NEW COLD-LEVEL UTILIZATION SCHEME FOR CASCADE THREE-LEVEL RANKINE CYCLE USING THE COLD ENERGY OF LIQUEFIED NATURAL GAS

by

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The topic of this study is the intermediate fluid vaporizer gasification system for a liquefied natural gas floating storage regasification unit. To reduce the loss of heat exchange, the primary distributary cascade three-level Rankine cycle is optimised based on the cascade three-level Rankine cycle that uses the cold energy of liquefied natural gas to generate power. The optimized primary distributary cascade three-level Rankine cycle is then compared with the original cascade three Rankine cycle established under the same conditions. Then, a secondary distributary cascade three-level Rankine cycle is proposed. Results show that under a liquefied natural gas flow of 175 t/h, the primary distributary cascade three-level Rankine cycle system exhibits a maximum net output power of 4130.72 kW and an exergy efficiency of 23.78%, which is higher than that of the typical cascade three-level Rankine cycle. Moreover, the net output power and exergy efficiency of the primary distributary cascade three-level Rankine cycle system increased by 3.71% and by 3.84%, respectively. The secondary distributary cascade three-level Rankine cycle system exhibits a maximum net output power of 4143.75 kW and an exergy efficiency of 23.85%.

Key words: *liquefied natural gas, three-level Rankine cycle, power generation, distributary*

Introduction

The worldwide consumption of natural gas (NG) has rapidly increased given the efficient and clean combustion of NG. The NG liquefaction, however, consumes considerable power. When the liquefaction-to-storage temperature is –162 °C, the unit energy consumption of the liquefaction process reaches as high as 850 kWh/t. Thus, a considerable temperature difference exists between the terminal liquefied natural gas (LNG) and the ambient temperature, resulting in a large amount of available cold energy [1]. The cold energy of LNG is used in many applications, such as air separation, power generation, refrigeration, liquefied CO_2 and dry ice production, car refrigeration, and automotive air conditioning [2, 3]. Power generation with the cold energy of LNG is most effective use of cold energy because of its short industrial chain, environmental friendliness, and easy recovery [4].

Domestic and foreign scholars have made considerable progress in the research on the utilization of LNG cold energy. Yang *et al.* [5-7] proposed a segmentation model for the utili-

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zation of LNG cold energy. They then developed horizontal and cascade three-level Rankine cycles that are based on the proposed model. However, the power generation of the horizontal three-level Rankine cycle is low, and the exergy loss between the heat exchangers during the first and second Rankine cycle is high. Thus, they proposed a two-stage pumping optimization program. Rao *et al.* [8] used industrial waste heat to gasify refrigerant in a third level Rankine cycle. They proved that the cycle of thermal efficiency and work increases with the increase in evaporation pressure. However, given the high temperature of industrial waste heat, exergy loss is large and this method can only be used under specific conditions. Shi *et al.* [9] proposed a new type of gas-ammonia combined power generation system coupled with solar energy and LNG cold energy. The proposed system provides a new concept for the comprehensive application of fossil energy and renewable energy. Lee *et al.* [10] combined seawater and exhaust gas as the heat sources of a horizontal two-level Rankine cycle. Exhaust gas is used as the heat source of the first and second Rankine cycles. The thermal efficiency and cold utilization of the proposed system are relatively high, and the cycle is suitable for powering small fishing boats. However, the temperature of the exhaust gas remains high, thus increasing the exergy loss of the heat exchanger. In addition, the system has low net output power, W_{net}. Li *et al.* [11] proposed a cascade power utilization of solar energy and LNG organic Rankine cycle system, this system has two kinds of working fluids. The hot water heated by solar energy gasify first working fluids to work, the refrigerant after working out gasify the other refrigerant to work, and the second kinds of refrigerant after working out gasify LNG to work. The system realizes the combination of low temperature Rankine cycle power generation and direct expansion method. Bao *et al.* [12] proposed the two-stage condensed Rankine cycle system. Its net power output and thermal efficiency are better than those of the combined cycle. Suna *et al.* [13] proposed a novel Rankine power cycle which uses a mixture of hydrocarbons to recover the cold energy from LNG, and they found that the ethylene is more suitable than ethane to be used in the mixed working fluid. Kim *et al.* [14] proposed binary mixture working fluid cascade ORC utilizing LNG cold energy, and the optimum working fluid and process configuration are obtained via an optimization. Cui [15] established a five-level Rankine cycle that uses the cold energy of LNG to generate power. Although the system has an efficiency of 61%, it cannot be easily applied in practice give its complexity and requirements for numerous pieces of equipment.

