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Evaluation on energy performance in a low temperature district heating system integrated with organic Rankine cycle*

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Abstract: Combined heat and power (CHP) coal-fired plants and heat-only boilers are still working as main heat sources in North China. These provide high temperature water on the primary side of district heating (DH) systems. There can be large temperature differences between the primary side and secondary side deployed in low temperature district heating (LTDH) of buildings. In this paper, a LTDH system, integrated with an organic Rankine cycle (ORC) system, is presented and evaluated on how to utilize the limited temperature difference between the primary and secondary sides in a substation. Two cases are illustrated of the performance of two modes (series or parallel connection) and energy efficiencies of the configurations with or without an ORC system. The results showed that the integrated ORC system could provide sufficient power for the circulating pumps in the DH system. The integration of the ORC leads to only a very slight decrement on the supply water temperature. Generally, the series mode can generate the maximum output power from an integrated ORC system. The parallel mode showed more flexibility on the adjustment of output power from the ORC system, especially where domestic hot water is needed. When the cold tap water was used as a cooling stream in the condenser of an ORC system before preparing to be the domestic hot water, it is very helpful in improving the performance of the ORC and increasing the energy efficiency of the DH system.

Key words: District heating; Low temperature district heating (LTDH); Organic Rankine cycle (ORC); Energy evaluation; Pipe network

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1 Introduction

District heating (DH) is considered as a very energy efficient way to supply heat to high heat-density regions. In Europe, DH systems have been widely used with various heat sources and technologies to provide efficient, environmental and economic benefits (Lund et al., 2014). Development of the 4th generation DH involves a multitude of technical innovations to reduce fossil fuel consumption and carbon dioxide (CO_2) emissions. For instance, the inclusion of combined heat and power (CHP), thermal energy storage (TES) and/or heat pump in DH has provided high overall thermal efficiencies of DH systems and flexible cogeneration, balancing heat, and power demands (Münster et al., 2012; Antonio et al., 2016; Ommen et al., 2016); industrial surplus heat and/or renewable energy have been working as the main sources on the primary side (Holmgren, 2006; Lund et al., 2010; Hassine and Eicker, 2013; Sartor et al., 2014). There has been more retrofitting of existing buildings to make more energy-saving, energy-efficient buildings (Abel, 1994; Chwieduk, 2001; Thomsen et al., 2005; Karlsson and Moshfegh, 2007; Thyholt and Hestnes, 2008) even to the point of zero emission (Catto, 2001; Zhu et al., 2009a, 2009b) on the secondary side.

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A number of recent reports and studies have described low temperature district heating (LTDH) systems to meet the challenges facing new heat sources and energy-efficient buildings. Coupling with the low supply water temperature and well-designed DH networks, systems with LTDH will obtain lower return water temperature and less network heat loss. Most research papers put significant emphasis on the integration of heat tanks and heat pumps in LTDH. Brand and Svendsen (2013) showed that DH supply temperature can be considerably decreased in a LTDH system with special low-temperature heat exchangers and storage tanks to set an operation temperature of 22 °C. They suggested that the use of the bypass flow in bathroom floor heating is a cost-effective solution exploiting the heat that would otherwise be lost in the LTDH network (Brand et al., 2014). Østergaard and Svendsen (2016) suggested that critical radiators must be replaced to obtain the full potential of LTDH after evaluating the actual radiator sizes and heating demands in four existing Danish single-family houses from the 1930s. Elmegaard et al. (2016) compared the energy and exergy efficiency of the conventional DH and LTDH at different temperatures. Moreover, the annual heating cost for solutions that utilize electricity for the heat pump was calculated. Tol and Svendsen (2015) presented a method for the dimensioning of the LTDH piping networks operating with a control philosophy of supplying heat. Baldvinsson and Nakata (2016) presented a high spatial resolution-based method for design and operation of a LTDH system. Köfinger et al. (2016) described the development of economically and ecologically optimized concepts for LTDH networks using four representative case studies in Austria. Colmenar-Santos et al. (2016) assessed the economic impact following as a result of the conversion of conventional stations to cogeneration plants connected to a system for DH and refrigeration. Yang et al. (2016a) analyzed and modified a system with decentralized substations to avoid the risk of Legionella and to reduce the average return temperature. The performances of five different substation configurations in single-family houses supplied with ultra-LTDH (46 °C at the heat plant and around 40 °C at the substations) were investigated (Yang et al., 2016b). Zhang et al. (2016) investigated the energy, economy, and exergy performances of the proposed domestic hot water systems in various configurations to fit with LTDH.

In North China, the DH system downtown serves both residential and commercial buildings, whose current peak space heating density is approximately 45–55 W/m^2 on average (Zhang et al., 2016). The incensement of LTDH demand for low-energy and/or passive solar buildings has been accompanied by an accelerated process of urbanization and building efficiency in large cities such as Beijing and Tianjin (Xiong et al., 2015). Nowadays biomass, waste (MOHURD, 2010) or industrial surplus heat (Li et al., 2016) is used to replace a share of the fossil fuel, but coal is the major resource in China's energy complex. The combined heat and power plants, gas-fired boilers, and coal-fired boilers serve 43.4%, 10.8%, and 40% of the DH area, respectively in China in 2010 (Lecompte et al., 2015), and still are the main DH heat sources in North China. These kinds of heat sources are especially suitable for providing a high supply temperature on the primary side of DH systems considering the higher amount of heat per unit mass of water. Meanwhile, higher supply temperature means a smaller amount of hot water flow and accordingly lower power consumption of circulating pumps on the primary side of a DH system. Furthermore, pipelines usually have good thermal insulating properties, which promises extremely small heat loss on the primary side of a DH system. Thus, there must be larger temperature difference between both sides of a heat exchanger in a DH substation where higher supply temperature is adopted on the primary side and lower heating temperature is adopted on the secondary side.

The organic Rankine cycle (ORC) is one of the most economic and efficient ways to convert low-grade thermal energy to electricity. Several ORC waste heat recovery plants are already in operation to generate power levels of 100 W to 1 MW (Chen et al., 2013). The optimal selections of working fluids in an ORC corresponding to the heat source temperature level are shown in Table 1 (Bao and Zhao, 2013; Saadatfar et al., 2013). Quoilin et al. (2013) investigated the performance of ORC systems in application, measuring thermal efficiency, working fluid pump efficiency, and back work ratio (BWR). China's research on and application of ORC technology is also developing rapidly (Feng et al., 2015; Wu et al., 2015).

Note that a proper organic fluid can be chosen to work efficiently according to certain DH operating temperatures. The working temperatures of ORC shown in Table 1 coincide with the supply temperature (e.g. 120 °C) on the primary side and the return temperature (e.g. 30 °C) on the secondary side of a DH system. To take full advantage of the large temperature difference between the two sides of a DH system in North China, a new LTDH system is proposed in this paper, where an ORC system is integrated between the primary side and secondary side in a substation. The output power of an ORC can be directly used to drive the circulating pump in the substation.

In this study, a LTDH system integrated with ORC is proposed which has not been suggested before. To improve the performance of an ORC and the thermal efficiency of DH, two connection modes (series or parallel) were discussed with a simplified model on how the conventional heat exchangers can work properly with the ORC devices. Then, two cases are presented to illustrate the practical configurations that a DH system can integrate ORC devices in a substation. The energy efficiencies and influence on supply water temperature of a substation between different modes and configurations are evaluated. The results show that integrated ORC can exploit the temperature difference between the two sides of DH and keep the high thermal efficiency of the whole system simultaneously.

