Ejector – Vapor Compression Refrigeration System Using Exergy analysis and R134a

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Abstract: The inevitable deletion of fossil fuels and the requirements of environment protection, the drawback of air conditioner with vapor compressor have become conspicuous. Recently, researchers are studying refrigeration and air conditioning principles using thermal power such as ejector system or absorption chillers… This paper proposed a comprehensive investigation of a combined ejector – vapor compression refrigeration system using exergy analysis. The exergy analysis is used to determine various losses from various parts of the system. The results show that ejector is the most irreversible part in the system (34.36%). That mean, the optimization of ejector's geometry to reduce the irreversibility is very important to improve the ejector system performance. Besides, the effect of operating parameters on the exergy loss also researched.

Index term: exergy analysis, combined ejector – vapor compression system, irreversible.

Nomenclature

e – Exergy at a point (kJ/kg)

h – specific enthalpy (kJ/kg)

m – mass flow rate (kg/s)

P – Pressure (bar)

s – Specific entropy (kJ/kgK)

T – temperature (K)

Q – Heat transfer rate (kW)

W – work transfer rate (kW)

I – exergy loss rate (kW)

c – Velocity of flow (m/s)

η – Nozzle efficiency

ω – ejector's entrainment ratio

Subscripts

1, 2 … – cycle points

0 – reference point

Is – isentropic process

g – Generator

c – condenser

Comp – compressor

p – pump

e – Evaporator

i – intercooler

Ex 1, ex 2 – expansion valve 1, expansion valve 2

ej – ejector

m – mixing point

d - diffuser

I. INTRODUCTION

Many researchers have used the first law of thermodynamics to analyze the energy of the system. In this paper, use analysis the integrated system of ejector-compressor air conditioning according to the second law of thermodynamics, also known as exergy analysis. Exergy is defined as the greatest amount of theoretical work when a system changes from the initial state to the death state. When a system is in thermal and mechanical equilibrium with its surroundings, it is said that the system is in death state. When the system is in death state, energy cannot be exploited, although it still has some specific value.

According to the second law of thermodynamics, the spontaneous processes always take place in a certain direction in practice: increase entropy and decrease exergy. Entropy and exergy are thermodynamic parameters used to determine the irreversibility of any real processes. Thus, exergy losses are an important parameter used to evaluate the effectiveness of a thermodynamic system. To optimize the energy of the system, exergy losses need to be minimized.

Alexis [1] analyzed the exergy of ejector system using water as a refrigerant. The study shows ejectors and condensers that the two devices have the largest exergy losses. When the generator pressure is 6 bars, the condenser temperature is 44-50 ^{o}C; the evaporator temperature is 4-8 C, the exergy efficiency remains at 0.17.

Pridasawas [2] conducted an exergy analysis of the ejector systems powered by solar energy. The irreversibility of the components in the system varies depending on the operating conditions of the system.

The collectors and ejectors are the two most exergy losses. With the given operating conditions such as solar radiation 700W/m, the evaporator temperature 10°C, cooling capacity 5kW, the environmental temperature 30°C, the optimum temperature of the generator is 80°C to the total exergy losses in the system is the smallest.

Yiping Dai et al. [3] conducted an exergy analysis to optimize the combined system produce electricity and refrigeration. The system is the combination of the Rankine cycle and the ejector cooling cycle. The results show that the two components cause the most exergy losses are the heat addition process and the ejector. Optimal exergy performance with given condition is 0.27.

Syed A. Tirmizi et al. [4] conducted an energy analysis for a single ejector system with different refrigerants. For the same operating conditions, COP is highest when using R717 and lowest when using R114. Exergy analysis also shows that the two components for the most exergy losses are the ejector and the generator.

II. SYSTEM ANALYSIS

A. Principle of the system:

Fig. 1. Describes the principle of ejector refrigeration system combined with vapor compression cycle. The system operates at four different pressure levels including: generator pressure, condenser pressure, intercooler pressure and evaporator pressure. Two sub-cycles in the system are connected to each other by intercooler. The intercooler acts as an evaporator for ejector Cycle and a condenser for vapor compression cycle.