The LNG floating storage regasification unit (LNG-FSRU) system is usually used at sea, and the pressure to deliver NG is usually higher than the NG pressure at the export of the intermediate fluid vaporizer (IFV) system on land. According to the American standards for LNG pressure, gas pressure should reach more than 7 MPa when LNG is being transported over long distances [16]. The LNG is in a supercritical state during transport.

This study focuses on the IFV regasification system of LNG-FSRU. The LNG is gasified under supercritical gasification pressure, and seawater is used as a heat source. This study is based on the idea that increasing a hot fluid in the LNG heat exchanger, the heat transfer temperature difference between the hot fluid and the cold fluid is smaller than that of the original two heat exchangers in the heat exchanger and takes a way of distributary to reduce the exergy loss of the LNG heat exchanger. A new scheme, the primary distributary cascade three-level Rankine cycle (PDCRC), is proposed to address the problem of the high exergy loss of the LNG evaporator in the third level Rankine cycle. Another new scheme, which is the secondary distributary cascade three-level Rankine cycle (SDCRC) is proposed on the basis of the previous scheme. Detailed thermodynamic analysis is conducted to optimize the LNG-FSRU cold energy generation system.

Composition of the PDCRC and SDCRC system

The molar composition of LNG is selected as follows: 95% methane, 3% ethane, and 2% propane. The gasification pressure is 8 MPa, which is supercritical pressure. Only reference [5] has previously proposed the three-level Rankine cycle power generation system using LNG cold energy during steaming. The original cascade three-level Rankine cycle system is shown in fig. 1. The preliminary HYSYS simulation was performed to identify the best combination of the refrigerants under supercritical pressure for the cascade three-level Rankine cycle. The efficiency and exergy loss of the components in the system with the given assumptions and system parameters were analysed. Given the high exergy loss of LNG evaporator 3 (as shown in tab. 5), a PDCRC system was proposed, as shown in fig. 2. The proposed system increases a fluid that flows from the shunted refrigerant in the second Rankine cycle in the LNG evaporator 3. The exergy loss of LNG evaporator 3 is reduced because the temperature of the fluid is lower than that of the original fluid in LNG evaporator 3. Furthermore, an improved scheme for LNG evaporator 2, the SDCRC scheme, was proposed by using the same optimization and improvement method. The diagram of the SDCRC system is shown in fig. 3.

