Bulgarian Chemical Communications, Volume 50, Issue 2, (pp. 228 – 236) 2018

Optimization scheme for a typical longitudinal three-level Rankine cycle cold energy power generation system for recycling liquid gas

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Received: October 15, 2017; Revised: January 2, 2018

This study investigates an intermediate fluid vaporizer in gasification systems for liquefied natural gas floating storage regasification units. A heated longitudinal three-level Rankine cycle system that uses the cold energy of liquefied natural gas to generate power was optimized. The system was then compared to the original longitudinal three-level Rankine cycle system established under the same conditions. Results showed that under a liquefied natural gas flow of 175 t/h, the net power output and exergy efficiency of the new system increase by 10.3% and 15.3%, respectively.

Keywords: Liquefied natural gas, Three-level Rankine cycle, Power generation, Heated

INTRODUCTION

Gasification of liquefied natural gas (LNG) before use releases a large amount of cold energy [1]. With the increase in the production and consumption of LNG for LNG cold energy utilization, research has focused on LNG and its application in power generation, air separation, liquefied carbon dioxide and carbon dioxide production, seawater desalination, refrigeration, and low-temperature culturing [2]. LNG cold energy generation is the most important measure to take full advantage of high-grade LNG cold energy [3]. Other methods of cold energy power generation include direct expansion, secondary media, joint, mixed media, and Brayton cycle and gas turbine utilization methods [4]. In the area of LNG cold energy power generation, the focus has shifted to low-temperature organic fluids composed of multi-stage Rankine cycles for the maximization of the application of LNG cold energy [5].

Li et al. [6] proposed cascade power utilization of solar energy and an LNG organic Rankine cycle system, which involves two types of working fluids: the working fluid in which hot water heated by solar energy is gasified first and the second working fluid in which the refrigerant is gasified after LNG becomes operable. The system combines low-temperature Rankine cycle power generation and the direct expansion method. Sun et al. [7] also established a Rankine cycle cold energy power generation system involving two methods, namely, the vaporization of Rankine cycle working fluid by solar heated water as the heat source and the direct expansion method. The system increases the temperature of the refrigerant entering the turbine and improves the net output of the system. However, solar heating is affected by the weather, and many auxiliary equipments are required to provide sufficiently hot water when the system handles large amounts of LNG. Wenji Rao[8] used industrial waste heat as the heat source in a Rankine cycle and concluded that with the increase in evaporation pressure, thermal efficiency and power increase; however, industrial waste heat at high temperatures results in large differences in heat transfer temperature and consequently leads to a loss of a significant amount of energy. Given that these systems require volume or heat sources, they should be applied on land only.

Many scholars have studied systems that are applicable to water bodies. Hongchang Yang [9-11] proposed an LNG cold energy utilization segmentation model and established a horizontal and vertical three-level Rankine cycle according to the model. They also put forward an improved optimization scheme for existing problems on the basis of the concept of dual pumping. Chao Zhang [12] used a mixed refrigerant composed of methane, propane and ethane as circulating working fluids. The results showed that the ratio of the mixed working media provides an important contribution to the maximum net output. However, in the actual operation of the proposed system, the optimal mass fraction ratio of the refrigerants is difficult to determine, and the requirements for stable operation are relatively high. Guobiao Cui [13] used an LNG cold energy segmentation model and established five horizontal Rankine cycle power generation systems using LNG cold energy; all levels of the circulating heat originated from seawater, and the system efficiency reached 61%. In practice, systems

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based on a five-level Rankine cycle are complex and thus difficult to apply. Sangick Lee [14] used seawater as the heat source in the first-level Rankine cycle and exhaust gas as the heat source in the second-level Rankine cycle of a horizontal two-level Rankine cycle power generation system; the results indicated that this system is suitable for small-scale power fishing vessels, but the energy loss of the heat exchanger is considerable due to the high exhaust temperature. Junjiang Bao [15] proposed a two-stage condensed Rankine cycle system whose net power output and thermal efficiency are better than those of combined Rankine cycle systems. Li Boyang [16] adopted an LNG carrier and designed a set of systems that uses LNG cold energy and flue gas waste heat after natural gas combustion to generate electricity. This system involves only one level of Rankine cycle, and the working fluid is high-temperature flue gas. The system can save 2.77 million yuan per year. LNG carriers are transported year round; hence, their heat sources (i.e., high-temperature flue gas) can be stably supplied. In addition, the system involves only a quarter of a complete Rankine cycle and thus requires minimal power. These existing studies imply the need to improve turbines before working fluid temperature can benefit the net power output.