As the generator receives heat and vaporized the refrigerant, a saturated vapor refrigerant with a high pressure goes into the ejector, and creates a primary flow that enters the nozzle. This primary flow entrains the secondary flow from the intercooler to produce an effect of compression. The total mixing refrigerant from the primary and secondary flows is then condensed at condenser. After leaving the condenser, a part of liquid refrigerant returns to generator, and the rest passes through the intercooler. In the intercooler, vapor refrigerant in the vapor compression cycle releases heat to the liquid refrigerant in the ejector Cycle. Then a liquid refrigerant in the intercooler evaporates, and is entrained into the ejector to form a closed cycle. Meanwhile, in the intercooler, the vapor refrigerant in the vapor compression cycle condenses before going through the throttle valve, evaporator and compressor to perform the conventionally vapor compression cycle. The T-s diagram is shown in Fig. 2.

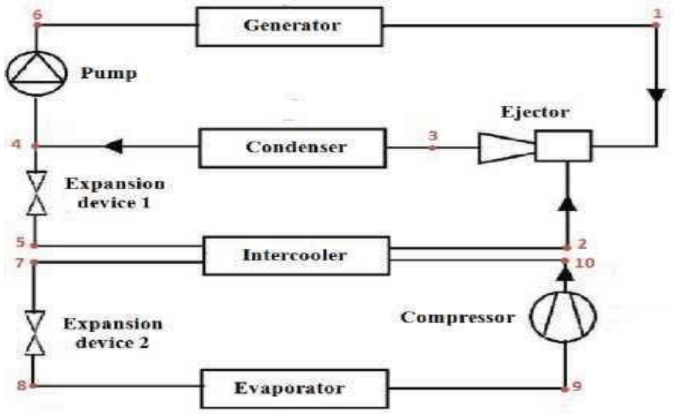


Figure 1. The combined Ejector-Vapour Compression System

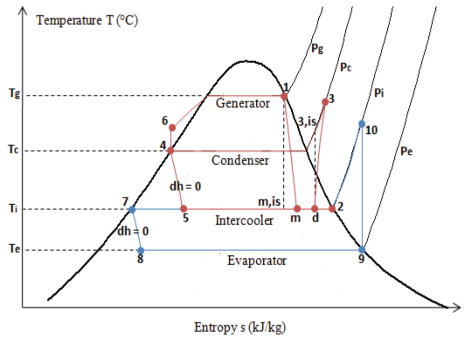


Figure 2. T-s diagram of combined system

Fig. 2. Shows the T-s graph of the combined refrigeration cycle with the numbered positions corresponding to the points on the first graph. Assume that the steam generated from the generator is directly put into the ejector without being superheated. The efficiency of this process can be chosen 0.9 [1]. Due to potential energy is transformed into kinetic energy, which corresponds to the reduction of the primary flow's pressure; the flow rate reaches the sonic speed in the nozzle throat, and then exceeds the sonic speed. After leaving the nozzle, primary flow entrains secondary flow (point (2)) and performs mixing process (point (d)). After blending, shock wave occurs inside a mixing chamber. The pressure increases (point (3)), and the speed suddenly drops to subsonic. In the diffuser, the refrigerant flow's kinetic energy is transformed into potential energy. Turbo charging process from point (d) to (3) is the irreversible process; (3, is) is the assumed point that the isentropic hypertension occurs. The performance of diffuser can be selected 0.8 [1]. After the mixing process, the vapor leaves the ejector in a superheated state before put into

The condenser to become a saturated liquid state (point 4). Part of the liquid is returned to the generator in a sub-saturated liquid state (point 6). The rest passes through the throttle valve to achieve a wet saturated vapor state (point 5) before it enters the intercooler.

