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Steady state behavior of a booster heat pump for hot water supply in ultra-low temperature district heating network

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1. Introduction

Reducing the use of fossil fuels and its related emissions can be achieved by increased use of renewable energy sources and efficiency improvement in existing energy utilization technologies. According to the EU Action 2030 Climate and energy framework [1], the European aim is to increase the share of renewable energy to 32% and improve energy efficiency by 32.5% by 2030. A number of individual member countries have more ambitious targets, e.g. Denmark which aims to phase out fossil fuels by 2050 [2]. According to the Heat Roadmap Europe 2 study, the share of district heating (DH) in the European Union (EU) could potentially increase to 50% of the entire heat demand by 2050 [3]. In countries with cold climate and higher heating needs, DH has begun to play an important role in reducing energy use in buildings. Thus, DH systems are expected to contribute to this aim by increasing the energy efficiency of the heat supply and enabling the exploitation of renewable heat sources [4–6]. At the same time, domestic hot water (DHW) has a significant share in building’s energy consumption. Households were responsible for 26% of the total final energy consumption in the EU in 2018. After space heating, the greatest contribution to energy consumption in EU households (up to 15%) is hot water [7,8]. Efficient DHW supply will have great potential in energy saving and system efficiency improvement.

The DH temperature was decreased with the implementation of heat pumps and a general idea to increase the system efficiency by decreasing the DH temperature [9]. And the supply temperature of low temperature DH (LTDH) can be as low as just above 50°C. Østergaard and Andersen [10] studied the impacts of different low-temperature heat sources in the DH system. LTDH had positive economic benefits in building’s energy consumption. Households were responsible for 26% of the total final energy consumption in the EU in 2018. After space heating, the greatest contribution to energy consumption in EU households (up to 15%) is hot water [7,8]. Efficient DHW supply will have great potential in energy saving and system efficiency improvement.

The DH temperature was decreased with the implementation of heat pumps and a general idea to increase the system efficiency by decreasing the DH temperature [9]. And the supply temperature of low temperature DH (LTDH) can be as low as just above 50°C. Østergaard and Andersen [10] studied the impacts of different low-temperature heat sources in the DH system. LTDH had positive effects on efficiencies of heat generating equipment, grid losses and the potential of heat recovery from various waste heat sources. As LTDH leads to less heat losses in the DH network it is therefore also economically beneficial [11]. Köfinge et al. [12] concluded that
LTDH networks can be an economic, ecologic and energetic advantage for the supply of space heating and DHW. The availability and economic conditions of low temperature heat sources (such as excess heat, heat pumps and the return line of existing HTDH networks) is a key factor for facilitating LTDH networks, since it is required for balancing the higher investment costs on the demand side. Ultra-low-temperature district heating (ULTDH), with the supply temperature lower than the LTDH, is unable to independently generate DHW meeting the comfort or hygienic rule [13]. In order to further elevate the DHW temperature, additional booster device is required. An evaluation for the possibility of an ULTDH system with individual heat pumps for low-energy buildings has been made by Knudsen and Petersen [14], the result show that the system is capable of providing energy cost savings for the end-user.