The liquefied natural gas floating storage regasification units (LNG-FSRU) system is usually placed offshore and thus requires long-distance land transport of natural gas; its delivery pressure should reach 7 MPa or higher [17], at which point LNG approaches a supercritical state. Power generation utilization using LNG gasification cold energy should be based on the heat source conditions of LNG gasification in land or on-board environments to consider LNG high-grade cold energy generation. Given that the operating environment of LNG-FSRU lacks a stable high-temperature heat source, LNG-FSRU in the supercritical state of LNG gasification in a cold energy power generation program cannot be directly obtained from the existing LNG gasification of cold energy power generation program.

The LNG of an intermediate fluid vaporizer (IFV) system in LNG-FSRU is gasified in a supercritical state. The current work proposes two forms of the heated longitudinal three-level Rankine cycle power generation scheme according to the gasification parameters of LNG-FSRU and using seawater as the only heat source. The two proposed schemes are suitable for the cold energy utilization of LNG-FSRU regasification systems because they increase the temperature of the working fluid in the turbines and thus increase turbine efficiency. This work provides a solution for optimizing LNG-FSRU cold energy generation systems.

Composition of the heated longitudinal three-level Rankine cycle system

The molar composition of LNG is as follows: 95% methane, 3% ethane, and 2% propane. The gasification pressure is 8 MPa, which is supercritical pressure. Only one reference [9] previously proposed a three-level Rankine cycle power generation system that uses LNG cold energy during steaming. The two forms of the heated longitudinal three-level Rankine cycle power generation system proposed in the present study are shown in Figures 1 and 2. These proposed systems improve the inlet refrigerant of turbines 1 and 2 through the heat transfer between the working fluid and seawater temperature.

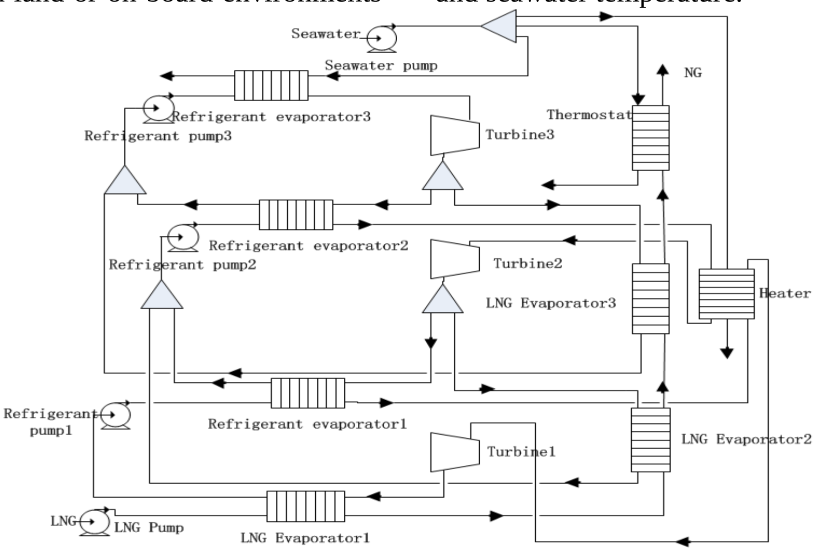

Fig.1. System diagram of the heated longitudinal three-level Rankine cycle (Form 1)

