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# Development and Performance Analysis of A Multi-Temperature Quasi-Cascade Refrigerating System

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## ABSTRACT

Low boiling point of gasoline always makes it evaporate into gas status during storage, transportation and charging. The direct emission of great deal of gasoline gas to atmosphere not only means a great deal of energy waste but also causes environmental pollutions. The situation is same in case of low boiling point components of rock oil or other low boiling point materials such as alcohol. There thus appears the requirement for the recycling of such gas. A multi-temperature quasi-cascade refrigerating system was developed for this purpose by the way of condensation.
A multi-temperature quasi-cascade refrigerating system, its working principle, structure, flow process and so on are introduced. The influences of some design parameters, such as evaporating temperature, condensing temperature, selection of refrigerants, the temperature differences, on the performances of the system are also presented. The losses of the system are analyzed in the end of this paper.

*Key Words*: multi-temperature, quasi-cascade, refrigerating system, gasoline gas

## 1. INTRODUCTION

According to the monthly oil market report issued by IEA in March 2010, the worldwide oil demand will grow by 1.8%, and will reach to 86,6mb/d, which is 70kb/d more than the anticipation in February.
In the other hand, huge oil demand also means huge gasoline waste. The gasoline waste causes the decrease in the amount and in the quality for various reasons, and causes a loss of economic value, during the entire process including production, discharge, transportation, loading, storage and sale. The gasoline waste can be divided into three main categories: the leakage, the mix and the evaporation (Zhao.1999). The volatility characteristic of oil and its products causes a serious evaporation waste during the exploitation, processing, which makes a great deal of energy waste, environment pollutions and the danger of fire and so on. According to the existing data, the evaporation waste occupies a considerable proportion in total gasoline waste (Chen.1999). Although there is not an exact data, many statistical materials show that the gasoline evaporation rate is between 2% and 3% in various countries (Yan.1999).
At present, there are three gasoline gas reclaiming programs: the gasoline gas pipe system program, the special apparatus program, and the gasoline gas pipe system and special apparatus combining program. The condensation is the special apparatus programs in the field of gasoline gas reclaiming programs. The special apparatus program is such a program that gasoline gas is transported to a special apparatus through pipes to be reclaimed. It requires the special apparatuses and the cost input including purchase, management, maintenance, operation, and energy consumption (Chang and Shu.2006).
Figure1shows the principle of condensation-type reclaiming apparatus. Different component of the gasoline gas has different saturated temperature. In the way of mechanical refrigeration, the gas is continuously cooled and its components are condensed into liquid status at different temperatures in different stages. (Feng 2008).