Table 1 Working fluids in ORC corresponding to the heat source temperature level

Working fluid	Evaporating temperature (°C)	Condensing temperature (°C)
R113, R134a	80–140	20–30
R1234yf, R123	60–80	10–30
R245fa	70–150	10–80

Data from Bao and Zhao (2013) and Quoilin et al. (2013)

2 Systems

2.1 Organic Rankine cycle system

The working principle of the ORC is: the working fluid is pumped to a boiler/evaporator where it is evaporated, and passed through an expansion device (turbine or other expander), and then through a condenser heat exchanger where it is finally re-condensed (Quoilin et al., 2013). In the ideal cycle described by the engine's theoretical model, the expansion is isentropic and the evaporation and condensation processes are isobaric. Refrigerants and hydrocarbons are commonly used as the working fluid, such as R1234yf, R134a, R245fa, and *n*-pentane. The selection of the working fluid is of key importance. A diagram of a basic ORC system integrated with the primary and secondary sides of a LTDH is shown in Fig. 1. Here, the high-temperature supply water on the primary side provides the heat used to evaporate the working fluid and the released heat in the condenser is taken away by the cold return water on the secondary side of the DH system. With a proper working fluid operating in the ORC, the thermal energy is transformed into useful work. Thus, ORC systems could be fit for a LTDH system to provide power if appropriate hot and cold water can be available. Generally, a 50–100 °C temperature difference can be expected in a LTDH system between the primary supply-water and secondary return-water to drive an integrated ORC to generate stable power. However, the integration of an ORC should not affect the heating performance of the DH system.

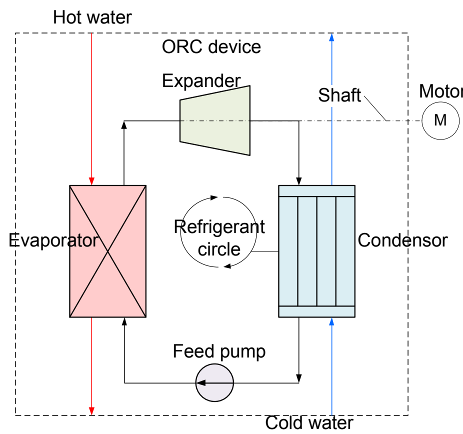

Fig. 1 An ORC system in DH

2.2 Low temperature district heating

A DH substation is a vital component in a DH system. It connects the primary heating side with the secondary heating side. The station normally has one primary circulating pump[1] (assuming that distributed variable-speed pumps are used), one secondary circulating pump[2], and heat exchangers (HEXs) as shown in Fig. 2. Here the circulating pumps are driven by the electric motors and used to circulate water in a DH system. The HEXs handle the heat transfer from the primary side into the secondary side of a DH system. If the heat loss from heat exchangers into the environment is not considered, the energy balance can be described as

$$H_{\mathrm{HEX}}^{1}=H_{\mathrm{HEX}}^{2}, \tag{1}$$

where H_{HEX}^{1} is the heat absorbed from the primary supply water; H_{HEX}^{2} is the heat released to the secondary heating water.

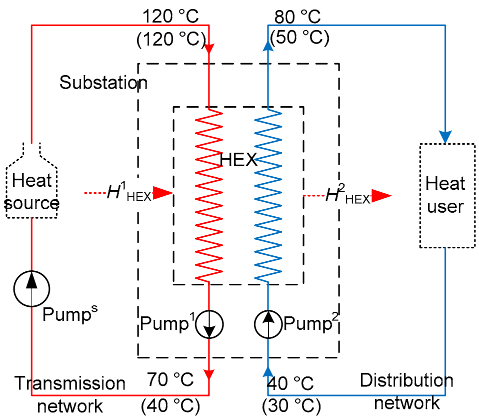

Fig. 2 A traditional substation in DH

Superscripts: 1: primary side; 2: secondary side; s: source

Energy-efficient buildings and LTDH systems are helpful for reducing the heat loss on the secondary side. For a conventional DH system, the heating temperature and the return temperature on the primary/secondary side could be 120 °C/70 °C and 80 °C/40 °C, respectively. However, for a LTDH system, the supply/return temperatures on both sides could be 120 °C/40 °C and 50 °C/30 °C, respectively as shown in Fig. 2. The heating temperature and the return temperature on the secondary side could be reduced to 45–55 °C and 25–35 °C, respectively (MOHURD, 2010). The ultralow supply/return temperature on the secondary side, such as 40 °C/20 °C has more benefit for decreasing heat loss while maintaining indoor thermal comfort (Yang et al., 2016b). Yet, an electrical heater or electricity-based heat pump must be used for domestic hot water in a LTDH system since it is required to be 55–60 °C to prevent Legionella bacteria growth (Tol and Svendsen, 2015; Baldvinsson and Nakata, 2016; Colmenar-Santos et al., 2016; Elmegaard et al., 2016; Köfinger et al., 2016).

In North China, residential heat demand is mainly for space heating in winter (Feng et al., 2015). The supply/return temperatures on the primary side and secondary side are usually 120 °C/70 °C and 80 °C/40 °C, respectively for the conventional high-temperature heating system. According to the design code for urban heating network in China, the heating water supply temperature can be 110–150 °C and the return temperature can be lower than 70 °C on the primary side if the combined heat and power plants and coal-fired boilers still serve as the main heat sources. The average heating energy demand is about 60–67 W/m^2 if no energy conversion is taken to improve energy efficiency in the existing residential buildings. It can be about 45–55 W/m^2 if energy conversion improvements can be performed for conventional buildings (MOHURD, 2010).

To apply LTDH in China, the supply temperature on the primary side must be reduced deliberately, which would have an adverse effect on the thermal efficiency of the combined heat and power plants or boilers. In addition, the benefit of keeping a higher supply temperature (such as 120 °C) on the primary side is that there can be a lower circulating flow rate which involves less power consumption of the circulating pumps. Especially for the large-scale DH systems with long transmission distance and a large number of heat users in North China, a higher supply temperature on the primary side means a much lower water flow rate. Therefore, keeping a higher supply temperature on the primary side is quite reasonable for the large-scale DH systems when the combined heat and power plants or coal-fired boilers are serving as main heat sources. The return temperature on the primary side should be reduced as much as possible (such as 40 °C), that is to say, the large-scale DH system should operate with large temperature difference and small flow rate on the primary side.

Moreover, researchers noticed that there is not too much heat loss from pipes to the environment during the transmission on the primary side since well-insulated big-diameter pipes are usually used. It means that the amount of heat loss during the transmission on the primary side is usually acceptable. In Denmark, the heat loss for transmission (primary side) and distribution (secondary side) was investigated in

2011 as 2% and 15%, respectively (Deng, 2013). For most of the DH systems in service in China, the heat loss on the primary side is much less than that of the secondary side where poorly insulated small-diameter pipes were often used in the buildings.

A new LTDH system integrated with ORC in substations is presented in this paper to meet the scenario where the high-temperature supply water and low-temperature return water are applied on the primary side and the low-temperature supply/return water are both applied on the secondary side. As shown in Fig. 2, for example, the supply/return temperatures on both sides could be 120 °C/40 °C and 50 °C/30 °C, respectively. The supply water on the primary side (120 °C) can serve as heat source for the evaporator of the ORC and the return water on the secondary side (30 °C) can serve as cold source for the condenser of the ORC.

2.3 LTDH integrated with ORC

Two configurations of new LTDH systems integrated with ORC are presented in this study: an ORC combined in series or parallel arrangement with a heat exchanger in a substation. Assuming that the supply/return temperatures on both sides of the DH system are 120 °C/40 °C and 50 °C/30 °C, respectively. The parallel connection mode is shown in Fig. 3.

In parallel connection mode as shown in Fig. 3, only part of the supply hot water (120 °C) on the primary side serves as heat source and flows into the evaporator of the ORC; part of the return water (30 °C) on the secondary side serves as cold source and flows into the condenser of the ORC. Not taking the heat and flow losses into account in the balance, the energy balance in the steady-state ORC system can be described as

$$H^1_{\mathrm{ORC}} = H^2_{\mathrm{ORC}} + W_{\mathrm{ORC}}, \tag{2}$$

where H^1_{ORC} is the heat absorbed from the primary side via the ORC evaporator; H^2_{ORC} is the heat released to the secondary side via the ORC condenser; W_{ORC} is the output power of the ORC, which can be used to drive the DH circulating pumps.