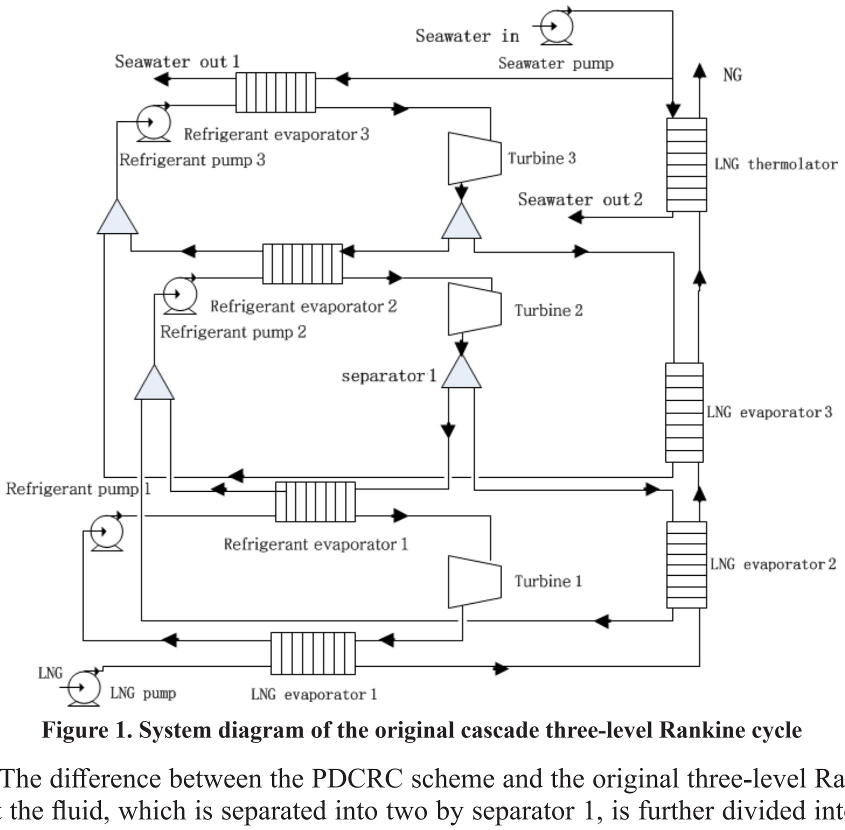

Figure 1. System diagram of the original cascade three-level Rankine cycle

The difference between the PDCRC scheme and the original three-level Rankine cycle is that the fluid, which is separated into two by separator 1, is further divided into three in the cascade three-level Rankine cycle. The multiseparated stream is introduced into the LNG evaporator 3 of the third level Rankine cycle for further heat exchange with the NG. It absorbs the cooling capacity of NG and then mixes with the other two fluids which have been cooled. Then, the mixed fluid flows into refrigerant evaporator 2.

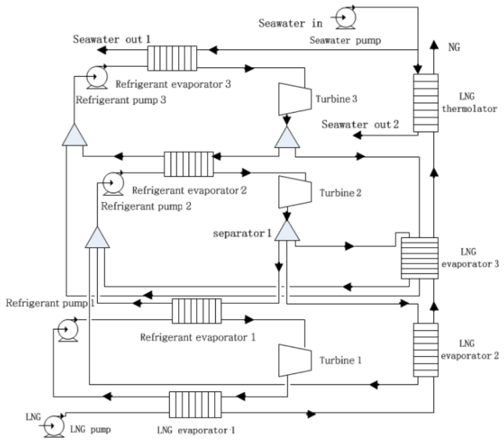

Figure 2. System diagram of the PDCRC

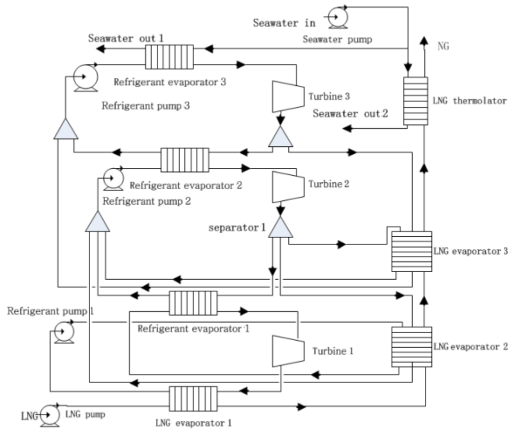

Figure 3. Diagram of the SDCRC

The SDCRC system is based on the PDCRC system. The refrigerant, which is pressurised by refrigerant pump 1, is first introduced into LNG evaporator 2 to enable additional heat exchange and then introduced to refrigerant evaporator 1.

Then, the thermodynamic analysis of the PDCRC system, the SDCRC system and the original cascade three-level Rankine cycle system was conducted. The three systems were also compared, and the optimal combination of refrigerant and parameter matching for the PDCRC system and the SDCRC system was determined.

Determination of the optimal combination of refrigerant and parameter matching

Selection of system parameters

For simulation calculations and analysis, the flow of LNG is assumed to be 175 t/h. The simulation calculation was conducted with the following settings:

- The condensed pressure of the circulating fluid is 110 kPa.
- The temperature of seawater, which is the heat source, is 20 °C. The ambient temperature is 25 °C.
- The minimum end difference of all heat exchangers is 5 °C.
- In all heat exchangers, the temperature of the hot fluid in addition to that of the refrigerant of the first Rankine level from the LNG evaporator 2 in the SDCRC is 2 °C.
- The efficiency of the turbine is 80% and that of the pump is 75%.
- The losses in the pressure and heat of all heat exchangers and pipes are ignored.
- The refrigerants, which are in the inlet of turbine, are in a saturated gas state.

The PDCRC system

Optimization of the refrigerant combination

The net output power and safety of the system should be considered during the selection of refrigerants. The adaption of the critical temperature of the refrigerants and temperature of the heat source should be considered as well. The selected refrigerant is important given its direct effect on the recovery rate of LNG cold energy [17].

The condensing temperatures of common refrigerants under 110 kPa are shown in tab. 1.

