-stage cycle. b) Multi-stage cycle. c) Cascade cycle.
single refrigerant still in use and the influence of the characteristics of refrigerant, the attenuation of its cycling performance along with the temperature increase is still unavoidable.

The cascade heat pump can effectively scope different refrigerants’ potential in their suitable working region to improve the thermodynamic perfect degree. Li et al. [19] tested the performance of BY3B/BY6 used in the cascade heat pump and the heating temperature was in the range 100–170 °C. However, the required heat source is 55 °C and one more cascade cycle is required for larger temperature lift as shown in Figure 11a. At present, there are also some conceives [20, 21] of heat pump systems based on multiple cascade cycles. However, these conceives are lack of application verification.

Through the coupling of absorption heat pump and compression heat pump can also achieve a large temperature lift as shown in Figure 11b. However, the total efficiency of this system is only about 150%. Essentially, these two types of large temperature-lift heat pumps are closed systems, which absorbs the heat energy of the low-temperature heat source (air source) and further supplies heat for steam generation through indirect heat exchange as the first steam-generation path.

4.3. Proposed Steam-Generation System

Through previous analysis in Section 3.1, the second steam-generation path has a relatively smaller irreversible loss and may have a better performance, and the corresponding system sketch is proposed in Figure 12a, b. This combined system mainly consist of a heat pump and water vapor compressor. The heat pump can supply heat for the evaporation process of water in FEOL, and the water vapor compressor can further compress the generated water vapor in BEOL. The evaporation process can be achieved through flash evaporation or falling film evaporation.

In that case, the Carnot efficiency of heat pump can also be improved due to a lower supply temperature of heat pump. However, the introduction of water vapor compressor may bring more work input and whether it is worth needs more considerations.

Essentially, the aim of the second path is to use the power consumption of the compressor in exchange for the improvement in heat pump performance. In this case, it is important to determine the parameters of the intermediate connection.

\[
\eta_{\text{boiler}} = \frac{\text{Desired output}}{\text{Required input}} = \frac{Q_{\text{total}}}{W_{\text{total}}} \tag{6}
\]

\[
Q_{\text{total}} = m_v \times (h_B - h_A) \tag{7}
\]

The work input mainly consists of the work of heat pump and compressor.

\[
W_{\text{total}} = W_{\text{HP}} + W_C \tag{8}
\]

Heat pump can absorb the heat from free heat source, and its performance is usually reflected by COP.

\[
W_{\text{HP}} = \frac{m_v \times (h_B - h_A)}{\text{COP}_{\text{Ta}\rightarrow\text{Tim}}} \tag{9}
\]

The COP of heat pump system is strongly affected by working fluid, the system components, and operating conditions. Also there is usually a higher COP value with smaller temperature lift value. Therefore, the following empirical formulas have been selected [22] to determine the COP values.

\[
\text{COP}_{\text{Ta}\rightarrow\text{Tim}} = a \cdot (\Delta T_{\text{lift}} + 2 \cdot b)^c \cdot (T_{\text{h, out}} + b)^d \tag{10}
\]

where \(\Delta T_{\text{lift}}\) represents temperature lift, \(T_{\text{h, out}}\) is the outlet temperature, and the fitting parameters are as follows: \(a = 1.4480 \times 10^{17}, b = 88.730, c = -4.9460, d = 0.0000\).
The input work of the compressor can be expressed as

$$W_C = \frac{m_v \times (h_{B,\text{is}} - h_{\text{in}})}{\eta_{\text{is}}}$$  \hspace{1cm} (11)

where $m_v$ is the mass flow rate of water vapor, $h_{B,\text{is}}$ is the isentropic enthalpy at the outlet state, $h_{\text{in}}$ is the enthalpy of the inlet water vapor, $\eta_{\text{is}}$ is the isentropic efficiency.

The isentropic efficiency of compressor can be influenced by many factors including but are not limited to the working parameters and types of the compressor, and herein is determined based on the investigation on the water vapor compressor$^{23}$ with a linear processing.