For a conventional DH system, H^1_{HEX} is the heat absorbed from the supply water on the primary side and H^2_{HEX} is the heat released to the heating water on the secondary side in a substation. The electricity from the grid, E_{pump}, should be used to drive the DH circulating pumps. Energy transfer diagrams of the DH substations with and without ORC are shown in Fig. 4. Assuming that there is no heat loss to the surroundings and both systems meet 100% of the consumer's heating needs, in the ideal case, the relationship of energy intake and energy expenditure of the DH substations with and without ORC are presented in Eqs. (3)–(5) based on the first law of thermodynamics:

$$H^1_{\mathrm{HEX}} + \Sigma E_{\mathrm{pump}} = H^1_{\mathrm{ORC}}, \tag{3}$$

$$H^2_{\mathrm{HEX}} = H^2_{\mathrm{ORC}}, \tag{4}$$

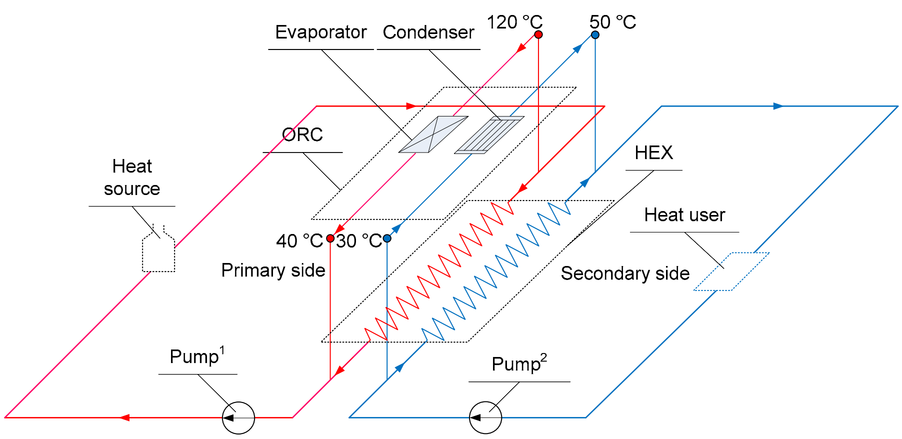

Fig. 3 A LTDH system integrated with ORC

$$\Sigma E_{\text{pump}} = W_{\text{ORC}}. \tag{5}$$

Fig. 5 shows the energy intake and energy expenditure of the DH substations with and without ORC. Here, the heat absorbed from the primary side of a conventional DH system, H^1_{HEX}, is less than that of a DH system with ORC, H^1_{ORC}. At the same time, part of the heat absorbed by ORC will convert into work, W_{ORC}, to replace the total electricity consumed by circulating pumps, ΣE_{pump}, in a conventional DH system. That is to say, the amount of energy required is equal for both of the DH substations with and without ORC, but no electricity from the grid is needed to drive the circulating pumps in a DH system with ORC. In the substation, the ORC system can take full advantage of the exergy stored in the high temperature difference between the primary and secondary sides of a LTDH system. Thus, the LTDH system integrated with ORC can be regarded as an energy cascade utilization system, by which the exergy destruction caused by the irreversible process of heat transfer in the heat exchanger of a conventional DH system can be avoided.

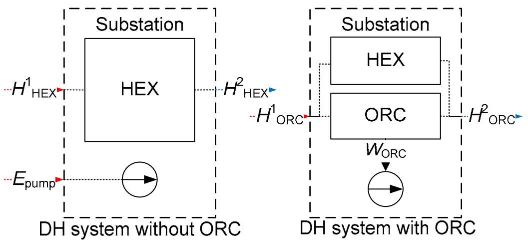

Fig. 4 Heat transfer and power consumption in two DH systems with and without ORC

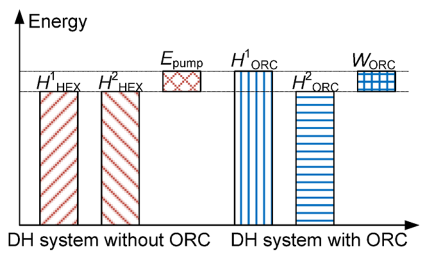

Fig. 5 Energy balance of two DH systems with and without ORC

In a substation, generally, the heat absorbed by the heat exchanger, H^1_{HEX}, is about two orders of magnitude greater than the electricity consumed by the circulating pump, ΣE_{pump}. Therefore, the application of ORC would not have a significant impact on the heating process. In the meantime, the electricity consumed by the circulating pump would be provided totally by the ORC output power. In the next sections, the detail of application of an ORC in a DH substation will be illustrated.

3 Methods

In a DH substation, a heat exchanger is used to handle the heat transfer from the main DH pipes into a secondary pipe network. The circulating pumps are driven by the electric motors in the conventional DH system and by ORC output power in a DH system integrated with an ORC. Thus, the heat exchanger, circulating pumps, and the ORC system are the main devices that would have effect on the heating process. The simplified thermal model for the devices is presented as follows.

3.1 Heat exchanger

A heat exchanger is necessary for a substation in a DH system with or without an ORC. If only space heating is provided, the conventional layout of a substation is as shown in Fig. 6. If domestic hot water is also provided in a DH system, the conventional layout of a substation is as shown in Fig. 7. Two heat exchangers are used here: the hot supply water on the primary side would pass through the HEX3 to heat the domestic hot water to achieve a suitable temperature (50–60 °C) first, and then it would pass through the HEX2 to heat the water for space heating.

For a heat exchanger, heat released from the supply hot water on the primary side is

$$H^1_{\text{HEX}} = M^1_{\text{HEX}} \bar{C}^1_{\text{p}} \left(T^1_{\text{in,HEX}} - T^1_{\text{out,HEX}} \right), \tag{6}$$

and for the heat exchanger, heat absorbed by the circulating water on the secondary side can be described as

$$H^2_{\text{HEX}} = M^2_{\text{HEX}} \bar{C}^2_{\text{p}} \left(T^2_{\text{in,HEX}} - T^2_{\text{out,HEX}} \right), \tag{7}$$

where $\bar{C}^1_{\text{p}}$ is the specific heat for the average temperature, ranging from a high $T^1_{\text{in,HEX}}$ to a low $T^1_{\text{out,HEX}}$.

$\bar{C}_{\mathrm{p}}^{2}$ is the specific heat for the average temperature, ranging from a low $T_{\mathrm{in,HEX}}^{2}$ to a high $T_{\mathrm{out,HEX}}^{2}$. $T_{\mathrm{in,HEX}}^{1}$ and $T_{\mathrm{out,HEX}}^{1}$ are the inlet and outlet temperatures of the hot supply water on the primary side passing through the heat exchanger HEX, respectively; $T_{\mathrm{in,HEX}}^{2}$ and $T_{\mathrm{out,HEX}}^{2}$ are the inlet and outlet temperatures of the cold return water on the secondary side passing through the heat exchanger HEX, respectively. M_{HEX}^{1} is the mass flow of the hot water passing through the heat exchanger HEX on the primary side. M^1 is the mass flow of the hot water on the primary side. M^2 is the mass flow of the cold water on the secondary side.

The ratio of heat loss $(H_{\mathrm{l,HEX}})$ to the total heat transfer (H_{HEX}^{1}) can be described as $\eta_{\mathrm{l,HEX}}$.

$$H_{\mathrm{l,HEX}}=\eta_{\mathrm{l,HEX}}H_{\mathrm{HEX}}^{1}, \tag{8}$$

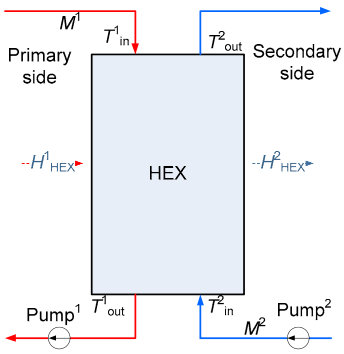

Fig. 6 DH system with one HEX (scenario 1)

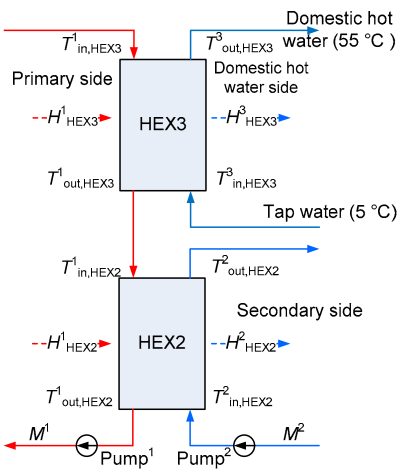

Fig. 7 DH system with HEX1 and HEX2 (scenario 3)
Superscript 3 represents domestic hot water side

$$H_{\mathrm{HEX}}^{2}=(1-\eta_{\mathrm{l,HEX}})H_{\mathrm{HEX}}^{1}. \tag{9}$$

Usually, for a well-insulated heat exchanger, the overall heat loss is less than 1% of the total heat transfer. For all of the heat exchangers mentioned in this study (HEX, HEX1, HEX2, and HEX3), we assumed that the ratio of heat loss $(H_{\mathrm{l,ORC}})$ to the total heat transfer (H_{HEX}^{1}) is 0.01 and the allowable minimum temperature difference during the heat exchange of hot and cold fluids is not less than 5 °C.

