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Theoretical and Experimental Research on A Multi-Temperature Quasi-Cascade Refrigeration System

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ABSTRACT

The research is based on a multi-temperature quasi-cascade refrigeration system used as a crucial part of a gasoline gas reclaiming unit which can condense and reclaim gasoline gas with three stage cycles. The system is similar with multi-temperature quasi-cascade refrigeration system which contains two refrigeration systems and can supply synchronously both -40℃ evaporating temperature and -80℃ evaporating temperature. The novel cycle has not only an evaporator in the second stage of the relatively high temperature gasoline gas reclaiming process but also has an evaporative-condenser used as the condenser for the relatively low temperature gasoline gas reclaiming process. The influence of operating parameters on the performance and the operating characteristics of the system has been analyzed based on the theoretical calculation results. The operating parameters, such as the components and the condensation temperature of the gasoline gas, the concentration of the gasoline gas, atmosphere temperature, the distribution of refrigerant flow rate between the evaporator and the evaporative-condenser of the high evaporating temperature refrigerating system, are taken into account. The system is also tested under different running conditions and experiment results are present in this paper. The more detail investigation will be proposed in the near future.

Keywords: multi-temperature, quasi-cascade, refrigeration system, gasoline gas

1. INTRODUCTION

According to the medium-term oil market report released by IEA in June 2009, the worldwide oil demand will grow by 0.6% every year between 2008 ~ 2014(about 540kb/d), which means growing from 85.8mb/d to 89mb/d.
At the same time, gasoline and its produces are the mixture of several kinds of hydrocarbons. The strong volatility of light hydrocarbons always makes it evaporated into gas state during exploitation, process, sale and use. 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. Therefore, it is important for the energy savings and for the environment protecting that reducing the gasoline gas emission to the atmosphere during the transport and use.
Nowadays, the reclaiming methods contain condensation, adsorption, absorption, membrane separation method and their combination method (Huang *et al*, 2006).
A gasoline gas reclaiming system supported by the Natural Science Foundation of Beijing was designed. The system including a multi-temperature quasi-cascade refrigeration system is a complex system consisting of a gasoline gas condensing subsystem, an energy (refrigeration) recycling subsystem and a secondary refrigerant subsystem. It is also a quasi-cascade refrigeration system containing two refrigeration cycles. The novel cycle has not only an evaporator in the second stage of the relatively high temperature gasoline gas reclaiming process but also has an evaporative-condenser used as the condenser for the relatively low temperature gasoline gas reclaiming process. Because these two evaporators can supply two different refrigerating temperatures, so the system is called multi-

temperature quasi-cascade refrigerating system. (It has been adjusted that the temperatures of high temperature stage evaporator and the evaporative- condenser are the same, so the initial and final running parameter of the refrigerants are the same)

The gasoline gas can be condensed by three stages in the system. The first stage, R22 vapor compression refrigeration cycle, supplies 3 ℃ condensing temperature to condense most of water vapor and a small amount of the high molecular hydrocarbons in the gasoline gas. The multi-temperature quasi-cascade refrigerating system is composed of the second and third stages. The temperature supplied by the second stage can cool the gasoline gas to -35℃ and make the low molecular hydrocarbons condensed, and the lower molecular hydrocarbons was condensed by the temperature, -75 ℃, supplied by the third stage.

2. THE CALCULATION EXAMPLE OF CONDENSATION GASOLINE GAS RECLAIMING SYSTEM LOAD

The load of the condensation gasoline gas reclaiming system will be calculated in the process that the gasoline gas temperature drops from 35℃ to 3℃; from 35℃ to -35℃; and lastly from 35℃ to -75℃ continuously. The system's processing capacity is designed to be 60m^3/h.

The gasoline gas concentration of the mixture is 30% and air relative humidity is 50%. In general, the volatilization of low-carbon material in the gasoline gas mixture is little. According to relative literature, the gasoline gas main components are kinds of light hydrocarbons, such as $C_4 \sim C_6$. Therefore, the load of reclaiming system will be calculated considering $C_4 \sim C_6$ only in following paragraph. The components of the gasoline gas are shown in the Table1.