Table 1. Condensing temperatures of common refrigerants under 110 kPa

R1150	R170	R23	R116	R1270	R290	R717	R134a	R152a	R600a
Units [°C]									
–102.64	–87.22	–80.53	–77.20	–46.16	–40.55	–31.44	–24.24	–22.61	–9.93

According to the principle of the use of cold cascade, the cold energy of NG, which is heated in the third level Rankine cycle, may also be used for desalination, cold storage and other cold energy use. Therefore, the temperature of the NG at the outlet of the LNG evaporator in the third level Rankine cycle was selected as approximately –45 °C [18, 19]. The minimum terminal temperature differences of the heat exchangers must be 5 °C. Table 1 shows that R290 and R1270 are the most suitable refrigerants for the third level Rankine cycle. The temperature of LNG increased by –158 °C after it is pressurised by the pump. When R290 was selected as a refrigerant of the third level Rankine cycle, the temperatures of the three-level Rankine cycle using the cold energy of the LNG cold energy ranged from –158 °C to –45.55 °C. When R1270 was selected as the refrigerant of the third level Rankine cycle, the temperatures of the three-level Rankine cycle using the cold energy of the LNG ranged from –158 °C to –51.16 °C. The R1150, R170, R23, R116, and R1270 can satisfy the requirements given in tab. 1. These refrigerants should be allocated to the two cycles, and the need to minimize the exergy loss of heat transfer caused by a large difference of temperature should be considered. Therefore, R1150 and R170 were selected as possible refrigerants for the first level Rankine cycle, and R23, R116 and R1270 were selected possible refrigerants of the second level Rankine cycle. When the refrigerant of the second level Rankine cycle was R1270, the refrigerant of the third level Rankine cycle can only be R290. Thus, 10 refrigerant combinations are possible.

In the HYSYS simulation, the ratio of the refrigerant, which is separated in the second level Rankine cycle by separator 1 into the refrigerant evaporator 1 of the first level Rankine cycle, was changed at intervals of 0.01 in the range of 0.01 and 0.99 (referred to as *ratio* in the succeeding sections). When the refrigerant of the second level Rankine cycle was R116, the system was not established because R116 is a dry fluid. Therefore, the outlet temperature of the cold fluid (the refrigerant of first level cycle) of the refrigerant evaporator 1 was higher than the condensation temperature of R116, causing temperature crossing in refrigerant evaporator 1. Consequently, R116 was not considered. The results of HSYSY simulation showed that the selected refrigerant of the third level Rankine cycle does not affect this ratio (refer to the detailed analysis below). When the system can be implemented, the ratios of different combinations of refrigerants are set as shown in tab. 2.

When different refrigerants were combines, the flow rates of the fluid with the given ratio and the refrigerant of LNG evaporator 2 were kept constant because the temperature and the flow rate of the cold fluid at the inlet and outlet of refrigerant evaporator 1 and the LNG evaporator 2 remained unchanged. The study took a example that the refrigerants of the first and second levels were R1150 and R23, respectively. The ratio cannot be less than 0.62 because of the following reasons: the incoming R23 stream in the LNG evaporator 3 was –80.53 °C. The temperature of the NG in the LNG evaporator 3 changed from –85.53 °C to –45.55 °C. Thus, NG could only be heated by the R23 stream to at most –80.53 °C. Even when the ratio was reduced to increase the flow

Table 2. Ratios of different combinations of refrigerants that make the system implemented

Different combinations of refrigerants	Ratio
R1150, R23, R290	0.62-0.66
R1150, R23, R1270	0.62-0.66
R1150, R1270, R290	0.38-0.39
R170, R23, R290	0.82-0.88
R170, R23, R1270	0.82-0.88
R170, R1270, R290	0.49-0.51

rate of R23, the increased flow rate could not increase the temperature of NG to more than –80.53 °C. When the ratio was increased, the flow of the hot stream (the refrigerant of third level cycle) in LNG evaporator 3 was reduced, and NG could not increase the temperature from –85.53 °C to –45.55 °C in LNG evaporator 3. Thus, when the ratio was less than 0.62, the process was not established. The ratio cannot be greater than 0.66 because as shown by HYSYS simulation, the flow rates of R23 in refrigerant evaporator 1 and LNG evaporator 2 were 141917.78 kg/h and 70394.75 kg/h, respectively. At this time, the flow rate of R23 at a given ratio accounted for the total flow rate of 0.668. Therefore, the flow rate of R23 could only be up to the total flow rate of 0.668, and the given interval is 0.01. Thus, the ratio could not exceed 0.66. This reason also explains why the refrigerant of the third level cycle does not affect the ratio.

