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Thermodynamic Performance Analysis of an Improved Two-Stage Organic Rankine Cycle

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Received: 17 September 2018; Accepted: 16 October 2018; Published: 23 October 2018

Abstract: In order to improve the two-stage organic Rankine cycle of two heat exchanges of exhaust gas, a two-stage organic Rankine cycle with a regenerator is proposed. Toluene, benzene, cyclohexane and R245fa were selected as the working fluids of the cycle. The thermal efficiency, exergy efficiency and net output power of the cycle were selected as the objective function of the system. The influence of the regenerative performance on the thermodynamic performance of the system was analyzed. The influence of the temperature change of the primary heat exchange outlet on the thermodynamic performance of the system is discussed. The research shows that the regenerator can increase the net power and thermal efficiency of the cycle output. For the selected working fluid, as the efficiency of the regenerator increases, the thermal efficiency of the cycle and the net output power increase. When the primary heat exchange outlet temperature of the exhaust gas increases, the net output power and the exergy efficiency of the cycle increase. For the selected working fluid, when the exhaust heat exchange outlet temperature was increased from 410 K to 490 K, the net output power of the cycle increased up to 10.76 kW, and the exergy efficiency increased up to 7.85%.

Keywords: organic Rankine cycle; exergy; thermodynamics process; diesel exhaust gas

1. Introduction

Diesel engines are widely used in transportation vehicles, industrial and agricultural machines and small power units [1]. However, around 50% of the fuel energy content is dissipated as waste heat [2]. Many researchers [3–7] believe that waste heat recovery is the most potential method to improve the thermal efficiency of diesel engines.

The organic Rankine cycle (ORC) has been proven to be the most promising technology for recovering diesel engine waste heat [7–11]. Many scholars have conducted in-depth research on the waste heat of diesel exhaust gas recovered by ORC, including simple systems [12,13], systems with preheating [14,15], and dual loop ORC systems. SHU et al. [16] proposed a dual loop organic Rankine cycle for recovering residual heat from internal combustion engines, and added reheaters to increase the efficiency of the cycle in high temperature and low temperature cycles, respectively. Yu et al. [17] presented a simulation model based on an actual organic Rankine cycle (ORC) bottoming system of a diesel engine, and proved that the thermal efficiency of a diesel engine can be improved up to 6.1%. Wang et al. [18] analyzed the static and dynamic properties of waste heat from exhaust gas recovered using a two-stage organic Rankine cycle, under five typical internal combustion engine conditions. Wang et al. [19] proposed a two-stage organic Rankine cycle system to absorb the energy of gasoline engine exhaust and cooling water, and the results showed that the net power of the low temperature cycle is higher than that of the high temperature cycle. Chen et al. [20] proposed a confluent cascade cycle-expansion ORC (CCE-ORC) system and proved that this cycle had the advantages of simple structure, small volume and high thermal efficiency compared with the traditional two-stage ORC. Their results showed that the engine peak thermal efficiency can be improved from 45.3% to 49.5% and the CCE-ORC system can generate 8% more net power compared with conventional two-stage ORC. Yang et al. [13] designed a dual loop organic Rankine cycle to recover high-temperature exhaust gas, engine cooling water, and residual heat from the turbocharger. At the engine rated condition, the dual loop ORC system achieved the largest net power output at 27.85 kW when the engine power was 247 kW. Yao et al. [21] designed a two-stage ORC to recover waste heat from a heavy-duty compressed natural gas engine (CNGE), and the results showed that the maximum power output increase ratio and the maximum brake specific fuel consumption improvement ratio were 33.73% and 25% compared with the original CNG engine. Huang et al. [22] proposed a novel two-stage organic Rankine cycle, using a high temperature cycle to exchange heat with the exhaust gas, and using the low temperature cycle for secondary heat exchange of the exhaust gas. They proved that the thermodynamic performance of this new cycle was superior to the traditional two-stage organic Rankine cycle. Li et al. [23] conducted a comprehensive analysis of the thermodynamic and economic performance of the organic Rankine cycle with a regenerator. The results show that the comprehensive economics of the ORC with a regenerator is better than the basic ORC when the heat source temperature is relatively high.