Table1.Volume fraction of components in the gasoline gas

Component	n-C_4	i-C_4	c-C_4	n-$C_4^=$	i-$C_4^=$	n-C_5	i-C_5	n-C_6	Nitrogen	Oxygen
Vol (%)	2.420	2.894	3.724	2.944	3.24	0.434	6.156	6.884	54.7	13.83

The calculation method is that to calculate the saturated steam pressure of hydrocarbons according to condensing temperatures, then compare them with actual partial pressures in the corresponding condensing temperature to judge if the hydrocarbons are condensed.

Three parameters method steam pressure equation (Tong.1996) is adopted to gain the saturated steam pressure of hydrocarbons in condensing temperatures,

$$\ln p_r = f^{(0)}(T_r) + \omega f^{(1)}(T_r) \tag{1}$$

where

$$f^{(0)} = 5.92714 - \frac{6.09648}{T_r} - 1.28862\ln T_r + 0.169347 T_r^6$$

$$f^{(1)} = 15.2518 - \frac{15.6875}{T_r} - 13.472\ln T_r + 0.43577 T_r^6$$

$$\omega = (-\ln p_c - 5.92714 + 6.09648\theta^{-1} + 1.28862\ln\theta - 0.1697347\theta^6)/(15.2518 - 15.6875\theta^{-1} - 13.4721\ln\theta + 0.43577\theta^6)$$

The physical parameters of calculated hydrocarbons is showed in table2.

Table2. Physical parameters

	T_c (K)	T_b (K)	p_c (kPa)
n-C_4	425.16	272.66	3796
i-C_4	408.1	261.4	3647
c-C_4	493.72	306.6	5095.1
n-$C_4^=$	443.7	284.0	4500
i-$C_4^=$	425.15	268.69	4330
n-C_5	469.6	309.2	3369
i-C_5	460.4	301.0	3380
n-C_6	507.44	341.88	3031

The calculated hydrocarbons potential heat values which gained by above table and calculating by equation (1) is shown in table3.

Table3. potential heat values

component	n-C_4	i-C_4	c-C_4	n-$C_4^=$	i-$C_4^=$	n-C_5	i-C_5	n-C_6

| potential heat value (kJ/mol) | 21.29 | 25.77 | 26 | 23.6 | 22.68 | 25.44 | 24.47 | 29.434 |

To ensure that the heat capacities at constant pressures of hydrocarbons are calculated accurately, a calculation method presented by Rihani and Doraiswamy (1965) is recommended. The calculation error of this method is often less than 2%~3%:

$$c_p = \sum_i n_i a_i + \sum_i n_i b_i T + \sum_i n_i c_i T^2 + \sum_i n_i d_i T^3 \qquad (2)$$

where the values of a_i, a_i, a_i and a_i can be calculated with looking up correlative tables.

Table4. Heat capacity at constant pressures

component	a_i	$b_i \times 10^2$	$c_i \times 10^4$	$d_i \times 10^6$
n-C$_4$	2.0064	8.5592	-0.4089	0.027892
i-C$_4$	-1.16971	9.8457	-0.5372	0.06
c-C$_4$	1.578	8.5452	-0.4788	0.01308
n-C$_4^=$	-5.829	11.1182	-0.7366	0.03876
i-C$_4^=$	-5.829	11.1182	-0.37614	0.03873
n-C$_5$	2.4009	10.6955	-0.37614	0.03049
i-C$_5$	-3.3959	13.0273	-0.08	0.05803
n-C$_6$	2.7954	12.8318	-0.8196	0.03308

The cold load of every condensation stage calculation results is shown in the table5.

Table5. the results of cold load calculation

Step	35℃～3℃	3℃～-35℃	-35℃～-75℃
Total load	1.80kW	2.75kW	3.75kW

The second stage refrigeration system supplies the cooling capacity not only condensing gasoline gas, but also removing the releases heat of the third stage, so the final cooling capacity supplied by the second stage should be add with the release heat of the third step.