For the other three refrigerant combinations of the first and second level cycle, the ratios exhibited a certain interval when the process was established because of the same reasons.

Results of refrigerant filtering

The HYSYS was used to calculate the net output power of the system under different combinations of refrigerants and ratios. The property package of refrigerants is Peng-Robinson. The net output power of the system is shown in fig. 4. The dryness of the refrigerant in the outlet of the three turbines under different combinations of refrigerants when the net output power of the system reaches maximum is shown in fig. 5.

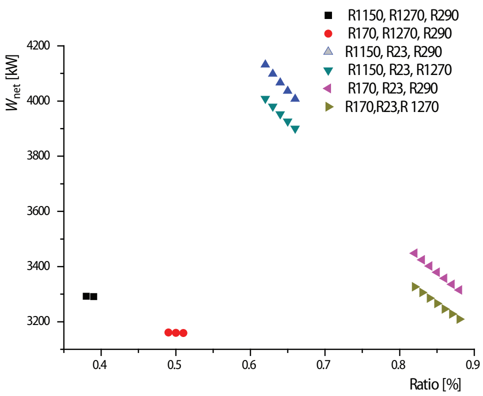

Figure 4. Net output power of the system

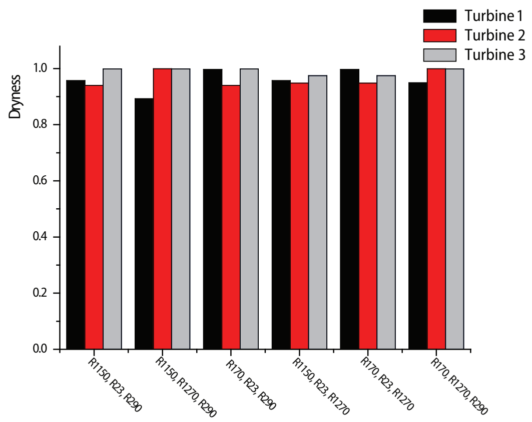

Figure 5. Dryness of the different combinations of refrigerants

Figure 4 shows that when combination of refrigerants was R1150, R23, and R290 and the ratio was 0.62, the system produced the highest net output power of 4130.72 kW. Figure 4 also shows that under the different combinations of refrigerants, the net output power of the system continuously decreased when the ratio increased. The combination of refrigerants R1150, R23, and R290 was considered because of the following reasons. The net output power of the system is determined by three turbines (the power changes at the pump are negligible). Because the parameters of R1150 in the turbine 1 remained unchanged, the output power of turbine 1 was unchanged. Because the flow rate of R23 in the refrigerant evaporator 1 remained the same, the total flow rate of R23 before shunting decreased when the ratio increased, thus, the output power of turbine 2 decreased. As a result of the increased ratio, the total flow rate of R23 before shunting decreased, and the flow rate of R23 in the refrigerant evaporator 1 and that of the R23 in LNG evaporator 2 remained constant to exhibit the same decrement of the flow rate of the R23 stream from LNG evaporator 3 and that of R23 in the refrigerant evaporator 2 (the total flow rate of the R23 before shunting). In addition, the decrement of heat transfer caused by the decrement of the flow rate of R23 in refrigerant evaporator 2 was higher than the decrement of heat transfer caused by the decrement of the flow rate of R23 in the LNG evaporator 3. The heat transfer of the R23 and R290 streams in refrigerant evaporator 2 were numerically equal, and the decrement of the heat transfer of the hot R23 stream was equal to the increment of the heat transfer of the hot R290 stream in LNG evaporator 3. Therefore, the decrement of the heat transfer of the hot R290 stream in refrigerant evaporator 2 was higher than the increment of the heat transfer of the hot R290 stream in LNG evaporator 3. The change in the heat transfer per unit mass of the hot R290 stream in refrigerant evaporator 2 and the hot stream R290 in LNG evaporator 3 was the same. The decrement of the flow rate of the hot R290 stream in refrigerant evaporator 2 was higher than the increment of flow rate of the hot R290 stream in LNG evaporator 3. Therefore, the total flow of R290 and the output power of turbine 3 decreased. For the other combinations of refrigerants, the reasons of that are same as previously given. Figure 5 shows that when the combination of refrigerants was R1150, R23, and R290, the dryness of the three turbine export refrigerants were high. Therefore, considering the net output power of the system and the three turbine outlet dryness, system performance was optimal when the refrigerants of the three level Rankine cycles were R1150, R23, and R290.

