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A NEW CYCLE AND ITS COMPUTER OPTIMIZATIONON ON SINGLE STAGE AMMONIA COMPRESSION REFRIGERATION

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ABSTRACT

This paper proposes a new cycle. An auxiliary circuit is added to the common ammonia single compression refrigeration cycle. A part of the refrigerant evaporates and makes liquid ammonia super-subcooled. The coefficient of performance (COP) and flow rate ratio on the new cycle are analyzed and calculated on a personal computer as the evaporating temperature in subcooler (TIS) varies. The results show that the COPs of new cycle will increase by 3-6% compared with the common cycle. The TIS has an optimum value where the increment of COP is maximum. An equation for calculating the optimum TIS is proposed. The paper further discusses how the new cycle is carried out in different cases. It is quite evident that the new cycle is feasible and energy saving can be obtained.

INTRODUCTION

Subcooling of the liquid refrigerants can improve the coefficient of performance (COP) in the compression refrigeration cycle. It can be carried out by the intercooler in the two stage compression refrigeration. But subcooling is seldom used in single stage compression ammonia refrigeration plant because subcooling by water is not very efficient and brings little benefit. This paper proposes a method of super-subcooling. An auxiliary circuit is added to common cycle. The super-subcooling of the liquid refrigerant is carried out by this branching. The COPs and volume flow rates of the new cycle in various conditions are calculated and analyzed using a personal computer. The relationships between the parameters are discussed. The paper further discusses the operating mode of this new cycle in the different cases of industrial refrigerating plants.

NEW CYCLE AND ITS PRESSURE-ENTHALPY CHART

The proposed new cycle is composed of a common cycle and an auxiliary circuit. The auxiliary circuit consists of a subcooler, an expansion valve and a special compressor. Fig.1 shows the diagram of the cycle. A part of the refrigerant in the subcooler evaporates and makes the liquid refrigerant subcooled. Suppose that the evaporating temperature in the subcooler is TIS and that the temperature of liquid refrigerant after subcooling is 5 °c higher than the TIS or $T_5 = TIS + 5 = T_4 + 5$. Fig.2 is the pressure-enthalpy chart of the new cycle in Fig.1.

Let the refrigerating capacity of the new cycle be Q, the mass flow rate in the condenser be G, and the mass flow rate in the auxiliary circuit be A. Then the mass flow rate in the evaporator is given by (G-A). Neglecting the loss of heat transfer, the energy balance of the subcooler is given by,

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$$G*H_3=A*H_{1'}+(G-A)*H_5 \quad (1)$$

Where H is the enthalpy of the refrigerant. From equation (1):

$$A=G(H_3-H_5)/(H_{1'}-H_5) \quad (2)$$

The mass flow rate in the condenser is the sum of the evaporator flow rate and subcooler flow rates,

$$G=Q/q_n+A=Q/(H_1-H_6)+G(H_3-H_5)/(H_{1'}-H_5) \quad (3)$$

where q_n is the refrigerating capacity per kilogram refrigerant in the new cycle. The relation between G and Q can be obtained from equation (3) as,

$$G=Q/(H_1-H_6)*(H_{1'}-H_5)/(H_{1'}-H_3) \quad (4)$$

COPS OF THE NEW CYCLE VARY WITH THE TIS

It is easy to understand that the COP of the new cycle will vary with the different TIS in the subcooler. In order to find the pattern of variation the new cycle is compared with a common single compression cycle in the same refrigerating capacity (Q) and the same condensing and evaporating temperature. A common cycle showed in Fig.2 is 1-2-3-m-1.

The COP of the common cycle is[1] :

$$COP_m=q_m/W_m=(H_1-H_m)/(H_2-H_1) \quad (5)$$

Where the q_m is the refrigerating capacity per kilogram refrigerant in the common cycle. W_m is the compression work per kilogram refrigerant in the common cycle.