In this paper, the two-stage organic Rankine cycle of the two heat exchanges of the exhaust gas were improved. According to the characteristics of high temperature of the expander outlet, an organic Rankine cycle with a regenerator was proposed. The influence of regenerator efficiency on the thermal performance of the cycle was analyzed. At the same time, the influence of the primary heat exchange outlet temperature of exhaust gas on the thermal performance of the cycle was analyzed.

2. System Model

The schematic diagram of the two-stage organic Rankine cycle by adding a regenerator is shown in Figure 1. The cycle consists of a high temperature cycle and a low temperature cycle. The high temperature cycle is used to absorb the exhaust heat for the first time, and the low temperature cycle absorbs the exhaust heat for the second time. Because the outlet temperature of the high temperature circulating expander is very high, a regenerator is set at the outlet of the expander to further absorb heat.

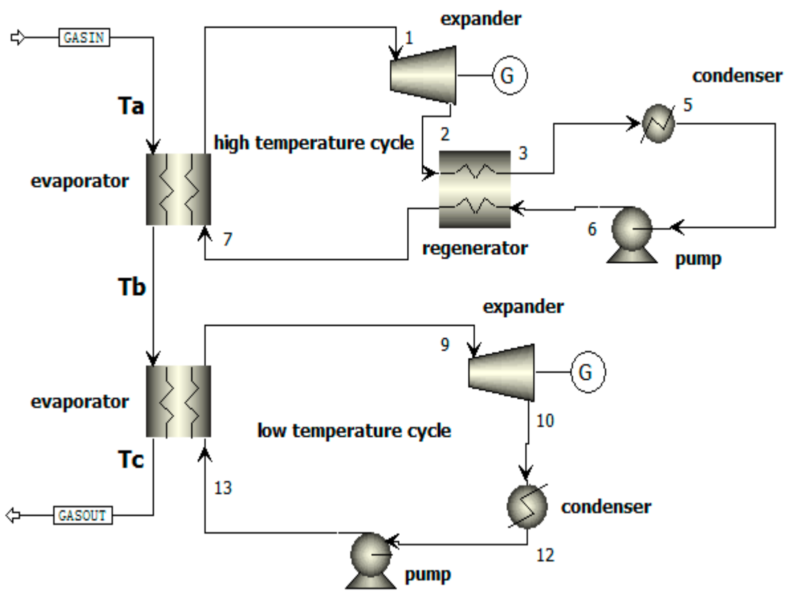

Figure 1. Schematic diagram of the dual loop ORC system.

The object of recovering waste heat in this paper is Cummins' six-cylinder in-line heavy-duty diesel engine, and the specific parameters are shown in Table 1. According to the temperature of the heat source, the most suitable working fluid for the cycle is selected. Because the temperature of the exhaust gas is high, ordinary refrigerant is easily decomposed by heat, and it is not suitable for use in the research process. Therefore, in the research process, toluene, benzene and cyclohexane were selected as the working fluids for the high temperature cycle. After preliminary calculation, according to the range of low temperature circulating heat source temperature, R245fa was selected as the working fluid for the low temperature cycle. The composition of the diesel exhaust and the ratio of each component are shown in Table 2 [24]. The thermal properties of the high and low temperature circulating working fluids are shown in Table 3.

Table 1. Main technical parameters of the diesel engine.

Parameter	Value
Displacement	13 L
Maximum torque	2500 N·m
Exhaust gas mass flow	0.75 kg/s
Rated power//Rotation speed	412 kW/2100 rpm
Exhaust gas temperature	653 K

Table 2. The composition of the exhaust gas.

Composition	Molecular Weight (g/mol)	Fraction
O_2	32.00	0.1483
CO_2	44.00	0.0436
N_2	18.01	0.0620
H_2O	28.01	0.7461

Table 3. Properties of the working fluids.

