

**Enhancing the efficiency of power generation through the utilisation of
LNG cold energy by a dual-fluid condensation Rankine cycle system**

Fei Wang^a, Panfeng Li^b, Limei Gai^a, Yujie Chen^c, Baikang Zhu^a, Xianlei Chen^d, Hengcong
Tao^a, Petar Sabev Varbanov^e, Farooq Sher^f, Bohong Wang^{a,*}

^a National-Local Joint Engineering Laboratory of Harbour Oil & Gas Storage and
Transportation Technology/Zhejiang Key Laboratory of Petrochemical Environmental
Pollution Control, Zhejiang Ocean University, No.1, Haida South Road, 316022, Zhoushan,
P.R. China

^b ENN (Zhou Shan) LNG Co., Ltd., Zhoushan 316021 P.R. China

^c School of Mechanical Engineering, Beijing Institute of Petrochemical Technology, Beijing,
102617, China

^d Zhoushan Institute of Calibration and Testing for Quality and Technology Supervision,
Zhoushan 316012 P.R. China

^e Sustainable Process Integration Laboratory – SPIL, NETME Centre, Faculty of Mechanical
Engineering, Brno University of Technology – VUT Brno, Technicka 2896/2, 616 69, Brno,
Czech Republic

^f Department of Engineering, Nottingham Trent University, Nottingham NG11 8NS, United
Kingdom

* Corresponding author: wangbh@zjou.edu.cn (Bohong Wang)

Abstract

As a clean energy source with high calorific value and low pollution, liquefied natural gas (LNG) has gained much attention and increased fast in the current energy market. It also has considerable cold energy resources that can be used to generate electricity during the regasification process. To fully utilise the cold energy of LNG, a double-Rankine cycle power generation system that incorporates heat exchange between LNG cold energy utilisation and a propane-ethylene cycle working medium is proposed and optimised. The optimisation is based on the Process Integration method, which uses Pinch Analysis to develop a Heat Exchange Network. Upon a specified LNG flow rate of 15.6 kg/s and natural gas delivery pressure of 7.85 MPaG, a retrofit case of the optimised LNG cold energy system generates a power of 1917.21 kW. A 28.6 % increase in power generation efficiency compared with the existing case. The result showed that by employing the Process Integration method, this study maximises the use of LNG cold energy through heat exchange with various working media, effectively addressing power generation efficiency issues. This approach is important in reducing power generation costs, minimising environmental impact, and advancing resource sustainability. Furthermore, it serves as a valuable reference for enhancing power generation efficiency by utilising LNG cold energy.

Keywords: Pinch Analysis; Heat exchanger network; LNG; Propane-ethylene cycle; Power generation

1 Introduction

The increasingly severe environmental pollution issues and the escalating disparity between energy supply and demand have garnered heightened attention towards reducing greenhouse gas emissions and fostering sustainable development. Natural gas (NG) is an important environmentally friendly energy source with a high caloric value and low pollution, which is widely used in industrial systems. By 2022, the total consumption of the global LNG market has reached 394 billion m³, up 5.9 % from 372 billion cubic meters in 2021 [1]. The International Energy Agency (IEA) expects a new global LNG demand of 26 billion m³ in 2023 [2], and the LNG supply capacity will increase by 23 billion m³ per year [3]. To facilitate LNG storage and transportation, NG needs to be converted into LNG and regasified for other purposes. Based on the energy released by the regasification process of LNG at temperatures ranging from -162 to 5°C, which is about 830-860 kJ/kg [4]. In the current scenario, the insufficient utilisation of LNG cold energy is often overlooked. But it's a valuable resource with vast potential for various applications, ranging from industrial processes to environmental cooling systems. However, due to limited awareness, infrastructure constraints, and technological barriers, a substantial portion of LNG cold energy remains untapped. Yadav et al. [5] proposed a cogeneration system in response to this problem. This system is based on the Organic Rankine Cycle that utilises the cold energy released from the regasification of 5 million tons of LNG per year, generating electricity at 5 °C, capable of producing 15.8 MW of power, with a maximum energy efficiency of 58 %. To further underscore the significance of utilizing

cold energy the utilisation, the value of cold energy is 250 kWh/t [6], and the annual global LNG consumption is 39.64 billion tons [7]. If all LNG can be fully utilised within one year, it is possible to recover a total of 9.91 trillion kWh of cold energy.

Efficient energy utilisation, emissions reduction, and enhanced economic benefits for the industrial system can be achieved by effectively recycling LNG cold energy. The main methods of utilising cold energy in LNG include cryogenic power generation cycles [8], air separation [9], seawater desalination [10], cryogenic carbon dioxide capture [11] based on CCUS (carbon capture, utilisation, and storage) [12], cold energy storage [13], data center cooling [14], clathrate hydrate-based desalination [15], and cold chain for food transportation [16]. Among them, low-temperature power generation is one of the most effective methods for the sustainable utilisation of cold energy [17]. At present, methods for LNG cold energy power generation primarily include the direct expansion cycle method (DC) [18], the organic Rankine cycle method (ORC) [19], and combined cycle methods (CC) [20].

The DC method involves pressurising LNG with seawater to facilitate its regasification, subsequently driving an expansion turbine for power generation [21], as shown in **Fig. 1**. This method boasts a simple process and low initial investment, but it suffers from a low cold energy utilisation rate and limited power generation capacity. The ORC method involves the conversion of LNG cold energy through a condenser with a specific refrigerant, utilising the temperature difference between LNG and the environment to drive the refrigerant in a steam

power cycle for external power generation [21], as presented in **Fig. 2**. This method has a simple process and low investment. However, it has a low cold energy utilisation rate and limited power generation capacity. The CC method combines the advantages of the aforementioned two methods, offering a broad range of applications and a high cold energy utilisation rate, as illustrated in **Fig. 3**. However, it involves a more intricate process and requires a higher investment.

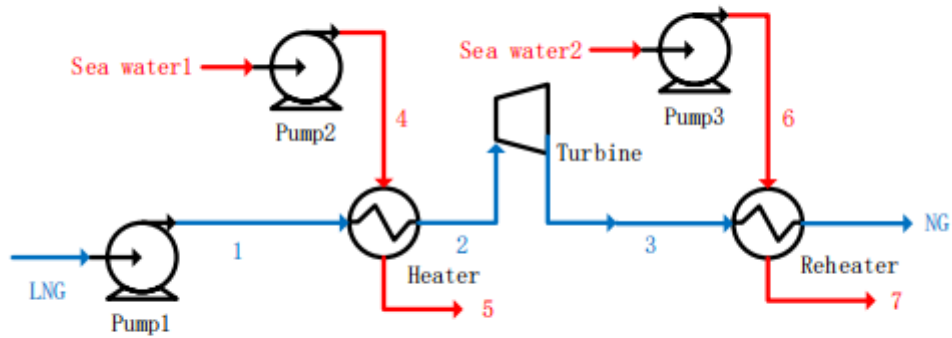


Figure 1. A schematic of the DC method (adapted from Bao et al. [22])

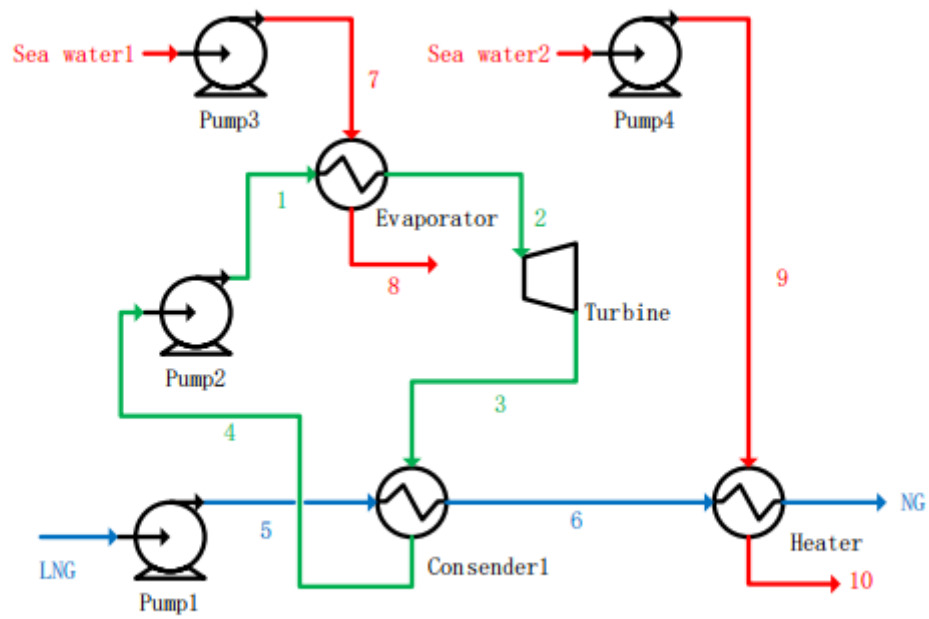


Figure 2. A schematic of the ORC method (adapted from Bao et al. [22])

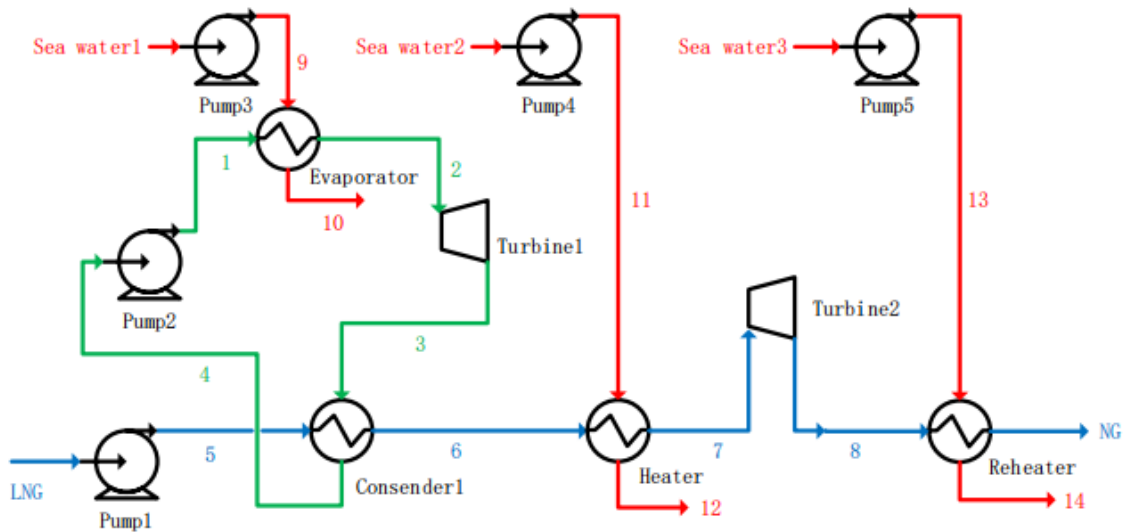


Figure 3. A schematic of the CC method (adapted from Bao et al. [22])

99 Regarding the design and optimisation of power generation systems in the LNG cold energy
100 power generation sector, it is noteworthy that a substantial portion of the existing academic
101 literature predominantly centers on LNG cold energy power generation with a basis in the
102 Rankine Cycle utilising an intermediate refrigerant. These studies primarily emphasise the
103 working medium used in circulation and the optimisation of circulation system configurations.
104 In a single Rankine Cycle power generation system, the condensation properties of the
105 circulating working medium have an impact on the heat transfer compatibility between the
106 circulating working medium and LNG in the condenser, subsequently influencing the
107 efficiency of cold energy generation for LNG. To enhance the power generation efficiency of
108 LNG in cold energy applications using a pure working medium, a substantial temperature
109 difference in heat transfer between LNG and the pure working medium is necessary. Between
110 early 1979 and 1982, the Osaka Natural Gas Company in Japan utilised propane as the working
111 medium in the organic Rankine cycle [23]. They utilised the cold energy from LNG for cycle
112 power generation, aiming to maximise the utilisation of cold energy within LNG. This was
113 achieved by modifying the working fluid's condensation temperature to attain optimal system
114 performance. Daniarta and Imre [24] introduced that the low-temperature Organic Rankine
115 Cycle is an effective solution for utilizing cold energy in power generation. And propane as a
116 working fluid in a single cryogenic ORC generates the highest net output power and efficiency.
117 He et al. [13] proposed a novel LNG cold energy cascade utilisation system named CES-ORC-
118 DC-LNG, integrating Cryogenic Energy Storage (CES), Organic Rankine Cycle (ORC), and

Direct Cooling (DC) to utilise LNG cold energy across low, middle, and high-temperature ranges, respectively. LNG is first pumped to 30 MPa, and the final reduced pressure of LNG to 7 MPa for power generation before heating for natural gas distribution. The LNG flow rate specified in the paper for the system configuration is 3600 kg/h. Ethylene serves as the primary component of the mixed working fluids in the system. The net power output of the system is 14.46 kW.

