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The Theoretical Analysis and Selection of Suitable Refrigerants Working in The Combined Ejector-Vapour Compression System

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Abstract

With the innovation of air conditioning and cooling technology in order to make use of environmentally-friendly and renewable energy, ejector system has attracted much attention. ejector system is advantageous for its simplicity, reliability with low installation and operation cost; yet the main disadvantage is its low COP. The integration of ejector with vapor compression cycle can help improve ejector system's COP. In this article, the theoretical analysis is made in order to select the ideal refrigerants for the integration cycle. In addition, the effects of generator, evaporator, condenser and intercooler are considered. The results indicate that the ideal refrigerant pair for integration cycle is R600 or R600a on the ejector sub-cycle and R134a or R32 on the vapor compression sub-cycle

***Keywords**:* Ejector, Vapour Compressor, Refrigerant.

Nomenclature

COP	coefficient of performance
P	pressure (bar)
T	temperature (°C)
Q	heat rate (kW)
W	work rate (kW)
c	velocity (m/s)
h	specific enthalpy (kJ/kg)
η	isentropic efficiency
m	mass flow (kg/s)
s	specific entropy (kJ/kgK)
ω	flow entrainment ratio
x	quality

Subscripts

1, 2 …	cycle locations
m	mixing
d	diffuser
is	isentropic
n	nozzle
c	condenser
e	evaporator
g	generator
i	intercooler
p	pump
comp	compressor

1. Introduction

Refrigeration and air conditioning techonology has contributed significantly to the quality of human life. The common current refrigeration techonology uses air conditioner with a vapor compressor to vaporize chemically synthesized refrigerants. The advantage of this refrigeration principle is that the device is effective, inexpensive and compact compared to other air cooling principles.

However, in recent years, due to the inevitable depletion of fossil fuels and the requirements of environmental protection, the drawbacks of air conditioners with vapor compressor has become obvious. Refrigerant leakage into the environment could cause destruction of the ozone layer and worsen the greenhouse effect.

Recently, scientists are studying refrigeration and air conditioning principles using thermal power. Among these solutions, an air conditioning system using an ejector has many advantages such as compact device, simple and inexpensive production process, low operating costs, and eco-friendly refrigerants. In addition, the ejector can function with low heat grades such as solar energy or heat emitted from factories.

The weak point of the refrigeration and air conditioning cycles using an ejector is low COP. To improve the COP of this cycle, the ejector is combined into the vapor compression cycle. We can then both save energy and increase COP compared to the vapor-compressor refrigeration cycle.

Sokolov and Hershgal [1] first presented a plan to combine a compressor and an ejector to improve the COP of the ejector refrigeration. As for the first option, the compressor is directly connected to partly increase the refrigerant pressure before the refrigerant flows into ejector. The second option is that the compressor and ejector cycles are separated and work together as a cascade refrigeration system. The empirical and theoretical analyses shows that option 2 have more benefits: the system is more stable and free of oil suctioned from compressor into the ejector. In 1993, Sokolov and Hershgal [2] kept developing models integrating the ejector and the compressor cycle using refrigerant R114, in which the intercooler works as a heat and substance exchanger. They proved the system could work using solar energy. Similar systems are developed by Arbel and Sokolov [3] using refrigerant R142b.

Sun [4] set up an combined model including two separated cycles using solar energy. Water was used as a refrigerant in the ejector Cycle, while R134a was used in vapor compression cycle. The theoretical analysis results showed that the system could increase the COP by 50% compared to the traditional ejector cycle, and reduce the power consumption nearly twice as high as the traditional vapor compression cycle does. Sun [5] continued to evaluate the system using single or dual refrigerants. The results showed that the use of dual refrigerants would increase COP more than that of single-component refrigerants. In this case, the use of water in ejector Cycle and R21 in the vapor compressor cycle would maximize efficiency. Vidal and Colle [6] analyzed a similar system with refrigerants R141b for ejector cycle and R134a for and compressor cycle. The analytical result from TRNSYS-EES software was 105 m^2 flat plate collector the area for 10.5 kW, with 19°C intercooler temperature and 0.89 COP.

Rusly [7] presented a theoretical analysis to select refrigerants for the combined system between ejector cycle and compressor cycle. The analytical results pointed out that that the use of R245fa in ejector cycle and R141b in compressor cycle would achieve the highest efficiency.