Zvingilaite et al. [15] proposed the concept of a booster heat pump (BHP) included in the district heating substation to provide DHW by lifting the temperature of the district heating water decentrally. Raising the temperature from district heating sources locally at the user side makes it possible to decrease the supply temperature and the related heat loss of the district heating system. As BHP is used at local point for DHW supply, which enable DH system easy to use lower level heat source (much possibility for introducing lower level temperature renewable heat) and no worry about DHW supply. It is a flexible way meet the heat demand of the building. The BHP is part of a DH substation, which is normally connected in parallel with the space heating cycle in the DH system as a DHW production unit, as depicted in Fig. 1. The DH supply enters the BHP system at a forward temperature. One part of the flow is used as the heat source in the evaporator and cooled down, and the other part is heated up in the condenser to provide the demand temperature which is used for heating up DHW connected via a heat exchanger or storage into the stratified hot water tank. The outlet flow from the BHP evaporator is mixed with either the return flow from the DHW heat or the outlet from the stratified tank. The mixed flow then returns to the DH return line. During discharging of the stratified storage tank, the cold return flow from the DHW heat exchanger is fed into the bottom of the stratified storage tank. Østergaard and Andersen [16] indicated that applying BHPs enables the DH system to operate at substantially lower temperature levels. Elmegaard et al. [17] investigated the integration of space heating and hot water supply in low temperature district heating. The highest performance with 25% exergy efficiency was reached by an R134a heat pump with hot water storage on the district heating side. Ommen et al. [18] proposed a prospective solution for an ULTDH system with BHP, which achieves 12% performance improvement compared with the case that ULTDH system supplied by central HPs without BHP. Zühlsdorf et al. [19] demonstrated an increase in the thermodynamic performance of a BHP, which was achieved by choosing mixed working fluids. This resulted in higher investment cost compared to the pure fluids, but they could considerably improve the thermodynamic performance of the overall heat supply system while being economically competitive to pure fluids. The related study by Meesenburg et al. [20] found that investment and maintenance cost limit the feasibility of ULTDH. ULTDH could be feasible if the cost of decentral units could be lowered or if the investment cost of the central heating unit was significantly lower compared to LTDH. Dermentzis et al. [21] conducted a complete and comprehensive simulation study on heating and distribution systems in a new residential district. The results showed that the distribution systems combined with a return flow HP (a water-to-water HP with the return flow of the floor heating as a source and the water tank for DHW as a sink, like booster heat pumps) is efficient. A BHP in a low-heat consumption density area is a sustainable solution, which can be coupled with solar PVs and allows reducing CO2 emissions by up to 17.5% compared with the business as usual scenario with the temperature regime in the main network 118/70 [22]. A heat pump used for domestic hot water preparation in single-family houses using the ULTDH network as a source (at a temperature of 20 °C) has been reported by Reiners et al. [23]. Result shows that the COP value of this system is 3.4 when the source return temperature is 17 °C. From references [10,17,18,23,24], it can be observed that BHP may have much perspective for use in the ULTDH system. Further investigation about the operating performance of the system is necessary to help the practical application of the system better.

In this study, experimental research was conducted on a proposed BHP test rig. The main purpose is to verify the performance of applying the BHP in ULTDH network for DHW applications. Evaluation of the performance of BHP with R134a as refrigerant was carried out by varying the DH forward temperature and the condenser outlet temperature. Furthermore, the effects of forward water temperature on coefficient of performance (COP) and exergy efficiency was determined at different operating conditions. The main novel contributions of the paper are the following, (i) Highlight of steady state thermal behavior of a BHP for use in ULTDH networks. (ii) Determination of optimal working condition of the proposed BHP in the studied range. (iii) Exergy destruction of each component is analyzed as well as exergy efficiency of BHP used in the proposed system, which assists further improvement of each component under different working conditions. Based on exergy analysis, potential directions for performance improvement are discussed. (iii) Provision of a method for better matching of the secondary water flow rate and the forward temperature of the BHP system. COP based on experimental data for the BHP system have been presented.

Details of the experimental BHP test rig and methods applied for the analyses are presented in Section 2. The experimental results for steady-state thermodynamic behavior together with the exergy analysis of the BHP cycle based on the experimental results are shown in Section 3. A discussion of the operation improvement by applying optimal matching of the water flow rates in condenser and evaporator is presented in Section 4. Section 5 presents the conclusion of the present study.

2. Methods

2.1. Experimental set up

Fig. 2 shows the experimental setup of the BHP system. The design of the present test rig was developed consistent with the integration of the BHP in a DH system on building level. The test rig
includes four sub-systems: a secondary water cycle to illustrate the
ULT district heating system (designed for the future application in
district heating system), in a building, a refrigerant cycle (BHP) with
R134a as working medium, a shunt-system and a cooling system.
Both heat exchangers in the BHP cycle was configured for counter
flow operation between refrigerant and the secondary fluid. Low
temperature water source temperature is first adjusted via the heat
exchanger (HEX) with the Shunt system. And then, an electrical
heater was used to heat the water forward temperature and ensure
that the BHP behavior can be tested at different steady conditions.
The water at controlled forward temperature was pumped to both
the inlet of condenser and evaporator with the same temperature
(forward temperature), and complete the heat exchange with
refrigerant fluid in each heat exchanger. Thus, the hot water from
the outlet of the condenser becomes the boosted supply temper-
ature, intended domestic hot water consumption in a building. To
complete the cycle, the water flows from the condenser and the
evaporator outlet are mixed, and then the mixed stream flow to the
heat exchanger (HEX) and Heater 1 in turn, to keep the temper-
ature for the next heat exchange cycle.

Both inlet and outlet temperatures of the condenser and the
evaporator of water loop were monitored by Type-T thermocouple.
The refrigerant temperatures at the inlet and outlet of the
condenser and outlet of evaporator were also measured by ther-
mosouples. Pressure sensors placed at the inlet of the evaporator
and the outlet of the condenser were used to measure the inlet
and outlet pressure of the compressor. Four mechanical valves (V1, V2,
V3 and V4) were used for controlling the water flow rate through
the condenser, evaporator, heat exchanger 1 and heat exchanger 2.
As shown in Fig. 2. The rate of compressor power consumption was
monitored based on the current and voltage data. H1, H2, H3 and
H4 are manual ball valves used to set the flow within the district
heating loop. H1S and H2S are manual ball valves used to direct the
flows within the shunt system loop.