Working Fluid	T_{cr} (K)	P_{cr} (MPa)	Molecular Weight (g/mol)	GWP	ODP
toluene	591.75	4.126	92.138	Very low	0
benzene	562.02	4.906	78.112	Very low	0
cyclohexane	553.64	4.075	84.161	Very low	0
R245fa	427.16	3.651	134.05	950	0

3. Methods

3.1. Thermodynamic Model

Figure 2 is the T-s diagram of a two-stage cycle with a regenerator. Among them, (a) is the T-s diagram of the high temperature cycle, and (b) is the T-s diagram of the low temperature cycle. The thermodynamic model is established for the cycle by the first and second laws of thermodynamics.

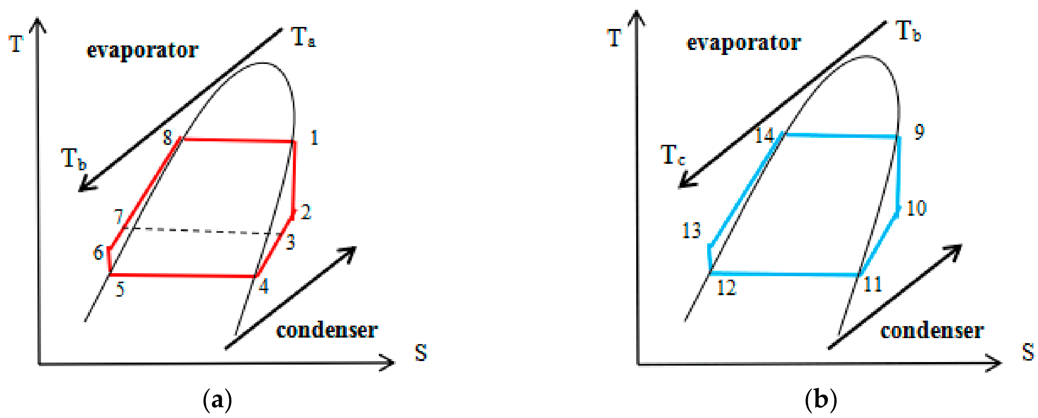

Figure 2. T-s diagram of the dual loop ORC system with regenerator. (**a**) High temperature cycle T-s diagram; (**b**) Low temperature cycle T-s diagram.

The net out power and exergy loss of the high temperature cycle expander are:

$$W_{th} = \dot{m}_h \bullet (h_1 - h_2), \quad (1)$$

$$I_{th} = \dot{m}_h \bullet [h_1 - h_2 - T_0(s_1 - s_2)], \quad (2)$$

where T_0 represents the surrounding temperature, which is 20 °C, $\dot{m}_h$ represents the mass flow rate of the high temperature circulating working fluid, h is the enthalpy value of the state point, s is the entropy value of the state point.

The cooling load and exergy loss of the circulating condenser with high temperature are:

$$Q_{ch} = \dot{m}_h \bullet (h_3 - h_5), \quad (3)$$

$$I_{ch} = \dot{m}_h \bullet [h_3 - h_5 - T_0(s_3 - s_5)], \quad (4)$$

The power consumed by the working fluid pump with the high temperature is:

$$W_{ph} = \dot{m}_h \bullet (h_6 - h_5), \quad (5)$$

Approximating the working fluid pump as a reversible adiabatic process, so:

$$I_{ph} = 0, \quad (6)$$

The efficiency of the regenerator can be expressed as:

$$\varepsilon = \frac{T_2 - T_3}{T_2 - T_6}, \quad (7)$$

According to the conservation of energy in the regenerator,

$$h_7 = h_6 + (h_2 - h_3), \quad (8)$$

The exergy loss of the regenerator is:

$$I_{INT} = \dot{m}_h \bullet T_0[(s_7 - s_6) - (s_2 - s_3)], \quad (9)$$

The heat absorption of the working fluid in the high temperature cycle evaporator and the exergy loss are:

$$Q_{eh} = \dot{m}_h \bullet (h_1 - h_7), \quad (10)$$

$$I_{eh} = E_{xa} - E_{xb} - \dot{m}_h \bullet [h_1 - h_7 - T_0(s_1 - s_7)], \quad (11)$$

where E_x represents the exergy value of the state point.