3. ANALYSIS OF THE INFLUENCING FACTORS ON SYSTEM LOAD

3.1 The influence of gasoline gas components, concentrations and atmosphere temperature on the performances of multi-temperature quasi-cascade refrigeration system

As Figure1 shows, the differences between the saturated steam pressure of high and low molecular weight hydrocarbons is enormous. The high boiling point of high molecular weight hydrocarbons makes it condensed easier than low molecular weight hydrocarbons which has a low boiling point and is hard to be condensed, so the components of gasoline gas must be test rigorously in the early term of design for a suitable condensation gasoline gas reclaiming system.
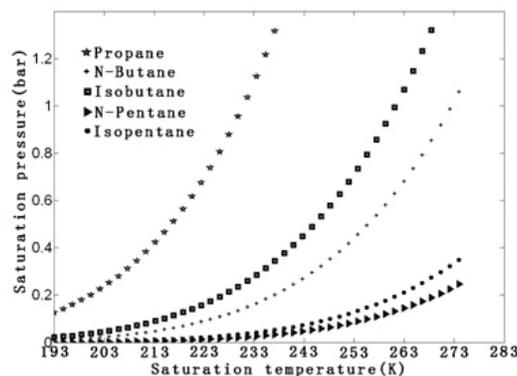

Figure 1: Saturated steam pressures

Most of hydrocarbons begin to be condensed under 0℃ and atmosphere temperature keeps above 0℃ all year round, so the changes of atmosphere temperature influence on the gasoline gas very little. But the changes of atmosphere temperature have a great influence on the refrigeration system. the atmosphere temperature decrease means the condensing temperature decrease, which is benefit for COP improvement.

3.2 The influence of refrigerant distribution on the performances of multi-temperature quasi-cascade refrigerating system

Assuming that the high temperature stage condensing temperature is 318K, the low temperature step evaporating temperature is 193K, the high temperature step superheat degree is 13K, the subcooling degree is 12K, degree of superheat of the low temperature cycle is 30K, and subcooling degree is 12K.

As shown in Figure2, the COP increases coherent with k (the mass flow ratio of the refrigerants through the high and the low temperature stage evaporators) increases, which leads to the conclusion that COP is the monotone increasing function of k. Similarly, the COP increases as j (the mass flow ratio of the refrigerants through the high temperature step evaporator and the evaporative-condenser) increases, as shown in Figure 3.
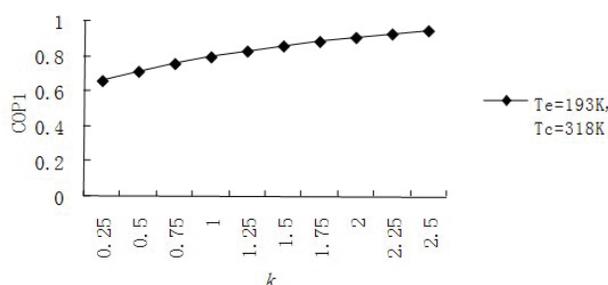

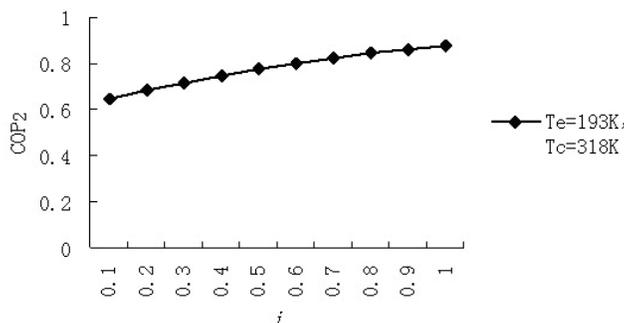Figure 2: COP1 variation of multi-temperature cascade refrigerating cycle with mass flow ratio k

Figure 3: COP2 variation of multi-temperature cascade refrigerating cycle with the mass flow ratio j

4. APPARATUS SELECTION AND ANALYSIS OF THE EXPERIMENT RESULT

According to the refrigeration system design parameters, compressors are selected. The result is shown in Table 6.

