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Guoyong Yang
Guangdong Meizhi Compressor Limited, China, People's Republic of, yangguoy@chinagmcc.com

Cheng Zhang
Guangdong Meizhi Compressor Limited, China, People's Republic of, zc@chinagmcc.com

Yanping Wu
Guangdong Meizhi Compressor Limited, China, People's Republic of, wuyp2@chinagmcc.com

Siqing Liao
Guangdong Meizhi Compressor Limited, China, People's Republic of, liaosq@chinagmcc.com

Development of a New Dual-Cylinder Rotary Compressor for VI System

Guoyong Yang[1*], Cheng Zhang[2], Yanping Wu[3], Siqing Liao[4]

Guandong Meizhi Compressor Co. Ltd. R&D Section,
Foshan City, Guangdong Province, China, 528333

[1]Phone: +86-757-22325058, E-mail: yangguoy@chinagmcc.com
[2]Phone: +86-757-22329089, E-mail: zc@chinagmcc.com
[3]Phone: +86-757-22325073, E-mail: wuyp2@chinagmcc.com
[4]Phone: +86-757-22329158, E-mail: liaosq@chinagmcc.com

ABSTRACT

On the vapor compression refrigeration system, Vapor injection (VI, the phase separator type injection or the internal heat exchanger type injection) compression cycle's superiority over non-injection cycle has been well known. VI system produces the high heating/cooling capacity, and its power consumption is less than the non-injection system. But if a VI compression cycle uses a single rotary compressor, there is the problem that refrigerant injection increases the indicated power by mixture loss. If we use a two-stage rotary compressor, indicated power also increases because of its two times exhaust process. To solve these problems, we developed a new dual-cylinder rotary compressor for VI systems, one of the cylinders is used to compress the gas from the evaporator, and the other is used to compress the gas from the phase separator (flash-tank). Its design method is discussed and its performance under different conditions is analyzed.

1. INTRODUCTION

The demands of people about the efficient energy consumption are increasing continuously by reason of the exhaustion of fossil fuel and global warming. To meet these demands, most governments are strengthening the regulation of the CO2 emission and standard about certification of products and each company is spurred to the development of new technology for the high efficiency. Also, the demand for high efficiency in heating and cooling system is increasing by a growing customer's recognition about the energy conservation. On the vapor compression refrigeration system, Vapor injection compression cycle's superiority over non-injection cycle has been well known. VI system produces the high heating/cooling capacity, and its power consumption is less than the non-injection system. Unfortunately, other problems arise, such as mixture loss on refrigerant injection into the compression chamber of a single-stage compressor. If a two-stage compression cycle uses a single rotary compressor, there is the problem that refrigerant injection increases the indicated power by mixture loss (Sekiya *et al.*, 2005), compared to that of ideal isentropic compression.

Another solution for the two-stage compression cycle is the use of a two-stage compressor. A two-stage flash tank cycle (FTC) contains two additional components compared to the standard vapor compression cycle, namely an additional expansion valve and a flash tank separator. The two-stage flash tank cycle plotted on a P-h diagram is shown in Figure 1. In a two-stage flash tank cycle the condensed liquid leaving the condenser is first throttled through the first stage expansion device and the two-phase refrigerant (state point 4) enters in the flash tank separator. In the flash tank the liquid and vapor components of the two-phase mixture are separated. The saturated vapor refrigerant (state point 5) is injected at an intermediate compressor suction port and the saturated liquid refrigerant (state point 6) is throttled through the second stage expansion device prior to entering the evaporator. After entering the compressor, the saturated vapor refrigerant mixing with the exhaust gas form the first stage, and then was compressed by the second stage together. In a two-stage flash tank cycle, usually we use a two-stage rotary compressor, because most of the gas is compressed two times, the indicated power increases because of its two times exhaust process.