From the perspective of research on optimisation in circulation system configurations, the dual Rankine cycle system builds upon the single Rankine cycle by combining two separate Rankine cycles operating under different pressures and temperatures. It utilises the temperature gradient to transfer thermal energy at temperatures closer to the heat source, thereby reducing thermal energy loss. The waste heat produced by the first cycle can be utilised by the second cycle, enabling more efficient energy recovery and further improving energy utilisation efficiency. Liu et al. [25] found that at a heat source temperature of 130 °C, the exergy efficiency of the two-stage Organic Rankine Cycle system is increased by 9.4 % compared to a single-stage Organic Rankine Cycle system. The dual Rankine cycle system can utilise thermal energy more effectively, reduce thermal energy loss, optimise the energy conversion process, and improve energy utilisation by recovering waste heat. Moreover, the thermal efficiency of the dual Rankine cycle system is 10% to 15% higher than that of the traditional single-cycle system [26].

To improve the cycle efficiency, the idea of series two-stage Rankine cycle power generation system is proposed. Bao et al. [22] introduced a two-stage condensing Rankine cycle system, utilising propane as the working fluid. This system demonstrated significant enhancements, increasing the net power output, thermal efficiency, and exergy efficiency by 45.27 %, 42.91 %, and 52.31 %, respectively, when compared to the traditional combined cycle for LNG cold energy generation methods. In addition to the Rankine cycle study using the same cyclic working medium, the study of the double Rankine cycle system with different cyclic working mediums was also conducted. Yuan et al. [27] aimed to enhance the performance of LNG cooling power generation systems and developed a two-stage condensing Rankine cycle system. They optimised the composition of the mixed medium to determine the highest achievable net power output while simultaneously minimising the total investment cost for the ideal mixed medium. After considering both economic considerations and power efficiency, they recommended employing the first level in ethylene and the second level in ethane as the primary components for the two-stage condensing Rankine cycle system. Bao et al. [28] connected the two-stage Rankine cycle separately and optimised the key parameters and working mediums of eight different schemes. The results indicated that the circulatory system with the highest net power output utilised CH_2F_2 (Difluoromethane) as the working medium in the top cycle and C_2H_4 (ethylene) as the working medium in the bottom cycle.

An effective insight-based process optimisation method is proposed to enhance the increase the energy recovery efficiency of the dual-Rankine cycle LNG cold energy power generation

system. LNG cold energy typically refers to a low-temperature energy source generated during the regasification of LNG into natural gas. Process Integration, aimed at optimising energy and resource utilisation in industrial production processes, can combine energy and heat from various production processes to minimise energy and resource wastage. By incorporating LNG cold energy into other industrial processes, it can be efficiently harnessed to fulfil cooling or freezing needs in those processes, thus reducing additional energy consumption. This energy integration approach can enhance energy efficiency in industrial operations, cut down on energy expenses, lower carbon emissions, and reduce other environmental impacts, thereby promoting the sustainability of production processes.

The aim of the Process Integration method is to effectively utilise energy, enhance energy utilisation efficiency and recovery rate. Heat Integration is widely used in processing industries (such as chemical, petrochemical, etc.) and power generation industries, and it also demonstrates considerable potential to reduce the overall energy demand and emissions across the site [29]. Pinch Analysis originated from heat recovery and has been applied to a wide range of Process Integration problems. One of the primary objectives of the Pinch Technology approach is to prioritise internal heat exchange between processes or fluid flows, thereby reducing the need for external utilities while staying within the target investment cost. It was originally developed as a method to optimise the energy efficiency of plant processes, primarily to determine the heat recovery target for the heat exchanger network (HEN) within the plant process [30]. Over the past 45 years, Pinch Analysis has evolved and matured, with its main

contribution being the extension new areas [31]. to various energy-intensive processes. This has helped the chemical engineering industry in reducing both capital and operating costs. Wang et al. [32] introduced the use of Pinch Analysis methods to ascertain fundamental process heat requirements, identify energy recovery opportunities, and promptly determine the specific energy targets for cooling and heating facilities. Also, in 2018, a significant review of the Pinch Methodology was published, emphasising the importance of key steps, including positioning, decomposition, synthesis, and design [31].

In addition, Total Site Heat Integration (TSHI) is proposed based on Pinch Analysis to recover energy across multiple plant processes through a unified utility system [33]. Researchers have utilised TSHI to enhance thermal energy optimisation for individual processes and have progressively applied it to optimise energy across entire functional fields. Varbanov et al. [34] proposed an extended Total Site methodology that includes industrial, commercial, residential, and renewable energy integration. TSHI can also be effectively employed to investigate energy conservation issues within eco-industrial parks, providing opportunities for energy recycling [35]. Butturi et al. [36] proposed the potential for energy recovery within eco-industrial parks. Klemeš et al. [37] proposed the importance of enhancing heat transfer to optimise Heat Exchange Networks. Varbanov et al. [38] proposed an improved Total Site heat recovery target and identified the minimum temperature difference ($\Delta T_{\min}$). Boldyryev et al. [39] proposed a design framework for Total Site energy recovery and optimised both the heat transfer area and capital cost of the HEN. Wang et al. [40] proposed a method to determine the heat exchanger

and thus optimise the HEN. Wang et al. [41] proposed a HEN modification framework to analyse the original HEN and select the appropriate heat exchanger and material type, so as to determine the feasible modification scheme. Via Total Site Heat Integration, a design framework is developed to reduce energy consumption and enhance energy efficiency in the region [42].

Most scholars have focused only on the cycle configuration optimisation design and the selection of optimal working fluids in their studies on the Rankine cycle power generation system, which create research gaps. Thus, on the basis of prior studies, energy demand planning can be balanced and adjusted. By integrating the TS method, predicated on the amalgamation of LNG cold streams and hot streams, and utilising advanced heat exchange network optimisation techniques, it is feasible to substantially elevate the efficiency of power generation. By analysing and optimising energy demand, using the TSHI method to optimise the LNG cold energy dual Rankine cycle power generation system can significantly reduce energy consumption, improve the efficiency of energy recovery and use, promote the planning and management of energy systems, help to balance and adjust the demand for different energies (cold streams, hot streams), and can also drive the development of new technologies and processes, such as efficient heat exchanger design, heat recovery technologies, etc.

According to the Process Integration method of the above review, the important elements in advantages which are particularly suitable for our problem have been summarised as follows:

(1) By selecting the appropriate minimum heat transfer temperature difference, which offers substantial energy recovery at a minimal investment cost, one can maximise energy utilisation, enhance production efficiency, reduce energy costs, and reduce utility usage; (2) During the optimisation and transformation of a Heat Exchanger Network, by selecting a suitable heat exchanger and confirming its parameters, the heat transfer area is then optimised to achieve the optimal heat transfer efficiency at a certain investment cost; (3) Optimising resource utilisation does not only focus on a single process but also comprehensively considers the energy demand and supply of the whole system to ensure the optimal distribution and utilisation of different streams; (4) By minimising energy waste, providing effective solutions to energy integration problems, supporting sustainable production, and aiding in addressing pressing environmental concerns, it also can offer economic and environmental benefits throughout the entire production process lifecycle.

As discussed above, this paper introduces the feasibility of recovering and utilising LNG cold energy, identifying the generation of electricity using LNG cold energy as one of the most effective methods. When designing LNG cold energy power generation systems, many researchers predominantly employ Rankine cycle based on intermediate refrigerant. The process design and optimisation of LNG cold energy power generation systems are primarily studied from two perspectives: optimising the cycle working medium and optimising of cycle configuration. Initially, researchers searched for the optimal cycle working medium when establishing the power generation system and identified propane or ethylene as the primary

cycle working mediums. To improve cycle efficiency, some have subsequently designed a series of two-stage Rankine cycle systems to further use cold energy. In these systems, whether utilising pure or mixed working mediums, propane or ethylene remains the preferred choice as the cycle medium for the majority.

While the optimal design of cycle fluid and configuration can enhance power generation efficiency, there are inherent limitations in the composition of the working fluid and the established cycle structure. Therefore, this paper proposes the Pinch Analysis within the Process Integration approach to retrofit existing HEN for cold energy power generation. By introducing the Pinch Analysis, this paper aims to overcome these limitations and offer a novel approach to enhance the power generation efficiency of LNG cold energy power generation systems. The energy integration of LNG cold energy and the Rankine cycle system for power generation not only effectively utilises cold energy resources but also fulfills the cooling demand of the circulating working medium during the Rankine cycle, thereby reducing additional energy consumption. By analysing the energy recovery of the original system and the required utility amounts, a Heat Exchange Network is established, and the heat recovery target is determined. According to the optimisation method for HENs studied by predecessors, it is concluded that this paper should retrofit the best HEN scheme based on the determined minimum heat transfer temperature difference, suitable heat exchangers, and comprehensive consideration of investment costs. The goal is to optimise power generation efficiency at the lowest investment cost by identifying energy recovery opportunities.

266

267 This study aims to: (1) Utilise Pinch Analysis to identify energy recovery opportunities,
268 determine heat exchange requirements, and assess utility consumption; (2) Establish and
269 optimise the Heat Exchange Network, with heat exchange area and investment cost of the heat
270 exchanger as variables, analyse variations among different schemes, and determine the optimal
271 solution. The primary contribution of this study lies in its adoption of a Process Integration to
272 propose a parallel dual-fluid Rankine cycle power generation system. This system approach to
273 take full advantage of LNG cold energy and the circulating working medium for heat transfer.
274 Additionally, it optimises the HEN retrofit to enhance power generation efficiency. The
275 findings from this study offer valuable insights for improving the power generation efficiency
276 of LNG cold energy power generation systems.

277

278 The remainder of this work is structured as follows. Section 2 outlines our method for analysing
279 how to improve the power generation efficiency of the system and introduces a dual-fluid
280 condensation Rankine cycle system. Section 3 discusses the energy integration optimisation of
281 the system, presents the performance indicators for optimisation, and compares the retrofit
282 scheme of the HEN through Pinch Analysis. And optimal Heat Exchange Network schemes for
283 enhancing the power generation efficiency of the system are deliberated. Finally, conclusions
284 drawn from this study are presented in Section 4.

2 Methodology

In this section, a dual-fluid condensation Rankine cycle system is developed. The efficiency of the LNG cold energy utilisation power generation system is typically low. To ensure the efficient use of LNG cold energy resources for power generation, HEN was established and modified by Pinch Analysis. The following study is conducted, and the specific research framework is illustrated in **Fig. 4**.