All actions including power supply, mechanical valve, electric
heater, water pump and heat pump in the BHP system were
launched via the master control box shown in the picture on the
right of Fig. 3. The Flex Logger software [25] was used for data
monitoring and recording. Main information of the components are
listed in Table 1. Specifications of each instrument are listed in
Table 2.

2.2. Test conditions

The forward temperature (inlet temperature of the condenser
and evaporator) was varied to be 30 °C, 35 °C, 40 °C and 45 °C. As
the proposed BHP system is used to provide domestic hot water, the
main test case is designed to achieve 60 °C supply temperature
(outlet water temperature of condenser) by varying the ratio of
water flow rates in the evaporator. The conditions with 60 °C outlet
water temperature of the condenser were defined as predominant
conditions of the present study. In addition, the test range was also
extended to the other conditions exclude 60 °C outlet temperature.
Table 3 shows predominant conditions.

2.3. Data processing

2.3.1. Coefficient of performance (COP) of BHP

Due to the heat loss, the temperature difference of heat transfer
in evaporator and the condenser, pressure drop and several other
irreversible factors, the actual COP of the BHP cannot reach the
value in ideal cycle condition. Therefore, the relationship between
the water temperature and the COP value of the BHP needs to be
calculated further for the BHP system. The COP of the BHP can be
calculated as,

\[ \text{COP} = \frac{\dot{Q}_c}{W} \] (1)

\[ \dot{Q}_c = c_p m_e (t_{co} - t_{ci}) \] (2)

\[ W = \sqrt{3} I U \cos \phi \] (3)

where \( \dot{Q}_c \) is the heat flow obtained from the condenser; \( W \) is the
power consumption of the compressor; \( c_p \) is the specific heat
capacity of water; \( m_e \) is the water mass flow rate in the condenser;
\( t_{ci} \) is the inlet water temperature of the condenser; \( t_{co} \) is the
outlet water temperature of the condenser; \( I \) is the electric current of
the compressor motor; \( U \) is the voltage of the compressor motor, which
is constant at 400 V; \( \cos \phi \) is the power factor, which was calculated
as 0.9 according to Danfoss compressor prototype [26].

2.3.2. Lorenz COP and lorenz efficiency

The Lorenz COP is defined by Lorenz as [28],

\[ \text{COP}_{Lor} = \frac{T_H}{T_H - T_L} \] (4)

where \( T_L \) is the logarithmic mean low temperature (for heat pumps,
evaporator); \( T_H \) is the logarithmic mean high temperature (for heat
pumps, condenser) in K, defined as,

\[ T_H = \frac{T_{ci} - T_{co}}{\ln \left( \frac{T_{co}}{T_{ci}} \right)} \] (5)

\[ T_L = \frac{T_{co} - T_{eo}}{\ln \left( \frac{T_{eo}}{T_{co}} \right)} \] (6)

The Lorenz efficiency (\( \eta_{Lor} \)) of the BHP is defined as the ratio
between the actual COP value and Lorenz COP, which can be used as
a parameter to reflect the potential performance of the heat pump.
The expression can be written as,

\[ \eta_{Lor} = \frac{\text{COP}}{\text{COP}_{Lor}} \] (7)

2.3.3. Exergy evaluation

Compared to a conventional energy analysis, the exergy analysis
can identify the location of inefficiencies and quantitatively
characterize the thermodynamic imperfection of processes and the possibility for thermodynamic improvement of heat pump systems [29]. Enthalpy value of the state points have been determined and compared based on the tested parameters, and found that the error between the actual condition with pressure drop and the assumption condition without pressure drop is less than 3.2%. Therefore, in order to simplify the calculation process and to specify the exergy destructions in the system, the following assumptions were made:

1) Kinetic and potential energy and exergy losses are not considered; 2) All the components operate in steady state conditions; 3) The dead state was defined as $T_0 = 25 \, ^\circ\text{C}$ and $p_0 = 1 \, \text{bar}$.