3.2 Circulating pump

Circulating pumps are arranged on both sides of a DH system to guarantee water circulation. Conventionally, the circulating pumps are driven by electric motors. The electricity consumed by the circulating pumps, E_{pump} (kW), can be written as

$$E_{\mathrm{pump}}=\frac{P_{\mathrm{pump}}M_{\mathrm{pump}}}{1000\eta_{\mathrm{pump}}\rho}, \tag{10}$$

where η_{pump} is the pump efficiency; P_{pump} is the pressure head of a circulating pump, Pa; M_{pump} is the mass flow rate of water, kg/s; ρ is the density of water, kg/m^3.

Generally, higher pump efficiency (η_{pump}=70%–80%) can be expected for high capacity water pumps on the primary side. Lower pump efficiency (η_{pump}=60%–70%) is for the low capacity water pumps on the secondary side.

For a DH system with an ORC, the expander output power can be used directly to drive the circulating pumps. No electric generator and/or electric motor are needed in the substations. If the supply of ORC output power far exceeds the energy consumption in the pumps, then the excess power can be used to drive the heat pumps, etc. In this study, the output power of the ORC is only used to drive the circulating pumps.

3.3 Organic Rankine cycle

Two configurations of new LTDH systems, an ORC combined in series or parallel arrangements with a heat exchanger in a substation, are shown in Fig. 8 and Fig. 9, respectively.

The supply hot water on the primary side serves as heat source for the ORC evaporator and the secondary return water serves as cold source for the ORC condenser. If we take the heat loss into account, the energy balance on the steady-state ORC system can be written as

$$H^1_{\mathrm{ORC}} = H^2_{\mathrm{ORC}} + W_{\mathrm{ORC}} + H_{\mathrm{l,ORC}}, \tag{11}$$

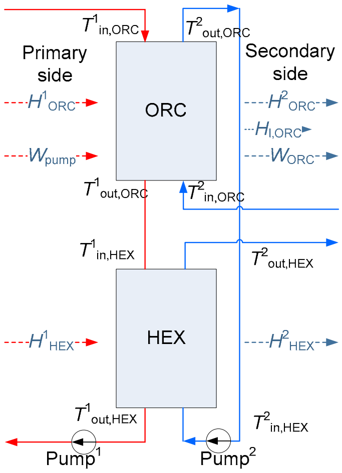

Fig. 8 DH system with HEX and ORC in series connection (scenario 2)

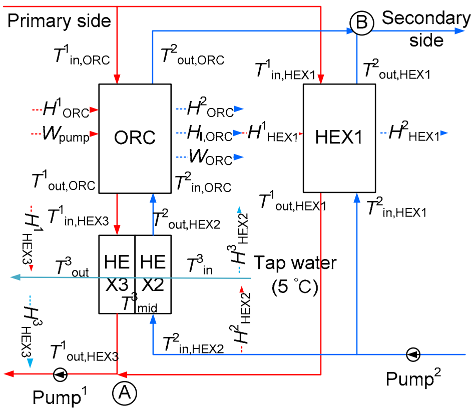

Fig. 9 DH system with HEXs and ORC in parallel connection (scenario 4)

T: temperature; H: heat; W: power. Subscripts: in: inlet; out: outlet; l: heat loss; mid: middle

The cycle efficiency of ORC, $\eta_{\mathrm{w,ORC}}$, is mainly due to the working fluid, system scale, and operating temperature. For the high supply temperature (120 °C) on the primary side and low return temperature (30 °C) on the secondary side of a DH system, up to 16% of the heat absorbed is turned into the output power of the ORC (W_{ORC}) (Quoilin et al., 2013). In this study, the performance of an ORC system was carried out on the basis of practical reference research instead of theoretical calculation. A lower ratio of the output power generated to the heat absorbed, 10%, for an ORC system is adopted here:

$$W_{\mathrm{ORC}} = \eta_{\mathrm{w,ORC}} H^1_{\mathrm{ORC}} = 10\% H^1_{\mathrm{ORC}}. \tag{12}$$

The heat losses of an ORC system ($H_{\mathrm{l,ORC}}$) to the environment were estimated to be equal to 1% of the heat transferred from the primary side to the secondary side:

$$H_{\mathrm{l,ORC}} = \eta_{\mathrm{l,ORC}} (H^1_{\mathrm{ORC}} - W_{\mathrm{ORC}}). \tag{13}$$

According to the investigation of Quoilin et al. (2013), if R245fa is used as the working fluid of an ORC system and the temperatures of heat/cold sources are about 90–150 °C/30 °C, the power consumed by the working fluid pump is equal to 4%–8% of the expander output power. The temperatures of hot supply water on the primary side and cold return water on the secondary side in a DH system are exactly fit for the operation of ORC.

In this study, the temperatures of heat source and cold source for ORC were 100–120 °C and 10–30 °C, respectively. If the ratio of the power consumed by the working fluid pump ($W_{\mathrm{p,ORC}}$) to the expander output power (W_{ORC}) is described as $\eta_{\mathrm{p,ORC}}$, then

$$W_{\mathrm{p,ORC}} = \eta_{\mathrm{p,ORC}} W_{\mathrm{ORC}}. \tag{14}$$

The net output power of ORC ($W_{\mathrm{net,ORC}}$) can be written as

$$W_{\mathrm{net,ORC}} = W_{\mathrm{ORC}} - W_{\mathrm{p,ORC}}. \tag{15}$$

Hereinafter, the value of $\eta_{\mathrm{p,ORC}}$ is assumed to be 8% according to Quoilin et al. (2013).

4 Case study

An ORC combined in series or parallel arrangements with a heat exchanger in a substation are shown in Fig. 8 and Fig. 9, respectively. Two cases were carried out to explain the operation and feature of the two configurations of DH systems with an

ORC. Comparison between the performances of the DH substations with and without ORC is presented in case 1 (an ORC combined in series with a heat exchanger) and case 2 (an ORC combined in parallel with a heat exchanger), respectively.

A larger temperature difference between the heat and cold sources yields a higher power output when keeping all else unchanged. In the following cases, the temperature differences between the heat and cold sources of the ORC systems were over 80 °C.

4.1 Case 1: in series connection

Case 1 was performed on designing a substation for a hypothetical DH area, 100000 m^2. Assuming that only space heating was required for the area and the average heat load was 50 W/m^2, the total heating demand of the area was 5000 kW. Fig. 6 shows the diagrammatic sketch of a substation with only one HEX (scenario 1). Fig. 8 shows the diagrammatic sketch of a substation with an ORC combined in series with a HEX (scenario 2).

4.1.1 Substation without ORC (scenario 1)

Here, in design operating conditions, the supply/return water temperature on the primary side was assumed to be 120 °C/40 °C and the water flow rate was 54 t/h (15 kg/s); the supply/return water temperature on the secondary side was 50 °C/30 °C and the water flow rate was 216 t/h (60 kg/s); the ratio of heat loss to the total heat transfer in a heat exchanger, $\eta_{\mathrm{l,HEX}}$, was 1%.

For the substation shown in Fig. 6, the calculated parameters based on the energy balance (Eqs. (6)–(9)) of the HEX are presented in Table 2.