The COP of the new cycle is :

$$COP_n=Q/W=\frac{G(H_1-H_6)(H_{1'}-H_3)/(H_{1'}-H_5)}{(G-A)(H_2-H_1)+A(H_{2'}-H_{1'})}$$
$$=\frac{(H_1-H_6)(H_{1'}-H_3)}{(H_{1'}-H_3)(H_2-H_1)+(H_3-H_5)(H_{2'}-H_{1'})} \quad (6)$$

Where W is the power which the new cycle consumes. The increment of the COP of the new cycle compared with the common cycle is :

$$IN_{cop}=(COP_n-COP_m)/COP_m*100\% \quad (7)$$

VOLUME FLOW RATES OF THE NEW CYCLE VARY WITH DIFFERENT TIS

It is clear that the volume flow rate of the new cycle will vary with the different TIS. Because the capacity of the compressor is determined by the volume flow rate of the refrigerant, the change in the volume flow rate of the new cycle is a key factor. It decides whether the new cycle is practicable or not.

The volume flow rate of the common cycle without subcooling is:

$$V_m = G_m * v_1 = Q * v_1 / (H_1 - H_m) \tag{8}$$

The sum of the volume flow rate of the evaporator and auxiliary circuit in the new cycle is :

$$\begin{aligned} V_n &= (G-A) * v_1 + A * v_{1'} \\ &= Q * v_1 / (H_1 - H_6) + G * v_{1'} * (H_3 - H_5) / (H_{1'} - H_5) \\ &= [Q / (H_1 - H_6)] [v_1 + (H_3 - H_5) * v_{1'} / (H_{1'} - H_3)] \end{aligned} \tag{9}$$

Where v_1 and $v_{1'}$ are the specific volume of refrigerant vapour at point 1 and 1'. The ratio of the volume flow rate of the new and common cycles is :

$$V_n / V_m = (H_1 - H_m) [1 + (H_3 - H_5) * v_{1'} / v_1 / (H_{1'} - H_3)] / (H_1 - H_6) \tag{10}$$

The evaporating pressure in the subcooler is higher than that in the evaporator. So the volumetric efficiency of the compressor in the auxiliary circuit is higher than that of the compressor for the evaporator. The volume flow rate of two different specific volumes can be added in equation (9) because the actual volume flow rate of the new cycle will be slightly less than the result of equation (9). This will not affect the conclusion of analysis in the end.

On the other hand, from the point of view of application, the relation between the main compressor and the auxiliary compressor is a important factor. That is to say , the ratio of the volume flow rate of two part compressors must be considered.

The volume flow rate of the main compressor (for the evaporator) is:

$$V_z = (G-A) * v_1$$

and the volume flow rate of the auxiliary compressor (for the subcooler) is:

$$V_f = A * v_{1'}$$

The ratio of the volume flow rates of the main compressor and the auxiliary compressor is :

$$\begin{aligned} V_z / V_f &= (G-A) * v_1 / A / v_{1'} \\ &= (H_{1'} - H_3) * v_1 / v_{1'} / (H_3 - H_5) \end{aligned} \tag{11}$$

MATHEMATICAL MODEL AND COMPUTER OPTIMIZATION

With the changes of temperature conditions, the calculation by hand will be enormous and tedious. Fortunately, more and more equations of refrigerant thermodynamic properties are well developed. So that a personal computer can be used in the calculation of optimization of this cycle.

The saturation pressure of refrigerant vapour can be calculated by following equation[2]:

$$P=4.46558+0.166907t+24.3664(t/100)^2 +16.1561(t/100)^3+3.4276(t/100)^4 \quad (12)$$

Where P is pressure (bar) and t is temperature (°C)..

The following equations[3] can be used for refrigerant enthalpies:

$$H_l=180.885+4.6225t+0.287168(t/10)^2 +13.836(t/100)^3+5.36124(t/100)^4 \quad (13)$$

$$H_g=1443.36+1.11051t-0.87654(t/10)^2 -32.7365(t/100)^3+11.9649(t/100)^4 \quad (14)$$

Where H_l is enthalpy of liquid refrigerant, KJ/KG.
and H_g is enthalpy of vapour refrigerant, KJ/KG .