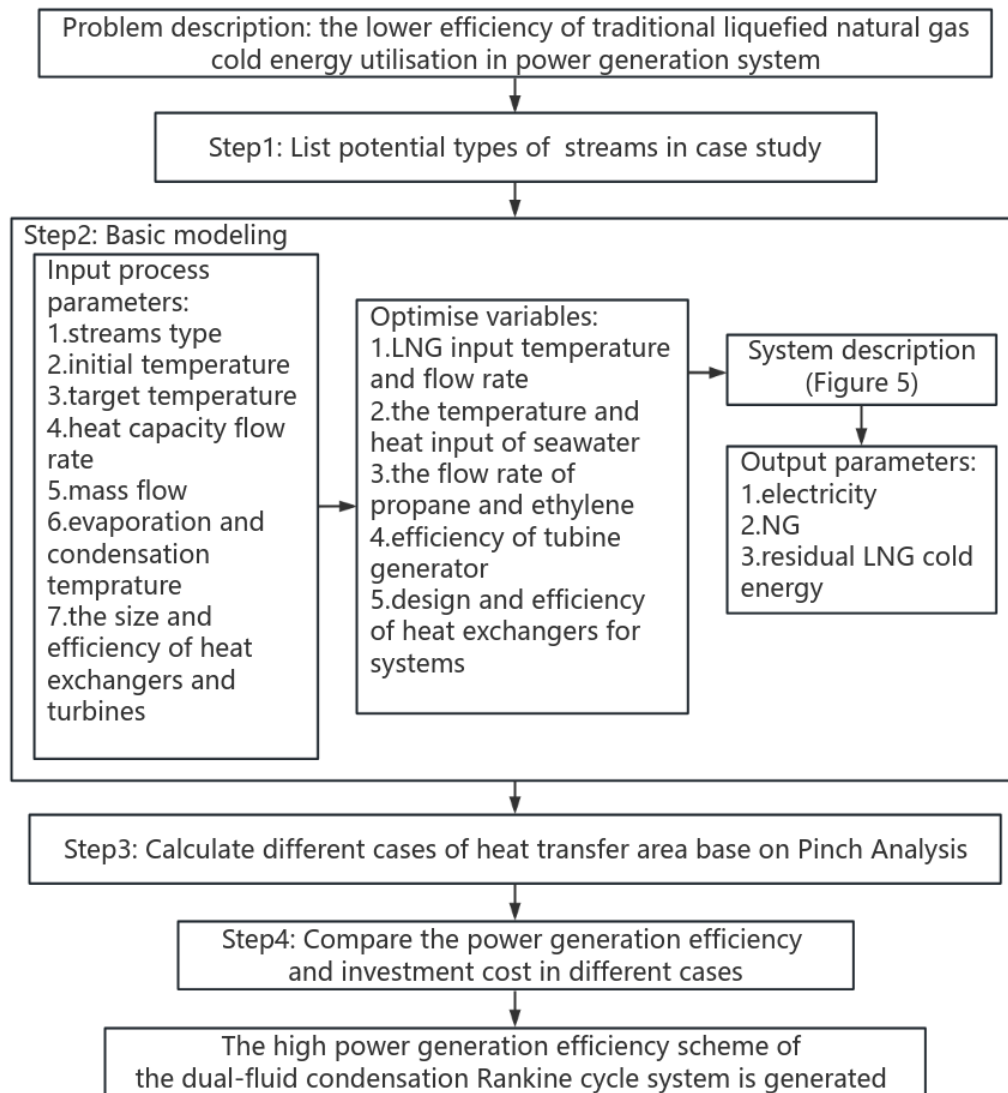


Figure 4. The framework of the proposed method.

The methods for improving power generation efficiency can be categorised into five parts. In the first part, the system structure is described by establishing the power generation system with potential streams data. In the second part, the Grand Composite Curve (GCC) is drawn, and the energy target is determined using the Pinch Analysis. The third part focuses on optimising the Heat Exchange Network structure to achieve the energy target. Subsequently, the fourth part conducts energy analysis, and the fifth part considers investment costs. Through these steps, the proposed system's maximum power generation efficiency scheme is determined.

2.1 System structure description

In this sub-section, a dual-fluid condensation Rankine cycle system through the utilisation in LNG cold energy is developed to enhance the efficiency of power generation. Therefore, the proposal includes the generation of potential streams and the establishment of the system structure. Further details regarding the selection of the Rankine cycle as the cycle structure for cold energy generation can be found in Ref. [43].

2.1.1 List potential types of streams

Firstly, material and energy balances are established. The material balance relies on the mass flow rate of streams, while the energy balance is calculated based on temperature, heat load, and material balance. In circulating systems, material loss due to leakage is typically minimal,

but heat loss is permissible. To ensure the reliability of balance data, it is essential to consider appropriate streams temperature and heat load data. The initial and target temperatures of the hot and cold streams are known parameters determined according to the substance's thermodynamics and process design requirements. Required streams data include the temperature range, type (cold or hot), and thermodynamic information, including heat capacity flow rate (CP) in kW/K. Subsequently, streams data is extracted, and the heat load (ΔH) is obtained by multiplying the mass flow rate, specific heat capacity value, and temperature change value of the streams import and export.

2.1.2 System structure generation

To simplify the analysis and based on the First Law of Thermodynamics, the following assumptions are proposed in this study: (1) Except for the pump, the pressure drop in all streams is negligible; (2) The heat and friction losses of all streams are ignored; (3) All streams circulate in a steady state; (4) LNG remains in its liquid state throughout the cycle, with no consideration for evaporative losses; (5) The cold energy of LNG is sufficient to provide the heat required for the evaporation of the circulating medium; (6) The circulating medium can circulate stably within the system and can operate at different temperatures; (7) The working fluid was evaporated as saturated steam in the evaporator and condensed as a saturated liquid in the condenser; (8) Equipment and pipe materials can withstand the low-temperature environment.

The study adopts the structure of a parallel double-fluid Rankine cycle power generation

system, as retrofitted in Ref. [28]. This choice is motivated by the recognition that a two-stage Rankine cycle is more efficient compared to a single Rankine cycle power generation. The series system circulation is interconnected through an intermediate heat exchanger, and the condenser circulating at the bottom of the series system facilitates the recovery of cold energy from LNG. However, in the parallel system, two condensers on the left and right sides enable the recovery of cold energy from LNG at different temperatures. Therefore, a more flexibility parallel system can be chosen, allowing for the use of different circulating working fluids in the left and right cycles.

The schematic diagram of the proposed dual-fluid condensation Rankine cycle power generation system is shown in **Fig. 5**. This system mainly consists of a buffer tank, a holding tank, two expansion generator sets, seven heat exchangers and six booster pumps. Firstly, the LNG flow as a cold stream inflow system, which is primarily used to cool the circulating working media (ethylene and propane) and recover energy. The seawater flow as a hot stream to heat the circulating medium ensures the continuous operation of the system. Next, the dual-fluid condensation Rankine cycle system is proposed to design two Rankine cycles, one for the ethylene cycle and the other for the propane cycle, aiming to recycle LNG cold energy and generate electricity. In the ethylene cycle, low-pressure ethylene is used for heat exchange with LNG in the LNG evaporator. The liquefied ethylene is then transferred to the ethylene buffer tank (V-0501). The booster pump is pressurised into the LNG heating exchanger and heat exchange with LNG. The incoming propane/ethylene heat exchanger (E-0508) is vaporised by

ethylene. After that, the ethylene is routed to the seawater/ethylene heat exchanger (E-0505) and then supplied to the ethylene expansion generator set (K-0501) for electricity generation. The expanded low-pressure ethylene is subsequently directed to the LNG evaporator to complete the cycle. In the propane cycle, low-pressure propane is utilised for heat exchange with both natural gas and propane in the LNG heating heat exchanger and the propane/ethylene heat exchanger, respectively. Following that, the liquefied propane is pressurised by the booster pump and then pumped into the seawater/propane heat exchanger (E-0504), where it vaporises after being reheated by seawater. Subsequently, the gaseous propane is directed to the propane expansion generator set (K-0502) to generate electricity, and the expanded low-pressure propane is transmitted to the LNG heating heat exchanger and the propane/ethylene heat exchanger. This completes the cycle. As we can see from the above, the system can eventually generate electric energy, NG (heated LNG), and residual LNG cold energy (using hot utilities).

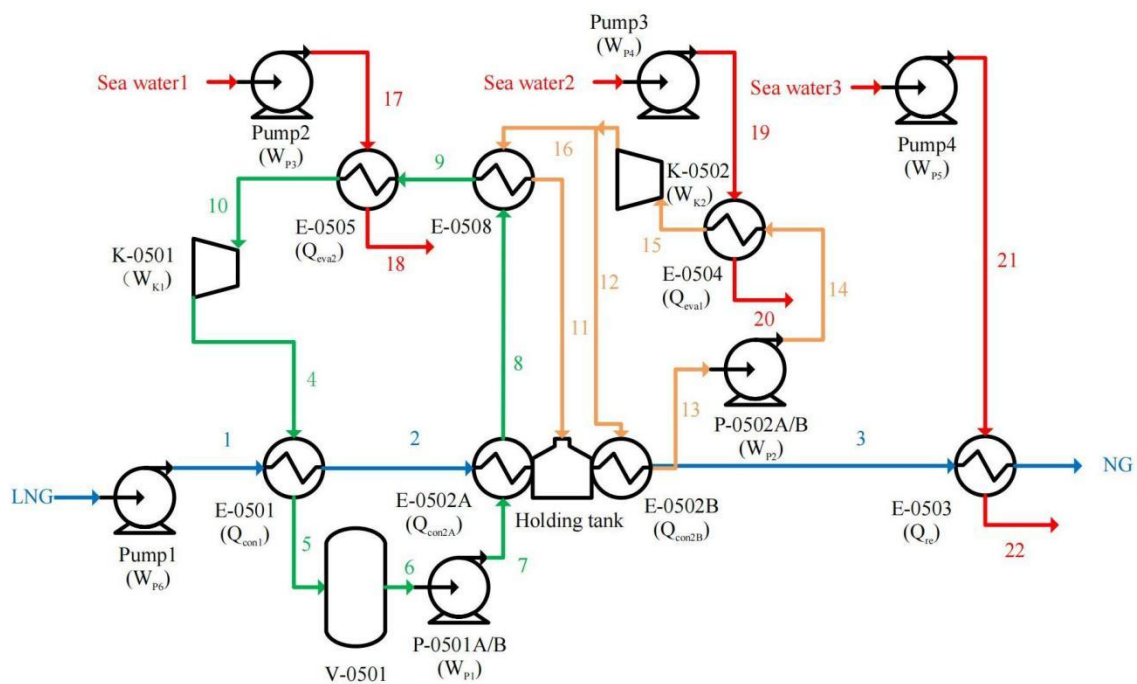


Figure 5. The dual-fluid condensation Rankine cycle power generation system schematic.

2.2 Pinch Analysis methodology

In the first part, the LNG cold energy power generation cycle system has established. In the second part, the proposed system is analysed using the Pinch Analysis method in Process Integration. Pinch Analysis plays a crucial role in identifying the optimal energy transfer path to maximise the efficiency of energy conversion [44]. By pinpointing the threshold between two energy values, it enables the assessment of potential losses in the energy conversion process. This analysis helps in devising measures to enhance efficiency. Pinch Point is the minimum vertical difference between the hot composite curve and the cold composite curve, essentially representing the minimum heat transfer temperature difference, and denoted as $\Delta T_{\min}$. In the second part, the method involves leveraging the constraints imposed by $\Delta T_{\min}$ values to create Total Composite Curve (TCC) diagram and Grand Composite Curve (GCC) diagram. These diagrams are instrumental in establishing energy targets for the system.

2.2.1 Minimum heat transfer temperature difference constraints

A correlation exists between $\Delta T_{\min}$ in the heat exchanger and the total utility load of the system [45]. A zero-temperature difference implies the need for an infinite heat exchanger area, a practically unattainable scenario. The choice of $\Delta T_{\min}$, which represents the temperature difference, plays a crucial role. If the value of $\Delta T_{\min}$ increase, more total utilities are required, and the investment cost is higher, so within a certain range, it is necessary to choose as small a value as possible to achieve the maximum energy efficiency. This methodology ensures that

the energy recovered from process heat exchanges is maximised, thereby diminishing energy consumption and the reduction in operational costs. A smaller $\Delta T_{\min}$ value leads to greater hot and cold utility requirements, but it maximises the effective use of energy. Higher energy recovery from the stream heat exchange results in lower energy consumption and, consequently, reduced operating costs. However, selecting a smaller $\Delta T_{\min}$ value also increases the size of heat exchangers, which, in turn, raises the investment cost due to the expanded heat exchange area. Thus, there exists a trade-off between operating costs and investment costs, and an optimal $\Delta T_{\min}$ value needs to be determined.

2.2.2 Draw total composite curve diagram

The Total Composite Curve comprises the Hot Composite Curve (HCC) for heat streams and the Cold Composite Curve (CCC) for cold streams. With the established $\Delta T_{\min}$ value, predictions can be made regarding energy recovery and utility consumption.