Points 7, 8, 9, 10 represent refrigerant states in the vapor compression cycle. After leaving the intercooler, the refrigerant is in a saturated liquid state (point 7). Then the refrigerant runs through the throttle valve to perform a adiabatic process and achieve a wet saturated vapor state (Point 8). After passing through the evaporator, the refrigerant is completely vaporized into a dry saturated vapor state (point 9). It is then taken into the compressor to reach a superheated vapor state (point 10).

B. Energy balance equations:

The equation for energy balance at the mixing point of the ejector:

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3 \tag{1}$$

The equation for mass conservation and the law of Impulse at the mixing section of the ejector:

$$m_1 c_m = (m_1 + m_2) c_d \tag{2}$$

The nozzle's isentropic efficiency is defined as:

$$\psi_n = \frac{h_1 - h_m}{h_1 - h_{m,is}} \tag{3}$$

The velocity of the primary flow through the nozzle is Calculated as:

$$c_m = \sqrt{2.(h_1 - h_m)} \tag{4}$$

$$\psi_d = \frac{h_d - h_{3,is}}{h_d - h_3} \tag{5}$$

The velocity of the secondary flow is written as:

$$c_d = \sqrt{2.(h_3 - h_d)} \tag{6}$$

The pump increases the enthalpy of the liquid condensate going to the generator, so:

$$h_6 = h_4 + (P_6 - P_4).v_4 \tag{7}$$

Energy balance in the intercooler (assume that heat exchange effectiveness is 100%)

$$m_7 (h_{10} - h_7) = m_2 (h_2 - h_5) \tag{8}$$

The ejector's entrainment ratio is the ratio between the primary and the secondary mass flow rates:

$$\xi = \frac{m_2}{m_1} \tag{9}$$

Base on energy balance equations, Q_e, Q_g, W_p, W_{comp} are calculated:

$$Q_e = m_7 (h_9 - h_8) \tag{10}$$

$$Q_g = m_1 (h_1 - h_6) \tag{11}$$

$$W_p = m_1 (h_6 - h_4) \tag{12}$$

$$W_{comp} = m_7 (h_{10} - h_9) \tag{13}$$

C. Exergy analysis:

The exergy at a point is defined as:

$$e = (h - h_0) - T_0 (s - s_0) \tag{14}$$

Where h_0, s_0, T_0 are enthalpy, entropy and reference temperature (environmental temperature).

The exergy loss at steady state, in each component is calculated by the following equation:

$$I_g = Q_g \left(1 - \frac{T_0}{T_1}\right) - m_1 (e_1 - e_6) \tag{15}$$

$$I_c = m_3 (e_3 - e_4) - Q_c$$

$$I_{ej} = m_1 (e_1 + e_2) - m_3 e_3 \tag{16}$$

$$I_{ex1} = m_3 e_4 - m_2 e_5 \tag{17}$$

$$I_p = W_p - m_1 (e_4 - e_6) \tag{18}$$

$$I_i = m_9 (e_{10} - e_7) - m_2 (e_2 - e_5) \tag{19}$$

$$I_{ex2} = m_9 (e_7 - e_8) \tag{20}$$

$$I_e = Q_e \left(\frac{T_0}{T_e} - 1\right) - m_9 (e_9 - e_8) \tag{21}$$

$$I_{comp} = W_{comp} - m_9 (e_9 - e_{10}) \tag{22}$$

The total exergy loss is the sum of the exergy loss of each component.

Methodology:

A mathematical model is implemented in the EES software. The following assumption is used in the model:

+ The refrigerant is R134a for both sub-cycles.
+ Cooling capacity is 3.5 kW.
+ Generator temperature: 90°C
+ Condenser temperature: 45°C
+ Intercooler temperature: 25°C
+ Evaporator temperature: 5°C
+ Reference temperature: 20°C
+ Cooling water mean temperature: 30°C

Based on the system analysis and equations (1) – (22), a program is developed to calculate the cycle. The flow chart is presented in Fig. 3. The refrigerant properties are obtained Directly from EES's data.