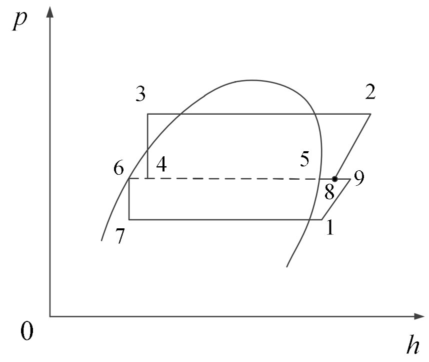

Figure 1: Two-stage compression refrigeration/heat pump system

2. MODEL OF NEW DUAL-CYLINDER ROTARY COMPRESSOR VI SYSTEM

The new dual-cylinder rotary compressor in this paper is a rotary compressor provided with two cylinders, each of which is provided with an independent suction port. The two cylinders woke synchronously by the same electric motor.

The schematic diagram of the new dual-cylinder rotary compressor VI system plant and its corresponding thermodynamic diagram are shown in Figure 2. In this new dual-cylinder rotary compressor VI system, the condensed liquid (state point 3) leaving the condenser is first throttled through the first stage expansion device and the two-phase refrigerant (state point 4) enters in the flash tank separator. In the flash tank the liquid and vapor components of the two-phase mixture are separated. The saturated vapor refrigerant (state point 5) is sucked in by one of the cylinders (the subordinate cylinder) and compressed directly to the exhaust pressure (state point 8) and the saturated liquid refrigerant (state point 6) is throttled through the second stage expansion device prior to entering the evaporator (state point 7), and then, after evaporation (state point 1), is sucked into the other cylinder (the main cylinder) and compressed to the exhaust pressure (state point 9). The gas discharged from the two cylinders is discharged after mixing in the compressor (state point 2), and then is condensed (state point 3). Because the gas is compressed independently, mixed losses and two exhaust losses are avoided. The state point 3’ is assumed to be obtained by direct throttling of the state point 3, in fact it does not exist in this system. By assuming the state point 3’, we get the basic cycle (state point 1-9-3-3’) under the same conditions.

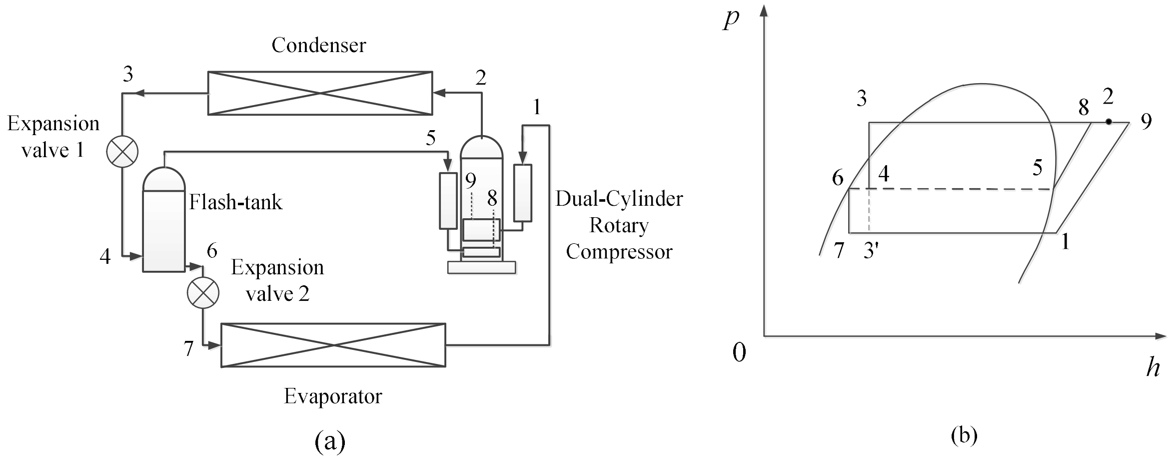

Figure 2: New dual-cylinder rotary compressor VI system

3. THERMODYNAMIC ANALYSIS

For system shown in Figure 2, mathematical is presented as below:
i. The refrigerating capacity per unit weight in evaporator is

$$q_o = h_1 - h_7 \tag{1}$$

ii. The power consumption for the main cylinder is

$$w_{1-9} = \frac{h_{9s} - h_1}{\eta_{s,1\text{-}9}\eta_{\mathrm{m}}\eta_{\mathrm{mo}}} \tag{2}$$