2.2.3 Draw grand composite curve diagram

The Grand Composite Curve is a graphical representation used for the analysis and design of the Heat Exchange Network. It serves as an extension of the TCC. By analysing GCC, one can identify the Pinch Point to determine the optimal placement of heat exchangers and facilitate heat exchange between different streams. Additionally, by considering the required utility energy consumption and the effective energy recycling area, strategies can be developed to reduce energy consumption, minimise energy costs, and enhance the efficiency of the overall system process.

2.3 Heat exchanger network structure optimisation

Pinch Analysis mentioned in the previous section is an important tool to design and retrofit a Heat Exchange Network [46]. This method can identify key positions within the Heat Exchange Network and place different heat exchangers at these key positions. Also, this enables the system to more effectively transfer energy from the hot stream to the cold stream by leveraging the temperature difference. This not only aids in optimising the heat exchanger layout but also effectively recovers energy within the system, thereby reducing energy consumption and investment costs. The subsequent steps involve optimising the Heat Exchange Network to meet the energy target. This optimisation is conducted considering constraints such as streams temperature, hot utility, and the selection of appropriate heat exchangers. The third part is dedicated to establishing the optimal scheme for the Heat Exchange Network. This network design method aims to optimise energy transfer, recovery, and distribution within an energy system [47]. Given that energy transfer and recovery in industrial and chemical processes frequently involve multiple streams requiring heating or cooling, it is crucial to configure and design heat exchangers effectively [48]. This optimisation process enhances energy efficiency, reduces energy consumption, and minimises costs by mitigating energy losses. The following outlines the approach to optimising the HEN, considering streams temperature constraints, hot utility constraints, and the selection of appropriate heat exchangers.

2.3.1 Stream temperature constraints

To optimise the retrofit of the Heat Exchange Network, extracting streams data from the system's process flow is essential. The initial and target temperatures of both hot and cold

streams are predetermined parameters based on the material's thermodynamics and process design requirements. During optimisation, adjustments are made to the temperature of the streams segment. Simultaneously, it is crucial to consider that the process streams should not be excessively segmented. Excessive segmentation can amplify network complexity and constraints, potentially hindering heat recovery opportunities. Striking a balance is necessary to ensure an efficient and practical HEN. Therefore, the selection of logistics temperatures should take into account the following constraints: (1) Check and rectify data near the Pinch Point, ensuring that heat cannot be transferred through the Pinch Point; (2) Ensure that the inlet temperature of the heat exchanger for both hot and cold streams is equal to the initial temperature of the streams; (3) Ensure that the outlet temperature of the heat exchanger for hot streams is equal to the inlet temperature plus the reduced temperature, while the outlet temperature of the cold streams used for heat exchange is equal to the inlet temperature plus the rising temperature. Both depend on the heat load for temperature conversion.

2.3.2 Hot utilities constraints

The proposed system utilises LNG cold energy, and LNG terminals are typically situated along the coast [49]. Given the geographical location and a focus on minimising investment costs, seawater is employed as the hot utility for heat transfer. To safeguard against the pollution of seawater by the cold pollution, the allowable temperature difference for heat transfer in seawater is restricted to within 5 °C [50].

2.3.3 Heat exchanger selection

Heat exchangers have a variety of specifications, including specific operating temperatures and heat exchange areas [51]. To select a suitable heat exchanger specification for this study case, it is necessary to compile a list of all available heat exchanger specifications that can be considered.

2.4 Energy analysis

To retrofit HEN schemes with high power generation efficiency, energy analysis is essential. Thermal efficiency typically refers to the efficient conversion of input energy into useful work or electricity during the energy conversion process [52]. In this paper, thermal efficiency is defined as the ratio of energy effectively converted into electrical energy from the total heat absorbed in the proposed system. This includes heat derived from the cryogenic cold energy of a portion of the LNG and the cooling capacity of the circulating working medium. Here, η_{th} represents the thermal efficiency of the system; W_{net} denotes the electricity generated by the system; and Q_{tot} indicates the amount of absorbed heat recovered by the system. By utilising the thermal efficiency formula in the energy analysis process, we can effectively illustrate the energy conversion efficiency in the LNG cold energy power generation system. This formula reveals how much of the total heat absorption is ultimately converted into useful electric energy. The specific formula is as follows.

According to the mass and energy conservation principles, the equations for each equipment of the dual-fluid condensation Rankine cycle power generation system are presented in **Table**

1. The net power output of the proposed system is given in **Eq. (1)**. The total heat absorption of the proposed system is presented in **Eq. (2)** and the thermal efficiency of the proposed system is given in **Eq. (3)**. **Eqs. (1) to (3)** are commonly used to calculate the energy analysis [22].

$$W_{\text{net}} = W_{K1} + W_{K2} - W_{P1} - W_{P2} - W_{P3} - W_{P4} - W_{P5} - W_{P6} \quad (1)$$

$$Q_{\text{tot}} = Q_{\text{re}} + Q_{\text{eva1}} + Q_{\text{eva2}} \quad (2)$$

$$\eta_{\text{th}} = W_{\text{net}} / Q_{\text{tot}} \quad (3)$$

Table. 1. The energy analysis equation of each equipment of the dual-fluid condensation Rankine cycle system.

Symbol	Equipment	Equation
E-0505	Evaporator	$Q_{\text{eva2}} = m_{\text{ETH}}(h_{10} - h_9) = m_{\text{SW}}(h_{18} - h_{17})$
E-0504	Evaporator	$Q_{\text{eva1}} = m_{\text{P}}(h_{15} - h_{14}) = m_{\text{SW}}(h_{20} - h_{19})$
K-0501	Expansion generator	$W_{K1} = m_{\text{ETH}}(h_4 - h_{10})$
K-0502	Expansion generator	$W_{K2} = m_{\text{P}}(h_{16} - h_{15})$
E-0501	Condenser	$Q_{\text{con1}} = m_{\text{LNG}}(h_2 - h_1) = m_{\text{ETH}}(h_5 - h_4)$
E-0502A	Condenser	$Q_{\text{con2A}} = m_{\text{LNG}}(h_3 - h_2) = m_{\text{ETH}}(h_8 - h_7)$
E-0502B	Condenser	$Q_{\text{con2B}} = m_{\text{LNG}}(h_3 - h_2) = m_{\text{P}}(h_{13} - h_{12})$
P-0501	Feed pump	$W_{P1} = m_{\text{ETH}}(h_7 - h_6)$
P-0502	Feed pump	$W_{P2} = m_{\text{P}}(h_{14} - h_{13})$
Pump2	Sea water pump	$W_{P3} = m_{\text{SW1}}(h_{17} - h_{\text{SW1_in}})$

Pump3	Sea water pump	$W_{P4}=m_{SW2}(h_{19}-h_{SW2_in})$
Pump4	Sea water pump	$W_{P5}=m_{SW3}(h_{21}-h_{SW3_in})$
Pump1	LNG pump	$W_{P6}=m_{LNG}(h_1-h_{LNG_in})$
E-0503	Reheater	$Q_{re}=m_{LNG}(h_{LNG_out}-h_3)=m_{SW3}(h_{22}-h_{21})$

Notes: m is the mass flow rate, kg/s; h is the mass enthalpy, kJ/kg.

2.5 Investment cost

Investment cost is another critical factor used to evaluate the feasibility of the HEN schemes with high power generation efficiency. Eqs. (4) to (6) are commonly used to calculate the investment cost [53].

$$\Delta T_{LMTD} = \frac{(T_{H_in}-T_{C_out})-(T_{H_out}-T_{C_in})}{\ln \frac{(T_{H_in}-T_{C_out})}{(T_{H_out}-T_{C_in})}} \quad (4)$$

$$A^{HEN} = \frac{Q}{U \Delta T_{LMTD}} \quad (5)$$

$$C_{INV} = \alpha_1 + \alpha_2 \times A^\beta \quad (6)$$

In these formulae, ΔT_{LMTD} is the log average temperature difference, K; T_{H_in} and T_{H_out} are the inlet temperature and the outlet temperature of the heat stream, respectively; T_{C_in} and T_{C_out} are the inlet temperature and the outlet temperature of the cold stream, respectively; A^{HEN} is the area of the Heat Exchange Network, m^2 ; Q is the heat delivered by the heat exchanger, kW; U is the total heat transfer coefficient, $kW/(m^2 \cdot K)$; C_{INV} is the investment cost, CNY; α_1 、 α_2 、 β is the investment cost parameters.

2.6 The high power generation efficiency scheme generate

Once a high power generation efficiency scheme is identified, the subsequent steps to obtain the optimal Heat Exchanger Network retrofit scheme are as follows: (1) Calculate the energy analysis and investment costs for various HEN retrofit schemes; (2) Evaluate the schemes based on key influencing factors such as the utilisation rate of LNG cold energy, power generation efficiency, and investment costs. Compare the schemes to determine the optimal retrofit solution.

3 Case study

In this paper, to ensure a more efficient recovery of LNG cold energy and to complete the power generation cycle through the expansion generator, a dual-fluid condensation Rankine cycle power generation system is proposed. This system is based on the enhancement of the cold energy utilisation rate of LNG by employing two different working fluids (ethylene and propane) with LNG in heat exchange. Subsequently, the LNG cold energy utilisation power generation system is optimised, and the optimal $\Delta T_{\min}$ is determined using the Pinch Analysis to maximise heat transfer between streams. A 25 °C seawater is used as a hot utility within the system. To protect the marine environment, the seawater can only be cooled in 5 °C. Finally, based on different heat exchanger specifications, the HEN is optimised. Depending on the selected $\Delta T_{\min}$, it maximises the utilisation of LNG cold energy, reduces the amount of seawater required, and achieves the most cost-effective heat transfer. Consequently, this approach offers a lower-cost and more energy-efficient means to improve power generation efficiency.

3.1 Basic parameter

The composition of the LNG is provided in **Table 2**. And the **Fig. 6** shows the specific heat capacity of LNG in the temperature range of -160 °C to -20 °C. The LNG has an initial temperature of -141.63 °C and an initial pressure of 7.85 MPa. In the dual-fluid condensation Rankine cycle power generation system, 15.6 kg/s of LNG is used as the cold stream in the heat exchanger section, and it is heated and evaporated for external transport, resulting in natural gas (NG) at 5 °C.

Table 2. Mole fractions of the LNG.

Component	Mole fraction(mol%)	Component	Mole fraction(mol%)
Methane	87.5	Ethane	5.5
Propane	2.1	i-sobutane	0.3
n-butane	0.5	i-pentane	0.56
n-pentane	0.44	Nitrogen	3.1

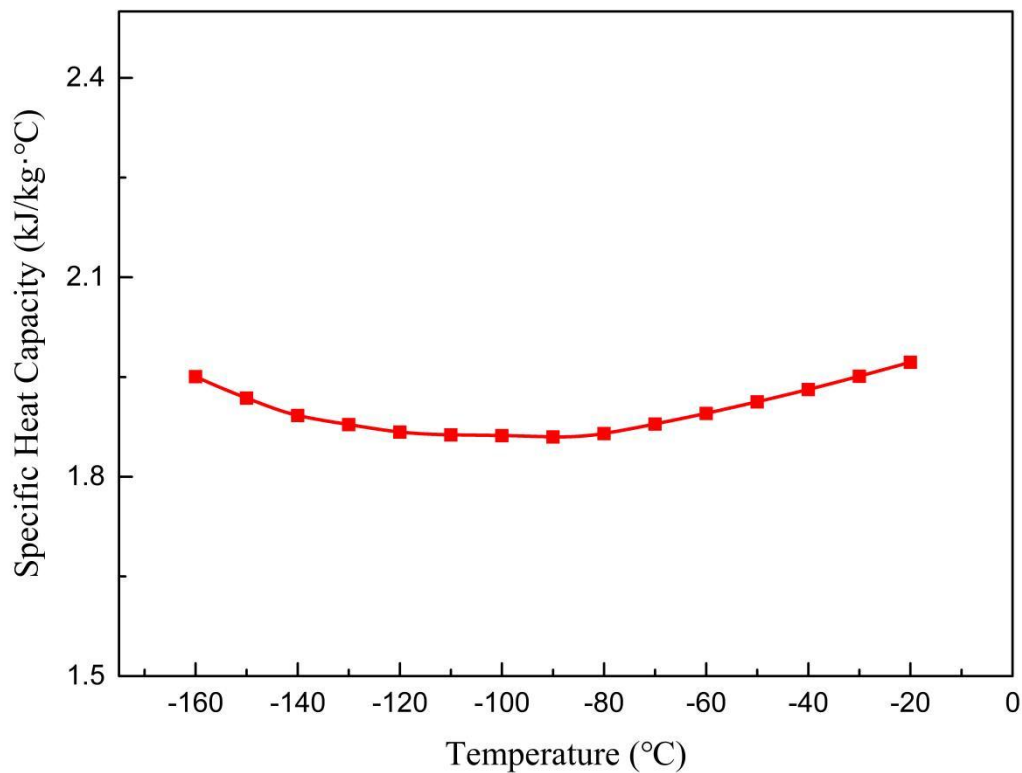


Figure 6. the Specific Heat Capacity of LNG in this case.