As shown in Table 2, heat absorbed via the heat exchanger on the secondary side was 5003.364 kW, which would meet the design requirement (5000 kW). In addition, electricity from the electrical grid was required to drive the circulating pump in the substation. Energy consumption of the pump on the secondary side is related to network layout and the operation condition of the DH system. In North China, energy consumption of a pump on the secondary side is ranged from 0.1 to 0.3 W/m^2. Therefore, the total energy consumption of the pump on the secondary side, E^2_{pump} was 10–30 kW for case 1. If distributed variable-speed pumps were adopted on the primary side, then there must be electricity consumption on the primary side. The energy consumption of the pump on the primary side is due to the hydraulic head requirement of the pump. Usually, the hydraulic head requirement of a primary distributed variable-speed pump is no more than 1.0 MPa. Assuming that the pump efficiency, η_{pump}, was 0.75, the water flow rate on the primary side was 54 t/h (15 kg/s), the density of water was 992.4 kg/m^3 (40 °C), and the hydraulic head requirement of the pump was 1.0 MPa, then the electricity required for the pump on the primary side can be calculated as

$$\begin{aligned} E^1_{\mathrm{pump}} &= \frac{p^1_{\mathrm{pump}} M^1}{1000\eta^1_{\mathrm{pump}}\rho^1} \\ &= \frac{1.0\times10^6\times15}{1000\times0.75\times992.4} \\ &= 20.153\ \mathrm{kW}. \end{aligned} \tag{16}$$

Thus, when distributed variable-speed pumps were adopted on the primary side, the total electricity consumption for driving pumps (ΣE_{pump} (kW)) on the primary and secondary sides in the substation was calculated as

$$\Sigma E_{\mathrm{pump}} = E^1_{\mathrm{pump}} + E^2_{\mathrm{pump}} = (30.2-50.2)\ \mathrm{kW}. \tag{17}$$

4.1.2 Substation with HEX and ORC in series connection (scenario 2)

For a substation with an ORC combined in series with a HEX (Fig. 8), the hot supply water on the primary side flows first into the ORC evaporator and then into the HEX. The calculated parameters based on the energy balance (Eqs. (11)–(15)) of the ORC are

Table 2 Energy balance in HEX of substation without ORC (case 1)

Network side	M^1_{HEX} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,HEX}}$ (°C)	$T^1_{\mathrm{out,HEX}}$ (°C)	H^1_{HEX} (kW)	E^1_{pump} (kW)
Primary	54.00	4.212	120.00	40.00	5053.903	≤20.153
Network side	M^2_{HEX} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,HEX}}$ (°C)	$T^2_{\mathrm{out,HEX}}$ (°C)	H^2_{HEX} (kW)	E^2_{pump} (kW)
Secondary	216.00	4.179	30.00	49.96	5003.364	10.00–30.00

presented in Table 3. Here, the energy conversion efficiency ($\eta_{\rm w,ORC}$) is 10%; the ratio of power consumed by the working fluid pump to the expander output power ($\eta_{\rm p,ORC}$) is 8%; the ratio of heat loss to the heat transferred from the primary side to the secondary side ($\eta_{\rm l,ORC}$) is 1%.

As shown in Table 3, the calculated outlet water temperature of the ORC condenser, $T^2_{\rm out,ORC}$ was 32.26 °C, which was only 2.26 °C higher than the designed return temperature (30 °C) on the secondary side. The low cooling temperature for the ORC condenser would help to keep the ORC operating stably.

After flowing out of the ORC evaporator, the hot water on the primary side flows into the HEX to heat up the return water coming from the ORC condenser on the secondary side. For the substation shown in Fig. 8, the calculated parameters based on the energy balance (Eqs. (6)–(9)) of the HEX are presented in Table 4.

As shown in Table 4, the calculated heating temperature at the secondary side ($T^2_{\rm out,HEX}$) was 49.69 °C, which was 0.31 °C lower than the designed heating temperature (50 °C).

The results above showed that the integration of an ORC would have a little effect on the amount of heat transfer in the DH system. Thus, the integration of an ORC in series with HEX can meet the heating requirements without changing the designed heating parameters on the primary side.

However, if the heating temperature at the secondary side was strictly required to be 50 °C, it would help to increase the supply temperature or flow rate on the primary side slightly, which will be discussed in the next section.

According to the values shown in Tables 3 and 4, the total heat released from the supply hot water on the primary side to the ORC and HEX (H^1) is

$$H^1 = H^1_{\rm ORC} + H^1_{\rm HEX} = 5049.9\ \text{kW}. \tag{18}$$

The heat absorbed by the return water on the secondary side (H^2) is

$$H^2 = H^2_{\rm ORC} + H^2_{\rm HEX} = 4936.022\ \text{kW}. \tag{19}$$

Comparing the DH system without and with ORC shown in Fig. 6 and Fig. 8, the heat absorbed by the return water on the secondary side in scenarios 1 and 2 was 5003.364 kW and 4936.022 kW, respectively. The impact of the integration of an ORC (in series with the HEX in a substation) on the heat absorbed by the return water on the secondary side can be calculated by

$$\frac{5003.364 - 4936.022}{5003.364} = 1.35\%.$$

The total electricity consumption for driving pumps ($\Sigma E_{\rm pump}$=(30.2–50.2) kW) was lower than the net output power of ORC ($W_{\rm net}$=58.478 kW). The energy required to drive the circulating pumps on both primary and secondary sides of a DH system can be provided by the ORC output power. That is to say, the output power of an ORC can be used directly to drive

Table 3 Energy balance in a substation with an ORC combined in series with a HEX (for ORC, case 1)

Network side	$M^1_{\rm ORC}$ (t/h)	$\bar{C}^1_{\rm p}$ (kJ/kg)	$T^1_{\rm in,ORC}$ (°C)	$T^1_{\rm out,ORC}$ (°C)	$H^1_{\rm ORC}$ (kW)
Primary	54.00	4.238	120.00	110.00	635.700
Network side	$M^2_{\rm ORC}$ (t/h)	$\bar{C}^2_{\rm p}$ (kJ/kg)	$T^2_{\rm in,ORC}$ (°C)	$T^2_{\rm out,ORC}$ (°C)	$H^2_{\rm ORC}$ (kW)
Secondary	216.00	4.179	30.00	32.26	566.672
Device	$H_{\rm l,ORC}$ (kW)	$W_{\rm ORC}$ (kW)	$W_{\rm pump}$ (kW)	$W_{\rm net}$ (kW)	
ORC	5.721	63.560	5.085	58.478	

Table 4 Energy balance in a substation with an ORC combined in series with a HEX (for HEX, case 1)

Network side	$M^1_{\rm HEX}$ (t/h)	$\bar{C}^1_{\rm p}$ (kJ/kg)	$T^1_{\rm in,HEX}$ (°C)	$T^1_{\rm out,HEX}$ (°C)	$H^1_{\rm HEX}$ (kW)
Primary	54.00	4.204	110.00	40.00	4414.20
Network side	$M^2_{\rm HEX}$ (t/h)	$\bar{C}^2_{\rm p}$ (kJ/kg)	$T^2_{\rm in,HEX}$ (°C)	$T^2_{\rm out,HEX}$ (°C)	$H^2_{\rm HEX}$ (kW)
Secondary	216.00	4.178	32.26	49.69	4369.35

the circulating pumps in the substation without using a generator and a motor.

Thus, the ORC system can take advantage of the exergy stored in the high temperature difference between the primary and secondary sides of a LTDH system.

4.2 Case 2: in parallel connection

Case 2 was performed on designing a substation for a hypothetical DH area of 35 000 m^2. The average heat load for space heating was 50 W/m^2. The total space heating demand of the area was 1750 kW. Hot water is also needed in this area with a water flow rate of 7.2 t/h and water temperature 55–60 °C. Fig. 7 shows the diagrammatic sketch of a substation with only two HEXs (scenario 3). Fig. 9 shows the diagrammatic sketch of a substation with an ORC combined in parallel with a HEX (scenario 4).

4.2.1 Substation without ORC (scenario 3)

Here, the hot supply water on the primary side flows firstly into the HEX3 and then into the HEX2. In design operating conditions, the supply/return water temperature on the primary side was assumed to be 120 °C/40 °C and the water flow rate (M^1) was 24 t/h (6.67 kg/s); the supply/return water temperature on the secondary side for space heating was 50 °C/30 °C and the water flow rate (M^2) was 79.20 t/h (22 kg/s). The heat exchanger HEX2 was used to warm up the heating water on the secondary side to 50 °C for space heating. The heat exchanger HEX3 was used to heat up the tap water (5 °C) to 55–60 °C to supply domestic hot water (flow rate M^3=7.20 t/h). The ratio of heat loss to the total heat transfer in either HEX2 or HEX3 was assumed to be 1%.