$\eta_{s,1\text{-}9}$ is the isentropic efficiency of compression process 1-9, h_{9s} is the enthalpy assuming that compression process 1-9 is isentropic, η_{m} is the mechanical efficiency of the compressor, η_{mo} is the electrical efficiency of the compressor.
iii. The power consumption for the subordinate cylinder is

$$w_{5-8} = \frac{(m_5 / m_1)(h_{8s} - h_5)}{\eta_{s,5\text{-}8}\eta_{\mathrm{m}}\eta_{\mathrm{mo}}} \tag{3}$$

$\eta_{s,5\text{-}8}$ is the isentropic efficiency of compression process 5-8, h_{8s} is the enthalpy assuming that compression process 5-8 is isentropic.

iv. The cooling COP of the new dual-cylinder rotary compressor VI system is

$$\mathrm{COP_L} = \frac{q_o}{w_{1-9} + w_{5-8}} \tag{4}$$

v. As contrast, the cooling COP of the basic refrigeration cycle (base system) is

$$\mathrm{COP_{BASE}} = \frac{h_1 - h_{3'}}{w_{1-9}} = \frac{h_1 - h_3}{w_{1-9}} \tag{5}$$

In equations (2) and (3), $\eta_{s,1\text{-}9}$ and $\eta_{s,5\text{-}8}$ were calculated by the follow formula (Xu Shuxue and Ma Guoyuan, 2014)

$$\eta_s = 1 - 0.6[1 - (p_d / p_s)^{-0.3}] \tag{6}$$

p_d is the discharge pressure, and p_s is the suction pressure. The efficiency ($\eta_{\mathrm{m}}\eta_{\mathrm{mo}}$)for the compressor take a value of 0.70.
The following assumptions have been made to simplify the analyses:
(1) System operates in steady state.
(2) Heat transfer with the surroundings is negligible.
(3) No pressure drop occurs in evaporator, condenser, flash tank economizer, and their connecting pipelines.

Based on Equations (1) -(6) and the properties of the refrigerant, R410A, the variations of cooling performance with the intermediate temperature T_m (temperature of flash tank/state point 6) is calculated. The calculation conditions were: condensing temperature, T_k, of 45℃, evaporating temperature, T_o, of 3℃, 5℃, 7℃ under cooling condition, the superheat is 5℃ and the sub cooling is 5℃; The calculation results are plotted in Figure 3.

As shown in Figure 3, the cooling COP increase firstly and then decrease with the increasing of the intermediate temperature T_m. The intermediate temperature T_m which makes the VI system has the highest $\mathrm{COP_L}$ called the optimal T_m. The variations of the optimal T_m with the evaporating temperature T_o is plotted in Figure 4 (the condensing temperature T_k= 45℃, the superheat = 5℃ and the sub cooling = 5℃). As shown in Figure 4, the optimal T_m almost increased linearly with the increasing of the intermediate temperature T_m. Under the optimum intermediate temperature T_m, the variations of the $\mathrm{COP_L/COP_{BASE}}$ (the COP ratio of the new VI system to the base system) with the evaporating temperature T_o is plotted in Figure 5(the condition is same as Figure 4), and we can get that the lower the evaporation temperature, the greater the COP improvement.