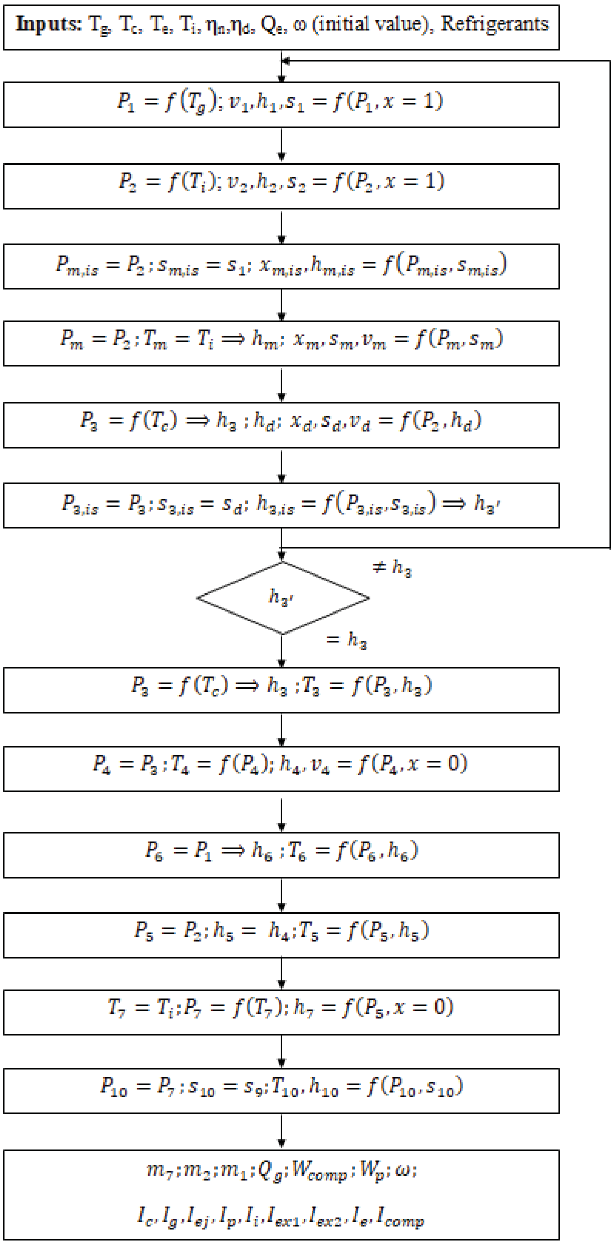


Figure 3. Flow chart of the program

III. RESULTS

In order to validate the present model, a comparison Between obtained numerical results from the present model and experimental data reported in the literature is made. Table 1 show a comparison between the predicted entrainment ratio by this model with the refrigerant R141b and the experimental data of Huang et al. [5]. The simulated results are in good agreement with the experimental data. The minimum error is 0.4% and the maximum error is under 12%.

TABLE 1. MODEL VALIDATION

T_g (°C)	T_e (°C)	T_c (°C)	ξ [5]	ξ	Error (%)
95	8	31.3	0.4377	0.424	-3.13
95	8	33	0.3937	0.378	-3.9878
95	8	33.6	0.3457	0.363	5.0043
95	8	34.2	0.3505	0.349	-0.4279
95	8	36.3	0.2814	0.303	7.6759
95	8	37.1	0.2902	0.287	-1.1027
95	8	38.6	0.2552	0.258	1.0971
95	8	41	0.2043	0.217	6.2163
95	8	42.1	0.1859	0.2	7.5847
90	8	33.8	0.3488	0.33	-5.3899
90	8	36.7	0.304	0.268	-11.842
90	8	37.5	0.2718	0.252	-7.2847
90	8	38.9	0.2246	0.227	1.06856
84	8	30.5	0.4241	0.377	-11.1059
84	8	33.6	0.3117	0.298	-4.3952
84	8	35.5	0.288	0.257	-10.7639
78	8	24.4	0.6227	0.549	-11.8356
78	8	26.9	0.4889	0.448	-8.3657
78	8	29.5	0.3922	0.364	-7.1902
95	12	33.1	0.4989	0.503	0.8218
95	12	34.5	0.4541	0.459	1.0790
95	12	38.7	0.3503	0.349	-0.3711
95	12	39.3	0.304	0.335	10.1973
90	12	32	0.5422	0.507	-6.4920
90	12	36	0.4034	0.384	-4.8091
90	12	39.5	0.2946	0.301	2.1724
84	12	28.9	0.635	0.583	-8.1889
84	12	32.4	0.479	0.449	-6.2630
84	12	36	0.3398	0.344	1.2360
78	12	25.7	0.7412	0.692	-6.6378

Exergy loss ratio of each components of the system.