Specific heat (KJ/KG-K) of ammonia at constant pressure is :

$$C_p=2.543+0.004643((t-10)+0.05015(t-10)^2) \quad (15)$$

for $10 <= t <= 52$ °C

The final temperature due to isentropic compression can be calculated by basic equation:

$$T_2=T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n}$$

Where T_2 is the discharge temperature of the compressor, K.
T_1 is the suction temperature of the compressor, K.
P_2 is the discharge pressure, Pa.
P_1 is the suction pressure, Pa.
n is the isentropic exponent.

With these equations, according to the practice of ammonia compression refrigeration plants in subtropics area, three temperature conditions (A-Ice making in winter and B-Ice making in summer, C-Air conditioning in summer) are selected. Equations (5-7) and (10-11) are solved using a computer program, as TIS varies. The control flow statements are used in the program, which make repeat calculation to be carried out, the step value of variable (TIS) is set to 1 °C. The results are shown in Tables 1 and 2 and Figures 3-5. In order to make the tables clear, a step value of 2 °C is shown in Tables 1 and 2.

Table 1. IN_{cop} vs. TIS under Different Conditions

Condensing Temperature °C	30	40	40
Evaporating Temperature °C	-15	-15	5
TIS \ IN_{cop} °C \ %	A	B	C
34		0.43	0.39
32		1.25	1.09
30		2.01	1.7
28		2.72	2.19
26		3.36	2.59
24	0.4	3.95	2.88
22	1.14	4.46	3.05
20	1.81	4.92	3.12
18	2.42	5.3	3.08
16	2.96	5.62	2.92
14	3.42	5.86	2.65
12	3.8	6.03	2.26
10	4.11	6.13	1.76
8	4.33	6.14	1.14
6	4.67	6.08	0.41
4	4.52	5.93	
2	4.48	5.7	
0	4.34	5.38	
-2	4.1	4.97	
-4	3.76	4.47	
-6	3.32	3.87	
-8	2.78	3.18	
-10	2.13	2.39	
-12	1.36	1.51	
-14	0.48	0.53	

CONCLUSIONS

Compared with the common single stage ammonia compression refrigeration cycle in the same condition, the COPs of the new cycle increase by 3--6 % under the three conditions of Table 1. The increments of the COPs vary with the TIS in a constant condition. The relationship between the increment of COP and TIS is non-linear. There is an optimum TIS where the increment of COP is maximum.

Table 2. TIS vs. Ratio of Volume Flow Rate under Different Conditions

Condensing Temperature °C	30		40		40	
Evaporating Temperature °C	-15		-15		5	
\ Ratio TIS \ °C \	V_n/V_m	V_z/V_f	V_n/V_m	V_z/V_f	V_n/V_m	V_z/V_f
34			0.996	1147	0.997	549
32			0.989	362	0.992	173
30			0.982	205	0.988	98
28			0.975	138	0.984	66
26			0.969	101	0.981	49
24	0.997	908	0.964	78	0.978	37
22	0.99	285	0.958	62	0.976	29.8
20	0.985	161	0.954	50.9	0.975	24.3
18	0.979	108	0.949	42.2	0.975	20.2
16	0.974	79	0.946	35.5	0.976	17
14	0.97	60.6	0.942	30.1	0.977	14.4
12	0.966	48	0.94	25.8	0.98	12.3
10	0.963	39	0.938	22.2	0.984	10.6
8	0.96	32.2	0.936	19.2	0.989	9.2
6	0.958	27	0.936	16.7	0.996	8
4	0.957	22.80	0.936	14.6		
2	0.957	19.4	0.937	12.8		
0	0.957	16.6	0.94	11.3		
-2	0.959	14.3	0.943	9.9		
-4	0.961	12.4	0.947	8.7		
-6	0.965	10.8	0.953	7.7		
-8	0.97	9.4	0.96	6.8		
-10	0.976	8.2	0.969	6		
-12	0.984	7.2	0.98	5.3		
-14	0.994	6.3	0.993	4.7		

The volume flow rates of the new cycle for all three conditions are less than that of the common cycle, i.e, the compressors which new cycle needs will be of less capacity for the same condition. The extra investment of the subcooler is not significant. So energy saving (saving of running cost) is obvious when the new cycle is applied.