Also the final expression of the boiler efficiency is as follows by Equation (6) and Equation (7)–(11)

$$\eta_{\text{boiler}} = \frac{(h_B - h_A)}{\text{COP}_{\text{is}} \times \text{COP}_{\text{is}} + \frac{h_B - h_A}{h_C}}$$  \hspace{1cm} (12)

4.4. Energy Evaluation on the Air-Source Heat Pump for Steam Generation

Figure 13a shows the input work of the second steam-generation path under different intermediate temperature. With the increase in intermediate temperature from 40 to 120°C, the input work of the heat pump increases from 388.0 to 2197.5 kJ kg$^{-1}$. In contrast, the input work of water vapor compressor decreases from 1972.4 to 2.4 kJ kg$^{-1}$. Moreover, the total input work increased initially and then decreased, with a minimum value of 1420.2 kJ kg$^{-1}$ at the intermediate temperature of 76°C. Correspondingly, Figure 13b further analyzed the variation of efficiency factor. With the increase in intermediate temperature from 40 to 120°C, the COP of heat pump decreases from 6.42 to 1.19, and the boiler efficiency increased initially and then decreased, with a maximum value of 1.86 at the intermediate temperature of 76°C. In fact, the circumstances on the intermediate temperature of 120°C is close to the first steam-generation path. Also, the fact that the optimal boiler efficiency is obtained at 76°C also verifies the advantages of the second steam-generation path in energy saving.

Meanwhile, it is clear that the efficiency factor of the heat pump and water vapor compressor has a significant influence on the whole system as shown in Figure 14.

In general, the current thermodynamic perfection of heat pump and isentropic efficiency of compressor are around 50–60%. Therefore, as indicated by the red dotted line frame, the maximum energy efficiency of the current system can only reach about 2.1.

5. Discussion and Perspective of the Steam-Generation System with Air-Source Heat Pump

5.1. Optimum Matching of Refrigerant Physical Properties with External Heat-Exchange Process

As mentioned earlier, due to the latent heat of water vapor that accounts for most of the enthalpy increment, the mainstream
heating temperature should be corresponded to the steam evaporation temperature to reduce temperature difference for heat transfer. However, there still exists appreciable irreversible heat-transfer loss especially in the sub-cooling section. Therefore, for heating facility based on the vapor compression heat-pump technology, utilizing the superheated and sub-cooled process of refrigerant for water heating at liquid and vapor state may further reduce the temperature difference. The following settings are made to evaluate the heat-load fitness between hot fluid stream (refrigerant) and cold fluid stream (water/steam): 1) For the high-temperature and medium-temperature evaporation scenarios, R245fa and R134a were correspondingly selected, which are widely used in the corresponding temperature zones. 2) To ensure adequate heat exchange, the phase-change heat value of water and refrigerant is equal through adjusting the mass flow rate of refrigerant. 3) Correspondingly, the starting point and ending point in the phase-change process are adjusted consistently.

As shown in Figure 15, the slope of R245fa’s T–Q curve is smaller than that of water/steam in the sub-cooled zone. Therefore, it is possible to use the super-cooled section of the refrigerant to heat water because sufficient temperature difference can still be maintained during this process. Similar trend also occurs in the overheating zone that R245fa’s T–Q curve has a smaller slope in the overheating zone. However, as the refrigerant’s condensation temperature is set near the evaporation temperature of water, it is difficult to use the overheated zone of refrigerant for superheated steam generation. Figure 16 further shows the improved heat-transfer process with the utilization of refrigerants’ sub-cooled section.

5.2. Ejector for Low-Temperature Steam Generation

In essence, the second steam-generation path can improve the heating efficiency of heat pump through reducing the evaporation temperature of the water vapor. However, limited by the special thermodynamic properties of steam as follows, the reduction of evaporation temperature is not unlimited by water vapor compressor: 1) When the evaporation temperature decreases, the density of saturated steam decreases and the compression work per unit mass increases. 2) When the temperature decreases, the saturation pressure of water vapor decrease significantly (the saturation pressure of water at 60°C is only about 0.02 MPa), which increases the airtightness requirement of the corresponding equipment and may have the leakage risk.