In order to calculate the state points of the thermodynamic cycle, the following assumptions have been made:

- The refrigerant cycle follows a conventional vapor compression cycle based on the reversed Rankine Cycle.
- Pressure losses on the refrigerant side of the condenser and evaporator were neglected.
- The expansion in the valve follows an isenthalpic process.
Further, the isentropic efficiency of the compressor was calculated from the measurement data as a measure of system performance and to evaluate the improvement potential of the cycle. For this reason, both the real discharge state (point 2) and the isentropic discharge state (point 2’) are determined based on the tested temperature and pressure data. Thus, based on the typical thermodynamic cycle of (Fig. 4a pressure–enthalpy diagram), an exergy evaluation was carried out for the exergy flows of the studied BHP shown in Fig. 4b. The measurements for the refrigerant cycle in the present study were temperatures on the condenser side, temperature at the evaporator outlet and pressures on both sides. The enthalpy value of the refrigerant at state point 2 and 3 represent the state of compressor outlet and condenser outlet, which can be calculated using REFPROP 9.1 [30] with the pressure and temperature data recorded. Table 4 shows the conditions of each case involved in the exergy analysis (these cases were chosen from typical cases above at each different forward temperature).

The exergy balance equations of the components were described as [31,32],

\[ E_{in} - E_{out} = \sum E_{d, component} \quad (8) \]

where, \( E_{in} \) is the exergy flow rate entering the system; \( E_{out} \) is the exergy flow rate leaving the system; \( E_{d, component} \) is the exergy destruction occurring in the component.

For a control volume at steady state the exergy rate balance reads,

\[ E_F = \dot{E}_P + \dot{E}_D \quad (9) \]

The exergy efficiency is given as,

\[ \eta_{ex} = \frac{\dot{E}_P}{E_F} \quad (10) \]

where,

\[ \dot{E}_P = m_c \cdot c_w \cdot \left( T_{co} - T_{ci} - T_0 \ln \frac{T_{co}}{T_{ci}} \right) \quad (11) \]

\[ E_F = W + \Delta E_{HS} \quad (12) \]
where $\eta_{\text{ex}}$ is exergy efficiency of the BHP system; $\dot{E}_{f}$ is exergy fuel, which is defined as the input exergy of BHP (exergy from power input of the compressor and the external heat source $\Delta H_{\text{ex}}$, which is district heating water that is cooled in the evaporator); $\dot{E}_{d}$ is exergy destruction, which is the increase in physical exergy of the heated water in the condenser; $\dot{E}_{d}$ is destruction of exergy.

The share $\gamma_{d,\text{component}}$ of the exergy destruction of a component is another indication of the contribution of the component to the overall irreversibilities.

$$\gamma_{d,\text{component}} = \frac{\dot{E}_{d,\text{component}}}{\dot{E}_{d,\text{total}}} \tag{13}$$

2.3.4. Energy balance and uncertainty

Energy flow of the BHP system should keep balance according to the law of conservation of energy, equation (14). The energy flow distribution of the BHP cycle is shown in Fig. 5.

$$\dot{Q}_e + \dot{W} = \dot{Q}_c + \dot{Q}_{\text{loss}} \tag{14}$$

The heat loss in the overall system was calculated in the present study. The energy imbalance rate has been defined as $e$,

$$e = \frac{\dot{Q}_e + \dot{W} - \dot{Q}_c}{\dot{Q}_e + \dot{W}} \times 100\% \tag{15}$$

Uncertainty analysis is expressed by the relative error of COP, which was determined by the related independent variables, and the error for it was obtained through the error propagation method. The uncertainty analysis was conducted according to the method proposed by Moffat [33], and use its derived expression (equation [16][34]) based on the variables involved, which can be calculated as below,

$$\text{COP} = \frac{\dot{Q}_c}{W} = \frac{c_{p}m_{c}(t_{ci} - t_{co})}{\sqrt{3}IU \cos \phi} \tag{16}$$

$$\frac{\Delta \text{COP}}{\text{COP}} = \sqrt{\left(\frac{\Delta t_{ic}}{t_{ic}}\right)^2 + 2\left(\frac{\Delta t_{c}}{t_{ci} - t_{co}}\right)^2 + \left(\frac{\Delta I}{T}\right)^2} \tag{17}$$

3. Results

3.1. Uncertainties

The energy imbalance in the tests was calculated based on equations (14) and (15). The energy imbalance ranged from 0.54% to 5.5%, and the uncertainty of COP (calculated based on equations (16) and (17)) ranged from 1.5% to 3.2% of the all 12 typical conditions (Fig. 6).