The ratio of the volume flow rate of the evaporator to the subcooler decreases with the drop of the TIS, the relation with TIS being in direct proportion. In other words, the less is the TIS, the less is the ratio.

On the basis of conservation of energy, the heat rejection effect of the condenser is the sum of the refrigerating capacity and the power the cycle consumes. The increase of the COP of the new cycle means that the power consumed is decreased in the same refrigerating capacity. Then the heat rejection effect of the condenser in the new cycle is less than that in the common cycle. That is to say, the COP of the actual running can be improved further when a common cycle is reset to a new cycle.

DISCUSSION

According to Table 1, the higher the compression ratio of the cycle, the greater is the improvement of the COP when the new cycle is used. In other words, the new cycle has more benefit (energy saving) to the condition of higher compression ratio.

Further analysis of Table 1 can show that the saturation pressure of the optimum TIS is nearly equal to the optimum inter-pressure of two stage compression. Therefore, the optimum TIS of the new cycle can be determined by means of the method of optimum inter-pressure in two stage compression (the saturation pressure of optimum TIS = $(P_2 * P_1)^{1/2}$) because the COP of the new cycle in vicinity of the optimum TIS varies slightly.

From Table 1 and 2, when new cycle runs at the optimum TIS, the capacity of the main compressor is about 20 times that of the auxiliary compressor. The new cycle is easily carried out if a refrigerating plant has many compressors. When the number of compressors is less in a plant, the new cycle can not be run in the optimum TIS. In this case , a suitable TIS can be selected because the COP of the new cycle will always increase for different TIS.

When a new plant is designed, the new cycle is carried out very easily because a smaller compressor or the high stage part of integral two stage compressor can be selected as the auxiliary compressor. To a new plant, the increase of first cost using new cycle will be very small (especial in larger plants), and can be ignored. In an auxiliary circuit , the automatic expansion valve can be used. So the management of new cycle is essentially the same as a common cycle. Therefore, it is quite evident that the benefit can be obtained when a new plant uses the new cycle.

For large industrial refrigeration plants and for the condition of higher compressure ratio, the cost of rebuilding the current system to the new cycle will be recovered from the improvement of COP in a very short period. It is relatively easy to estimate the benefit in the different cases by using the Table 1.