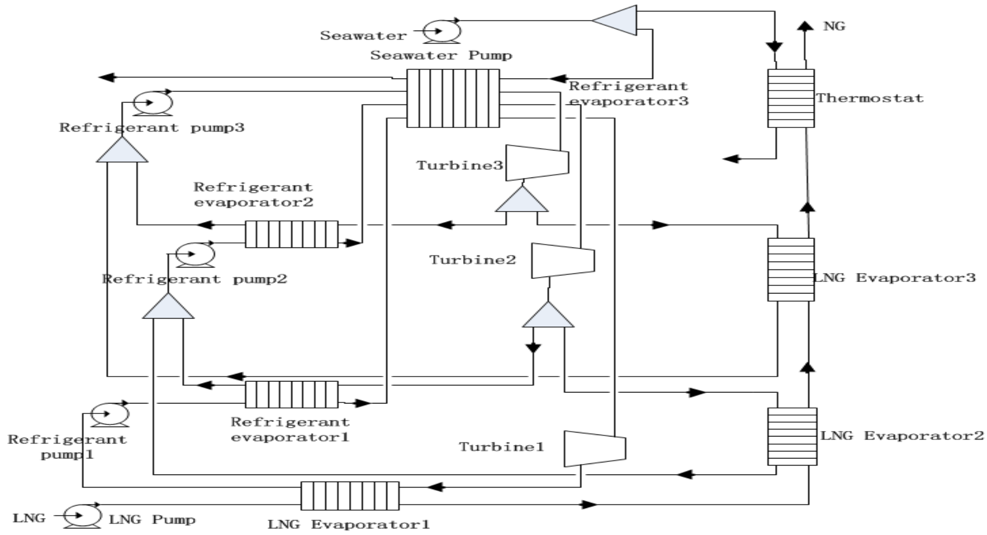

Fig. 2. System diagram of the heated longitudinal three-level Rankine cycle (Form 2)

The difference between Form 1 (Figure 1) and the original longitudinal three-level Rankine cycle is that in Form 1, the refrigerant entering turbines 1 and 2 is introduced into the heater for heat exchange with seawater to increase the temperature of the turbines and facilitate their operation.

The difference between Form 2 (Figure 2) and the original longitudinal three-Rankine cycle is that in Form 2, the refrigerant entering turbines 1 and 2 is introduced into the refrigerant evaporator 3 for heat exchange with seawater to increase the temperature of the turbines and facilitate their operation.

The calculation results showed that excluding that of refrigerant evaporator 3 and the heater, the equipment efficiency, optimum parameter matching, maximum net power output, and overall efficiency of the system are the same. Form 1 was used as an example to determine the optimal combination of refrigerant and parameters matching for heated longitudinal three-level Rankine cycle systems. Then, a thermodynamic comparison of the heated longitudinal three-level Rankine cycle system and the original longitudinal three-level Rankine cycle system was performed. The two forms of the heated longitudinal three-level Rankine cycle system were also compared in terms of energy loss, energy of equipment, and overall energy efficiency.

Determination of the optimal combination of refrigerant and parameter matching

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

- The condensed pressure of the circulating fluid was 110 kPa.
- The temperature of seawater serving as the heat source was 20 °C, and the output temperature of seawater was 15 °C. The ambient temperature was 25 °C.
- The minimum end difference of all heat exchangers was 5 °C.
- In all heat exchangers (except for heat exchangers whose hot fluids are seawater), the hot fluid outlet was subcooled at 2 °C.
- The efficiency of the turbine was 80% and that of the pump was 75%.
- The pressure and heat losses of all heat exchangers and pipes were ignored.
- The refrigerants at the outlet of the refrigerant evaporator were in a saturated gas state.

Optimization of refrigerant combination. In selecting the ideal working fluid, one should consider not only its effects on the net output and safety of the system but also whether its critical temperature matches the temperature of the heat source. The choice of refrigerant plays a key role in the recovery of the LNG cooling capacity of systems [18].