3.2. Coefficient of performance and Lorenz COP

The present work analyzed the steady-state behavior of a BHP system. Fig. 7a shows the log $p$-$h$ diagram for R134a in the BHP cycle and the operating points of some studied cases (case 2, 4, 8 and 10 in Table 3). These four chosen cases represented the basic operating range of the cycle at four different forward temperatures. It can be found from Fig. 7b that with the purpose of using the BHP to provide domestic hot water (outlet water temperature of condenser 60 °C), the evaporator temperature was within the range of 15.1 °C–24.3 °C, and the corresponding condenser temperature was within the range of 56.0 °C–63.3 °C. The cycle was slightly moved to the upper right with increasing forward temperature, and the difference between high-side and low-side was slightly increased.

Fig. 8 presents the test results of the BHP with different forward temperature as well as different water flow rates in the secondary cycle (corresponding to 12 typical conditions in Table 3). Data used in the analysis of this section is mentioned above for cases with 60 °C outlet water temperature. COP, heating capacity ($Q_e$) and power consumption ($W$) are shown in Fig. 8, respectively. It can be found that the COP value varies from 3.84 to 4.95 in the studied conditions, and that the heat capacity ranged from 12.4 kW to 16.4 kW. Both COP and heat capacity increased with increasing return temperature ($t_{eo}$, outlet temperature of evaporator) and increasing water flow rate due to the resulting increase of evaporator temperature and pressure. Fig. 8b shows that the heating capacity as well as electricity consumption of the BHP varied with return temperature $t_{eo}$. It was found that for increasing $t_{eo}$, the pressure in the evaporator increased. Therefore, the refrigerant mass flow rate increased at the same time, which made the power consumption of the compressor and the heating capacity increase as well. Fig. 8a also shows that COP yielded higher values at a forward temperature of 40 °C than for other forward temperatures, which is due to the special configuration of the BHP system as well as differences in dominant factors. All of these operating conditions are for the condenser outlet water temperature of 60 °C. The desired temperature difference between inlet and outlet of condenser decreases with the increase of forward temperature. This process is accompanied by the increase of water flow rate, so as to increase the heat transfer coefficient of the condenser. Heat transfer coefficient is the dominant factor of the BHP performance for this condition. The heat transfer rate of the condenser increased, and the COP also increased until the forward temperature reached 40 °C. When the forward temperature increased above 40 °C, the temperature difference between condenser inlet and outlet became smaller and smaller. The increase in the flow rate lead to an increase in the heat exchange of the condenser, and the flow rate of the refrigerant and pressure of the condenser increased accordingly. The compressor power increased more obviously, and thus BHP performance decreased. The compressor power increased more obviously than the heat transfer rate. The combined effect of above reasons made the system have an operating temperature with forward temperature of 40 °C that is better than other

Fig. 5. Energy flow and its distribution of BHP.
operating conditions. Fig. 8b shows that the pressure in the condenser and evaporator were affected by the forward temperature and the flow rate on each side.

Fig. 9 shows the distribution trend of COP correlation with variables (supply and return temperature, suction and discharge pressure). In total here involved a total of 40 specific cases, including 12 cases with 60 °C outlet water temperature mentioned above and 28 additional cases which were extended cases. As was expected, it was found from Fig. 9a and b that increasing evaporator temperature and decreasing condenser temperature benefitted the performance of the BHP. The COP value of range from 3.74 to 4.95 within the present test condition. Case 10 (Table 3) with COP = 4.95 yielded the best performance. Fig. 9 verified that keeping the return water temperature of the evaporator above 20 °C and thereby the evaporator pressure above 5 bar is required for operating with COP above 4.6 condition.

Fig. 10a shows the changes of BHP COP with the varying forward temperature and constant water flow rate for each side with water flow rates of 0.257 kg/s and 0.234 kg/s in condenser and evaporator, respectively. COP increased with the increase of forward temperature at constant water flow rates of condenser and evaporator, and it reached a peak point around a forward temperature of 42.5 °C.
and then decreased. The corresponding temperature difference between inlet and outlet of condenser and evaporator are shown in Fig. 10b.

Fig. 10c shows the pressure conditions of the cases mentioned above. The rate of change of the low pressure ($p_e$) curve is significantly lower than that of the high pressure side ($p_c$), which means that the forward temperature has a more significant effect on the high pressure. The compressor power consumption, the heating capacity and the cooling capacity are increasing with the increase of discharge pressure, shown as in Fig. 10d.

The Lorenz COP was calculated for each test case according to eq. (4). It ranged from 13 to 27. The Lorenz efficiency represent the ratio of the actual COP to the Lorenz COP (shown in Fig. 11) and ranged from 0.16 to 0.26. This is attributable to inefficiencies caused by non-isentropic compression and expansion as well as heat transfer at finite temperature differences.

It can be found that from Fig. 12 that the Lorenz COP is decreased with increasing pressure ratio $P_c/P_e$, which is consistent with the properties of the Lorenz cycle. On the other hand, the Lorenz efficiency was increasing with increasing pressure ratio. The same was not the case for the BHP COP.