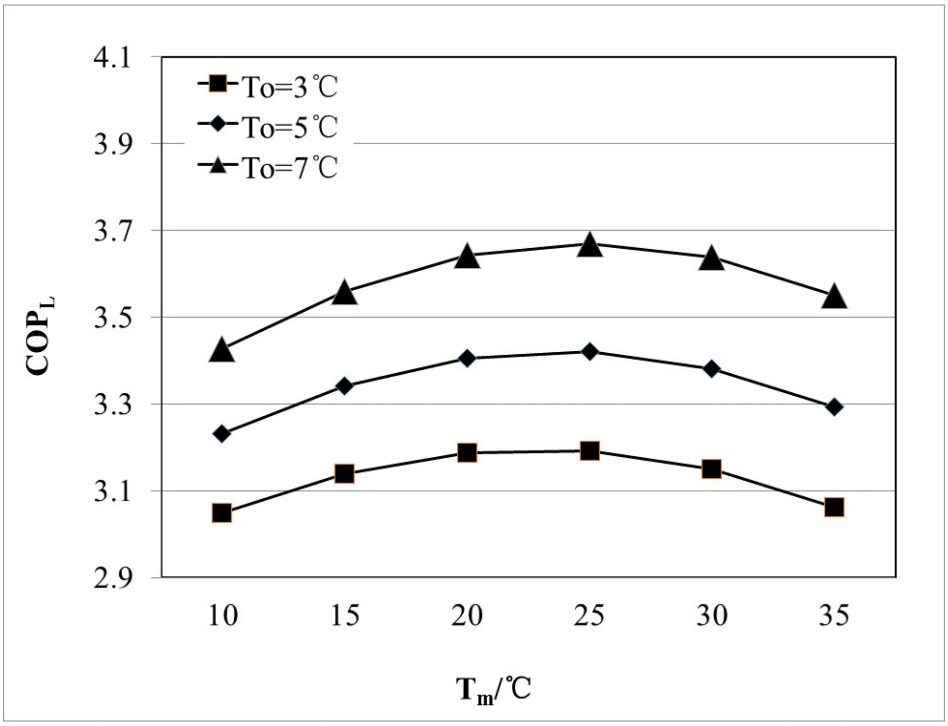

Figure 3: The variation of COP_L with T_m

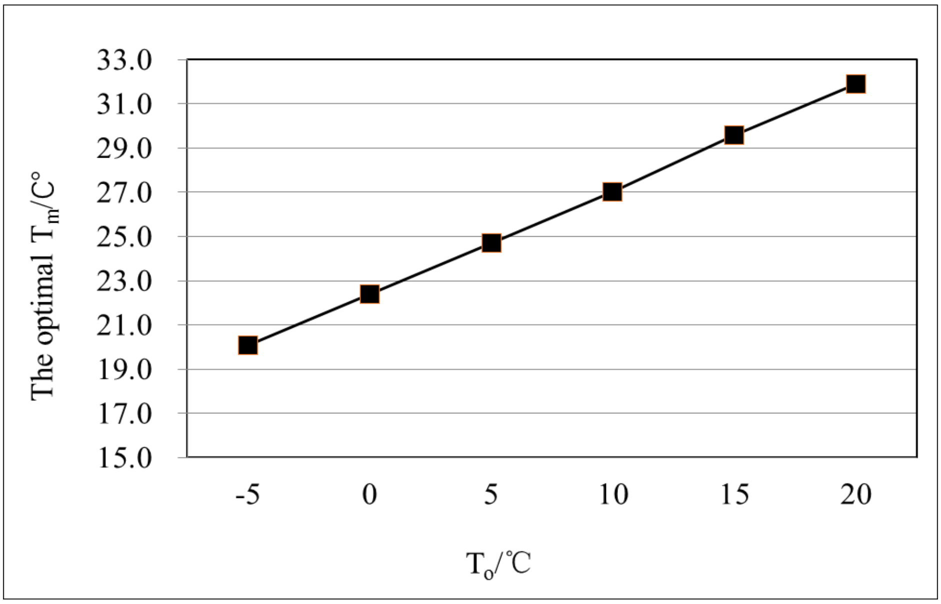

Figure 4: The variation of the optimal T_m with T_o

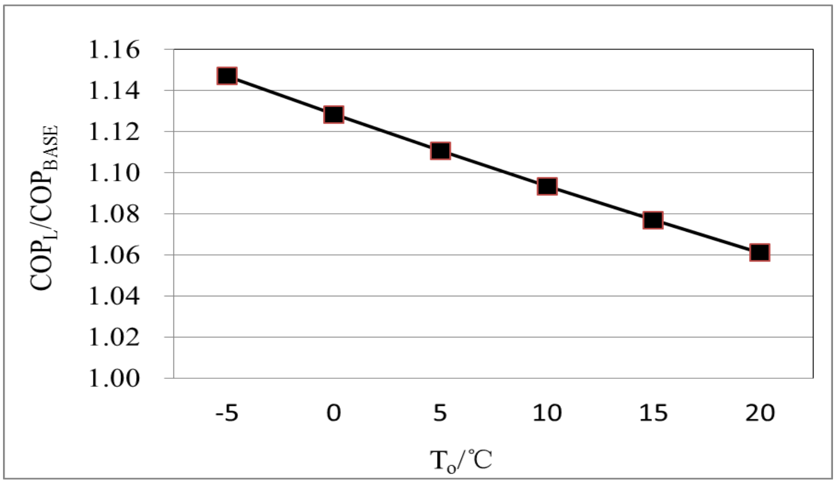

Figure 5: The variation of COP_L/COP_{BASE} with T_m (under the optimum intermediate temperature T_m)