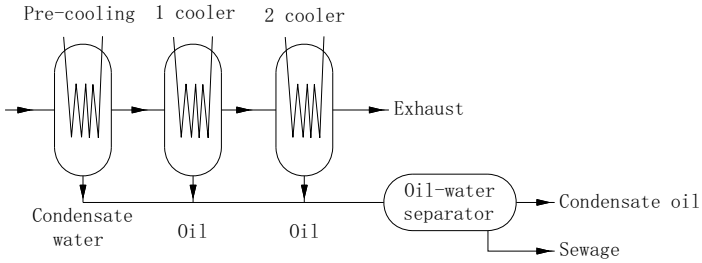


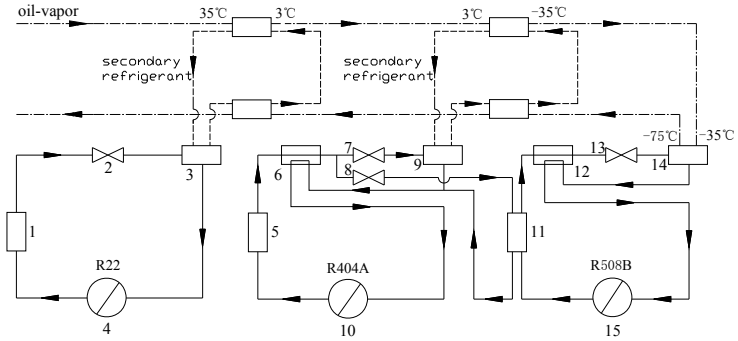

Figure 1: principle of condensation-type reclaiming apparatus

In the condensation-type gasoline reclaiming system, the condensing temperatures are usually determined according to the recovery coefficient and the organic content of exhaust emissions. In general, the temperature is from -40℃ to -110℃. To get such a low temperature, the dual/multi-stage compression refrigeration cycle or the cascade refrigeration cycle is employed, because it is difficult for the traditional single-stage vapor compression refrigeration cycle to reach such a low evaporation temperature and such a high pressure ratio, it is also difficult for a dual (multi)- stage compression refrigeration cycle, in such a low temperature, to find a proper refrigerant to meet the requirements that the evaporation pressure is not too low and the condensation pressure is not too high. Therefore, the cascade refrigeration cycle is widely used in the gasoline gas reclaiming process (Wang *et al.*2008).

## 2. THE MULTI-TEMPERATURE QUASI-CASCADE REFRIGERATING SYSTEM FOR GASOLINE GAS RECLAIMING

A gasoline gas reclaiming system containing a multi-temperature quasi-cascade refrigerating system was designed as shown in Figure 2. It is a complex system including a gasoline gas condensing subsystem, an energy (refrigeration) recycling subsystem and a secondary refrigerant subsystem. The quasi-cascade refrigerating system contains two refrigeration systems, a high temperature system and a low temperature system. The high evaporating temperature one not only has an evaporator for the second stage of the gas reclaimed process but also has another evaporator used as the condenser of the low evaporating temperature refrigeration system, an evaporative-condenser. These two evaporators provide two different refrigerating temperatures, so this system is called multi-temperature quasi-cascade refrigerating system. In all analysis below, it is assumed that the refrigerant temperatures in the evaporator and in the evaporative-condenser of the high evaporating temperature subsystem are same, and the inlet and the outlet statuses of refrigerant are same, too.

The gasoline gas can be condensed by three steps in this system. In the first step, a high evaporating temperature refrigeration cycle supplies a condensing temperature of 3 ℃ for the condensing of water vapor and pre-cooling (some high molecular hydrocarbons in the gasoline gas might be condensed, too). The multi-temperature quasi-cascade refrigerating system is used for the second and the third step. The cooling temperature supplied by the second step can cool the gasoline gas to -35℃ and make the low molecular hydrocarbons condensed, and the lower molecular hydrocarbons was condensed at -75 ℃, in the third step.




1—Air-cooled condenser；2—Throttle；3—Evaporator；4—Compressor；5—Air-cooled condenser；
6—Regenerator；7—High-temperature part throttle 1；8—High-temperature part throttle 2；
9—Evaporator；10—Compressor；11—Evaporative condensers（Plate heat exchanger）；
12—Regenerator；13—Throttle；14—Evaporator；15—Compressor

Figure 2: A gasoline gas reclaiming system containing a multi-temperature quasi-cascade refrigerating system

## 3. THE EFFECT OF THE DESIGN PARAMETERS ON THE SYSTEM PERFORMANCE

The state parameters in various operating condition gained by thermodynamic calculation software were adopted to analyze the effect on the performance of the multi-temperature quasi-cascade refrigerating system, such as the condensing temperature in the high-temperature part, the evaporating temperature in low-temperature part and the heat-transfer temperature difference of evaporative-condenser. The optimum condensing temperature was calculated according to the equation presented by Chen and Chen (2002), with the electrical efficiency of the two compressors of 0.75.

The overheating temperature degree is 30K and the sub-cooling temperature degree is 12K in the low-temperature part; the overheating temperature degree is 13K and the sub-cooling temperature degree is 12K in the high-temperature part. The heat-transfer temperature difference of evaporative-condenser is 8K (Assuming that $\triangle T$=6，7，8，9，10K , when discussing the effect of the heat-transfer temperature difference of evaporative-condenser on the system performance). The mass flow ratios of refrigerant though the high-temperature and the low-temperature evaporators are $k$=0.75 and $j$=0.2.