The net out power and exergy loss of the low temperature cycle expander are:

$$W_{tl} = \dot{m}_l \bullet (h_9 - h_{10}), \quad (12)$$

$$I_{tl} = \dot{m}_l \bullet [(h_9 - h_{10}) - T_0(s_9 - s_{10})], \quad (13)$$

The cooling load and exergy loss of the low temperature cycle condenser are:

$$Q_{cl} = \dot{m}_l \bullet (h_{10} - h_{12}), \quad (14)$$

$$I_{cl} = \dot{m}_l \bullet [h_{10} - h_{12} - T_0(s_{10} - s_{12})], \quad (15)$$

The power consumption of the low temperature cycle working fluid pump is:

$$W_{pl} = \overset{\bullet}{m}_l \bullet (h_{13} - h_{12}), \tag{16}$$

Approximating the working fluid pump as a reversible adiabatic process, so:

$$I_{pl} = 0, \tag{17}$$

The heat absorption of the working fluid in the low temperature cycle evaporator and the exergy loss are:

$$Q_{el} = \overset{\bullet}{m}_l \bullet (h_9 - h_{13}), \tag{18}$$

$$I_{el} = E_{xb} - E_{xc} - \overset{\bullet}{m}_l \bullet [h_9 - h_{13} - T_0(s_9 - s_{13})], \tag{19}$$

Total heat absorption of the evaporator:

$$Q_{tot} = Q_{eh} + Q_{el}, \tag{20}$$

The total net power output is:

$$W_{net} = W_{th} + W_{tl} - W_{ph} - W_{pl}, \tag{21}$$

The thermal efficiency of the cycle is:

$$\eta = \frac{W_{\text{net}}}{Q_{el} + Q_{eh}}, \tag{22}$$

The exergy efficiency of the cycle is:

$$\eta_{ex} = 1 - \frac{I_{th} + I_{ch} + I_{INT} + I_{eh} + I_{tl} + I_{el} + I_{cl}}{E_{xa}}, \tag{23}$$

3.2. Calculation Conditions

In the calculation process of the mathematical model, the calculation conditions are set as follows:

1. Assume that the system is stable, ignore pressure loss and heat loss in the pipeline [25];
2. According to the output power and the working pressure of the expander, setting the isentropic efficiency of the high and low temperature cycle expander to 0.8, and the isentropic efficiency of the high and low temperature cycle pump is 0.85 [25–28];
3. Both high and low temperature cycles are condensed at environment pressure (0.1 MPa);
4. Select the appropriate evaporation pressure according to the thermal properties of the working fluid. The high temperature cycle evaporation pressure is set to 2.5 MPa, and the low temperature cycle evaporation pressure is set to 1.6 MPa. The thermal properties of all working fluids are calculated by REFPROP 9.0.

4. Results

4.1. Effect of Regenerator Efficiency on Thermal Performance of Cycle

The pinch point temperature difference of the high temperature cycle evaporator is set to 30 °C, and the pinch point temperature difference of the low temperature cycle evaporator is 10 °C. Figures 3 and 4 show the relationship between the efficiency of the regenerator and the heat absorption of the high and low temperature cycle evaporators. As the efficiency of the regenerator increases, the heat absorption of the high temperature cycle evaporator decreases, and the heat absorption of the low temperature cycle evaporator increases. The reason is that the efficiency of the regenerator increases, and the temperature of the working fluid has increased before entering the high-temperature evaporator, thereby causing a decrease in the heat absorption of the working fluid in the high-temperature evaporator. As shown in Figure 5, when the regenerator efficiency increases, the high temperature evaporator exhaust gas outlet temperature Tb increases. When benzene is used as the high temperature working fluid, the T_b increases from 408 K to 429.2 K when the regenerator efficiency changes from 0 (without the regenerator) to 1; when the toluene is used as the high temperature working fluid, the T_b increases from 413.5 K to 455.5 K; when cyclohexane is used as a high temperature working fluid, the T_b increases from 421.9 to 465.6 K. Because the temperature of the primary heat exchange outlet of the exhaust gas increases, that is, the temperature of the exhaust gas entering the low temperature circulating evaporator increases, thereby causing an increase in the heat absorption of the working fluid in the low temperature circulating evaporator.