The SDCRC system

The system parameters and assumptions were the same as those of the PDCRC, and the preliminary results for refrigerant selection were the same as the latter. Ten different combinations were possible. Refrigerant R290 was used in the third level Rankine cycle of the improved system to ensure that the NG export temperature of the third level Rankine cycle in two distributary cascade three-level Rankine cycle systems was same. Thus, six refrigerant combinations are obtained.

In the HYSYS simulation, the temperature range encompassed the inlet temperature of the refrigerant of the first level Rankine cycle in LNG evaporator 2 to the inlet temperature of NG in LNG evaporator 2. The value of the outlet temperature of the first level Rankine circulating refrigerant in LNG evaporator 2 was set in increments of 0.5 °C. When the temperature was provided, the ratio of refrigerant in the second level Rankine cycle was separated by the separator 1 to the refrigerant evaporator 1 of the first level Rankine cycle at intervals of 0.01 in the range of 0.01 and 0.99 (referred to as *new ratio*). The simulation results showed that when the refrigerant of the second level Rankine cycle was R116, the system process was not established, thus, R116 was not considered. In addition, the refrigerant of the third level Rankine cycle still did not affect this new ratio because of the same reasons as those revealed by the analysis of the PDCRC system. The corresponding temperature range of the different refrigerant combinations is shown in tab. 3. The new ratios, which established the process under different combinations and different outlet temperatures of the first level Rankine circulating refrigerant in LNG evaporator 2, are shown in tab. 4.

Table 4 shows that the new ratio still existed in the range of values. The reason of it was the same as that of the PDCRC.

Table 3. Corresponding temperature range of the different combinations of refrigerants

Combinations of refrigerants	Temperature range
R1150, R23, R290	–104.5-107.6 °C
R1150, R1270, R290	–104.0-107.6 °C
R170, R23, R290	–89.21-92.22 °C
R170, R1270, R290	–88.95-92.22 °C

Table 4. New ratios that establish the system under different combinations and different outlet temperatures of the first level Rankine circulating refrigerant in LNG evaporators 2

Combinations of refrigerants	New ratios						
R1150, R23, R290	–105.0 °C	–105.5 °C	–106.0 °C	–106.5 °C	–107.0 °C	–107.5 °C	–
	0.62-0.66	0.62-0.67	0.62-0.67	0.63-0.67	0.63-0.67	0.63-0.67	–
R1150, R1270, R290	–104.5 °C	–105.0 °C	–105.5 °C	–106.0 °C	–106.5 °C	–107.0 °C	–107.5 °C
	0.38-0.39	0.38-0.40	0.38-0.40	0.38-0.40	0.38-0.40	0.38-0.40	0.38-0.40
R170, R23, R290	–89.7 °C	–90.21 °C	–90.71 °C	–91.21 °C	–91.71 °C	–92.21 °C	–
	0.82-0.88	0.82-0.89	0.82-0.89	0.83-0.89	0.83-0.89	0.83-0.89	–
R170, R1270, R290	–89.45 °C	–89.95 °C	–90.45 °C	–90.95 °C	–91.45 °C	–91.95 °C	–
	0.49-0.51	0.49-0.51	0.49-0.52	0.49-0.52	0.50-0.52	0.50-0.52	–

When the outlet temperature of the first level Rankine circulating refrigerant in the LNG evaporator 2 was provided, changing the new ratio only affected the net output power of the second and third level Rankine cycles. This result was the same as that for the PDCRC. Section *Results of refrigerant filtering* states that when the ratio is small, the net output power of the system is large. For the SDCRC, when the new ratio was small, the net output of the system was also large. Therefore, the minimum value of the new ratio was considered to calculate the net output of the system when the system was established. Figure 6 shows the net output power of the SDCRC system with different combinations of refrigerants and the different outlet temperatures of refrigerants of the first level Rankine cycles in LNG evaporator 2.