### 3.1 The selection of the refrigerant in low-temperature part

As shown in Figure 2, the refrigerant pairs R404A/R508B are adopted respectively in the high-temperature and in the low-temperature part of the multi-temperature quasi-cascade refrigerating system. R404A is a kind of typical moderate temperature refrigerant. There were two choices, R23 and R508B, when we chose the low temperature refrigerant early in the design.

The refrigerant thermal conductivity, viscosity, potential heat of vaporization, specific heat capacity and liquid density directly influence the performances of boiling evaporation and condensation heat transfer. The following figures compare the characteristics between R23 and R508B to determine the evaporating temperature in low-temperature part.

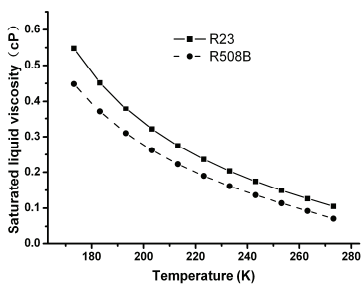



Figure 3: saturated liquid viscosity of R23 and R508B

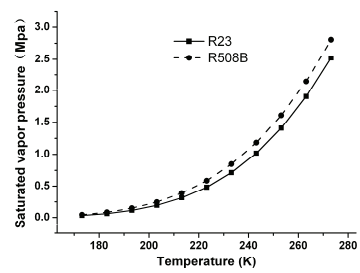



Figure 4: saturated vapor pressures of R23 and R508B

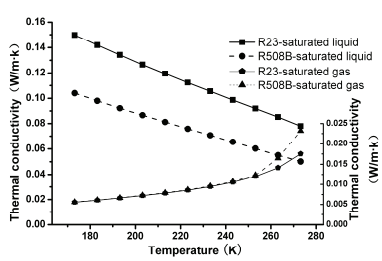



Figure 5: thermal conductivity of R23 and R508B

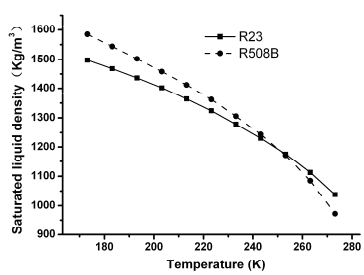



Figure 6: saturated liquid density of R23 and R508B

As shown in above figures, though the most properties of these two refrigerants are similar, the properties of fluidity and liquid density of R508B is better than R23 when used in the low temperature refrigerating. So R508B is more excellent in low temperature than R23.

In addition, the cycle performances of R23 and R508B are shown in Table 1, when the evaporation temperature of was -84.4℃, the condensing temperature was -35℃, the sub-cooling degree was 5.6℃, the suction temperature was

-17.8℃, the clearance volume of the compressor was 4% and the isentropic compression index was 0.7. In Table 1, the cooling capacities and the EERs of R508B and R23 were contrasted, and the result shown that the performance of R508B is better than R23. Otherwise the discharge temperature of R508B was lower than that of R23 in the same operating condition, which leads to the safe operation of the compressor and the benefit of good lubrication and long service life of the compressor (Liu and Hu.2007).

Table 1: The theoretical Performance of Low-temperature refrigerant in refrigerating cycle

| Refrigerant | R13 | R23 | R508B |
|---|---|---|---|
| Cooling capacity（%） | 100 | 104 | 138 |
| EER（%） | 100 | 90 | 98 |
| Discharge pressure（MPa） | 0.717 | 0.848 | 1.013 |
| Suction pressure（MPa） | 0.083 | 0.09 | 0.124 |
| Exhaust temperature（℃） | 92 | 138 | 85 |

In summary, considering the economy and security, R508B was selected as the refrigerant for the low evaporation temperature refrigeration cycle.