The selection of the most suitable refrigerant for combined systems is a relatively complex issue because it depends on such factors as thermodynamic properties, environmental impact, availability, cost and compliance with the current regulations. In this paper, a program that analyzes the thermodynamics of combined cycle was developed to evaluate the performance of the system. The program is written by EES software.

2. System Analysis

2.1 Principle of 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.

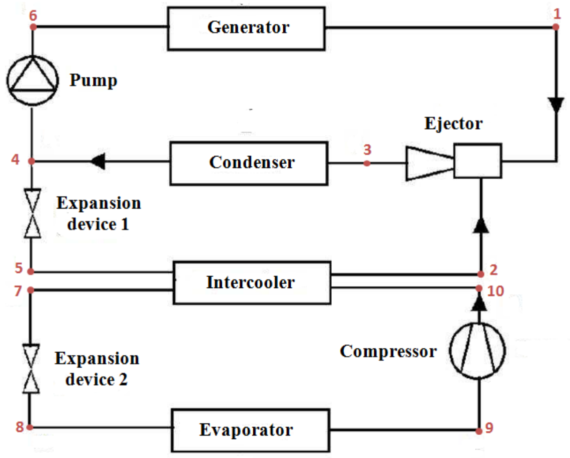

Fig. 1 The combined ejector-Vapour Compression system

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.

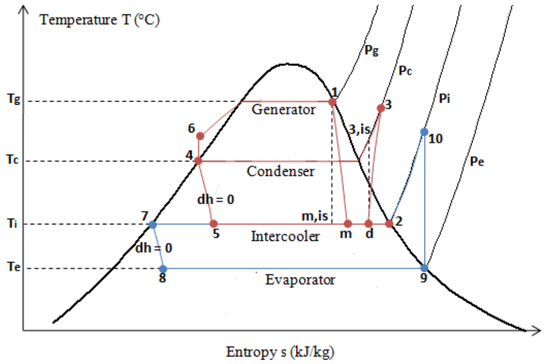

Fig. 2 T-s diagram of combined system

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. This steam going into the ejector is a dry saturated vapor that has a high pressure corresponding to point (1) on the T-s graph. After expanding in the ejector nozzle, vapor pressure dropped to a pressure corresponding to point (m) on the image. Point (m,is) is the assumed position if the decompression process is isentropic. The efficiency of this process can be chosen 0.9 [9]. 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 (b)). 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. Turbocharging 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 [9]. 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 evaporized into a dry saturated vapor state (point 9). It is then taken into the compressor to reach a superheated vapor state (point 10).

2.2 Equations

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

$$m_1h_1 + m_2h_2 = (m_1 + m_2)h_3 \qquad (1)$$

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

$$m_1c_m + m_2c_2 = (m_1 + m_2)c_d \qquad (2)$$

Assuming that the inlet area from the intercooler is large enough, the inlet velocity can be set to 0. Therefore, equation (2) can be written as:

$$m_1c_m = (m_1 + m_2)c_d \qquad (3)$$

The nozzle's isentropic efficiency is defined as:

$$\eta_n = \frac{h_1 - h_m}{h_1 - h_{m,is}} \qquad (4)$$

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

$$c_m = \sqrt{2.(h_1 - h_m)} \qquad (5)$$

The diffuser's isentropic efficiency is calculated as:

$$\eta_d = \frac{h_d - h_{3,is}}{h_d - h_3} \qquad (6)$$

The velocity of the secondary flow is written as:

$$c_d = \sqrt{2.(h_3 - h_d)} \qquad (7)$$

The pump increases the enthalpy of the liquid condensate going to the generator, therefore the enthalpy of point 6 is calculated as:

$$h_6 = h_4 + (P_6 - P_4)v_4 \qquad (8)$$

Assuming that energy loss is negligible in the intercooler, the energy balance equation can be written as:

$$m_7(h_{10} - h_7) = (h_2 - h_5)m_2 \qquad (9)$$

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

$$\omega = \frac{m_2}{m_1} \qquad (10)$$

Based on the energy conservation equations, the generator capacity, evaporator capacity, pump work and compressor work are calculated according to the following equations:

$$Q_e = m_7(h_9 - h_8) \qquad (11)$$

$$Q_g = m_1(h_1 - h_6) \qquad (12)$$

$$W_p = m_1(h_6 - h_4) \qquad (13)$$

$$W_{comp} = m_7(h_{10} - h_9) \qquad (14)$$

Mechanical COP is the ratio between the useful energy and total eletrical power:

$$COP_{me} = \frac{Q_e}{W_{comp} + W_p} \qquad (15)$$

Thermal COP is the ratio between the useful energy gained and the total input thermal energy:

$$COP_{th} = \frac{Q_e}{Q_g + W_{comp} + W_p} \approx \frac{Q_e}{Q_g} \qquad (16)$$