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

Table 1. Condensation temperatures of common refrigerants at 110 kPa

R1150	R170	R23	R116	R1270	R290	R717	R134a	R152a	R600a
-102.64℃	-87.22℃	-80.53℃	-77.20℃	-46.16℃	-40.55℃	-31.44℃	-24.24℃	-22.61℃	-9.93℃

LNG-FSRU systems also need to meet the requirements of daily life, such as fresh water resources and air conditioning. After recovery in three low-temperature Rankine cycles, LNG should have an adequate amount of cold energy to be used by other cold energy processes, such as desalination and cold storage. Therefore, LNG in three Rankine cycles is obtained after the recovery of the cold energy outlet temperature of about −45 °C [19,20]. Given that the minimum end of the heat exchanger in this work was set to 5 °C, the refrigerants R290 and R1270 shown in Table 1 were deemed as the most suitable for the third Rankine cycle. At the inlet of the system, the temperature of the LNG changed from −162 °C to −158 °C after it was pressurized by the pump. When R290 was selected as the refrigerant for the third Rankine cycle, the temperature of LNG cold energy utilization ranged from −158 °C to −45.55 °C. When R1270 was selected as the refrigerant for the third Rankine cycle, the temperature of LNG cold energy utilization ranged from −158 °C to −51.16 °C. The refrigerants R1150, R170, R23, R116, and R1270 meet the temperature range given in Table 1. The first and second Rankine cycles of the working fluid and the corresponding LNG inlet temperature should match as much as possible to reduce the exergy loss of the heat exchanger caused by large temperature differences. Therefore, R1150 and R170 were selected as possible refrigerants for the first Rankine cycle, and R23, R116, and R1270 were selected as possible refrigerants for the second Rankine cycle. When the refrigerant of the second Rankine cycle was R1270, the refrigerant of the third Rankine cycle could only be R290. Thus, 10 possible refrigerant combinations were identified.

In the HYSYS simulation, the net output of the system was calculated given the inlet temperatures of turbines 1 and turbine 2 (hereafter referred to as outlet temperature 1 and outlet temperature 2, respectively). The property package of the refrigerants was based on the Peng–Robinson equation. Under different working group combination schemes, the net power output of the system was hypothesized to reach the maximum point when the outlet temperature of the process was set to the maximum value. When the refrigerant of the second Rankine cycle was R116, refrigerant evaporator 1 of the first Rankine cycle showed a temperature cross. R116 is a dry fluid; thus, the difference between the cold fluid outlet temperature of refrigerant evaporator 1 and the condensing temperature of refrigerant R116 was large. This condition led to temperature crossing in refrigerant evaporator 1. Table 2 shows the ranges of outlet temperature 1 and outlet temperature 2 corresponding to the maximum outlet temperature 1 under the combinations of different refrigerants (without temperature crossing). The temperature interval of outlet temperature 2 is also given.

Results of refrigerant filtering

In HYSYS, a system simulation with different ranges of outlet temperature 1 and outlet temperature 2 was performed under different combinations of refrigerants. The net work output was calculated accordingly. The net output of the system and the dryness of the corresponding refrigerant in the turbine 1 outlet are shown in Figure 3. When outlet temperature 1 was taken as the maximum value, the dryness of turbines 2 and 3 was constant. Thus, only the outlet refrigerant dryness of turbine 1 was considered. Figure 4 shows the outlet refrigerant dryness of turbine 3 when the net power output of the system reaches the maximum under the combination of different refrigerants.

Figures 3 and 4 show that when the refrigerant combinations were R1150, R23, and R290, outlet temperature 1 was −16 °C, and outlet temperature 2 was −41 °C. The system ultimately produced the highest net power output of 4394.090 kW. The dryness of turbine 3 was also high. In the combination of different refrigerants other than R170, R23, R1270 and R170, R23, R290, the system achieved the maximum net work output when the outlet refrigerant dryness of turbines 1 and 2 was equal to 1. When the combinations of refrigerants were R170, R23, R1270 and R170, R23, R290, the outlet refrigerant dryness of system turbines 1 and 2 was less than 1.