### 3.2 The effect of the low-temperature stage evaporation temperature on the performances of multi-temperature quasi-cascade refrigeration cycle

As shown in Figure 8 and Figure 9, the pressure ratios of both the high and low temperature cycles decrease with the increases of evaporation temperatures at the optimal condensing temperature. The pressure ratio of high-temperature cycle decreases more sharply but is always higher than the low-temperature stage one. The pressure ratio of high-temperature stage is from 8 to 16, and the pressure ratio of low-temperature stage is from 4 to 10.

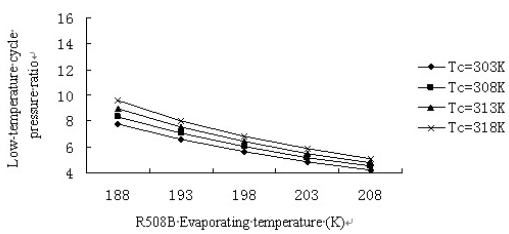



Figure 8: Pressure ratio of low-temperature part

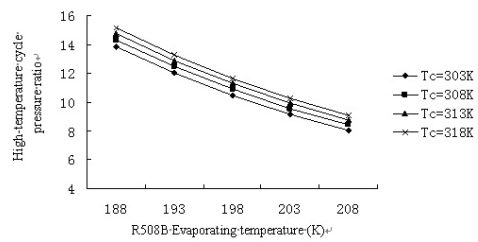



Figure 9: Pressure ratio of high-temperature part

In Figure 10 and Figure 11, it is shown that, with the increasing of the evaporating temperature of low-temperature part, the discharge temperature of the low-temperature stage compressor increases and the discharge temperature of the high-temperature stage compressor decreases. When *Te* is 188K and *Tc* is 318K, the discharge temperature of the high-temperature stage reaches to the highest value of 341.2K.

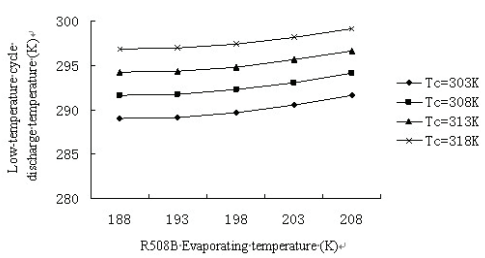



Figure 10: Discharge temperature of low-temperature part

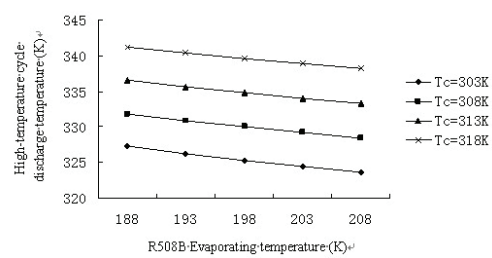



Figure 11: Discharge temperature of high-temperature part

Figure 12 and Figure 13 show that the *COP* varies with the change of evaporating temperature of low-temperature stage, here assuming that *k* is 0.75 and *j* is 0.2. It is observed that *COP* increases obviously with the evaporating

temperature increases. It is also observed form the influences of condensing temperatures that the less the difference between the condensing temperature and the evaporating temperature of low-temperature stage is, the more *COP* is.