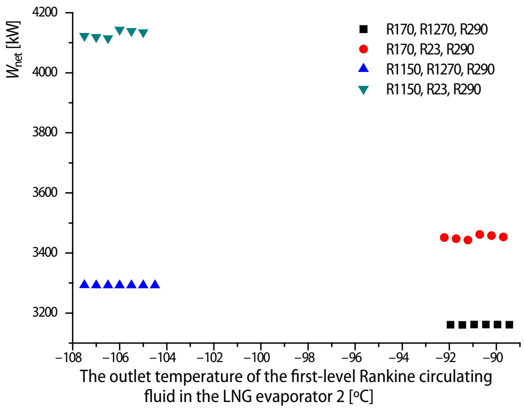

Figure 6. Comparison of the net output power of the SDCRC system

Under the different combinations of the refrigerants, the dryness of the three turbine export refrigerants in the SDCRC was the same as that in the PDCRC, as shown in fig. 5. Figure 6 shows that when the refrigerants are R1150, R23, and R290, the outlet temperature of the refrigerant of the first level Rankine cycle introduced into the refrigerant evaporator 2 was –106 °C and with a new ratio of 0.62, the maximum net output power of the system was 4143.75 kW. The dryness of the three turbine export refrigerants was high, indicating that the combination of R1150, R23, and R290 is the best combination of refrigerants for the secondary three-level Rankine cycle system.

Thermodynamic analysis and comparison of the PDCRC system and SDCRC system

When the process is optimal, the parameters of the PDCRC system and SDCRC system are shown in the fig. 7.

A dotted line is added on the basis of PDCRC to represent SDCRC, wherein the data in parentheses is the data of SDCRC. The same data in PDCRC and SDCRC only gives the data of PDCRC.

The definitions of exergy loss and exergy efficiency of equipments and system are shown in tab. 5.

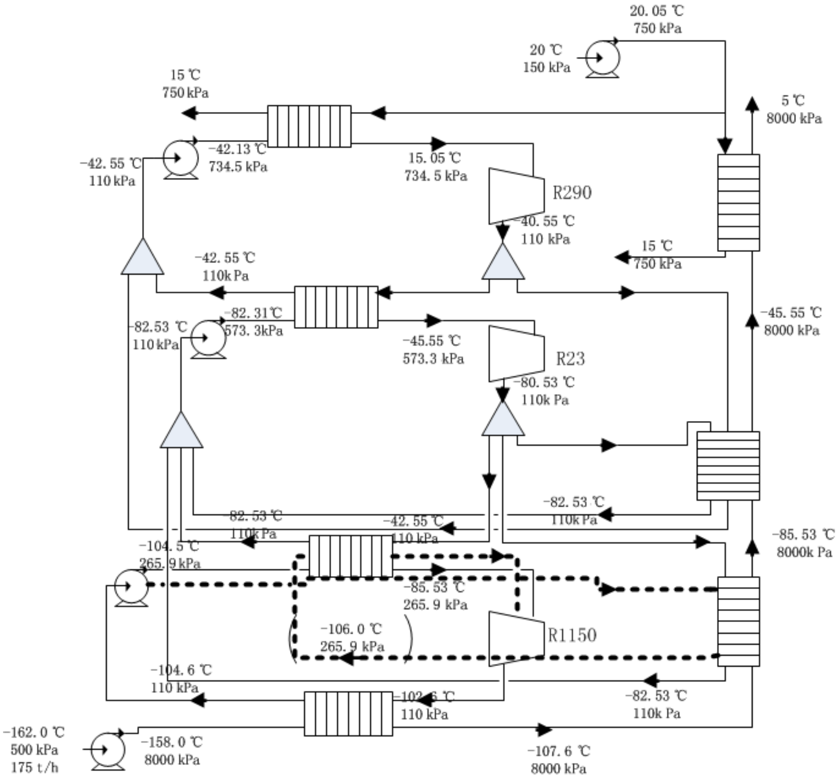

Figure 7. The data graph for the cycle

Table 5. Definitions of exergy loss and efficiency

Equipment	Exergy consumption	Income exergy	Exergy loss	Exergy efficiency
Pump	W_P	$m(ex_{out} - ex_{in})$	Exergy consumption – Income exergy	Income exergy/Exergy consumtion
Turbine	$m(ex_{in} - ex_{out})$	W_T	Exergy consumption – Income exergy	Income exergy/Exergy consumption
Heat exchanger	$m(ex_{1in} - ex_{1out})$	$m(ex_{2out} - ex_{2in})$	Exergy consumption – Income exergy	Income exergy/Exergy consumption

The exergy efficiency of a system is defined:

$$\eta_{ex} = \frac{W_{net}}{Ex_{LNG} + Ex_{SW}} \tag{1}$$

The results are shown in tab. 6.