For the substation shown in Fig. 7, the calculated parameters based on the energy balance (Eqs. (6)–(9)) of the HEX3 and HEX2 are presented in Table 5.

As shown in Table 5, the tap water was heated from 5 °C to 55 °C. The heating water on the secondary side was warmed from 30 °C up to 49.03 °C, which was slightly lower than the designed heating temperature (50 °C). Thus, the system shown in Fig. 7 can meet the space heating and domestic hot water requirements. The total heat absorbed for space heating and domestic hot water supplement from the HEX2 and HEX3 was 1749.600 kW. The return water temperature on the primary side was 42.25 °C. The system output can meet the design requirements.

As discussed in Section 4.1.1, electricity from the electrical grid was required to drive the circulating pump and the energy consumption of a heating pump on the secondary side is ranged from 0.1 to 0.3 W/m^2. Therefore, the total energy consumption of the heating pump on the secondary side, E^2_{pump} was 3.5–10.5 kW for case 2. In addition, a pump is arranged for the circulation of domestic hot water. Assuming that the electricity consumption of the domestic hot water pump (E^3_{pump}) is as proportional to the design flow as that of the space-heating water pump, then E^3_{pump} can be described as

$$E^3_{\text{pump}} = \frac{M^3}{M^2} E^2_{\text{pump}}. \tag{20}$$

The calculated E^3_{pump} was 0.318–0.955 kW (design flow: M^2=79.2 t/h, M^3=7.2 t/h). The total electricity consumption for driving pumps (ΣE_{pump} (kW)) on the secondary sides was calculated by

$$\Sigma E_{\text{pump}} = E^2_{\text{pump}} + E^3_{\text{pump}} = (3.8-11.5)\ \text{kW}. \tag{21}$$

Table 5 Energy balance in HEX3 and HEX2 in the substation without ORC (case 2)

HEX3	Primary side	M^1_{HEX3} (t/h)	$\bar{C}^1_{\text{p}}$ (kJ/kg)	$T^1_{\text{in,HEX3}}$ (°C)	$T^1_{\text{out,HEX3}}$ (°C)	H^1_{HEX3} (kW)
		24.00	4.234	120.00	105.04	422.300
	Domestic hot water	M^3_{HEX3} (t/h)	$\bar{C}^3_{\text{p}}$ (kJ/kg)	$T^3_{\text{in,HEX3}}$ (°C)	$T^3_{\text{out,HEX3}}$ (°C)	H^3_{HEX3} (kW)
		7.20	4.192	5.00	55.00	419.200
HEX2	Primary side	M^1_{HEX2} (t/h)	$\bar{C}^1_{\text{p}}$ (kJ/kg)	$T^1_{\text{in,HEX2}}$ (°C)	$T^1_{\text{out,HEX2}}$ (°C)	H^1_{HEX2} (kW)
		24.00	4.198	105.04	42.25	1757.300
	Secondary side (for space heating)	M^2_{HEX2} (t/h)	$\bar{C}^2_{\text{p}}$ (kJ/kg)	$T^2_{\text{in,HEX2}}$ (°C)	$T^2_{\text{out,HEX2}}$ (°C)	H^2_{HEX2} (kW)
		79.20	4.179	30.00	49.03	1749.600

Note that the electricity consumption of the distributed variable-speed pump on the primary side was not included in case 2. The reason is that less hot supply-water on the primary side flowing through the ORC evaporator will lead to less expander output power when an ORC is combined in parallel with a HEX. The expander output power will be only used to meet the pump energy consumption on the secondary side to drive space heating and domestic hot water circulating pumps.

4.2.2 Substation with HEX and ORC in parallel connection (scenario 4)

An ORC and a HEX working in parallel in a substation are especially fit for the needs of both space heating and domestic hot water. Two heat exchangers: HEX2 and HEX3 were adopted to heat up the tap water. As shown in Fig. 9, on the primary side, the hot supply water (M^1) flows in a bifurcate and the water flows will be oriented towards the ORC (M^1_{ORC}) and the HEX1 (M^1_{HEX1}), respectively. After flowing out of the ORC, the water flow $M^1{}_{\mathrm{ORC}}$ will pass through the HEX3. On the secondary side, the cold return water (M^2) will separate into two flows, which will be oriented towards the HEX2 (M^2_{HEX2}) and the HEX1 (M^2_{HEX1}). After flowing out of the HEX2, the water flow $M^2{}_{\mathrm{HEX2}}$ will pass through the ORC. In addition, the tap water (M^3) will flow in series into the HEX2 and HEX3 to supply domestic hot water for users. We assume that the ratio of heat loss to the total heat transfer in each heat exchanger was 1%. For the substation shown in Fig. 9, the calculated parameters of the ORC based on the energy balance (Eqs. (11)–(15)) are presented in Table 6.

As shown in Table 6, on the secondary side, the return water will be cooled first by the low temperature tap water to 10 °C in the HEX2 and then be warmed to 25.03 °C in the ORC. Here, lower cooling temperature will be of benefit for the capacity of the ORC module.

The calculated parameters based on the energy balance of the HEX1, HEX2, and HEX3 are presented in Table 7.

Table 6 Energy balance in the substation with an ORC combined in parallel with a HEX (for ORC, case 2)

Network side	M^1_{ORC} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,ORC}}$ (°C)	$T^1_{\mathrm{out,ORC}}$ (°C)	H^1_{ORC} (kW)
Primary	6.00	4.231	120.00	100.00	141.033
Network side	M^2_{ORC} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,ORC}}$ (°C)	$T^2_{\mathrm{out,ORC}}$ (°C)	H^2_{ORC} (kW)
Secondary	7.20	4.187	10.00	25.03	125.861
Device	$H_{\mathrm{l,ORC}}$ (kW)	W_{ORC} (kW)	W_{pump} (kW)	W_{net} (kW)	
ORC	1.269	14.102	1.128	12.974	

Table 7 Energy balance in a substation with an ORC combined in parallel with a HEX (for HEX1, HEX2, and HEX3, case 2)

Device	Side					
HEX1	Primary side	M^1_{HEX1} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,HEX1}}$ (°C)	$T^1_{\mathrm{out,HEX1}}$ (°C)	H^1_{HEX1} (kW)
		18.00	4.212	120.00	35.00	1790.100
	Secondary side (for space heating)	M^2_{HEX1} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,HEX1}}$ (°C)	$T^2_{\mathrm{out,HEX1}}$ (°C)	H^2_{HEX1} (kW)
		72.00	4.180	30.00	51.20	1772.320
HEX2	Secondary side (for space heating)	M^2_{HEX2} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,HEX2}}$ (°C)	$T^2_{\mathrm{out,HEX2}}$ (°C)	H^2_{HEX2} (kW)
		7.20	4.186	30.00	10.00	167.440
	Domestic hot water	M^3_{HEX2} (t/h)	$\bar{C}^3_{\mathrm{p}}$ (kJ/kg)	$T^3_{\mathrm{in,HEX2}}$ (°C)	$T^3_{\mathrm{out,HEX2}}$ (°C)	H^3_{HEX2} (kW)
		7.20	4.192	5.00	24.72	165.332
HEX3	Primary side	M^1_{HEX3} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,HEX3}}$ (°C)	$T^1_{\mathrm{out,HEX3}}$ (°C)	H^1_{HEX3} (kW)
		6.00	4.199	100.00	60.00	279.933
	Domestic hot water	M^3_{HEX3} (t/h)	$\bar{C}^3_{\mathrm{p}}$ (kJ/kg)	$T^3_{\mathrm{in,HEX3}}$ (°C)	$T^3_{\mathrm{out,HEX3}}$ (°C)	H^3_{HEX3} (kW)
		7.20	4.181	24.72	57.86	277.117

As shown in Table 7, the tap water was heated from 5 °C to 57.86 °C via the HEX2 and HEX3, which would meet the need of domestic hot water. The temperature of domestic hot water was 2.86 °C higher than in scenario 3. As shown in Fig. 9, the split hot supply water on the primary side flows (M^1_{ORC} and M^1_{HEX1}) through the ORC and the HEX1 separately at first and then joins at point A. The enthalpy of the return water at point A on the primary side, h^1_{A}, can be calculated by Eq. (22). Its temperature can be obtained via the property diagram of water. The parameters of the return water at point A on the primary side are listed in Table 8. As shown in Table 8, the return water temperature on the primary side was 41.25 °C, which was slightly higher than the design value (40 °C).

$$h^1_{\mathrm{A}} = \frac{h^1_{\mathrm{out,HEX3}} M^1_{\mathrm{HEX3}} + h^1_{\mathrm{out,HEX1}} M^1_{\mathrm{HEX1}}}{M^1_{\mathrm{HEX3}} + M^1_{\mathrm{HEX1}}}, \tag{22}$$

where $h^1_{\mathrm{out,HEX3}}$ and $h^1_{\mathrm{out,HEX1}}$ are the enthalpies of the hot supply water at the outlets of HEX3 and HEX1 on the primary side, respectively; M^1_{HEX3} and M^1_{HEX1} are the mass flows of hot supply water passing through HEX3 and HEX1 on the primary side, respectively.