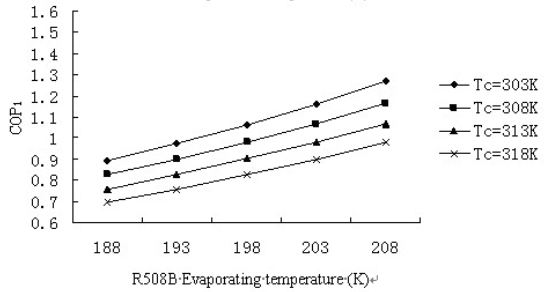



Figure 12: $COP_1$ vs R508B evaporating temperature

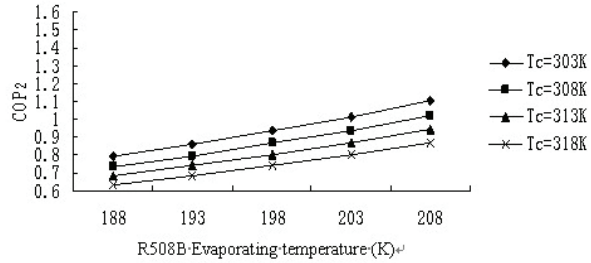



Figure 13: $COP_2$ vs R508B evaporating temperature

### 3.3 The effect of the condensation temperature of high-temperature stage on the performances of the multi-temperature quasi-cascade refrigeration cycle

Figure 14 and Figure 15 present that the pressure ratios of the high and the low-temperature cycles vary with the changes of the condensing temperatures. At their own optimal mid-condensing temperature, the pressure ratios of the high and the low-temperature cycle increase with the increases of the evaporating temperature. The pressure ratios of the low-temperature cycle is between 4 and 8, and the pressure ratios of the high-temperature cycle is between 7 and 14, which is higher than those of the low-temperature cycle.

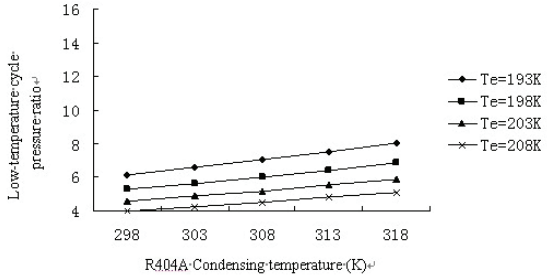



Figure 14: Pressure ratio of low-temperature part

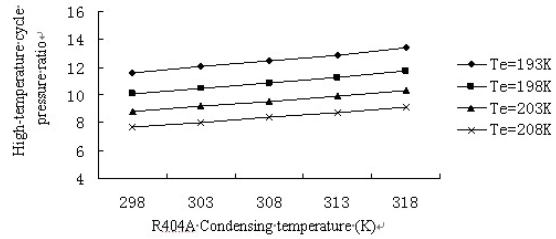



Figure 15: Pressure ratio of high-temperature part

In Figure 16 and Figure 17, it is shown that with the increase of the evaporating temperature, the discharge temperatures of both the high and the low-temperature stage compressors increase when other conditions are same. Especially when *Te* is 193K and *Tc* is 318K, the discharge temperature of the high-temperature stage compressor reach to 356.6K

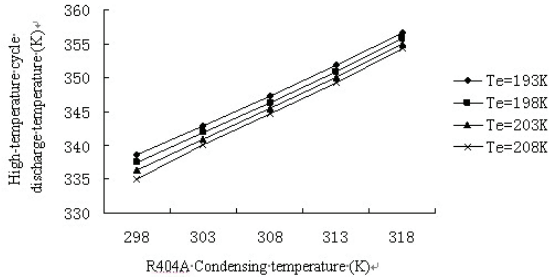



Figure 16: Discharge temperature of high-temperature part

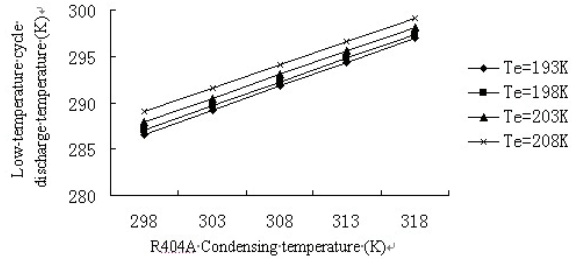



Figure 17: Discharge temperature of low-temperature part

As Figure 18 and Figure 19 shown, it is proved, by comparing with the *COP* values at four different evaporating temperatures, that the *COP* decreases with the increase of the evaporating temperature. And it is also proved that the less the difference between the condensing temperature and the evaporating temperature of the low-temperature part in the multi-temperature quasi-cascade refrigerating system is, the high the *COP* is.