Table 6 The compressors selection result

Compressor	Cooling Capacity (KW)	Power (KW)
The first stage	1.57	0.91
The second stage	13.55	13.53
The third stage	3.75	7.5

4.1 Analysis of the circulating pump operation pressure

As Figure 4 shows, the operation pressures of circulating pumps keep increasing, and the highest operation pressure of the second-level circulating pump reaches to 0.235MPa, which will cause the refrigerant circulation flow rate decrease and make heat transfer worse . The reason is that the refrigerant viscosity is too large at low temperature, which could be improved by adjusting the distribution of refrigerants.
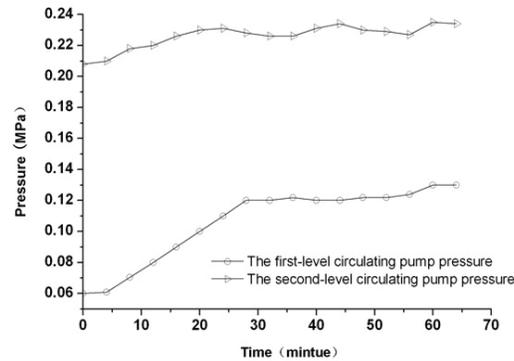
Figure 4: Operation pressure curves of circulating pumps

4.2 Analysis of the compressors operation

As Figure 5 shows, the first stage compressor operates properly. Its suction pressure keeps 0.3MPa, and discharge pressure keeps 1.2MPa. When the first condenser discharge temperature reaches the designed temperature, the first stage compressor is controlled to stop at once (20~32min, and 40~52min in Figure 5), and when the temperature is more than the designed temperature, the compressor works. By this way the system energy consumption will be reduced.
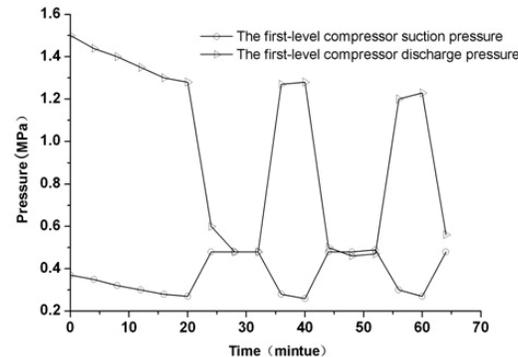

Figure 5: Suction and discharge pressure curves of first step compressor

Figure 6 shows the curves of suction and discharge pressure of the second stage compressor. R404A is adopted in the second stage refrigeration system. The load of second step refrigerating system is larger than others because it contains the load of condensing gasoline gas in the second step and the discharge heat of the third step. The second stage refrigerating system works properly. Its suction pressure keeps 0.1MPa (evaporating temperature is -45℃). Because of the low evaporating temperature, the pressure ratio of the second step compressor is large. So it is necessary to add a fan to the R404A compressor.
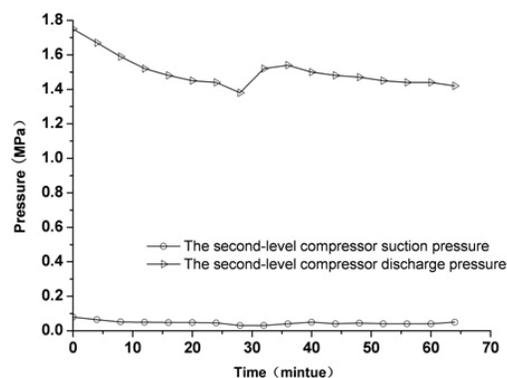

Figure 6: Suction and discharge pressure curves of second step compressor

Figure 7 shows the curves of suction and discharge pressure of the third stage compressor. It is shown that the third refrigerating system works properly and relative control system operates well. Because the low temperature stage of the multi-temperature cascade refrigeration system need a extremely low temperature (-45℃ in this system), it is necessary to lay up a evaporating temperature test equipment in the high temperature stage to ensure high temperature stage can run at a reasonable low condensing temperature, the third step compressor doesn't start, which can avoid the compressor burn failure at a high condensing temperature.
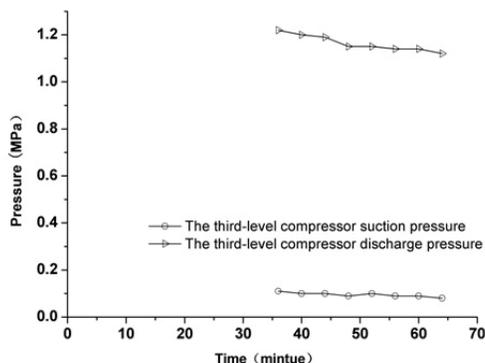

Figure 7: Suction and discharge pressure curves of third step compressor

4.3 Analysis of the stages outlet temperature

Figure 8 shows the curves of temperatures of the third stage condensing heat exchanger outlets. The system is tested in winter at an average temperature about 11℃. It is observed that the system meets the design requirements of every stage condensing temperature, but there also exist some problems:

4.3.1 The first refrigeration system is designed to removing the water vapor. According to the load calculation result presented in paragraph 2, the load of the first stage is less than others. To ensure the precision of the controller, the temperature decreases slowly. And the first condensing heat exchanger outlet temperature change delays in relation to the compressor start-stop.