Table 2. Ranges of outlet temperature 1 and outlet temperature 2 corresponding to maximum outlet temperature 1

Combinations of refrigerants	Range of the outlet temperature 1	Range of the outlet temperature 2	Temperature interval of the outlet temperature 2
R1150,R23,R290	-45.55℃~-16℃	-41℃~-80.59℃	4℃
R1150,R23,R1270	-51.16℃~-25℃	-41℃~-80.77℃	4℃
R170,R23,R1270	-51.16℃~-25℃	-67℃~-80.77℃	2℃
R170,R23,R290	-45.55℃~-16℃	-67℃~-80.59℃	2℃
R1150,R1270,R290	-45.55℃~-41℃	10.05℃~-46.8℃	8℃
R170,R1270,R290	-45.55℃~-41℃	-9℃~-46.8℃	4℃

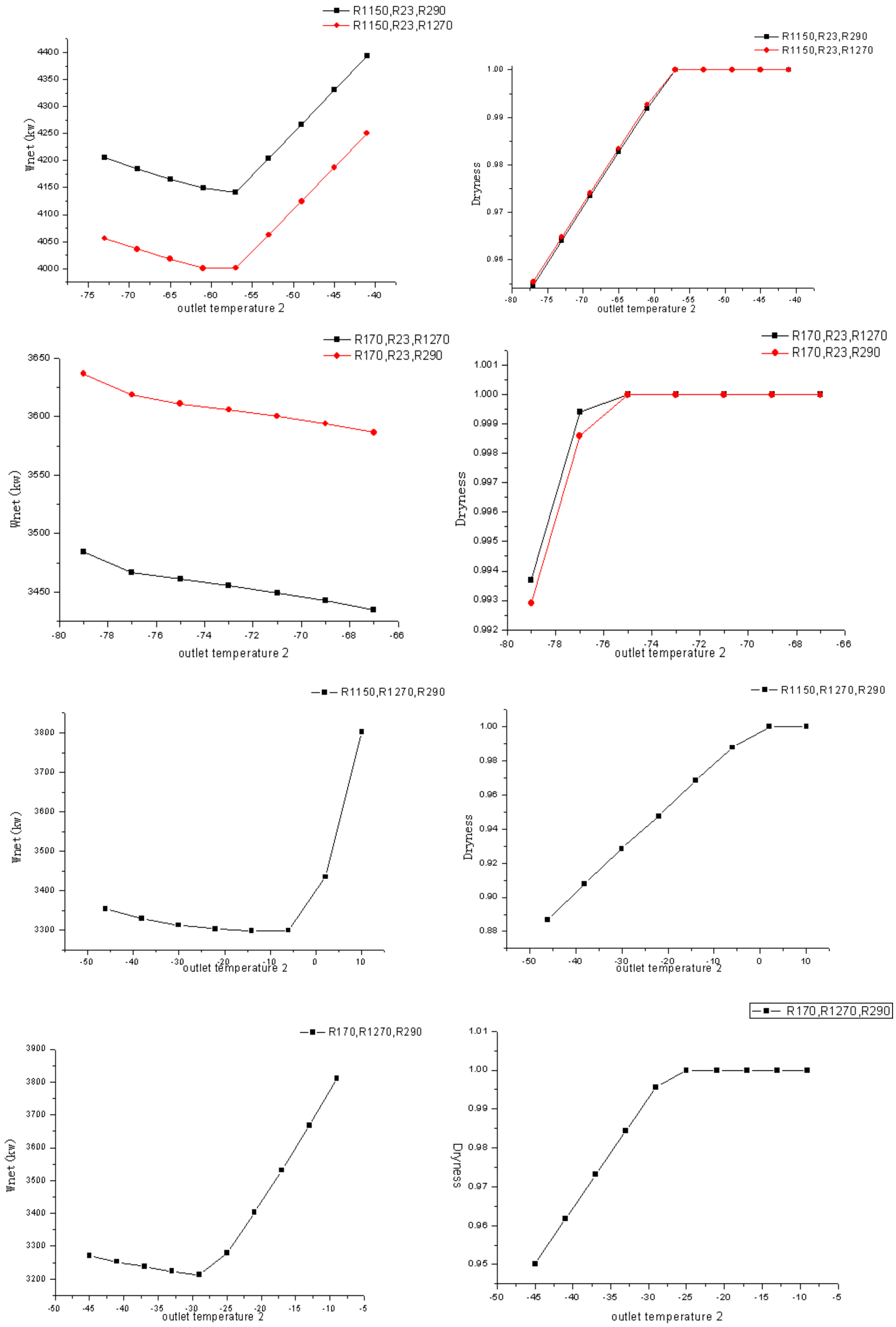

Fig. 3. Network output of the system (left) and dryness of turbine 1 (right) under different combinations of refrigerants