3.3. Heat transfer process and matching between condenser and evaporator

Fig. 13 shows how the temperature curves of the water and...
refrigerant streams through the condenser and the evaporator (case 4). The upper and bottom lines represent the refrigerant cycle corresponding to the temperature profile in the condenser and the evaporator, respectively. The two middle lines of the figure represent the corresponding secondary cycle (water).

Fig. 13 shows that when the pinch point temperature difference is kept constant, the outlet water temperature of the condenser can reach the needed supply temperature (60°C) with a higher slope of the condenser water curve, which enables a lower condenser temperature. In this way, as the condenser pressure can lower, COP of the BHP could be increased. On the other hand, a lower slope of the evaporator water curve is possible to increase the evaporator temperature, which also can increase the COP of the BHP. Fig. 13 includes an extra solution of the heat exchange process indicated by a dotted line, which illustrates a case with increasing condenser water slope and decreased evaporator water slope when the outlet temperatures are kept at 60 °C for the hot water and 20 °C for the evaporator outlet, with the same value of the pinch point difference. The energy balance of the heat transfer process of the water is given by.

\[
\int_{t_0}^{t'} c_{w}m_{w}dt = \int_{Q_0}^{Q'} dQ
\]

(18)

\[
\frac{dt}{dQ} = \frac{1}{c_{w}m_{w}t_0}
\]

(19)

where, \(t_0\) is the start temperature of water; \(t'\) is the temperature of water after heat exchanger process; \(Q\) is heat exchange amount during this heat exchange process.

Based on the analysis above, the slope of the water curve can be adjusted by changing the water flow rate through the heat exchangers, as shown in equation (19). The flow rate of water and temperature can be matched according to the curve trend. Increasing the flow rate reduces the slope of the water streams. This improvement approach is based on the given working conditions and aims at getting a better match of the temperature profiles in each heat exchanger. Case 7, 9, 10 and 11 have been analyzed in Fig. 14. Fig. 14a shows that a COP of 4.58 and an exergy efficiency of
29.4% was achieved with $t_f = 40 \degree C$, $F_c = 0.235$ kg/s, $F_e = 0.204$ kg/s. Fig. 14b shows a COP of 4.76 and an exergy efficiency of 30.1% with $t_f = 40 \degree C$, $F_c = 0.265$ kg/s, $F_e = 0.204$ kg/s. Compared with Figs. 14a and b achieves a better performance not only on the COP but also on exergy efficiency, which means that when the refrigerant evaporation temperature is kept approximately unchanged, decreasing the slope of the water can lower the refrigerant condensation temperature (from 57.5 \degree C decrease to 56.2 \degree C), which improves the performance of the BHP. The reduction of the condensation pressure benefited from the combined effect of the increase in the condenser water flow rate and the increase in the inlet water temperature/forward temperature. Similar observations can be made for the evaporator.

Compared to Fig. 14c and d, it can be found that due to the increase of water flow rate of the evaporator, the heat transfer in the evaporator was enhanced, which increased the evaporator temperature. The slope of the cooling water decreased and the refrigerant evaporation line moved upward, which increased the COP of BHP in this condition to 4.95. This comparison also verified that the analysis method shown in equation (18) can be used as a reference for finding the direction to improve the performance of the BHP integration in the district heating system only requiring a few parameters.

Effectiveness ($\varepsilon$) of the heat exchanger (condenser and evaporator) has also been involved to better show the heat transfer condition of each case above. The definitions of the effectiveness ($\varepsilon$) of a heat exchanger is that it is the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate \[\varepsilon = \frac{Q}{Q_{\text{max}}}\] (20) where, $Q$ is the actual heat transfer rate, $Q_{\text{max}}$ is the maximum possible heat transfer rate.

So, for the evaporator, $Q_{\text{max}}$ equals the heat transfer rate when the water outlet temperature of the evaporator equals the refrigerant inlet temperature of the evaporator. For the condenser, $Q_{\text{max}}$ equals the heat transfer rate when the water outlet temperature of the condenser equals to the refrigerant inlet temperature of the condenser (while considering the influence of the pinch point). Thus, corresponding to the cases (Case 7, 9, 10 and 11) discussed above in Fig. 14, the effectiveness, $\varepsilon$, for each case has been calculated based on equation (20). The $\varepsilon$ value of the evaporator is 0.759 and 0.768 for Case 7 and Case 9 (Fig. 14a and b), respectively, and the corresponding $\varepsilon$ values of the condenser are 0.751 and 0.851, respectively. The results reveal that the increase in the water flow rate of the condenser enhances the heat transfer on both sides. The effectiveness of the evaporator is 0.872 and 0.710 for Case 10 and Case 11 (Fig. 14c and d), respectively, and the corresponding $\varepsilon$ value of the condenser is 0.711 and 0.734, respectively. This result indicates that although the flow rate of the evaporator increased, the match of the flow rate, the heat transfer area and the other side has a comprehensive impact on heat transfer performance.