4.3.2 The second and third refrigeration cycles compose the dual temperature cascade refrigeration system. Two cycles run interdependently and are controlled together. It is observed that until the third step condensing temperature reach the set value, the third step compressor doesn't start. After it starts, the third step condensing heat exchanger outlet temperature decrease, which is the same as the second step condensing heat exchanger outlet temperature before the third step condensing temperature reach the set value.
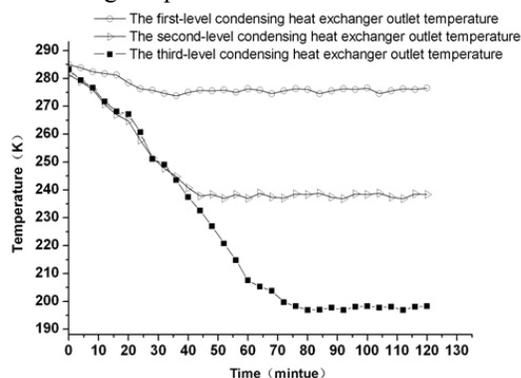

Figure 8: Temperature curves of condensing heat exchanger outlets

4.4 Analysis of the system energy losses

Electrical Measurement Instrument was employed to test the system operating energy consumption. In normal operation condition, the voltage of the whole system is 380V, and electrical current is 68A, and the power is 45kW (excepting the compressor, the system contains two low-temperature circulating pump for the refrigerant system).Every step operated well and achieved their design temperature parameters point in their operating

conditions. That means 60m^3 gasoline gas mixture which contain 30% gasoline gas will be reclaimed with 45kWh on 1 hour. According to the current international gasoline price, 84 U.S. dollars a barrel, economic benefits a lot, while the corresponding reduction of carbon dioxide emissions is remarkable.

At present, the major energy losses of this gasoline gas reclaiming system comprise the losses as following.

1. The two low-temperature circulating pump energy consumptions for the refrigerant system, especially the one for the second step system (-35℃). The refrigerant large viscosity causes the pump work pressure and the power consumption increased.
2. The final discharge temperature loss after the gasoline gas reclaiming. The final discharge temperature is little bit less of -20℃, although the cooling capacity of final exhaust is already reclaimed through two-stage refrigerant system. By the way, the gasoline gas mixture must keep a steady velocity through the condensing channel to ensure the steady reclaiming capacity. It is the key point to utilize the cooling capacity of final exhaust fluid without influences on gasoline gas flow rate.

In summary, the research will be focused on how to reduce circulating pump energy consumption and utilize the cooling capacity of final exhaust perfectly.

5. CONCLUSIONS

The load calculation methods adopted in the gasoline gas reclaiming system is feasible, and it is helpful to design a gasoline gas reclaiming system.

According to analysis, there are lots of influences factors. Therefore, the load should be calculated with consulting gasoline gas test results strictly.

It is proved that the system operates stability and meets the requirement of design parameters such as temperatures and pressures. The research of gasoline gas testing will carry on gradually.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

P_r	contrast pressure, $P_r = P/P_c$	(–)		**Subscripts**
P	total pressure	(bar)	i	a vapor component
T_r	contrast temperature, $T_r = T/T_c$	(–)	1	condition before condensation
ω	eccentricity factor	(–)	2	condition after condensation
θ	dimensionless temperature, $\theta = T_b/T_c$	(–)	c	critical point
COP	coefficient of performance,	(W/W)	b	ordinary boiling point
k	the refrigerant mass flow ratio through the high and low temperature parts	(-)		
j	the refrigerant mass flow ratio through the evaporative condensator	(-)		

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