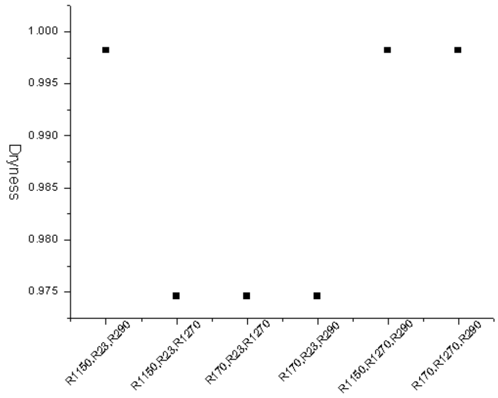

Fig. 4. Dryness of turbine 3 corresponding to the maximum output of the system under different combinations of refrigerants

However, in these two combinations, the system's maximum network output was lower than that under the combination of R1150, R23, R290. Thus, using only the outlet refrigerant dryness of turbine 3 in the comparison does not affect the results.

Figures 3 and 4 show that the optimal working combination was between R1150, R23, R290 and R1150, R23, R1270. The maximum net power output under the two combinations and the combinations of other refrigerants revealed obvious differences. The NG outlet temperature of LNG evaporator 3 was −45.55 °C when the working fluid of the third Rankine cycle was R290. The NG outlet temperature of LNG evaporator 3 was −51.16 °C when the working fluid of the third Rankine cycle was R1270. Therefore, the temperature span of the working combination R1150, R23, R290 was larger than the temperature span of the working combination R1150, R23, R1270. When the two working combinations were used under optimal conditions, the output of the working combination R1150, R23, R290 was large. Therefore, R1150, R23, and R290 were determined to be the best combination of refrigerants for the system.

DETERMINING THE OPTIMUM PARAMETER MATCHING

Figure 3 also shows that when outlet temperature 1 was given and outlet temperature 2 was taken as the maximum value, the net output of the system was maximum under the working combinations of R1150, R23, and R290. The influence of outlet temperature 1 on the net output of the system was also determined. In the calculation, the maximum value of outlet temperature 2 was calculated with a given outlet temperature 1. When the process was established and the working combinations were set to R1150, R23, and R290, the maximum outlet temperature 1 at different outlet temperatures 2 was determined. The results are shown in Table 3. Outlet temperature 1 was divided by the temperature interval of 3 °C. When the combination of refrigerants was R1150, R23, and R290 and the system reached the maximum outlet temperature 2 at different outlet temperature 1, the outlet refrigerant dryness of turbines 1 and 3 remained constant. Thus, only the outlet refrigerant dryness of turbine 2 needed to be considered. The net output of the system and the outlet refrigerant dryness of turbine 2 when the system reached the maximum outlet temperature 2 at different outlet temperatures 1 are shown in Figure 5. Figure 5 shows that when outlet temperature 1 reached the maximum value of −16 ℃, outlet temperature 2 also reached the corresponding temperature range with the maximum value of −41 ℃. At the same time, the system achieved the maximum net output of 4394.090 kW. This result indicates the correctness of the assumptions. When the outlet refrigerant dryness of turbine 2 was less than 1, the net power output of the system increased with the decrease in outlet temperature 1. When the outlet refrigerant dryness of turbine 2 was equal to 1, the net output of the system increased with the increase in outlet temperature 1. This phenomenon is consistent with that shown in Figure 3.

Therefore, when the system refrigerant combination was R1150, R23, and R290, outlet temperature 1 was −16 ℃, and outlet temperature 2 was −41℃. The net work output of the system reached the maximum of 4394.090 kW.