Therefore, an ejector is adopted for compression in lower temperature and pressure region as shown in Figure 17, which is so-called thermal compression. In this case, as shown B–E section, part of the generated steam will be continuously used for ejecting the low-pressure steam (I) without being exported, and thereby the compression work will increase.

It is clear that the ejection ratio will have a direct influence on the whole efficiency of the system, which is defined as

\[
R_{\text{ejection}} = \frac{m_I}{m_c} = \frac{m_B}{m_c}
\]  

(13)

Also the total boiler efficiency can be deduced as

\[
\eta_{\text{boiler}} = \left[\frac{(h_B - h_A)}{\text{COP}_{\text{B}}} \right] \frac{1 + 1/R_{\text{ejection}} \times (h_B - h_F)}{\text{COP}_{\text{E}} - \text{T}_I} \frac{\eta_{\text{is}}}{\text{COP}_{\text{F}}}
\]  

(14)

Furthermore, the boiler efficiency under different thermodynamic perfectness and the ejection rate is shown in Figure 18.
It is clear that the ejection rate has a significant influence on boiler efficiency. Essentially, with the increase in ejection ratio, there is an increase in the generated steam amount, which will result in a better efficiency. The ejection ratio is influenced by a lot of parameters such as the inlet pressure and the specific structure with a common range between 0 and 1, and based on an optimal estimation, the ejection ratio could reach more than 0.6.\cite{24}

In general, when the ejection ratio is above 0.56, the total efficiency can exceed 200%. However, as there are lack of ejector working in the zone, this idea is still in need of further verification.

6. Conclusion

In this article, we explore possible steam-generation paths based on heat-pump technology with basic thermodynamics analysis. Furthermore, the external matching schemes of the proposed steam-generation paths are discussed. With $T$–$Q$ diagram analysis and energy assessment, the optimal steam-generation path and the subsequent optimization space are discussed. The main results are listed as: 1) Direct steam-generation process in $T$–$h$ diagram is hard to achieve. In the open system, the main steam-generation methods can be divided into pressurization on the liquid side (the first steam-generation path) and on the vapor side (the second steam-generation path). 2) For the first steam-generation path, to reduce the temperature difference during the heat-transfer process, it is better to separate the superheated section from the phase-change section with different heat reservoirs. However, the heat-transfer temperature difference is still relatively higher to 12.99 $^\circ$C (the minimum heat-transfer temperature difference is 5 $^\circ$C). Compared with the first steam-generation path, the second steam-generation path which evaporates under a sub-atmospheric pressure with a heat-transfer temperature difference about 7.95 $^\circ$C. 3) It is possible to use the super-cooled section of the refrigerant to heat water to reduce the temperature difference. However, as the refrigerant’s condensation temperature is set near the evaporation temperature of water, it is difficult to use the overheated zone of refrigerant for superheated steam generation. 4) Based on heat pump for heating and compressor for compression, the second steam-generation path has an optimal efficiency with a maximum value of 1.86 at the intermediate temperature of 76 $^\circ$C, which also verifies the advantages of the second steam-generation path in energy saving. 5) Utilizing ejector may bring a positive influence on the steam-generation efficiency, but it mainly depends on the ejection ratio of the ejector.

Acknowledgements

This research was supported by the Key Project of the National Natural Science Foundation of China (52036004), and also China–Sweden International Collaborative project of National Natural Science Foundation of China (51811530019). The authors gratefully acknowledge the financial support from the Research Council of Norway and user partners of HighEFF (Centre for an Energy Efficient and Competitive Industry for the Future, an 8-year research centre under the FME-scheme). The authors also thank Dr. Lingji Hua, Dr. Bangjun Li, Dr. Di Wu, Dr. Jiatong Jiang, Dr. Zhao Shao, and Dr. Jintong Gao who supported and partly joined the research work.