To improve power generation efficiency, specific data support is required. Firstly, the input temperature and flow rate of LNG are determined. We also need to find the thermodynamic parameters of propane and ethylene. This information is used to establish the evaporation and condensation temperatures and to determine the specifications and efficiency of heat exchangers and expansion generators. Simultaneously, to avoid the failure of the dual-fluid condensation Rankine cycle power generation system, we stipulate that the inlet temperature of the cold stream is lower than the outlet temperature of the heat stream, and the outlet temperature of the cold stream is lower than the inlet temperature of the heat stream. The existing initial streams already in place involve propane, ethylene, and LNG. The specific stream parameters are presented in **Table 3**.

Table 3. Potential streams for the dual-fluid condensation Rankine cycle system.

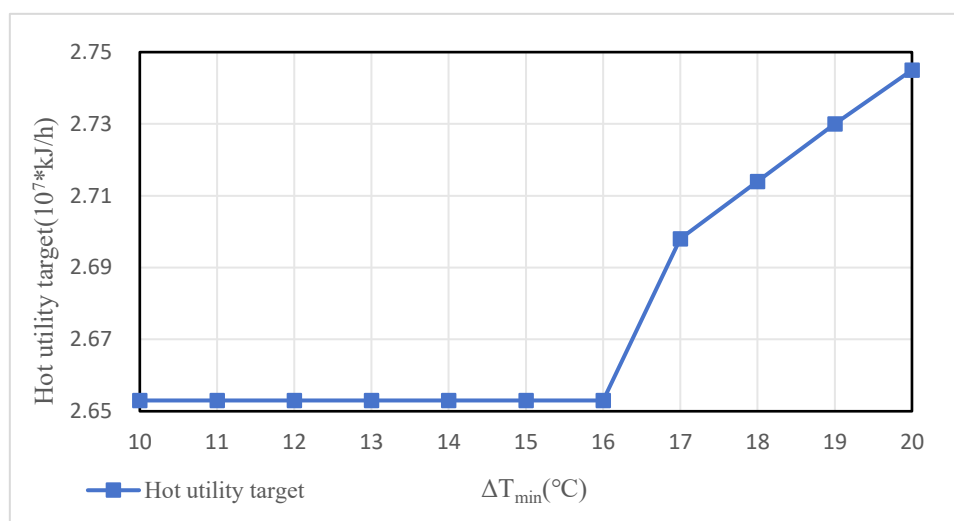
Stream	T _{in} (°C)	T _{out} (°C)	CP(kW/°C)	ΔT(°C)	Pressure(MPaG)
GER	-53.86	12.20	83.13	66.06	0.90
LER	-13.00	-21.61	138.98	8.61	0.79
GPR	-16.50	-19.76	223.38	3.26	5.04
LPR	-9.93	5.38	21.37	15.31	1.10
LNG	-141.63	-20.00	35.33	121.63	7.85
NG	-20.00	5.00	37.55	25.00	7.81

Notes: T_{in} is the inlet temperature of streams (°C); T_{out} is the outlet temperature of streams (°C); CP is the heat capacity flow rate (kW/°C); ΔT is the temperature difference between the inlet temperature and the outlet temperature.

3.2 Pinch Analysis

In the developed integrated structure, the primary objective of the Pinch Analysis is to maximise energy recovery within the process heating system and minimise the need for external utilities to provide heating and cooling energy. This approach reduces energy consumption costs. To economise the system, no external cold utility is used. Consequently, this constraint was applied to determine the value of ΔT_{min}. As depicted in **Fig. 7**, when ΔT_{min} exceeds 16.1 °C, a larger ΔT_{min} results in a greater demand for hot utilities. Therefore, the aim is to select a smaller ΔT_{min} value to maximise effective energy use and minimise operating costs. When ΔT_{min} is less than 16.1 °C, even though the requirement for hot utility remains constant, smaller ΔT_{min} values result in larger heat exchangers and increased heat exchange

547 area, leading to higher investment costs. Consequently, the optimal $\Delta T_{\min}$ determined is 16.1 °C.



548
549 **Figure 7.** The diagram illustrates the relationship between Pinch Point temperature difference
550 and utility.

551
552 The existing streams include propane, ethylene, and LNG. The specific data for these streams
553 are provided in **Table 4**.

554 **Table 4.** Streams parameter list for system used in the study.

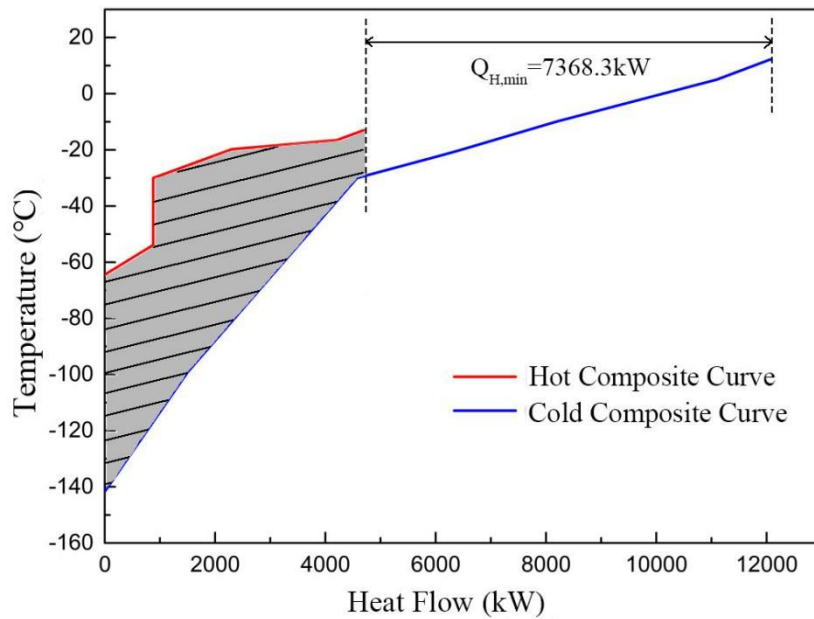
Stream	Type	$T_{\text{in}} (^\circ\text{C})$	$T_{\text{out}} (^\circ\text{C})$	CP(kW/K)	m(kg/s)	Pressure(MPaG)
1	Cold	-141.63	-98.95	35.33	15.60	7.85
2	Cold	-98.95	-20.00	44.43	15.60	7.83
3	Cold	-20.00	5.00	37.55	15.60	7.81
4	Hot	-53.86	-64.42	83.13	53.70	0.72
7	Hot	-13.00	-30.00	138.98	53.70	0.83
8	Cold	-30.00	-21.61	145.96	53.70	0.79

9	Cold	-21.61	12.20	134.84	53.70	0.90
12	Hot	-16.50	-19.76	223.38	121.40	5.04
14	Cold	-9.93	5.38	21.37	121.40	1.10
16	Hot	-16.50	-19.76	223.38	121.40	5.04
17	Hot	25.00	21.00	1065.08	253.59	0.40
19	Hot	25.00	24.70	1065.08	253.59	0.40
21	Hot	25.00	21.60	1065.08	253.59	0.40

555

556 Using the stream data provided in **Table 4**, the Hot Composite Curve and Cold Composite

557 Curve are plotted for $\Delta T_{\min}$ of 16.1 °C, as depicted in **Fig. 7**.



558

559 **Figure 8.** Total Composite Curves.

560 **Fig. 8** displays the HCC and CCC of the HEN in the dual-fluid condensation Rankine cycle

561 power generation system. In the terminology of Heat Integration, this is a typical Threshold

Problem. Heat recovery occurs in the overlapping section of the HCC and CCC, which is the area within the HEN where heat transfer is necessary. In this area, 4696.82 kW of energy is recovered. In the sections where the HCC and CCC do not overlap, there is no heat exchange between the hot and cold streams, requiring an external utility of 7368.3 kW for these parts of the HEN. The Grand Composite Curve is an extension of the Total Composite Curves. The TCC represents the energy balance between the hot and cold streams in the process, while the GCC combines multiple hot and cold streams to provide a comprehensive overview of the energy flow within the entire system.

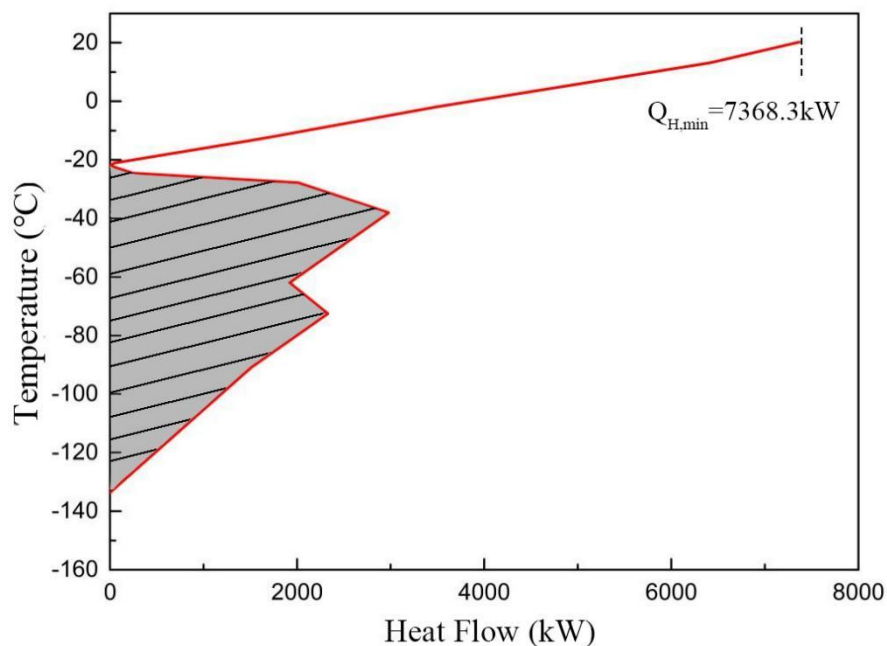


Figure 9. Grand composite curve.

To determine the exact location of the Pinch Point, the heat flow was decreased by $\Delta T_{\min}/2$, and an equivalent amount was added to the cold streams, drawing the GCC as shown in Fig. 9. The points where heat flow is 0 and intersects the longitudinal axis are Pinch Point. The Pinch Point temperature was determined to be -21.95 °C, with the upper-end temperature at -13.9 °C

and the lower end temperature at -30 °C. The shaded areas shown in the figure are all located below the Pinch Point, which is the heat exchange area of cold and hot streams. For unrecovered energy above the Pinch Point, a heating load of 7368.3 kW for hot utilities is required.

3.3 Improving the HEN

Based on the schematic of the dual-fluid condensation Rankine cycle power generation system, the Heat Exchange Network diagram has been drawn as shown in Fig. 10.