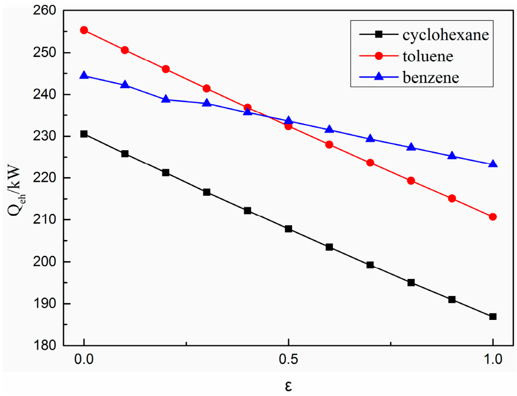

Figure 3. Effect of regenerator efficiency on heat absorption of high temperature cycle evaporator.

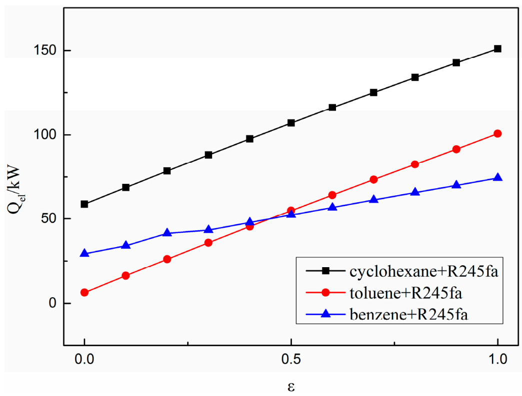

Figure 4. Effect of regenerator efficiency on heat absorption of low temperature cycle evaporator.

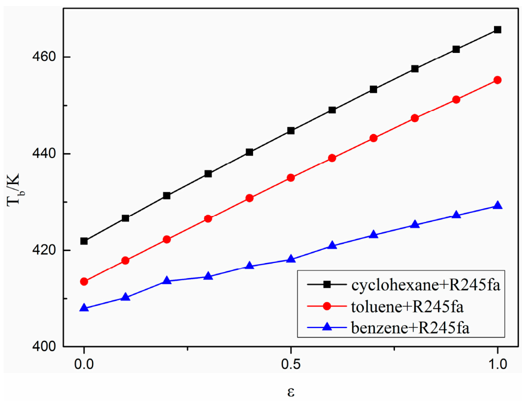

Figure 5. Effect of regenerator efficiency on the primary heat exchange outlet temperature of exhaust gas.

As shown in Figure 6, when the efficiency of the regenerator increases, the net output power of the cycle also increases. As the T_b increases, the heat absorption of the working fluid in the low temperature evaporator increases, and the required low temperature working fluid mass flow increases, resulting in an increase in the net power of the low temperature circulating output, and finally the total net power of the cycle output is increased. With toluene as a high temperature cycle working fluid, when the efficiency of the regenerator increases from 0 to 1, the net output power of the cycle increases from 42.8 kW to 57.16 kW. When benzene is used as the working fluid for the high temperature cycle, the net output power of the cycle increases from 48.66 kW to 55.49 kW. With cyclohexane as the working fluid for the high temperature cycle, the net output power of the cycle increased from 46.96 kW to 60.99 kW. Figure 7 shows the effect of regenerator efficiency on the cycle thermal efficiency. With toluene used as a high temperature cycle working fluid, when the regenerator efficiency increases from 0 to 1, the cycle thermal efficiency increases from 16.36% to 18.35%. When benzene is used as a high temperature cycle working fluid, the cycle thermal efficiency increases from 17.77% to 18.66%. When the high temperature cycle working fluid is cyclohexane, the cycle thermal efficiency increases from 16.32% to 18.05%. Therefore, it can be seen from Figures 6 and 7 that increasing the efficiency of the regenerator can increase the net output power and thermal efficiency of the cycle.