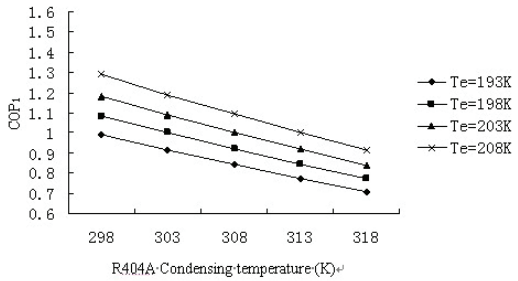



Figure 18: $COP_1$ vs the R404A condensing temperature

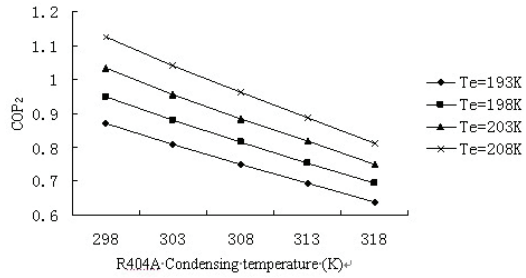



Figure 19: $COP_2$ vs R404A condensing temperature

### 3.4 The effect of the heat transfer temperature difference of the evaporative-condenser on the performances of multi-temperature quasi-cascade refrigeration cycle

The heat transfer temperature difference of the evaporative-condenser also influences the performance of the multi-temperature quasi-cascade refrigeration cycle. Usually the heat transfer temperature difference is between 5 ~10K. In this paper, the parameter values are calculated with the temperature differences of $\triangle T$=6，7，8，9，10K and keeping the other conditions same as above. The results are shown in Figure 20~25.

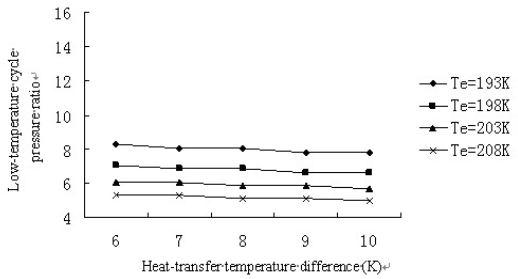



Figure 20: Pressure ratio of the low-temperature part

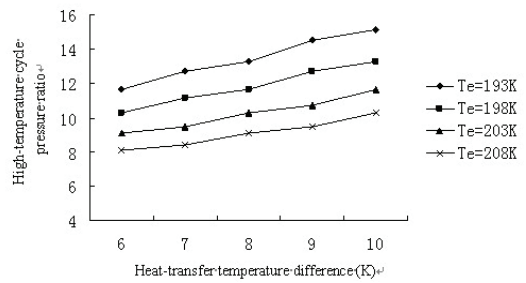



Figure 21: Pressure ratio of the high-temperature part

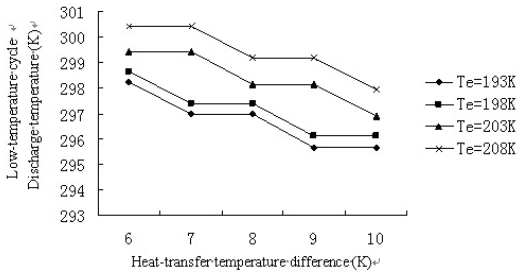



Figure 22: Discharge temperature of the low-temperature part vs the heat-transfer temperature difference of evaporative-condenser

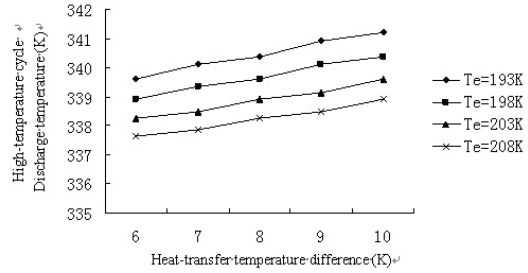



Figure 23: Discharge temperature of the high-temperature part vs the heat-transfer temperature difference of evaporative-condenser