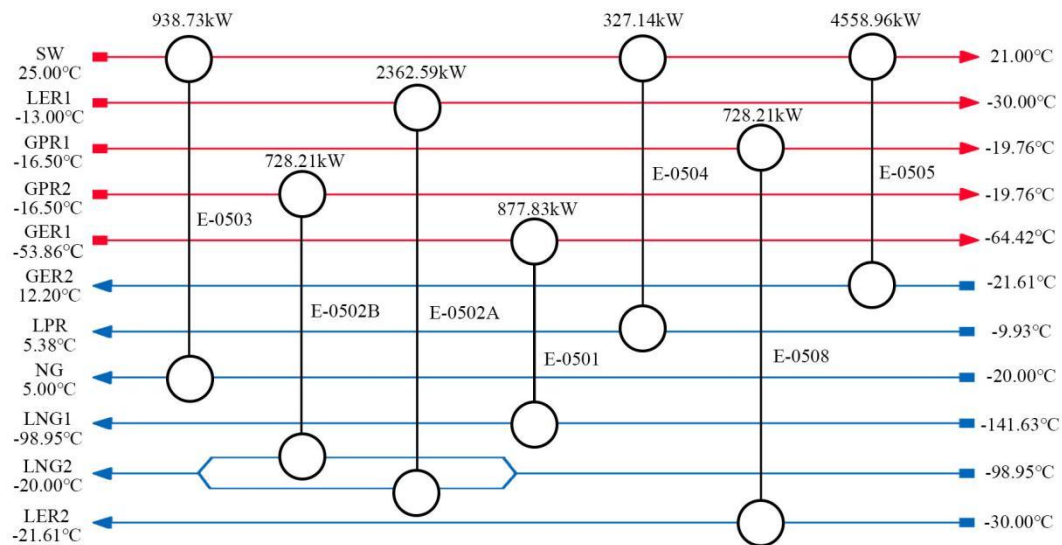


Figure 10. Grid diagram of the existing HEN in case study.

Obviously, in the Heat Exchange Network of Fig. 10, heat exchange between the streams is achieved using only four basic heat exchangers. To facilitate public engineering heat exchange, three additional external heat exchangers have been introduced. This situation necessitates further optimisation of the Heat Exchange Network. The aforementioned Pinch Analysis is

instrumental in identifying the most suitable heat sources to maximise energy efficiency and system performance. In the optimisation of the heat transfer network, various variables can be manipulated to increase degrees of freedom, including the inlet and outlet temperatures of the heat exchangers, loads, and heat transfer areas. This flexibility allows for the transformation of the HEN into different configurations. Typically, low-temperature cold streams should exchange heat with high-temperature heat sources to maximise the recovery and utilisation of heat energy in low-temperature cooling streams. This approach helps enhance the energy efficiency of the system, reduce energy wastage, lower energy costs, and minimise environmental impact.

The chosen heat exchanger is the shell-and-tube type heat exchanger and adopts a countercurrent configuration, in which the heat stream and cold stream in the shell-and-tube heat exchanger flow in opposite directions. The temperature profiles in the heat exchanger is a graphical representation that shows the temperature versus heat load change of the streams (hot or cold) as it passes through the heat exchanger. The graph will show that the temperature of the cold stream gradually increases, while the temperature of the hot stream gradually decreases. The following **Fig.11** shows the heat transfer in heat exchangers of existing HEN.

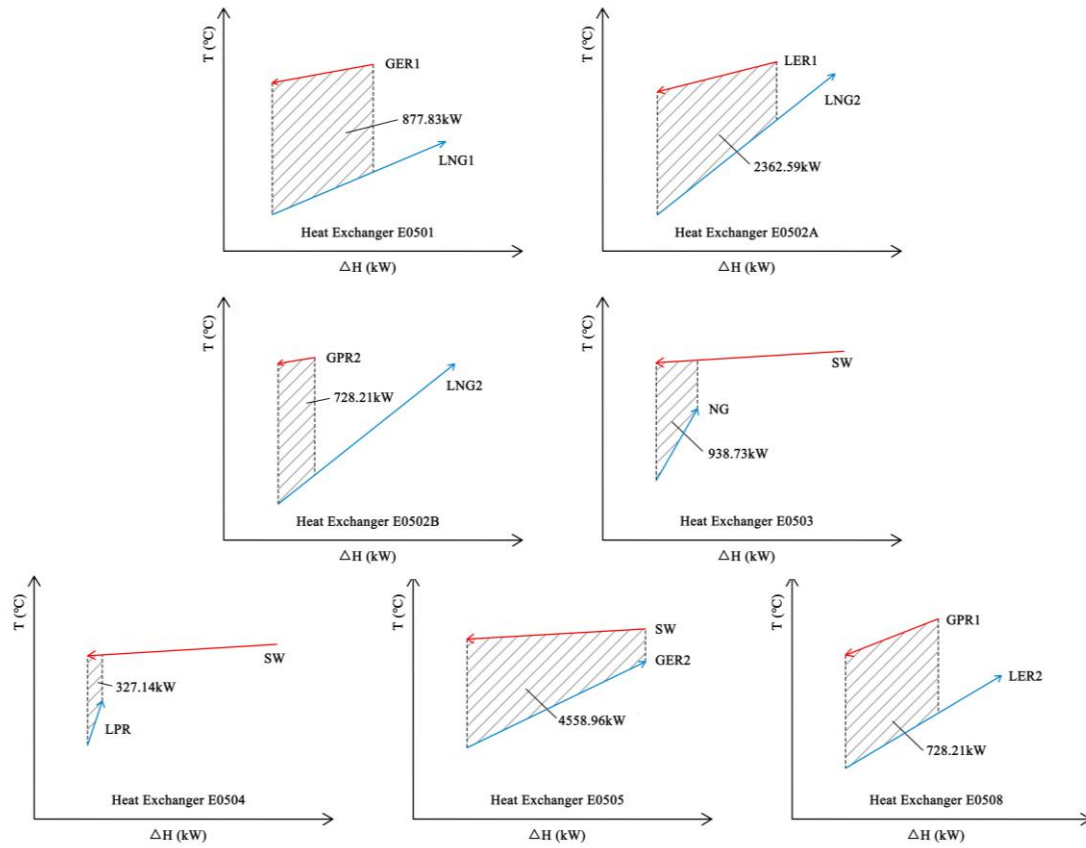


Figure 11. Heat Transfer in Heat Exchangers of existing HEN.

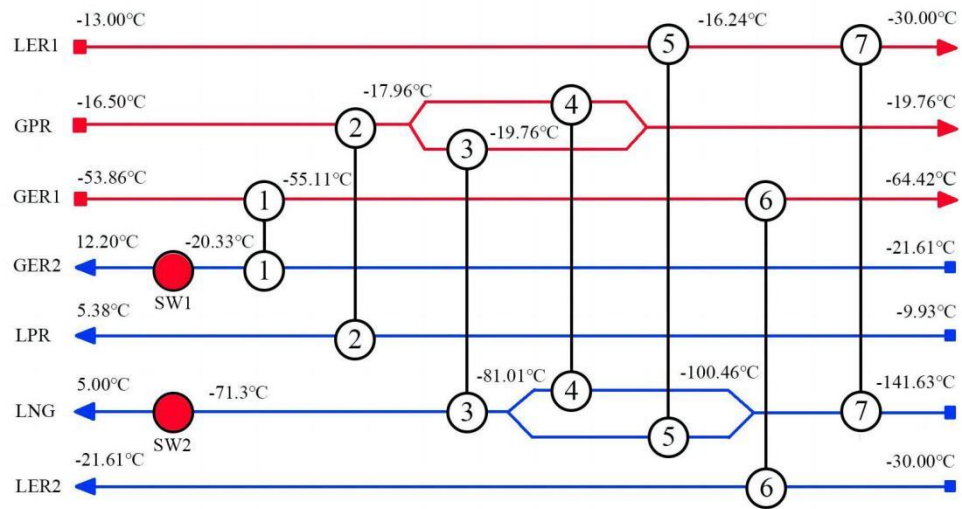
According to the Pinch Analysis, when $\Delta T_{\min}$ is 16.1 °C, the following three HENs are retrofitted, with hot streams located in the top half of the grid diagram and cold streams positioned below them. Firstly, it is established that the transformation requirement for the Heat Exchange Network is to enhance heat exchange with the aim of improving energy efficiency, reducing investment costs, and meeting increased heat load demands. Subsequently, based on the heat exchangers with distinct performance indices outlined in **Table 5**, the maximum number of heat exchangers is determined by the minimum heat transfer area, and the heat exchange configurations for different streams were established. Simultaneously, it is ensured that all three schemes incorporate some heat exchangers with identical heat exchange areas and

heat load, facilitating heat exchange and investment cost calculations and enabling the proposal of three distinct heat exchange schemes. **Fig. 12** depicts the modified HEN in the developed dual-fluid condensation Rankine cycle system.

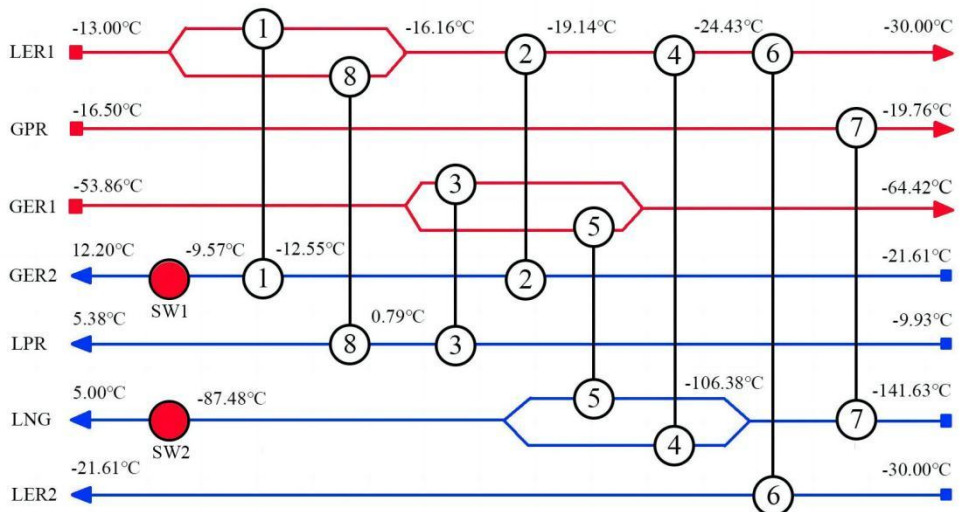
Table 5. The performance indicators of different heat exchangers used in the study.

Exchanger name	Area(m ²)	Heat recovered/ Load (kW)
1	6.85	78.42
2	9.41	488.83
3	7.83	299.64
4	5.86	299.64
5	1.98	299.64
6	8.48	514.58
7	3.85	1492.24
8	3.67	206.46
9	7.57	94.75

(a)



(b)



(c)

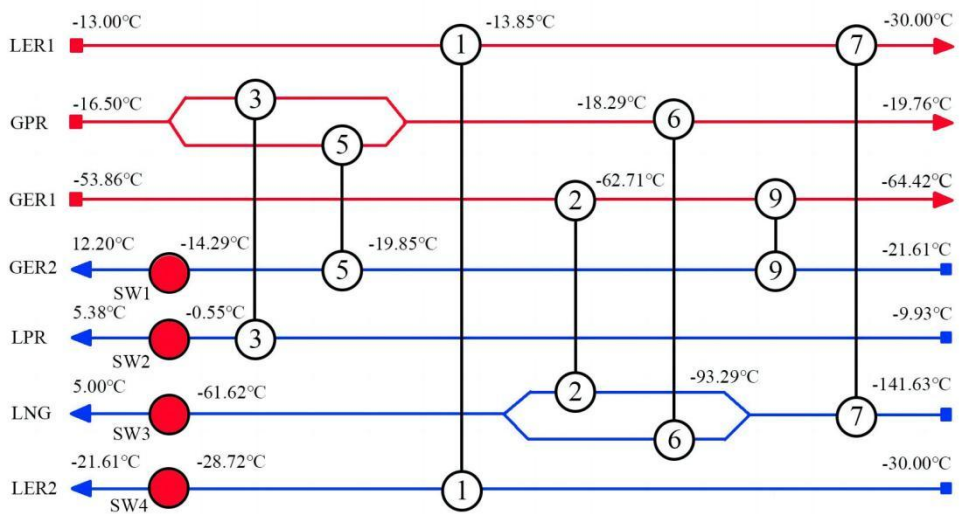


Figure 6. Grid diagram of the HEN retrofit results in case study: (a) case 1; (b) case 2; (c) case 3.