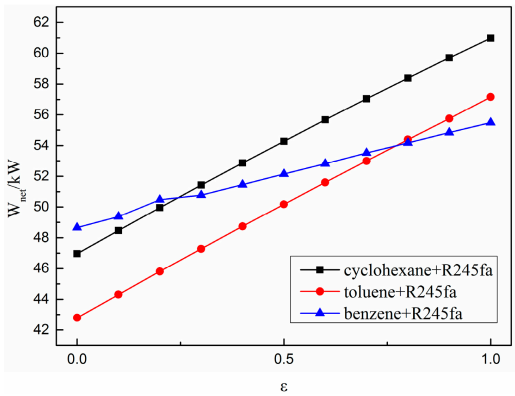

Figure 6. The effect of regenerator efficiency on the net output power of the cycle.

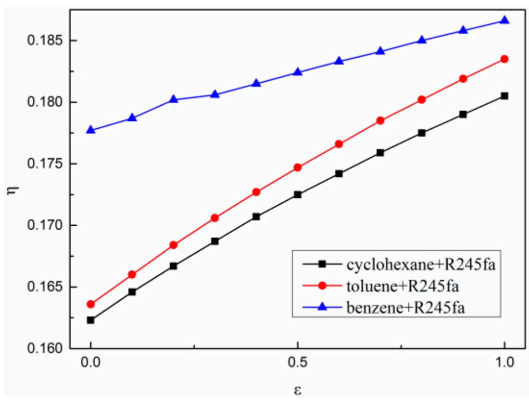

Figure 7. The effect of regenerator efficiency on the thermal efficiency of the cycle.

4.2. Effect of T_b on Cycle Thermal Performance

The primary heat exchange outlet temperature of the exhaust gas (T_b) has an important influence on the thermal performance of both the high and the low temperature cycles, so it's necessary to analyze the impact of T_b. During the study, the pinch point temperature difference of the low temperature cycle evaporator was set to 10 °C.

As shown in Figure 8, as the T_b increases, the total heat absorption Q_{tot} of the high and low temperature circulating evaporators increases. Because of the increase of T_b, the heat absorption of the high temperature cycle evaporator has a certain amount of decrease, while the heat absorption of the low temperature circulating evaporator increases, and the total heat absorption increases. Figures 9 and 10 show the effect of T_b on the net output power and the exergy efficiency of the cycle. As shown in Figures 9 and 10, toluene is used as a high temperature circulating working fluid, when the T_b increases from 410 K to 490 K, the net circulating power increases from 50.47 kW to 61.23 kW, and the overall exergy efficiency of the cycle increases from 36.82% to 44.67%. When benzene is used as the high temperature cycle working fluid, the net output power of the cycle increases from 51.83 kW to 62.14 kW, and the overall exergy efficiency of the cycle increases from 37.81% to 45.33%. When cyclohexane is used as the working fluid for high temperature cycle, the overall exergy efficiency of the cycle increased from 37.48% to 45.12%.

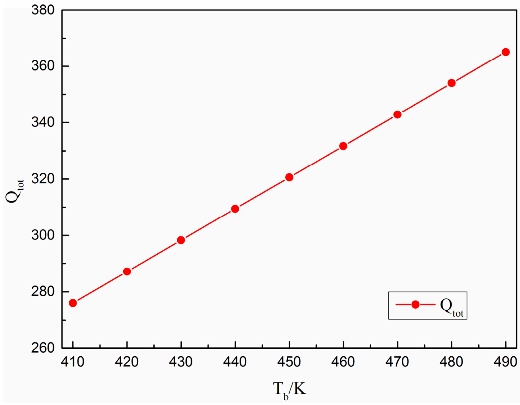

Figure 8. Effect of T_b on total heat absorption of high and low temperature cycle evaporators.

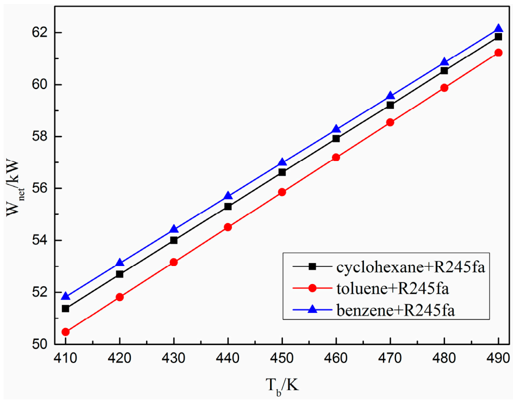

Figure 9. Effect of T_b on the net output power of the cycle.