Table 3. Maximum outlet temperature 2 at different outlet temperatures 1 under the combination of refrigerants R1150, R23 and R290

Outlet temperature 1	-43℃	-40℃	-37℃	-34℃	-31℃	-28℃	-25℃	-22℃	-19℃	-16℃
Maximum outlet temperature 2	-50℃	-50℃	-50℃	-50℃	-50℃	-50℃	-50℃	-50℃	-46℃	-41℃

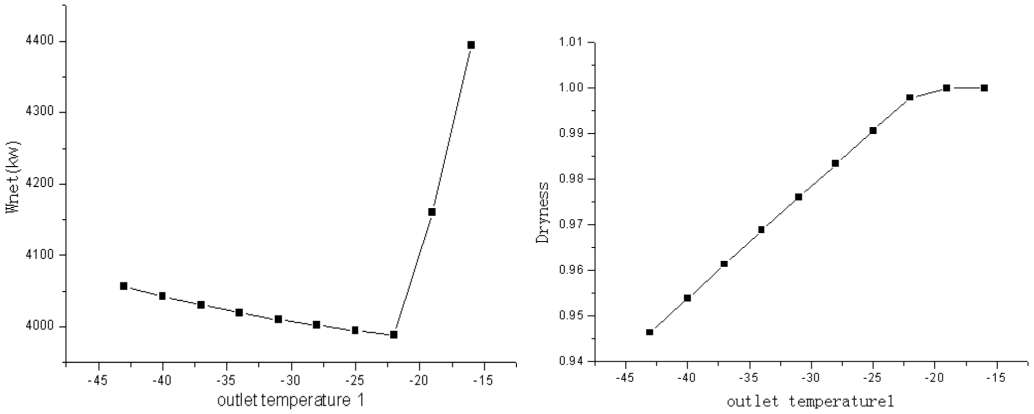

Fig. 5. Net output of the system and outlet refrigerant dryness of turbine 2

Table 4. Comparison of exergy loss and net power output of the heated longitudinal three-level Rankine cycle and the original longitudinal three-level Rankine cycle

Program	Original longitudinal three-level Rankine cycle		Heated longitudinal three-level Rankine cycle	
Equipment	Exergy loss (kW)	Exergy efficiency	Exergy loss (kW)	Exergy efficiency
LNG evaporator 1	3266.63	65.9%	3203.43	72.9%
LNG evaporator 2	624.70	79.7%	116.19	92.2%
LNG evaporator 3	1845.55	65.6%	1513.29	67.9%
LNG thermolator	1649.63	13.1%	1649.63	13.1%
Refrigerant evaporator 1	435.94	91.9%	130.91	97.9%
Refrigerant evaporator 2	750.75	85.1%	724.29	85.1%
Refrigerant evaporator 3	1296.09	38.9%	1216.76	38.9%
Refrigerant pump 1	3.69	45.0%	6.67	45.0%
Refrigerant pump 2	9.95	61.5%	9.60	61.5%
Refrigerant pump 3	36.17	61.4%	33.96	61.4%
LNG pump	967.69	11.6%	967.69	11.6%
Seawater pump	168.86	88.8%	170.59	88.8%
Turbine 1	277.64	69.6%	492.30	70.8%
Turbine 2	666.82	72.1%	715.27	72.2%
Turbine 3	1396.99	75.7%	1311.48	75.7%
Heater			699.30	82.6%
Exergy loss of the system (kW)	13397.1		12262.06	
Net output power of system (kW)	3982.92		4394.090	
Exergy efficiency of the system	22.9%		26.4%	
Refrigerants	R1150, R23, R290		R1150, R23, R290	

Thermodynamic analysis and comparison of heated longitudinal three-level Rankine cycle power generation systems

The definitions of exergy loss and exergy efficiency of the equipment and system are similar to those in ref. [15].