As shown in Fig. 9, the split return water flows (M^2_{HEX1} and M^2_{HEX2}) on the secondary side passing through the HEX2 and the HEX1 separately at first and then joins at point B for space heating. The enthalpy of the supply water at point B on the secondary side can be calculated by Eq. (23). Its temperature can be obtained via the property diagram of water. The parameters of the supply water at point B are listed in Table 8.

The temperature of the supply water on the secondary side was 48.82 °C, as shown in Table 8, which was slightly lower than the design value (50 °C). The temperature was 0.21 °C lower than that in scenario 3 (49.03 °C, Table 5).

$$h^1_{\mathrm{B}} = \frac{h^2_{\mathrm{out,ORC}} M^2_{\mathrm{ORC}} + h^2_{\mathrm{out,HEX1}} M^2_{\mathrm{HEX1}}}{M^2_{\mathrm{ORC}} + M^2_{\mathrm{HEX1}}}, \tag{23}$$

where $h^2_{\mathrm{out,ORC}}$ and $h^2_{\mathrm{out,HEX1}}$ are the enthalpies of the cold return water at the outlets of ORC and HEX1 on the secondary side, respectively; M^2_{ORC} and M^2_{HEX1} are the mass flows of cold return water passing through ORC and HEX1 on the secondary side, respectively.

A comparison of the parameters for space heating, domestic hot water, and the supply/return water on the primary side with and without ORC in case 2 is given in Table 9.

As shown in Table 9, for the temperatures of the return water on the primary side (T^1_{out}), the space heating water (T^2_{out}) and the domestic hot water (T^3_{out}) on the secondary side, there were only small differences between the DH substations with and without ORC in case 2 if the water flows were kept constant. For the DH substation without ORC, the total heat absorbed for space heating and domestic hot water was calculated as

$$H^2 + H^3 = 1749.600 + 419.200 = 2168.800\ \text{(kW)}.$$

For the DH substation with an ORC combined in parallel with a HEX, the total heat absorbed for space heating and domestic hot water supply was calculated as

Table 8 Parameters of the primary return water at point A and the secondary supply water at point B

Point A	M^1_{HEX3} (t/h)	$T^1_{\mathrm{out,HEX3}}$ (°C)	$h^1_{\mathrm{out,HEX3}}$ (kJ/kg)	h^1_{A} (kJ/kg)	T^1_{A} (°C)
	6.00	60.00	251.558	173.206	41.25
	M^1_{HEX1} (t/h)	$T^1_{\mathrm{out,HEX1}}$ (°C)	$h^1_{\mathrm{out,HEX1}}$ (kJ/kg)		
	18.00	35.00	147.089		
Point B	M^2_{ORC} (t/h)	$T^2_{\mathrm{out,ORC}}$ (°C)	$h^2_{\mathrm{out,ORC}}$ (kJ/kg)	h^1_{B} (kJ/kg)	T^1_{B} (°C)
	7.20	25.04	105.412	204.846	48.82
	M^2_{HEX1} (t/h)	$T^2_{\mathrm{out,HEX1}}$ (°C)	$h^2_{\mathrm{out,HEX1}}$ (kJ/kg)		
	72.00	51.20	214.783		

$$H^2 + H^3 = 1730.431 + 442.904 = 2173.335 \text{ (kW)}.$$

Then, the impact of ORC on the total heat absorbed for space heating and domestic hot water supply was

$$\left|\frac{2168.800 - 2173.335}{2168.800}\right| = 0.209\%.$$

The result showed that the integration of ORC in a DH substation would have only a minimal effect on the space heating and domestic hot water supply.

In addition, electricity from an external power grid was needed to drive the circulating pumps in a DH substation without ORC (scenario 3), while the net output power of the ORC was enough for pump driving in scenario 4.

In case 2, the return water (for space heating) on the secondary side was cooled down to 10 °C by the tap water in the HEX2, and this showed positive effects on the performance and operational reliability of the ORC. Accordingly, the tap water was preheated in the HEX2.

5 Discussion

5.1 Primary supply water temperature

The integration of an ORC in a DH substation will take full advantage of the high temperature difference between the primary and secondary sides of a DH system. The expander output power of an ORC can be used to drive the circulating pumps for saving electricity from the grid. Considering that the DH heat supply is generally two orders of magnitude greater than the energy consumption of the circulating water pumps, the integration of an ORC in a DH substation will have a minimal impact on space heating and domestic hot water supply.

However, the results of the case studies showed that if an ORC was integrated, the total heat absorbed on the secondary side for space heating and domestic hot water would be slightly decreased when the total heat released from the primary side is kept constant. If the total heat absorbed on the secondary side for space heating and domestic hot water needed to fully meet the users' requirement, there must be more heat released from the primary side to compensate for the heat transferred into work via ORC.

There are three ways to increase the heat released from the primary side. The first way is to increase the flow rate of the supply water on the primary side, which will lead to higher power consumption of the circulating pumps and even hydraulic imbalance. The second is to decrease the temperature of the return water at the primary side, which will lead to the improvement of the heat efficiency at the primary side, the need of a larger heat transfer area and stable return temperature on the secondary side. The third way is to slightly raise the temperature of the hot water on the primary side.

The third way was adopted to keep the total heat absorbed on the secondary side to fully meet the users' requirement. Case 1 (a substation with an ORC combined in series with a HEX) was taken as an example. As shown in Table 3, the calculated heating temperature at the secondary side was 0.31 °C lower than the designed heating temperature (50 °C) when the expander output power was 63.560 kW. To keep the heating temperature at the designed value (50 °C), the temperature of the hot water on the primary side

Table 9 Comparison between DH substations with and without ORC (case 2)

DH system	On the primary side				On the secondary side			
	M^1 (t/h)	T^1_{in} (°C)	T^1_{out} (°C)	H^1 (kW)	M^2 (t/h)	T^2_{in} (°C)	T^2_{out} (°C)	H^2 (kW)
Without ORC	24.0	120	42.25	2191.100	79.2	30	49.03	1749.600
With ORC*	24.0	120	41.25	2211.000	79.2	30	48.82	1730.431

DH system	Domestic hot water				Substation	
	M^3 (t/h)	T^3_{in} (°C)	T^3_{out} (°C)	H^3 (kW)	E_{pump} (kW)	W_{net} (kW)
Without ORC	7.2	5	55.00	419.200	3.8–11.5	–
With ORC*	7.2	5	57.86	442.904	3.8–11.5	12.8

* ORC combined in parallel with a HEX in a substation

must be raised. For the substation shown in Fig. 8, the calculated parameters based on the energy balance of the ORC and HEX are presented in Table 10.

As shown in Table 10, the temperature of the hot supply water ($T^1_{\mathrm{in,ORC}}$) on the primary side should only be 1.2 °C higher to keep the heating temperature at the designed value (50 °C). The small increment for the hot supply water on the primary side will not decrease the heat efficiency of the combined heat and power plants or gas-fired/coal-fired boilers. In addition, if industrial surplus heat or renewable energy serves as the heat source, a suitable ORC system should be selected to fit the temperature of the hot supply water (80–85 °C) on the primary side (Bao and Zhao, 2013). In this situation, the expander output power of the ORC can be used to drive not only the circulating pumps but also the heat pump. For a LTDH, heat pumps are usually adopted to heat domestic hot water (Yang et al., 2016c). The study on a low temperature ORC will be carried out in our future work.