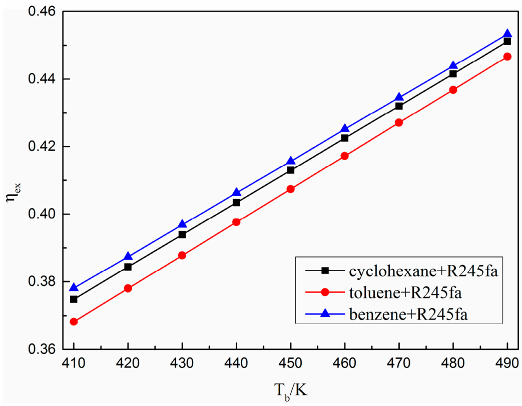

Figure 10. Effect of T_b on the exergy efficiency.

As can be seen from Figure 11, increasing T_b can reduce the cooling load of the high temperature cycle. As shown in Figure 11, with toluene as the working fluid, when the T_b changes from 410 K to 490 K, the cooling load of the high temperature cycle is reduced from 185 kW to 124.1 kW. When benzene is used as the working fluid, the cooling load of the high temperature cycle is reduced from 199.3 kW to 133.7 kW. When cyclohexane is used as the working fluid, the cooling load of the high temperature cycle is reduced from 221.9 kW to 148.9 kW. Because the outlet temperature of the high-temperature cycle expander is relatively high, the high-temperature waste heat at the outlet of the expander is utilized by setting a regenerator, and the cooling load of the high-temperature cycle condenser is reduced.

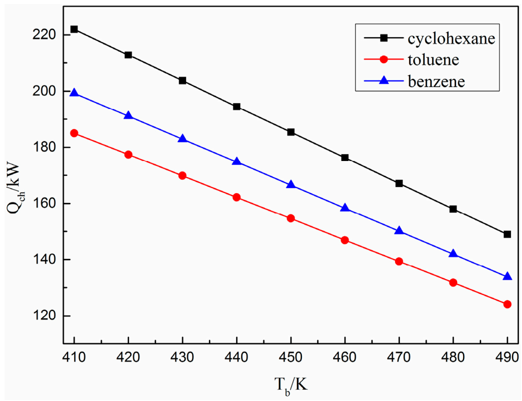

Figure 11. Effect of T_b on cycle cooling load.

5. Conclusions

In this paper, a two-stage organic Rankine cycle with a regenerator is designed to realize the cascade utilization of exhaust heat. The influence of regenerative heat on the thermal performance of the cycle was analyzed. At the same time, the influence of the primary heat exchange outlet temperature of exhaust gas on the thermal performance of the cycle was analyzed. The major conclusions are listed as follows:

- Setting the regenerator can increase the net output power and thermal efficiency of the cycle. For the selected working fluid, when the regenerator efficiency increases from 0 to 1, the net output power of the cycle can be increased up to 14.26 kW, and the thermal efficiency can be increased up to 1.99%.
- When the primary heat exchange outlet temperature of the exhaust gas increases, the net output power and the exergy efficiency of the cycle increase. For the selected working fluid, when T_b is increased from 410 K to 490 K, the net output power of the cycle can be increased up to 10.76 kW, and the exergy efficiency can be increased up to 7.85%.
- The efficiency of the regenerator affects the primary heat exchange outlet temperature of the exhaust gas. When the efficiency of the regenerator increases, the primary heat exchange outlet temperature of the exhaust gas also increases.

Author Contributions: X.L. Mathematical modeling and modification of manuscript; T.L. Analysis, writing of article and reply to comments; L.C. Organized literature and data.

Funding: The authors wish to acknowledge the financial support of the Natural Science Foundation of Tianjin (No. 16JCZDJC31400).

Acknowledgments: Thanks to Cao Gu from China North Vehicle Research Institute for giving us diesel engine related data.

Conflicts of Interest: The authors declare no conflict of interest.

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