Table 2 shows the thermodynamic parameters and exergy at each point in the cycle and table 3 presents the input exergy, output exergy, and exergy loss and exergy loss ratio of Each component in the system.

TABLE 2. THERMODYNAMIC PARAMETERS AND EXERGY AT EACH POINT IN THE CYCLE

State	t (°C)	p (bar)	h (kJ/kg)	s (kJ/kgK)	m (kg/s)	e (kJ/kg)
1	90	32.47	277.4	0.871	0.07338	31.0
2	25	6.658	264.2	0.9207	0.02546	3.223
3	45.56	11.61	274	0.9157	0.09884	14.51
4	45	11.61	115.8	0.423	0.09884	1.989
5	25	6.658	115.8	0.4184	0.02546	0.6496
6	45.33	32.47	116	0.4184	0.07338	2.174
7	25	6.658	86.4	0.3243	0.02096	0.1395
8	5	3.499	86.4	0.3286	0.02096	-1.128
9	5	3.499	253.4	0.929	0.02096	-10.03
10	27.44	6.658	266.7	0.929	0.02096	3.276
0	20	5.721	79.32	0.3006	-	0

Ejector has the biggest exergy loss (34.36%), followed by condenser (23.00%), compressor (17.79%), evaporator (11.98%)…

The friction of the flow inside the nozzle, the non-ideal adiabatic expansion causes the significant loss in ejector. So, optimal geometry design is needed to reduce the irreversibility in ejector.

TABLE 3. EXERGY INPUT, EXERGY OUTPUT, EXERGY LOSS AND EXERGY LOSS RATIO OF EACH COMPONENT

No.	Component	Exergy input (kW)	Exergy output (kW)	Exergy loss (kW)	Exergy loss ratio (%)
1	Generator	2.248	2.115	0.1695	5.41
2	Condenser	1.237	0.5161	0.7211	23.00
3	Ejector	2.511	1.434	1.077	34.36
4	Expansion valve 1	0.1966	0.01654	0.18	5.74
5	Pump	0.01361	-0.01361	0.02722	0.87
6	Intercooler	0.06574	0.06553	0.00021	0.01
7	Compressor	0.2789	-0.2789	0.5578	17.79
8	Expansion valve 2	0.002924	-0.02364	0.02657	0.85
9	Evaporator	0.1888	-0.1866	0.3754	11.98

Effect of operating parameters.

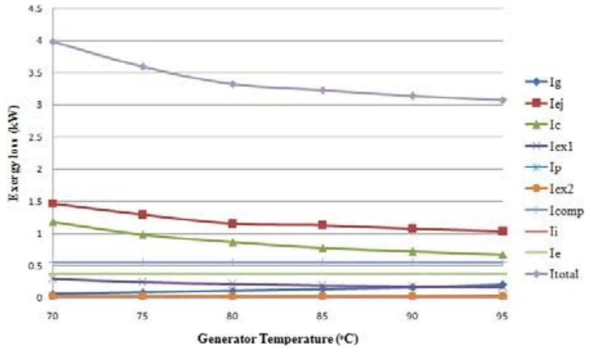
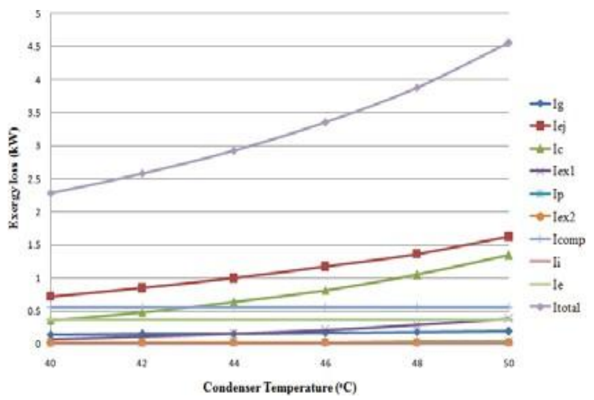