Fig. 14. Temperature varies with transferred heat in the cases of: (a) Case 7, (b) Case 9, (c) Case 10, (d) Case 11.
3.4. Exergy efficiency and exergy destruction

The exergy efficiency was calculated for selected cases according to the equations described in section 2.3.3. The relevant measurement data for the exergy analysis of all 4 cases is shown in Table 4. Table 5 presents the exergy product (i.e. heat output from the condenser), exergy flow rate taken up from the heat source, and exergy efficiency corresponding to the cases in Table 4. These results reflect with the same outlet water temperature (60°C) of the condenser, the higher forward temperature the higher exergy flux, exergy from heat source as well as exergy efficiency. In addition, the isentropic efficiency of the compressor has been calculated based on equation of  \( \eta_{is} = W_{is} / W \) [36]. The isentropic efficiency is 54.01 %, 56.83 %, 58.35 % and 62.58 % corresponding to Case 2, Case 4, Case 8 and Case 10, respectively. It shows that isentropic efficiency of the compressor increased with the increasing of forward temperature.

Fig. 15a shows case 0.8 (in Table 5) and its exergy flows in the BHP cycle. The compressor was the main contributor to the exergy destruction, with 51.5% of the overall exergy destruction. The input energy of compressor is electricity, which is converted into, enthalpy increase whereby exergy is destroyed. Since the exergy destruction in the compressor is the biggest contributor to the overall exergy destruction, the system may be improved by improving the isentropic efficiency of the compressor, i.e. by reducing flow losses in internal valves and piping. The second highest share of exergy destruction was found for the condenser (19.3%), followed by the evaporator (16.5%) and the expansion valve (12.6%). Note that the losses assigned to the expansion valve also included pressure losses in the filter drier placed directly prior to the valve. Which means, in the extended application of the BHP, the exergy destruction in the two heat exchangers can be reduced by adapting the water flow rates, such that the temperature gradients on the water and refrigerant side match more closely. A better match between the temperature glides in the sink and source also results in a reduced pressure ratio for the compressor, since the condensation temperature is reduced, while the evaporation temperature increases. Thereby, reducing the exergy destruction in the two heat exchangers is expected to also reduce the exergy destruction in the compressor. Accordingly, optimizing the water flow rates in the heat source and sink allows to optimize the overall BHP performance. Fig. 15b shows the exergy product, exergy flow rate from power input of compressor and from external heat source, which reflect an obvious increase of the exergy product as the forward temperature increased.

Exergy destruction shares of each component for four cases with different forward temperature were listed in Fig. 16. Due to the increase of forward temperature, isentropic efficiency of the compressor increased, a decreased lift of the compressor was observed, the share of exergy destruction of the compressor decreased correspondingly. With increasing forward temperatures, the exergy destruction ratio of the condenser increased. As the water outlet temperature of the condenser was kept at 60 °C for all cases involved, thus, with the increase of water inlet temperature (forward temperature), the water flow rate of the condenser should be increased. So, water temperature difference between inlet and outlet of the condenser decreased (combined with equation (15)), which made the exergy destruction of the condenser increase. The variation of condenser exergy destruction share reveals that condenser side is more sensitive to the water inlet temperature than that of the evaporator.

4. Discussion

This paper mainly shows the performance of the BHP used for ULTDH caused by the same forward temperature and different secondary (water) flow rates under the present test conditions. The performance of BHP was strongly dependent on the both operating conditions on both sides ( evaporator and condenser) of the cycle.

The results from the exergy analysis and heat transfer process analysis in section 3 have shown that minimizing the exergy destruction in the heat exchange processes can help to improve the overall BHP performance by minimizing the exergy destruction in the heat exchange processes. In addition, as exergy destruction of condenser side is more sensitive to the forward temperature (inlet water temperature) than that of the evaporator, it is particularly important to match the condenser reasonably in the BHP system.

The exergy destruction in the compressor is the biggest contributor to the overall exergy destruction of the BHP system. Improvement of working conditions of the compressor in the BHP system and reducing the proportion of its exergy destruction can be further studied. Optimizing compressor design to improve efficiency should be the vital research point, but it was outside the scope of the present work.