The results are shown in Table 4. As shown in Table 4, the heated longitudinal three-level Rankine cycle scheme reduced the exergy loss of the three LNG evaporators and the three refrigerant evaporators in comparison with the original longitudinal three-level Rankine cycle. The increase in the exergy efficiency of LNG evaporator 2 was most obvious. The exergy loss of the three turbines of the heated longitudinal three-level Rankine cycle increased, but the exergy efficiency was not reduced and even slightly increased. In terms of the performance of the entire system, the heated longitudinal three-level Rankine cycle improved by 10.32% relative to the original three-level Rankine cycle.

Table 5. Comparison of exergy loss and exergy efficiency of related equipment in the two forms of the heated longitudinal three-level Rankine cycle systemand exergy efficiency of the systems

Equipment	Heated longitudinal three-level Rankine cycle system（Form 1）		Heated longitudinal three-level Rankine cycle system（Form 2）	
	Exergy loss (kW)	Exergy efficiency	Exergy loss (kW)	Exergy efficiency
Refrigerant evaporator 3	1216.76	38.9%	1916.07	30.3%
Heater	699.30	82.6%		

Moreover, the exergy efficiency of the former increased by 15.3%, and the total exergy loss decreased by 8.5%.

Contrast of the two forms of the heated longitudinal three-level Rankine cycle system

A comparison of the exergy loss and exergy efficiency of related equipment in the two forms of the heated longitudinal three-level Rankine cycle system and the exergy efficiency of each form are shown in Table 5.

The number of heat exchangers in Form 2 was less than that in Form 1. Thus, refrigerant evaporator 3 was a four-stream, complicated heat exchanger. As shown in Table 5, the exergy efficiency of the new heat exchanger was relatively large, and the exergy efficiency of refrigerant evaporator 3 improved even with a large number of heat exchangers in Form 2. In actual processes, the appropriate form is selected according to specific circumstances, such as funds and area.

CONCLUSIONS

To improve the inlet refrigerant temperature of turbines and ultimately enhance turbine performance, this work proposed two forms of the heated longitudinal three-level Rankine cycle power generation system. A thermodynamic exergy analysis of the existing longitudinal three-level Rankine cycle was performed. The optimal refrigerant combination and parameter matching for the heated longitudinal three-level Rankine cycle power generation system were determined. In both forms, the parameter matching, exergy efficiency, exergy loss, and net output of the system were the same. The specific conclusions are as follows:

(1) For the heated longitudinal three-level Rankine cycle, the ideal combination of refrigerants was R1150, R23, and R290. The turbine inlet temperature of the second-level Rankine cycle was −16 ℃, whereas that of the first-level Rankine cycle was −41 ℃. The net power output reached the maximum of 4394.09 kW, and the efficiency was 26.4%.

(2) Compared with those of the original three-level Rankine cycle, the net output and exergy efficiency of the heated longitudinal three-level Rankine cycle increased by 10.3% and 15.3%, respectively, and the total exergy loss decreased by 8.5%.

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ОПТИМИЗАЦИОННА СХЕМА НА ТИПИЧНА НАДЛЪЖНА СИСТЕМА С РАНКИНОВ ЦИКЪЛ НА ТРИ НИВА, ИЗПОЛЗВАЩА СТУДЕНА ЕНЕРГИЯ, ЗА ГЕНЕРИРАНЕ НА ЕНЕРГИЯ ЗА РЕЦИКЛИРАНЕ НА ТЕЧЕН ГАЗ

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Постъпила на 15 октомври, 2017 г.; коригирана на 2 януари, 2018 г.

(Резюме)

В настоящата статия се изследва междинен течен изпарител за газификационни системи за подвижни регазификационни устройства за съхраняване на втечнен природен газ. Оптимизирана е нагреваема система, базирана на надлъжен Ранкинов цикъл на три нива, използваща студената енергия на втечнен природен газ за производство на енергия. Системата е сравнена с оригиналната надлъжна система с Ранкинов цикъл на три нива при същите условия. Установено е, че при поток на втечнения газ от 175 t/h, нетната изходна мощност и ефективността на ексергията на новата система нарастват съответно с 10.3% и 15.3%.