2.3 Develop a simulation program

Based on equations (1) to (16) and the cycle's thermodynamic properties shown in the T-s diagram, a calculating program is developed to simulate the integrating system using EES software. The inputs are generator temperature, condenser temperature, evaporator temperature, intercooler temperature, diffuser's isentropic efficiency, nozzle's isentropic efficiency, evaporator capacity, and refrigerants for the sub-cycles. The flow chart of the simulation program is shown in Fig. 3.

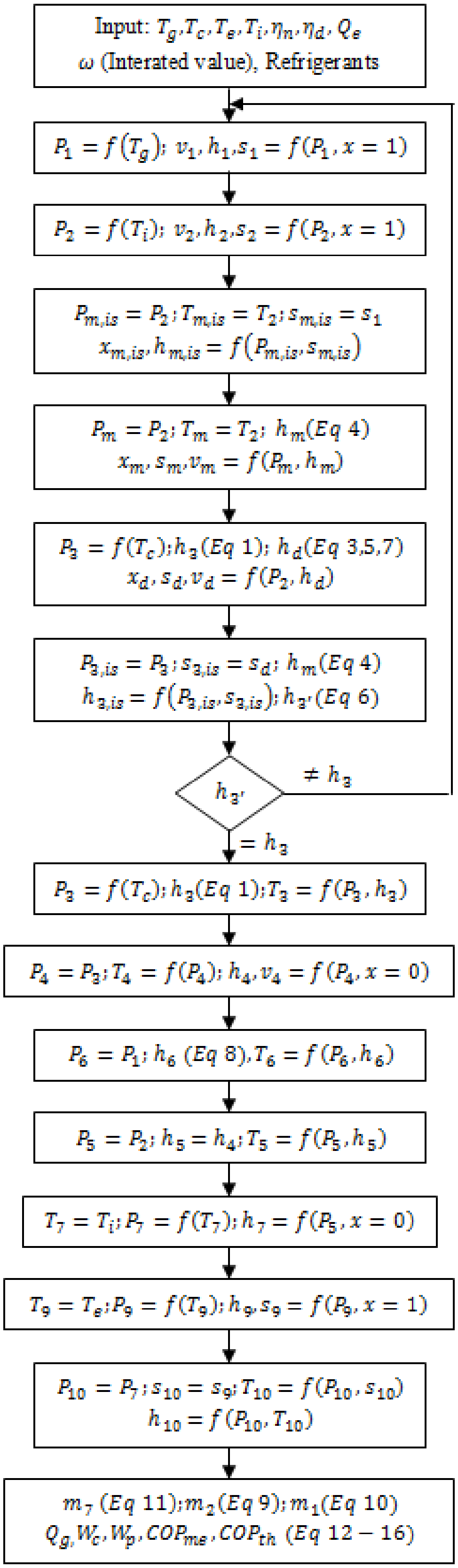

Fig. 3 Flow chart of the simulation program

3. Refrigerant Criteria

Refrigerants play a crucial role in the ejector refrigeration system. Many studies indicate that selection of refrigerants can affect the ejector's performance and design. The selection of refrigerants suitable for the combined system is challenging due to such factors as thermodynamic characteristics, chemical properties, environmental impacts, cost, compatibility and availability.

3.1 Environmental impacts

To assess the environmental impacts of refrigerants, two parameters were used: ODP and GWP. ODP is the ozone depletion potential, with base R11 (OPD = 1); GWP is the global warming potential, with base CO_2 (GWP = 1). According to these two parameters, many refrigerants were banned such as CFC group, or potentially banned with HCFC group being eliminated in 25 years for developing countries. Nowadays, HFC, mixing and natural refrigerants attract a large amount of attention.

3.2 Safety

Two major parameters used to assess the safety of refrigerants are toxicity and flammability. According to ASHRAE, there are 2 classes of toxicity (A – no toxicity; B – toxicity) and 3 groups of flammability characteristics (1 – no flame in normal condition; 2 – high flammability limit; 3 – low flammability limit)

3.3 Thermodynamic characteristics

The phase change in the generator can happen only if the refrigerant's critical temperature is higher than the generator temperature. Yet, a low critical temperature is necessary in order to reduce pump work and ensure high efficiency at a lower condenser's pressure.