Zühlsdorf et al. [37] have shown that, the exergy destruction associated to the fluid non-idealities during phase change is smaller for zeotropic mixtures than for pure fluids. Accordingly, future work will include experimental assessment of mixed refrigerants as a way to improve the performance of the BHP used in district heating systems. The results of the present study provides detailed insights in the thermal behavior of BHP. The heat transfer process curve analysis has shown that the secondary cycle flow rate control is a way to fit the cycle operation condition with the certain supply temperature (outlet temperature). Specially, it indicates the relationship between the secondary cycle temperature and the flow rate at the specific water supply temperature, which provide a reference to similar systems in choosing operation modes.

5. Conclusions

Based on the experimental results discussed above, the main conclusions of the present study on steady state behavior of the proposed BHP can be drawn as follows.

- The COP of the studied BHP system in steady state ranged from 3.84 to 4.95 for the return temperature varying from 19 °C to 31 °C. The highest COP value of 4.95 was obtained with the supply temperature of 60 °C, with forward temperature of 45 °C, secondary water flow rate of 0.257 kg/s and 0.234 kg/s in condenser and evaporator, respectively. Lorenz COP of the studies BHP is ranged from 13 to 27.
- In the adjustable inlet temperature applications, the COP value of the BHP system increases with the increase of forward temperature (inlet temperature of water cycle on both sides), which is mainly due to that the higher evaporation temperature reduced the lift of the compressor.
- Exergy analysis showed that the main exergy destruction occurs in the compressor, and that the exergy efficiency ranged from 24.9% to 33.4% within the studied cases using the BHP in the

<table>
<thead>
<tr>
<th>Case No.</th>
<th>( t_f )</th>
<th>W (kW)</th>
<th>( e_p ) (kW)</th>
<th>( e_{hs} ) (kW)</th>
<th>( \eta_{ex} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>29.3</td>
<td>3.14</td>
<td>0.799</td>
<td>0.057</td>
<td>24.9</td>
</tr>
<tr>
<td>4</td>
<td>35.0</td>
<td>3.24</td>
<td>0.966</td>
<td>0.089</td>
<td>29.0</td>
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<td>40.0</td>
<td>3.29</td>
<td>1.062</td>
<td>0.336</td>
<td>29.3</td>
</tr>
<tr>
<td>10</td>
<td>44.6</td>
<td>3.30</td>
<td>1.341</td>
<td>0.691</td>
<td>33.4</td>
</tr>
</tbody>
</table>
district heating system. Based on the exergy destruction shares of each component, the potential improvement component of the BHP system can be determined.

- The heat transfer process curve analysis is conducted may be used improving the match of water flow rates in both sides based on the heat transfer-temperature diagram and the heat transfer process of each exchanger. The exergy destruction in evaporator and condenser can be reduced by adapting the water flow rates, such that the temperature gradients on the water and refrigerant side match more closely.

**Declaration of competing interest**

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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**Nomenclature**

**Symbols**

- BHP: Booster Heat Pump
- COP: Coefficient of Performance
- \( c_p \): Specific heat capacity (J/(kg·K))
- DH: District heating
- DHW: Domestic hot water
- \( E \): Exergy flow (kW)
- \( e \): Heat loss percentage (–)
- \( h \): Specific enthalpy (kJ/kg)
- \( i \): Current (A)

**Subscripts**

- comp: Compressor
- c: Condenser
- component: Component
- d: Exergy destruction
- ExV: Expansion valve
- e: Evaporator
- ex: Exergy
- F: Fuel
- f: Forward
- H: High
- HS: Heat source
- i: Inlet
- isen: Isentropic
Greek Symbol

Δ Variation (−)

η Efficiency (%) 

ε Effectiveness of heat exchanger (−)

ṁ Mass flow rate (kg/s)

P Pressure (bar)

Q Heating capacity (kW)

T Temperature (K)

t Temperature (°C)

U Voltage (V)

W Compressor work (kW)

y Exergy destruction ratio (−)

Lor Lorenz

L Low

I Loss

o Outlet

P Production

r Refrigerant

sub Subcooling

sup Superheat

w Water

0 Dead state

1,2,3,4 State point

irr Irreversibilities

test error Error caused by the test

ρ Density (kg/m³)

ϕ Phase angle (−)

Credit author statement

Tingting Zhu: Conceptualization, Methodology, Investigation, Data curation, Writing — original draft. Torben Ommen: Methodology, Review & Editing. Wiebke Meesenburg: Writing - Review & Editing. Jan Eric Thorsen: Conceptualization, Methodology. Brian Elmegaard: Conceptualization, Writing - Review & Editing, Supervision.

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