5.2 Series vs. parallel circuits

In case 1, the calculation was carried out for a substation with an ORC combined in series with a HEX. As shown in Fig. 8, the hot supply water on the primary side flows first into the ORC evaporator and then into the HEX. In this condition, the net output power of the ORC was much higher than the power needed for driving circulating pumps on the primary and secondary sides. Therefore, an ORC combined in series with a HEX is suitable for a DH substation with high-energy consumption in pumps. If domestic hot water is required, tap water (5 °C) can be designed to absorb successively the heat from the return water on the secondary side. An improved configuration of the LTDH integrated with ORC in series connection can be organized as shown in Fig. 10. In this configuration, the temperature of water flowing into the ORC condenser will be colder, which is helpful for keeping the ORC system operating stably.

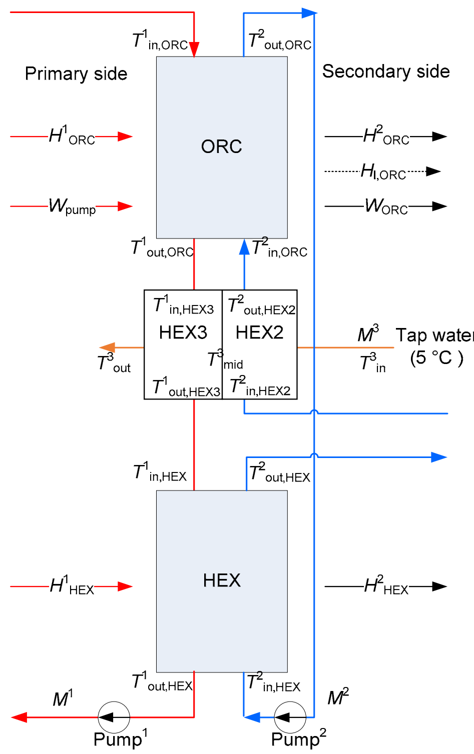

Fig. 10 An improved configuration for a LTDH integrated with an ORC in series

Table 10 Energy balance in a substation with an ORC combined in series with a HEX (case 1, Section 5.1)

For ORC	Primary side	M^1_{ORC} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,ORC}}$ (°C)	$T^1_{\mathrm{out,ORC}}$ (°C)	H^1_{ORC} (kW)
		54.00	4.240	121.20	111.20	635.900
	Secondary side	M^2_{ORC} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,ORC}}$ (°C)	$T^2_{\mathrm{out,ORC}}$ (°C)	H^2_{ORC} (kW)
		216.00	4.179	30.00	32.26	566.672
	ORC	$H_{\mathrm{l,ORC}}$ (kW)	W_{ORC} (kW)	W_{pump} (kW)	W_{net} (kW)	
		5.721	63.560	5.085	58.473	
For HEX	Primary side	M^1_{HEX} (t/h)	$\bar{C}^1_{\mathrm{p}}$ (kJ/kg)	$T^1_{\mathrm{in,HEX}}$ (°C)	$T^1_{\mathrm{out,HEX}}$ (°C)	H^1_{HEX} (kW)
		54.00	4.204	111.20	40.00	4489.870
	Secondary side	M^2_{HEX} (t/h)	$\bar{C}^2_{\mathrm{p}}$ (kJ/kg)	$T^2_{\mathrm{in,HEX}}$ (°C)	$T^2_{\mathrm{out,HEX}}$ (°C)	H^2_{HEX} (kW)
		216.00	4.178	32.26	50.00	4447.060

In case 2, the calculation was carried out for configuration where an ORC is combined in parallel with a HEX, where only part of the supply hot water on the primary side and return water on the secondary side flowed through the ORC devices. In such parallel circumstance, the ORC system and the HEX1 will have no interaction with each other. Designers can determine the share of water flowing into the ORC and the HEX1 flexibly according to the practical heating demand. In addition, the net output power of the ORC in case 2 (in parallel) will be less than that in case 1 (in series). Therefore, an ORC combined in parallel with a HEX is more suitable for a DH substation with low energy consumption in pumps.

6 Conclusions

In this paper, the idea that LTDH systems can integrate with an ORC to exploit the larger temperature difference between the primary and secondary sides of DH was proposed and analyzed. First, the simplified thermal models for the devices as heat exchanger, circulating pump, and ORC module were presented. Two configurations whereby an ORC system can be integrated with a LTDH system in series or parallel connection were proposed. Then two studies were carried out to analyze the thermodynamic performance of the LTDH systems integrated with or without an ORC. Finally, the effects on primary supply water temperature variation by the proposed systems were discussed. An improved configuration for series connection was also presented.

1. The large temperature difference between the primary and the secondary sides of a DH system (such as 120 °C to 30 °C) offers available energy for exploitation, and this is just suitable for an ORC. The expander output power of an ORC can be directly used to drive the local circulating pumps to reduce electricity consumption. The LTDH system with an ORC is an integrated cascade energy utilization system.

2. The integration of an ORC in the DH system has little impact on the heating supply temperature. The integration of an ORC in series and in parallel with the HEX in a substation led to a 1.35% and 0.209% decrease on the value of heat absorbed by the return water on the secondary side, respectively. The expander output power of the ORC was enough to drive the DH circulating pumps in these cases.

3. For the configuration where an ORC is combined in series with a HEX, the hot supply water on the primary side flows first into the ORC evaporator and then into the HEX. It is suitable for a DH substation with high-energy consumption by pumps. In case 1 (a hypothetical DH area, 100000 m^2), the calculated heating temperature on the secondary side was 0.31 °C lower than the designed value (50 °C). However, the net output power of the ORC was 58.478 kW for driving circulating pumps. If maintaining the heating temperature at the designed value (50 °C) is required, the temperature of the hot supply water on the primary side should only be raised by 1.2 °C.

4. For the configuration where an ORC is combined in parallel with a HEX, only part of the supply hot water on the primary side and return water on the secondary side flowed through the ORC devices. The ORC system and the HEX1 in parallel will not lead to interaction effects on their own thermodynamic performance. An ORC combined in parallel with a HEX is suitable for a DH substation with low energy consumption in pumps or with the need for domestic hot water. In case 2 (a hypothetical DH area, 35000 m^2), the calculated heating temperature at the secondary side was 1.18 °C lower than the designed heating temperature (50 °C). However, the temperature of domestic hot water was 2.86 °C higher than the designed value (55 °C). The net output power of ORC was 12.974 kW for driving circulating pumps.

In this paper, a new LTDH system with an ORC integrated in the substations was proposed and analyzed. The feasibility of the proposed system is theoretically reasonable. However, most ORC systems in operation cannot be commercially applied in this manner at present. The economic technical analysis will be added to consolidate the real practice of the proposed system in our work in the near future.

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中文概要

题　目：采用有机朗肯循环的低温区域供热系统性能分析

目　的： 在中国北方，需要采用更多的低温区域供热系统来适应节能型建筑的低能耗状况，而传统高温热源和低能耗建筑之间存在较大的能量品质差异。本文旨在研究一种新能源梯级利用系统，充分利用这一能量差异，并减少供热管网的热损失。

创新点： 1. 提出一种有机朗肯循环（ORC）与低温区域供热整合的系统，由有机朗肯循环提供水泵动力；2. 建立系统模型，模拟不同连接方式下系统的供热性能和节能效果。

方　法： 1. 通过理论建模，构建有机朗肯循环、热力站换热器及循环水泵之间的能量转换关系；2. 通过案例分析，比较换热器与 ORC 设备串联或并联两种供水和回水的组织方式的供热性能与节能性能。

结　论： 1. 对于低温区域供热系统，一次侧供水温度在 110～150 °C，二次侧回水温度为 30 °C 或更低温度时，可在热力站内整合 ORC 设备。2. ORC 设备输出功对供热的不利影响很小，ORC 设备可产生足够的动力满足热力站内水泵的能耗。3. ORC 与热力站换热器间可采用串联或并联的连接方式；对于热力站内水泵能耗大的场合，可采用串联方式，ORC 设备可最大限度地利用两侧温差；采用并联方式时，ORC 设备只与一部分一次侧供水和部分二次侧回水进行换热，并联设备之间的热力工况相互影响小。

关键词： 区域供热；低温供热；有机朗肯循环；能量评价；管网