Conflict of Interest

The authors declare no conflict of interest.

Data Availability Statement

Research data are not shared.
Keywords
heat pumps, large temperature lifts, steam generation, temperature–heat load diagrams

Received: December 28, 2020
Revised: February 15, 2021
Published online: March 8, 2021

[1] R. Saidur, J. U. Ahamed, H. H. Masjuki, Energy Policy 2010, 38, 2188.
[2] R. Liu, L. Chen, M. Liu, Q. Ding, Y. Zhao, IOP Conf. Ser.: Earth Environ. Sci. 2017, 94, 12058.
[3] T. Yue, X. Gao, J. Gao, Y. Tong, K. Wang, P. Zuo, X. Zhang, L. Tong, C. Wang, Y. Xue, Atmos. Environ. 2018, 184, 1.
[4] R. Saidur, E. A. Abdelaziz, A. Demirbas, M. S. Hossain, S. Mekhilef, Renewable Sustainable Energy Rev. 2011, 15, 2262.
[5] Y. Li, J. Liu, H. Han, T. Zhao, X. Zhang, B. Zhuang, T. Wang, H. Chen, Y. Wu, M. Li, Atmos. Environ. 2019, 213, 64.
[6] T. Jia, J. Huang, R. Li, P. He, Y. Dai, Renewable Sustainable Energy Rev. 2018, 90, 475.
[7] M. Amjad, G. Raza, Y. Xin, S. Pervaiz, J. Xu, X. Du, D. Wen, Appl. Energy 2017, 206, 393.
[8] V. Gaigalis, R. Skema, K. Marcinauskas, I. Korsakiene, Renewable Sustainable Energy Rev. 2016, 53, 841.
[9] Development and Sale of High Efficiency Steam Supply System “Steam Enhanced Heat Pump”, https://www.kobelco.co.jp/chinese/releases/1193477_15016.html (accessed: February 2011).
[10] M. Chamoun, R. Rulliere, P. Haberschill, J. Peureux, Int. J. Refrig. 2014, 44, 177.
[11] D. Wu, J. Jiang, B. Hu, R. Z. Wang, Energy 2020, 190, 116427.
[12] F. Bless, C. Arpagaus, S. S. Bertsch, J. Schiffmann, Energy (Oxford). 2017, 129, 114.
[13] H. Yan, B. Hu, R. Wang, Adv. Sustainable Syst. 2020, 2000118.
[14] B. Hu, D. Wu, R. Z. Wang, Renewable Sustainable Energy Rev. 2018, 98, 92.
[15] C. Qi, X. Wang, C. Miao, H. Feng, Q. Lv, Int. J. Heat Mass Transfer 2018, 119, 175.
[16] B. Hu, H. Yan, R. Z. Wang, Appl. Energy 2019, 255, 113851.
[17] S. Du, R. Z. Wang, Z. Z. Xia, Energy 2015, 80, 687.
[18] Z. Y. Xu, R. Z. Wang, C. Yang, Energy 2019, 176, 1037.
[19] X. Li, Y. Zhang, X. Ma, N. Deng, Z. Jin, X. Yu, W. Li, Appl. Therm. Eng. 2019, 159, 113895.
[20] B. Zühlendorf, F. Bühler, M. Bantle, B. Elmegaard, Energy Convers. Manage.: X 2019, 2, 100011.
[21] F. Bühler, B. Zühlendorf, T. Nguyen, B. Elmegaard, Appl. Energy 2019, 250, 1383.
[22] F. Schlosser, M. Jesper, J. Vogelsang, T. G. Walmsley, C. Arpagaus, J. Hesselbach, Renewable Sustainable Energy Rev. 2020, 133, 110219.
[23] W. U. Di, H. U. Bin, W. Ruzhu, J. Nanshan, L. I. Ziliang, J. Chem. Ind. 2017, 68, 2959.
[24] Y. Dai, J. Wang, L. Gao, Appl. Thermal Eng. 2009, 29, 1983.