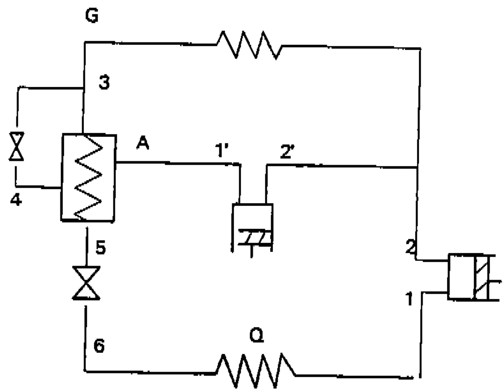

Fig. 1 New cycle

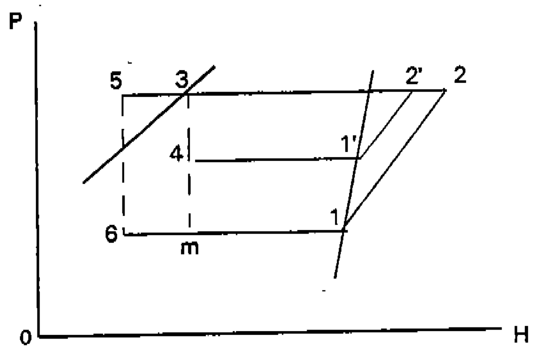

Fig. 2 Thermodynamic cycle

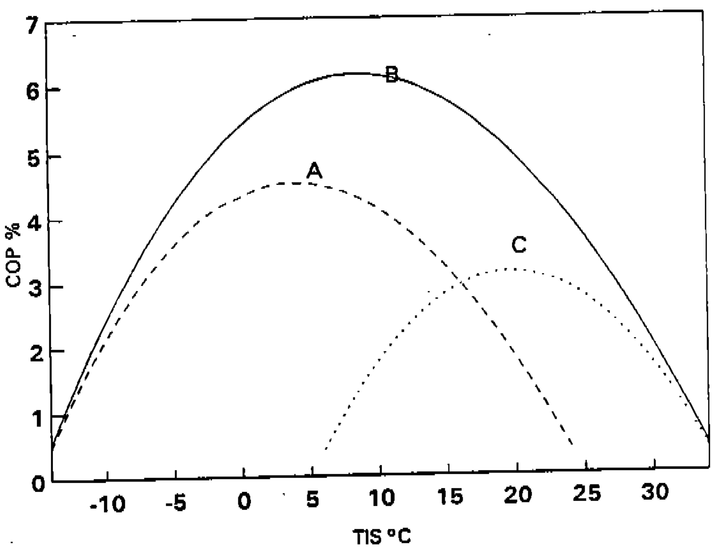

Fig. 3 INcop as TIS varies

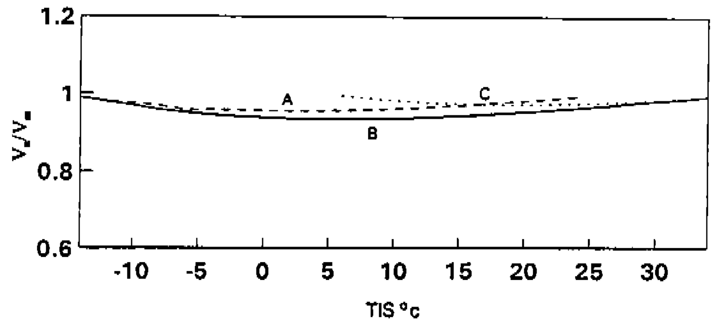

Fig. 4 Ratio of volume flow rate of new vs. common cycle

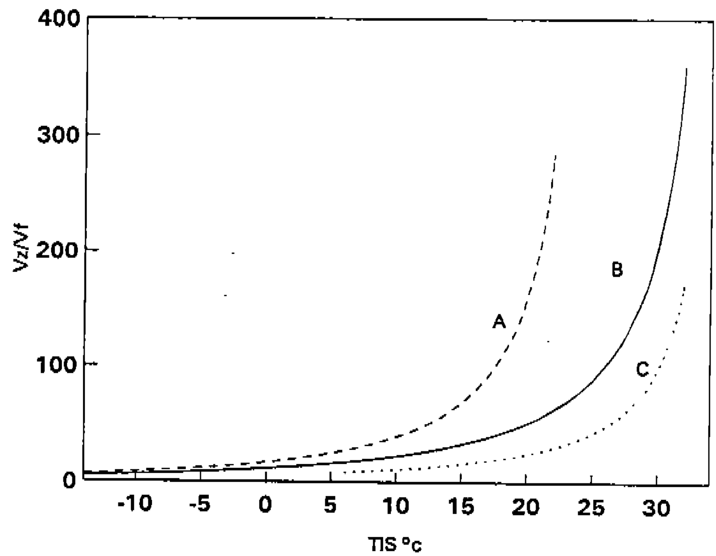

Fig. 5 Ratio of flow rate in new cycle

REFERENCES

1. ASHRAE Handbook 1989 Fundamentals.

2. V.K.Gupta & Manohar Prasad, 'Graphic estimation of design parameters for two-stage ammonia refrigerating systems parametrically optimized'. Mechanical Engineering Bulletin,1984, Vol.15. No.4.

3. V.K.Gupta & R S K Sinha, 'A computer simulation model for optimum performance of a three-stage ammonia refrigerating system with auxiliary compressors'. IE(I) Journal-ME, Vol 66,July 1985.