4. DESIGN OF THE NEW DUAL-CYLINDER COMPRESSOR

Based on the above analysis and the refrigerant R410A, a new dual-cylinder compressor used in this VI system is designed, for a residential air conditioner with 3500W target ability at rated condition. Consider the Chinese Annual Performance Factor (APF) requirements, the compressor is designed based on 5 conditions, showed in Table 1.

By the previous analysis, the optimum intermediate temperature of each working condition is different, and the design of the compressor is based on the comprehensive energy efficiency of this 5 conditions. The comprehensive energy efficiency is the weighted average COP_L of the 5 conditions, the contribution ratio is also showed in Table 1 respectively. The design result of displacement of the compressor is shown in Table 2.

Table 1: The refrigeration operating conditions for evaluating performance

Operating condition	The first	The second	The third	The fourth	The fifth
Condensing temperature (℃)	45.6	39.6	43.6	31.5	46.5
Evaporating temperature (℃)	10.7	17.8	0.8	2.9	-4.7
Superheat (℃)	9.3	7.2	5.3	5.1	6.7
Sub cooling (℃)	6.1	1.0	7.0	4.0	9.9
Ambient temperature (℃)	35	35	35	35	35
Contribution ratio	25%	33%	13%	10%	19%

Table 2: The design result of displacement

Parameters	The main cylinder	The subordinate cylinder
Displacement (cc)	10	0.9

5. EXPERIMENTAL RESULT OF THE NEW DUAL COMPRESSOR

Based on the design result, a new dual-cylinder rolling piston compressor was manufactured. At the same time, a comparative traditional compressor for the basic refrigeration cycle is also processed as a base compressor with a similar structure, and then we the tested this these two types of compressors. The test results of cooling COP of the 5 conditions are shown in Table 3, and we can get that the cooling COP increase 4.6%-9.1% under this 5 conditions. The comparison between theory and experiment of COP_L/COP_{BASE} is shown in Figure 6, the experimental results are in agreement with the theory, it shows that the compressor achieve the desired performance goals. In the system operation, the actual intermediate pressure has a great influence on the performance of the compressor. The variations of the COP_L with the intermediate pressure p_m (pressure of flash tank) about the first condition is plotted in Figure 7, it can be seen that when the intermediate pressure exceeds the optimum value, the COP_L is decreased dramatically because of the suction of liquid and this should be avoided in the application.

Table 3: The test result of cooling COP

Operating condition	The first	The second	The third	The fourth	The fifth
COP_{BASE} of the base compressor	3.824	6.597	2.897	4.951	1.847
COP_L of the new dual compressor	4.033	6.916	3.134	5.181	2.016
COP_L/COP_{BASE}	105.5%	104.8%	108.2%	104.6%	109.1%