Figure 4. Effect of generator temperature on exergy loss

Fig. 4. Represents the effects of generator on the exergy loss of each component as well as the total exergy loss of the system. In this case, condenser, evaporator and intercooler temperatures are respectively 45°C, 5°C, 25°C; the generator changes from 70°C to 95°C.

Fig. 5. Represents the effects of condenser on the exergy loss of each component as well as the total exergy loss of the system. The analysis is performed with generator, evaporator and intercooler temperatures respectively at 90°C, 5°C and 25°C. The condenser temperature changes within the range between 40°C and 50°C.



For ejector, when condenser temperature goes up, the entrainment ratio drop, the expansion process inside the ejector is performed badly.

Fig. 6. Illustrates the effects of evaporator temperature on the exergy loss of each component as well as the total exergy loss of the system. The analytical data is set up with generator, condenser and intercooler temperatures at respectively 90°C, 45°C and 25°C. The evaporator temperatures vary from 2°C to 12°C.

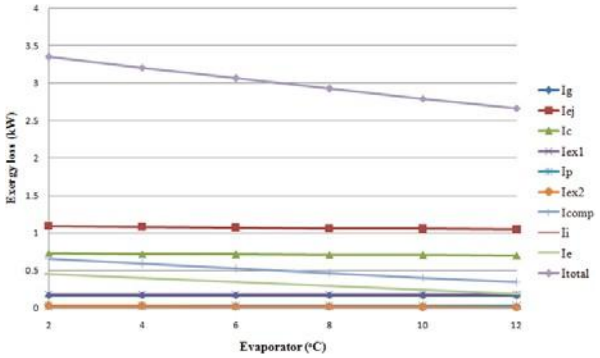


Figure 6. Effect of evaporator temperature on exergy loss

In this experiment, generator, condenser and evaporator temperatures are held constant respectively at 90°C, 45°C, 5°C; the intercooler temperature changes from 15°C to 35°C. The diagram Fig. 7. Represents the relationship between the intercooler temperature and exergy loss of each component as well as the total exergy loss of the system.

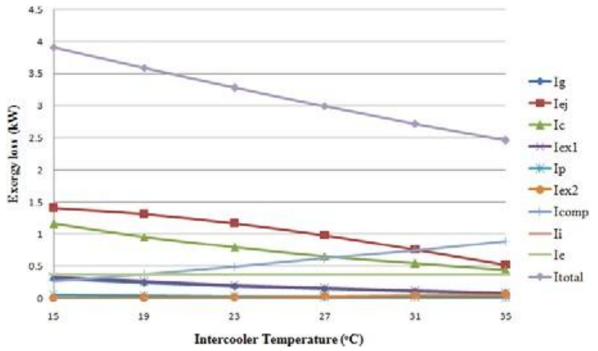


Figure 7. Effect of intercooler temperature on exergy loss

In this case, exergy loss of the components has different trends. The exergy loss of ejector and condenser decreases while the exergy loss of compressor increases. Because the exergy loss of ejector and condenser has bigger proportion, the total exergy loss has the same trend with the exergy loss of ejector.

IV. CONCLUSION

This paper, exergy analysis is used as a tool to analyze the performance of the ejector-vapor compression cycle. The effects of various operating parameters were investigated. In general, ejector has the biggest exergy lost in the system. Besides, generator, condenser, evaporator and intercooler temperature strongly affects on exergy loss. So, optimize the geometry ejector and choose the suitable operating parameters are two ways to reduce the exergy loss and improve the system's performance.

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