In the generator, the refrigerant requires high latent heat of vaporization in order to decrease the refrigerant's flow rate in the system. On the contrary, a low latent heat of vaporization helps to improve the system's performance in the evaporator of vapor compression sub-cycle.

Lower costs of piping and compressor asks for lower specific volume of the refrigerant. High thermal conductivity and low viscosity is needed to reduce friction.

3.4 Slope of saturation vapor line

Based on the characteristics of the saturation vapor line on the T-s diagram, refrigerants are grouped into 2 types: dry refrigerants and wet refrigerants. For wet refrigerants, the slope of saturation vapor line on the T-s diagram is negative. In this case, if the refrigerants pass through the ejector in a saturated vapor state, droplets can be formed after the expansion in the nozzle of the ejector, leading to poor performance. Thus, wet refrigerants need to be heated into superheat before going through the ejector.

For dry refrigerants, the saturation vapor line has positive slope on the T-s diagram. In this case, phase change rarely happens during the expansion in the ejector's nozzle.

3.5 Performance parameters

Two parameters COP_{me} and COP_{th} are important in assessing the economic – technical efficiency of the system. COP_{me} is compared with COP of the traditional vapor compression cycle to illustrate the energy saving ability of the combined system. COP_{th} is compared with COP of the single ejector cycle. In many cases, the input heat source is solar energy or waste heat source, the COP_{th} is only for reference because the input energy is cost-free. However, if the COP_{th} is too small, which means that the input energy is too high, initial investment for equipments should be considered.

3.6 Availability, cost and compatibility

They are also important factors in selecting refrigerants. Table. 1 lists the different refrigerants used in the ejector sub-system.

Table. 1 Summary of different refrigerants used in the ejector sub-system.

R-no	**Group**	**Dry/ Wet fluid**	**Molecular weight (kg/kmol)**	**Heat of vaporisation (kJ/kg)**	**$T_{critical}$ (°C)**	**$P_{critical}$ (bar)**	**Safety Group**	**GWP**	**ODP**
R22	HCFC	Wet	86,47	233,0	96	49,74	A1	1500	0,05
R123		Dry	152,89	167,20	183,7	36,68	B1	93	0,02
R141b		Dry	117,0	221,28	204,2	42,1	A2	630	0,01
R134a	HFC	Wet	102,0	205,88	101,1	40,7	A1	1600	0
R152a		Wet	66,0	319,59	113,5	44,92	A2	140	0
R245fa		Dry	134,0	182,66	154,1	36,4	B1	820	0
R290		Wet	44,9	-	96,7	42,48	A3	20	0
R717	Natural	Wet	17,0	1197,14	113,3	132,3	B2	< 1	0
R600		Dry	58,1	398,69	152,0	37,9	A3	0	0
R600a		Dry	58,1	382,5	134,7	36,4	A3	20	0
R1234ze	HFO	Dry	114,04	202,0	109,3	36,4	A2	6	0

4. Results and discussion

4.1 Effect of generator temperature

The Fig. 4 and Fig. 5 represent the effects of the generator temperature on COP_{th} and COP_{me}. In this case, condenser, evaporator and intercooler temperatures are respectively 40°C, 5°C, 15°C; and the generator temperature changes from 70°C to 100°C. The analysis is carried out with three single refrigerants R141b, R134a and R717.