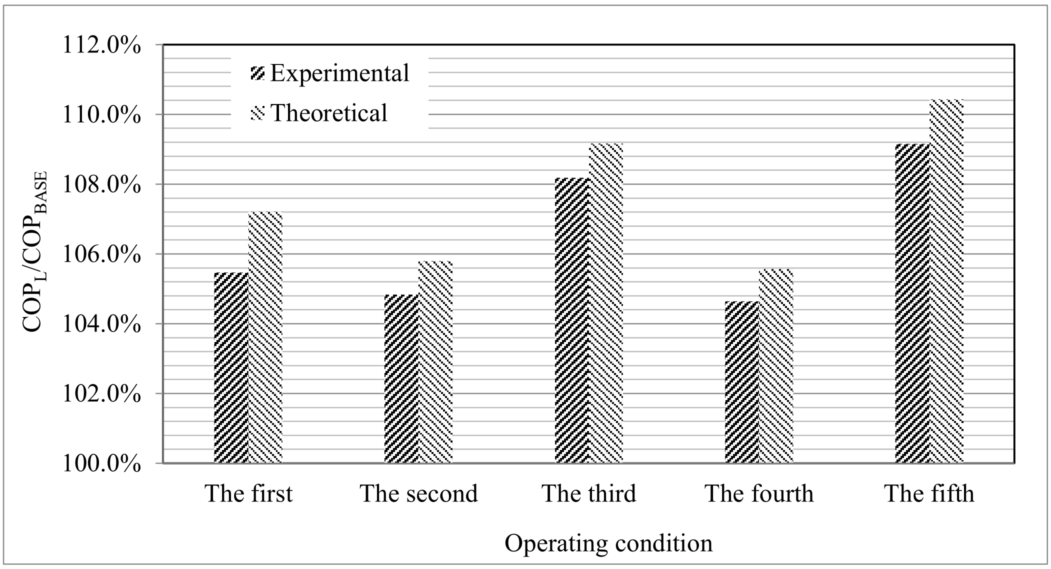

Figure 6: The Comparison between theory and experiment of COP_L/COP_{BASE}

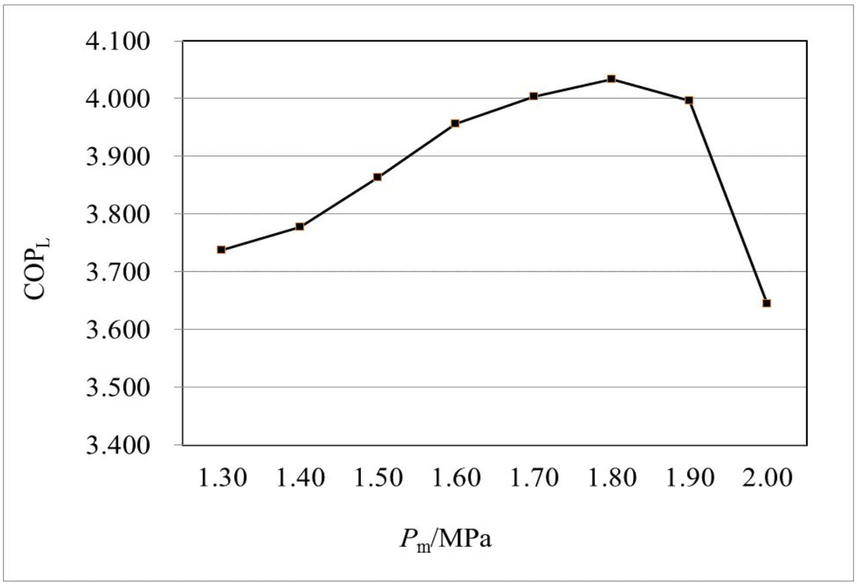

Figure 7: The variation of COP_L with the intermediate pressure p_m

6. CONCLUSIONS

A new dual-cylinder rotary compressor for VI systems is developed, one of the cylinders is used to compress the gas from the evaporator, and the other is used to compress the gas from the phase separator (flash-tank). Its design method is discussed and its performance under different conditions is analyzed:
(1) the cooling COP increase 4.6%-9.1% under the test conditions. The comparison between theory and experiment of COP_L/COP_{BASE} indicates the experimental results are in agreement with the theory.
(2) The variations of the COP_L with the intermediate pressure p_m (pressure of flash tank) indicates that when the intermediate pressure exceeds the optimum value, the COP_L is decreased dramatically because of the suction of liquid and this should be avoided in the application.

NOMENCLATURE

h	enthalpy	(kJ/kg)
m	mass flow rate	(kg/s)
p	pressure	(MPa)
p_d	discharge pressure	(MPa)
p_m	intermediate pressure	(MPa)
ps	suction pressure	(MPa)
T_k	condensing temperature	(℃)
T_o	evaporating temperature	(℃)
T_m	intermediate temperature	(℃)
w	compressor work	(kJ/kg)
COP_L	coefficient of cooling performance	(–)
COP_{BASE}	coefficient of heating performance	(–)
η_i	isentropic efficiency	(–)

Subscript

1, 2, 3, ...	state points shown in the figures

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Xu Shuxue, Ma Guoyuan, (2014). Experimental study on two-stage compression refrigeration/heat pump system with dual-cylinder rolling piston compressor. *Applied Thermal Engineering*, *62(2014)*, 803–808