3.3.1 Case 1

In **Fig. 12 (a)**, there are a total of seven heat exchangers. LNG cold energy is primarily utilised for heat exchange, with four heat exchangers being used to achieve a cold energy consumption of 2469.58 kW. During this process, LNG is only heated from -141.63 to -71.3 °C, which means that there is still 2356.82 kW of unused cold energy, resulting in a 44.62 % cold energy utilisation rate for LNG. For the heating requirements, seawater is used as a hot utility facility with a flow rate of 120.43 m³/h, resulting in a temperature change from 25 to 21 °C. To achieve a total of 4111.62 kW of heating, two utility heat exchangers were employed. Specifically, the hot utility located on stream GER2 utilised 132.85 m³/h of seawater, with the temperature changing from 25 to 20.1 °C. The following **Fig.13** shows the heat transfer in heat exchangers of the HEN retrofit by case 1.

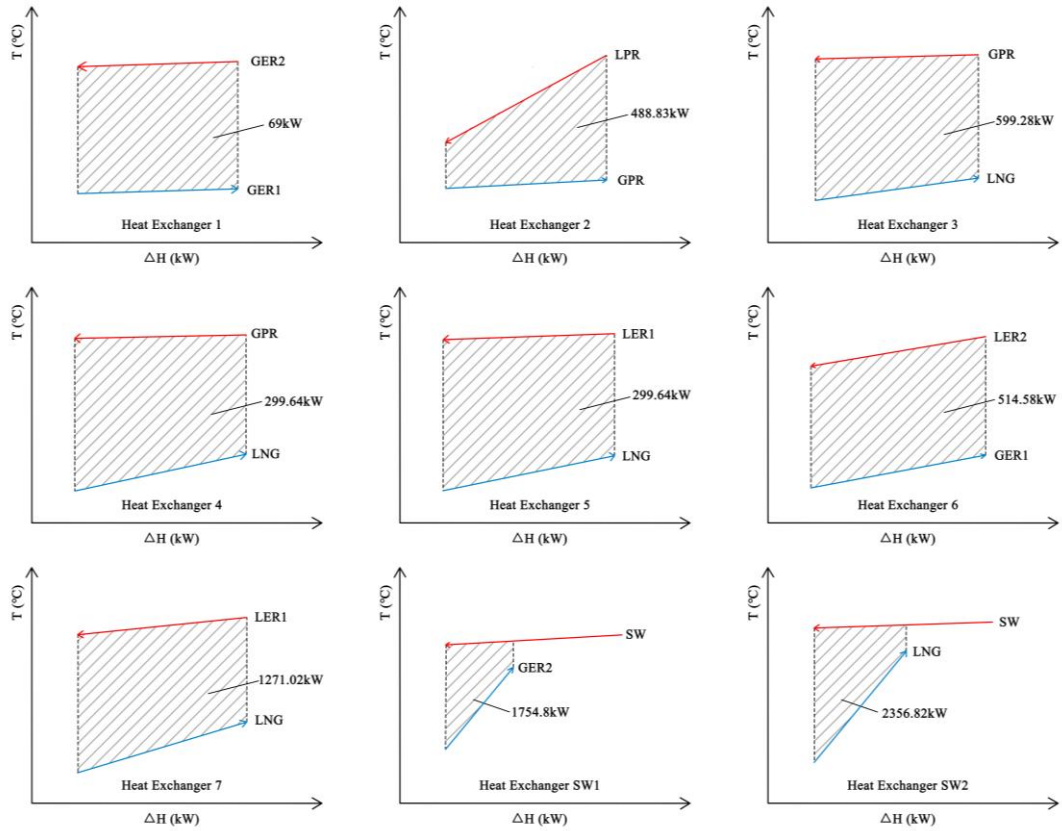


Figure 13. Heat Transfer in Heat Exchangers of the HEN retrofit by case 1.

3.3.2 Case 2

In **Fig. 12 (b)**, there are a total of eight heat exchangers. Three heat exchangers are employed in the LNG cold streams, achieving a cold energy consumption of 1671.69 kW. This means that 797.89 kW of LNG cold energy remains unused in comparison to the existing case, resulting in a 41.32 % utilisation rate for LNG cold energy. During this process, LNG is only heated from -141.63 to -87.48 °C, which means that there is still 2855.07 kW of unused cold energy. For the heating requirements, seawater is used as a hot utility facility with a flow rate of 104.41 m³/h, resulting in a temperature change from 25 to 21.7 °C. To achieve a total of 4111.62 kW of heating, two utility heat exchangers were employed. Specifically, the hot utility

located on stream GER2 utilised $158.64 \text{ m}^3/\text{h}$ of seawater, with the temperature changing from 25 to 20°C . The following **Fig.14** shows the heat transfer in heat exchangers of the HEN retrofit by case 2.

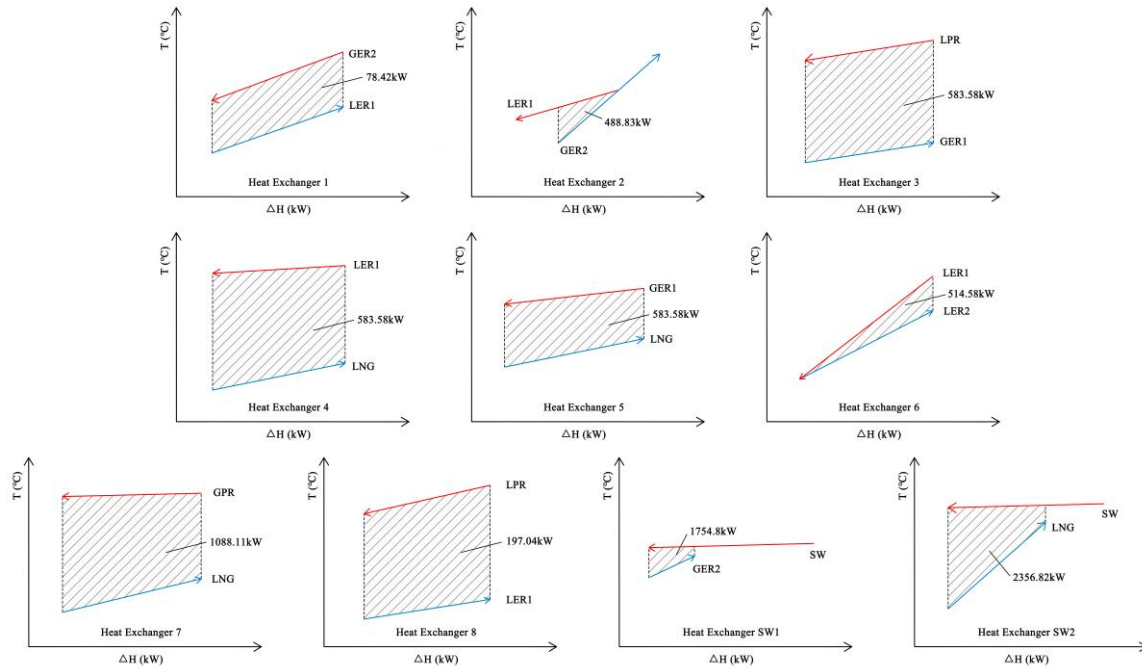


Figure 14. Heat Transfer in Heat Exchangers of the HEN retrofit by case 2.

3.3.3 Case 3

In **Fig. 12 (c)**, there are a total of seven heat exchangers. LNG cold energy is primarily utilised for heat exchange, with three heat exchangers being used to achieve a cold energy consumption of 2469.9 kW . During this process, LNG is only heated from -141.63 to -61.62°C , which means that there is still 2056.86 kW of unused cold energy, resulting in a 54.56% cold energy utilisation rate for LNG. For the heating requirements, seawater is used as a hot utility facility with a flow rate of $97.91 \text{ m}^3/\text{h}$, resulting in a temperature change from 25 to 20.9°C . To achieve a total of 4111.62 kW of heating, four utility heat exchangers were employed. Specifically, the

hot utility located on stream GER2, LPR, and LER2 utilised $19.53 \text{ m}^3/\text{h}$ of seawater with the temperature changing from 25 to 22.3°C , $135.71 \text{ m}^3/\text{h}$ of seawater with the temperature changing from 25 to 20.8°C , $25.14 \text{ m}^3/\text{h}$ of seawater with the temperature changing from 25 to 20.2°C , respectively. The following **Fig.15** shows the heat transfer in heat exchangers of the HEN retrofit by case 3.

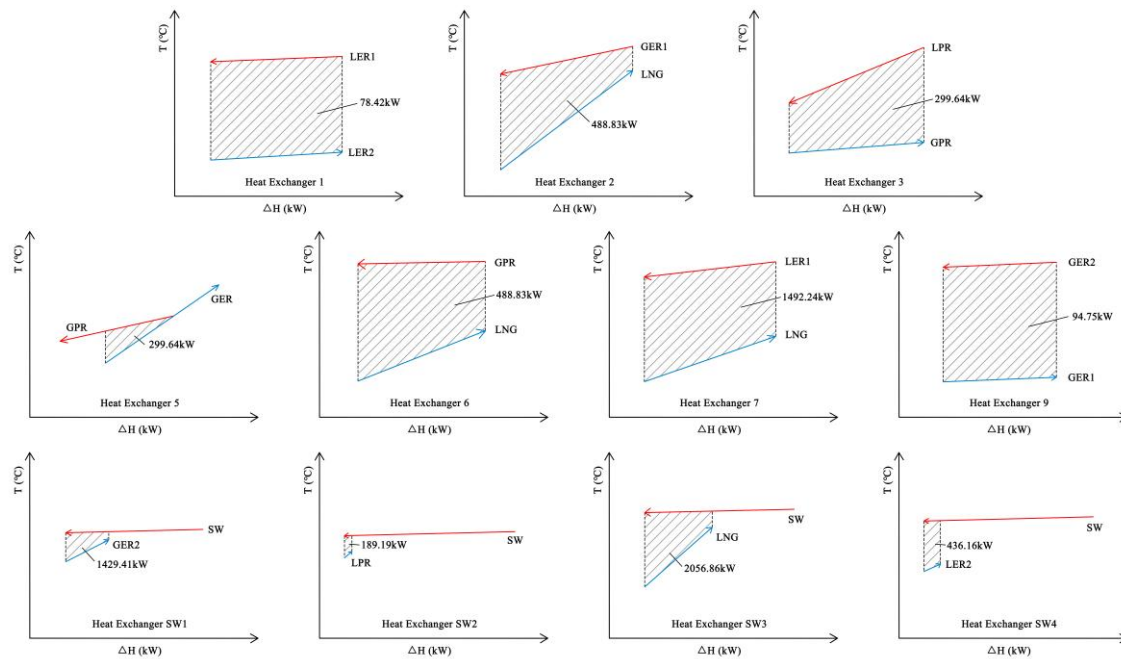


Figure 15. Heat Transfer in Heat Exchangers of the HEN retrofit by case 3.

Finally, for the existing case and the 3 modified ones, we can get the different Pinch temperature of each scheme can be clearly seen in **Fig.16**, which is -21.95°C , -18.19°C , -21.01°C , -22°C , respectively.