Table 6 shows that the PDCRC had a lower exergy loss at LNG evaporator 3 than the original cascade three-level Rankine cycle. This result showed the feasibility of the idea of proposing the PDCRC. The net output power of the PDCRC was 3.71% higher than that of the original three-tier Rankine cycle, and the exergy efficiency improved by 3.84%. Table 6 also shows that exergy loss and exergy efficiency in LNG evaporator 2 and in the system of the SDCRC improved relative to those of the PDCRC. These results showed the feasibility of the proposed system. The net output power of the SDCRC increased by 4.04%, and exergy efficiency increased by 4.15% compared with those of the original cascade three-level Rankine cycle.

Table 6. Comparison of exergy loss and net output power of the PDCRC, the SDCRC the original cascade three-level Rankine cycle

Program	Original cascade three-level Rankine cycle		PDCRC		SDCRC	
Equipment	Exergy loss [kW]	Exergy efficiency	Exergy loss [kW]	Exergy efficiency	Exergy loss [kW]	Exergy efficiency
LNG evaporator 1	3266.63	65.9%	3266.63	65.9%	3266.63	65.9%
LNG evaporator 2	624.70	79.7%	624.70	79.7%	612.08	80.3%
LNG evaporator 3	1845.55	65.6%	1564.06	70.9%	1540.29	71.4%
LNG thermolator	1649.63	13.1%	1649.66	13.1%	1649.66	13.1%
Refrigerant evaporator 1	435.94	91.9%	435.94	91.9%	448.55	91.8%
Refrigerant evaporator 2	750.75	85.1%	809.39	85.1%	814.34	85.1%
Refrigerant evaporator 3	1296.09	38.9%	1302.39	38.9%	1302.92	38.9%
Refrigerant pump 1	3.69	45.0%	3.69	45.0%	3.69	45.0%
Refrigerant pump 2	9.95	61.5%	10.66	61.7%	10.73	61.7%
Refrigerant pump 3	36.17	61.4%	36.29	61.5%	36.31	61.5%
LNG pump	967.69	11.6%	967.69	11.6%	967.69	11.6%
Seawater pump	168.86	88.8%	169.55	88.8%	169.60	88.8%
Turbine 1	277.64	69.6%	277.64	69.6%	277.64	69.6%
Turbine 2	666.82	72.1%	718.92	72.1%	723.31	72.1%
Turbine 3	1396.99	75.7%	1403.66	75.7%	1404.22	75.7%
Exergy loss of the system [kW]	13397.1		13240.85		13227.66	
Net output power of system [kW]	3982.92		4130.72		4143.75	
Exergy efficiency of the system	22.9%		23.78%		23.85%	
Refrigerants	R1150, R23, R290		R1150, R23, R290		R1150, R23, R290	

Conclusion

The PDCRC and the SDCRC system were proposed to decrease the exergy loss of the heat exchanger and to improve the exergy efficiency of the system. The proposed systems are based on the concept of primary and secondary distributaries. The two schemes were compared with the existing three-level Rankine cycle. The optimal combination of refrigerants and parameters that matched the two types of the distributary cascade three-level Rankine cycle system were obtained. The specific conclusions are as follows.

- The net output power of the primary distributary cascade three-level Rankine cycle reached the maximum value 4130.72 kW under dryness, the refrigerant combination of R1150, R23 and R290, and the ratio of 0.62 for the second level Rankine that circulates the refrigerant from the turbine to the refrigerant evaporator of the first level of the Rankine cycle. In addition, under different combinations of refrigerants and ratios, the net output power of the system decreased with the increase in the ratio.
- Compared with the original cascade three-level Rankine cycle, the net output power and exergy efficiency of the primary distributary cascade three-level Rankine cycle increased by 3.71% and 3.84%, respectively.

- The net output power of the secondary distributary cascade three-level Rankine cycle reached the maximum value of 4143.75 kW under dryness, the refrigerant combination of R1150, R23 and R290, the export temperature of –106 °C for the first level Rankine that circulates refrigerant in the LNG evaporator of the second level Rankine cycle and the ratio of 0.62 for the second level Rankine that circulates the refrigerant from the turbine to the refrigerant evaporator of the first level of the Rankine cycle.
- The net output power and exergy efficiency of the secondary distributary cascade three-level Rankine cycle increased by 4.04% and 4.15%, respectively, relative to those of the original cascade three-level Rankine cycle.

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