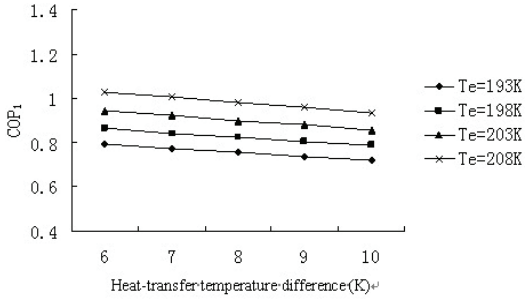



Figure 24: $COP_1$ vs heat-transfer temperature difference of evaporative-condenser

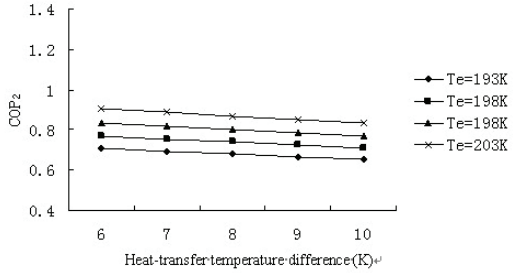



Figure 25: $COP_2$ vs heat-transfer temperature difference of evaporative-condenser

In Figure 20, it is shown that the change of the heat transfer temperature difference of the evaporative-condenser has little effect on the low-temperature cycle pressure ratio, which decreased faintly. It is also proved in Figure 21 that the high-temperature cycle pressure ratio increased tardily with the temperature difference getting lager.
In Figure 22 and Figure 23, it is proved that the discharge temperature of the low-temperature cycle compressor decreases and that of the high-temperature cycle compressor increases, at their own optimal mid-condensing temperature, with the increase of the heat transfer temperature difference of the evaporative-condenser.
As Figure 24 and Figure 25 shown, the *COP* value decreases slightly, when the heat transfer temperature difference of the evaporative-condenser gets lager.

## 4. ANALYSIS OF THE ENERGY LOSS

The system operating energy consumption was tested with some instruments. In normal operation condition, the voltage of the electric source is 380V, the total current is 68A, and the total power input is 45kW (in addition to the compressors, the system also contains two low-temperature circulating pumps for the second refrigerant system)
All refrigeration stages worked well and reached their designed temperature parameters point in their operating conditions.
At present, the major energy losses of this gasoline gas reclaiming system include the losses of the two low-temperature circulating pumps for the second refrigerant systems and the final exhaust temperature loss after the gasoline gas reclaimed. The final exhaust temperature is about -20℃, though the cooling capacity of final exhaust is reclaimed. It will be focused on how to reduce the circulating pump energy consumption and the reasonable use of the cooling capacity of final exhaust gas to improve the performance.

## 5. CONCLUSIONS

It can be obtained that the *COP* of the multi-temperature quasi-cascade refrigerating system decreases with the increase of the heat transfer temperature difference of the evaporative-condenser. The pressure ratio of the high-temperature part will be high when the condensing temperature is too high and the evaporating temperature of the low-temperature stage is too low.
In addition, although the low-temperature performances of R508 used for the low-temperature stage in the multi-temperature quasi-cascade refrigerating system is excellent and the *ODP* of R508B is 0, but R508B is too expensive, and its *GWP* is 12000. So we will focus on finding a new refrigerant which has excellent performances and environmentally friendly for the multi-temperature quasi-cascade refrigerating system.

## NOMENCLATURE

| | | |
|---|---|---|
| k | the refrigerant mass flow ratio through the high and low temperature parts | (-) |
| j | the refrigerant mass flow ratio through the evaporative condensator | (-) |
| $T_c$ | condensing temperature | (K) |
| $T_e$ | evaporating temperature | (K) |
| △T | total cost | (K) |
| COP | coefficient of performance | (W/W) |
| ODP | ozone depression potential | (-) |
| GWP | global warming potential | (-) |

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