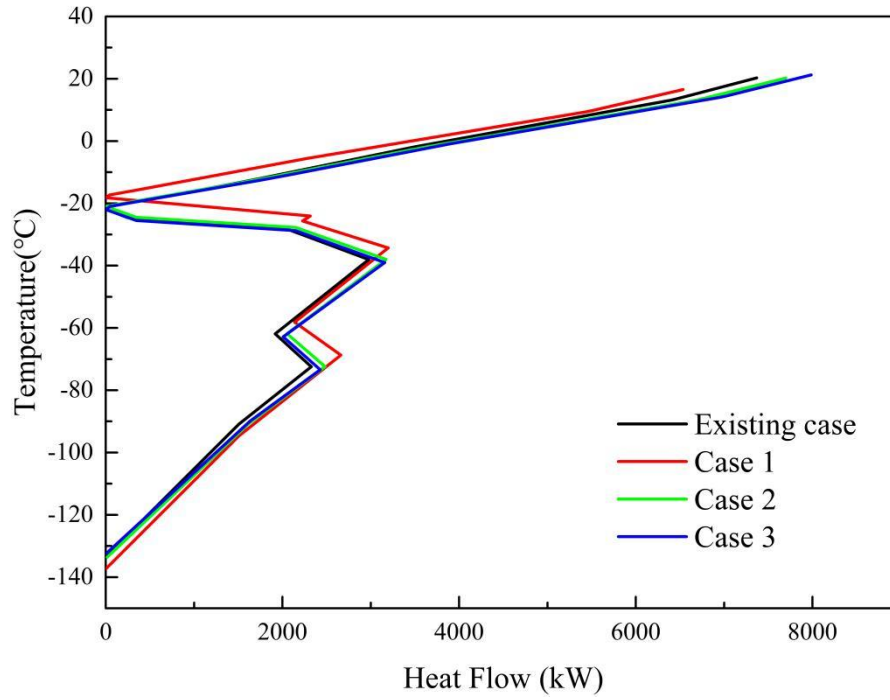


Figure 16. Grand composite curve in each case.

3.4 Energy analysis results

The mass flow rate of ethylene, propane, and LNG were 53.70s, 121.40 and 15.60 kg/s respectively. According to the formulas and component equations listed in Table 1, the energy analysis results for each of equipment within the dual-fluid condensation Rankine cycle system can be obtained. Specific data is presented in **Table 6**.

Table 6. Energy analysis results of each exchanger, expansion generator, and pump in the dual-fluid condensation Rankine cycle system.

Symbol	W(kW)	Q(kW)	m(kg/s)	Δh_i (kJ/kg)
E-0505	/	1815.60	53.70	33.81
E-0504	/	1858.63	121.40	15.31
K-0501	4547.42	/	53.70	84.68

K-0502	3167.98	/	121.40	26.09
E-0501	/	665.81	15.60	42.68
E-0502A	/	1231.62	15.60	78.95
E-0502B	/	1231.62	15.60	78.95
P-0501	2761.25	/	53.70	51.42
P-0502	1193.36	/	121.40	9.83
Pump2	1014.36	/	253.59	4.00
Pump3	76.08	/	253.59	0.30
Pump4	862.21	/	253.59	3.40
Pump1	317.77	/	15.60	20.37
E-0503	/	390.00	15.60	25.00

679

680 Based on **Eqs. (1) to (3)** as listed, the net power output of the originally designed system is
681 1490.37 kW, the total heat absorption is 3129.05 kW, and the thermal efficiency is 47.63 %.

682 When the LNG regasification rate is 56t/h, the power generation amounts to 7715.4 kW, with
683 the ethylene cycle power generation capacity is 4547.42 kW, and the propane cycle power
684 generation capacity is 3167.98 kW. The ethylene stream cycle capacity is 193.32 t/h, while the
685 propane stream cycle capacity is 437.04 t/h. The initial heat transfer area is 31.08 m². Regarding
686 the three HEN transformation schemes depicted in Figure 10. It's important to note that a larger
687 heat exchange area results in the recovery of more energy.

3.5 Investment cost result

Using Eqs. (4) to (6), we can calculate the investment cost of each transformation scheme. In Eq. (5), the total heat transfer coefficients for liquid to liquid, gas to liquid, and gas to gas are 1.8, 0.06 and 0.04 kW/m²·°C, respectively. Based on the provided Eq. (6), the parameters for the capital cost function, namely α_1 , α_2 , β , are 6.6, 5000, and 0.815, respectively. Using these values, the results for the HEN retrofit of different schemes can be found in Table 7.

Table 7. Comparison of the optimisation results subject to different schemes of HEN retrofit.

	Units	Existing case	Case 1	Case 2	Case 3
$Q_{H,min}$	kW	7368.3	4111.62	4111.62	4111.62
$Q_{C,min}$	kW	0	0	0	0
W_{net}	kW	1490.37	1917.21	1868.85	1979.38
Q_{tot}	kW	3129.06	3031.64	3331.28	2731.68
A^{HEN}	m ²	31.08	44.26	47.93	45.97
η_{LNG}	%	34.68	44.62	43.49	46.06
η_{th}	%	47.63	63.24	56.10	72.46
C_{INV}	¥	82295.65	109772.97	117135.77	113217.10

Notes: $Q_{H,min}$ is the minimum hot utility energy; $Q_{C,min}$ is the minimum cold utility energy.

While a larger heat exchanger area can recover more heat, it's important to consider that the investment cost is directly tied to the heat transfer area. When you need to expand the heat exchange area in tube and shell heat exchangers, this requires reinstalling pipes or adding more

shells, leading to an increased investment cost. Higher investment costs can render the transformation financially unfeasible. Simultaneously, within the circulating system, the circulating working medium incurs purchase costs only upon initial entry into the cycle, and there's no need for additional replenishment in subsequent cycles. Furthermore, the cold energy originally wasted during the regasification process of converting LNG into NG is utilised, the cost associated with purchasing LNG logistics can be disregarded. Consequently, only the pumping costs for running the fluids need to be considered in this paper.

3.6 The result of high power generation efficiency scheme

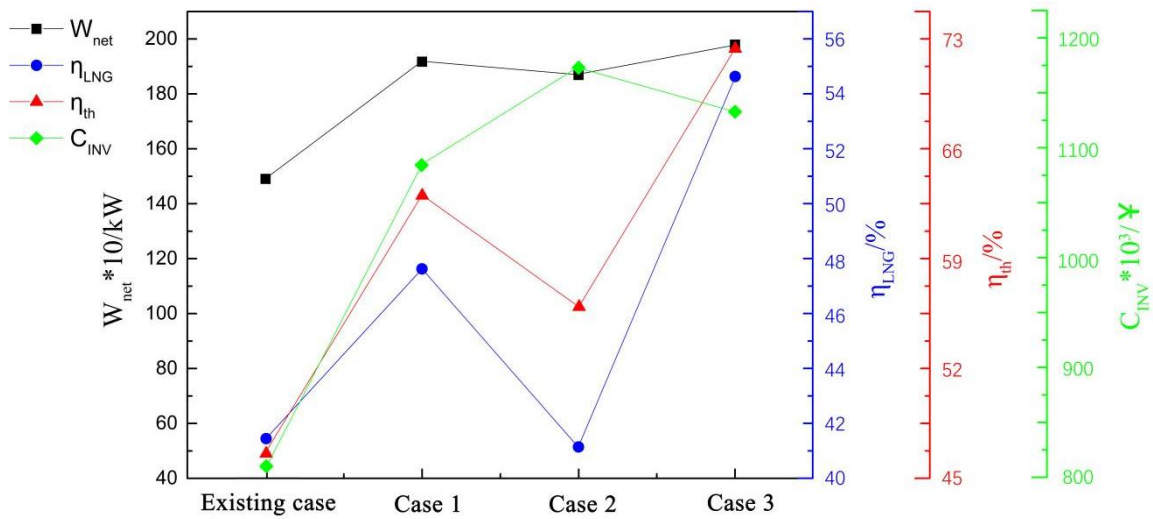


Figure 7. Comparison in net power output, LNG utilisation rate, thermal efficiency, and investment cost of different schemes.

According to **Fig. 17**, case 1 proves to be the optimal choice when prioritising a high net power output. Case 3 is the optimal selection when emphasising high LNG utilisation, while case 3 is the preferred choice for achieving high thermal efficiency. On the other hand, when aiming for low investment costs, case 1 stands out as the optimal option. In conclusion, considering the

combination of lower investment costs, high net power output, and efficient heat transfer, case 1 is selected as the optimal transformation scheme.

When the LNG regasification capacity is set at 48.96 t/h in the case 1, the power generation reaches 8142.24 kW, with the ethylene cycle power generation capacity is 4653.77 kW and the stream cycle capacity is 128.52 t/h, while the propane cycle power generation capacity is 3488.47 kW and the stream cycle capacity is 653.04 t/h.

4 Conclusion

This paper enhanced the efficiency of power generation through the utilisation of LNG cold energy by a dual-fluid condensation Rankine cycle system. The proposed approach integrates energy recovery techniques with enhanced power generation efficiency. The system offers the advantages of optimising the utilisation of LNG cold energy and lowering production costs through improved power generation efficiency.

This system can be implemented in LNG terminals or other facilities utilising LNG. With the enhanced power generation efficiency, the system can not only meet its own power needs but also provide surplus power to other facilities within the plant. But this paper has certain limitations as it does not take into account the power demands for running the fluids through the system. Additionally, the regasification and power generation processes for LNG may require substantial seawater resources, limiting the use of this system in certain areas.

The results indicate that the proposed dual-fluid condensation Rankine cycle system has been retrofitted reasonably. The optimised system achieves a net power output of 1917.21 kW, with a total heat absorption of 3031.64 kW, resulting in a thermal efficiency of 63.24 %. It also requires 4111.62 kW of external utility support, and the utilisation rate of LNG stands at 44.62 %. When the LNG regasification capacity in case 1 is 48.96t/h, the generated power is 8142.24kW. This represents a 28.6% increase in generating power efficiency compared to the existing case, where the LNG gasification capacity was 56.16t/h, resulting in a generating power of 7715.4kW. And this system generates approximately 8 million kWh of green electricity each year, leading to an annual reduction in carbon emissions of about 62.6 thousand tons, equivalent to planting around 563000 trees.

In the future, introducing the cascade utilisation of LNG cold energy into the system could enhance the recovery of cold energy resources more effectively. Expanding on the power generation system proposed in this paper, future research could delve into exploring an integrated system that encompasses cold, hot, and electric energy within a plant. The goal would be to achieve maximum energy recovery and realise cost savings.

Acknowledgements

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758 RESHeat", Grant Agreement # 956255.

759 **Appendix Nomenclature**

760 **Abbreviations**

761	LNG	liquefied natural gas
762	NG	natural gas
763	DC	direct expansion cycle
764	ORC	organic Rankine cycle
765	CC	combined cycle
766	HEN	heat exchanger network
767	TSHI	total site heat integration
768	GCC	grand composite curve
769	TCC	total composite curve
770	SW	sea water
771	GER	gas ethylene refrigerant
772	LER	liquefied ethylene refrigerant
773	GPR	gas propane refrigerant
774	LPR	liquefied propane refrigerant

775 **Subscripts**

776	eva1	evaporator 1
777	eva2	evaporator 2
778	K1	expansion generation 1

779	K2	expansion generation 2
780	con1	condenser 1
781	con2	condenser 2
782	P1	feed pump 1
783	P2	feed pump 2
784	P3	sea water pump 3
785	P4	sea water pump 4
786	P5	sea water pump 5
787	P6	LNG pump 6
788	net	net power
789	tot	total
790	th	thermal
791	re	reheater
792	ETH	ethylene
793	P	propane
794	1-22	the position of the number corresponding with in Fig.5
795	Symbols	
796	CP	heat capacity flow rate [kW/K,kW/°C]
797	ΔH	heat load [kW]
798	$\Delta T_{\min}$	minimum heat transfer temperature difference [°C]
799	η_{th}	thermal efficiency [%]

800	η_{LNG}	LNG efficiency [%]
801	W_{net}	net power output [kW]
802	Q_{tot}	total heat absorption [kW]
803	$\dot{m}$	mass flow rate [kg/s]
804	h	mass enthalpy [kJ/kg]
805	$Q_{\text{H,min}}$	minimum hot utility energy [kW]
806	$Q_{\text{C,min}}$	minimum cold utility energy [kW]
807	ΔT_{LMTD}	logarithmic mean temperature difference [K]
808	$\Delta T_{\text{H_in}}$	inlet temperature of the hot stream [K]
809	$\Delta T_{\text{H_out}}$	outlet temperature of the hot stream [K]
810	$\Delta T_{\text{C_in}}$	inlet temperature of the cold stream [K]
811	$\Delta T_{\text{C_out}}$	outlet temperature of the cold stream [K]
812	A^{HEN}	area of the heat exchanger network [m ²]
813	Q	heat delivered by the heat exchanger [kW]
814	U	total heat transfer coefficient [kW/(m ² ·K)]
815	C_{INV}	investment cost [¥]
816	$\alpha_1, \alpha_2, \beta$	investment cost parameters
817		

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