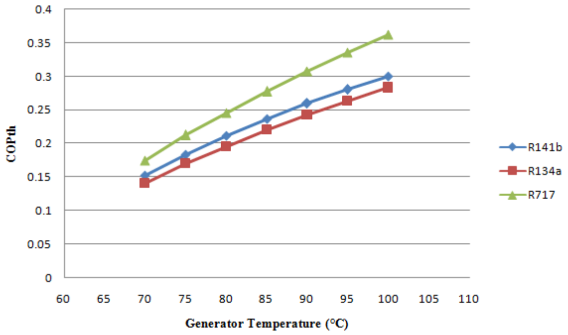

Fig. 4 Effect of the generator temperature on COP_{th}

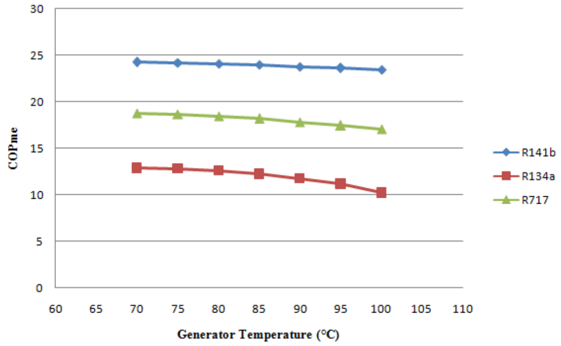

Fig. 5 Effect of the generator temperature on COP_{me}

Fig. 4 and Fig. 5 show that COP_{th} increases rapidly along with the generator's tempature, while COP_{me} has the tendency to slightly decrease. This is due to the fact that the generator temperature T_g, not having much effect on the vapor compression sub-cycle, has a direct effect on the ejector sub-cycle. When the generator temperature increases, the primary flow's pressure also increases, leading to the entrainment of a higher mass flow rate in the secondary flow. Entrainment ratio increases, and COP_{th} is thus raised. This process is similar to that in the single ejector cycle.

4.2 Effect of condenser temperature

Fig. 6 and Fig. 7 represents the effects of condenser temperature on COP_{th} và COP_{me} of the combined system. The analysis is performed with generator, evaporator and intercooler temperatures respectively at 80°C, 5°C và 15°C. The condenser temperature changes within the range between 40°C and 50°C. The chosen single refrigerants are R141b, R134a, R717.

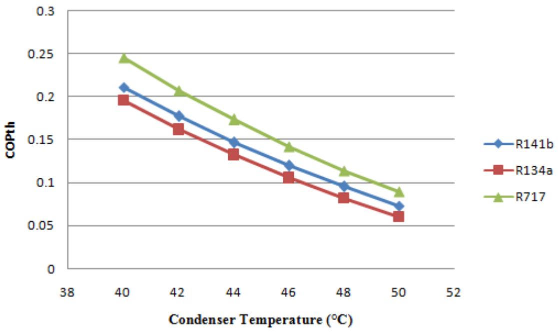

Fig. 6 Effect of the condenser temperature on COP_{th}

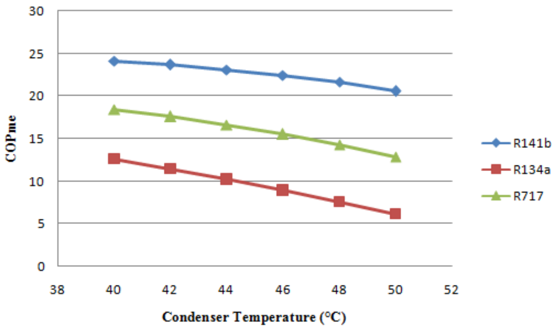

Fig. 7 Effect of the condenser temperature on COP_{me}

Similar to the effects of the generator temperature, the effects of condenser temperature on COP_{th} is quite strong while those on COP_{me} is minimal. Both COP_{th} and COP_{me} tend to decrease while condenser temperature rises. In ejector sub-cycle, this phenomenon is due to when condenser temperature increases, the primary flow rate rises more rapidly than does the secondary flow rate. This causes the entrainment ratio to drop, and thus COP_{th} also decreases. This process is similar to that in the single ejector cycle.

4.3 Effects of the evaporator temperature

Fig. 8 and Fig. 9 illustrate the effects of evaporator temperature on COP_{th} and COP_{me} of the system. The analytical data is set up with generator, condenser and intercooler temperatures at respectivly 80°C, 40°C và 15°C. The selected single refrigerants are R141b, R134a, R717.

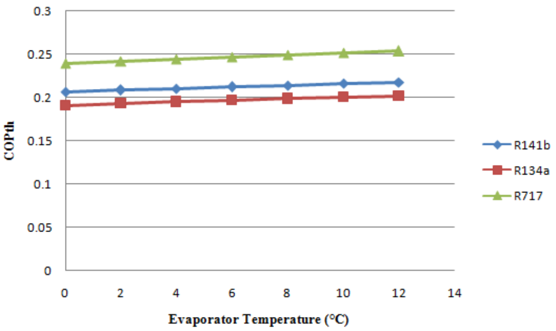

Fig. 8 Effect of the evaporator temperature on COP_{th}

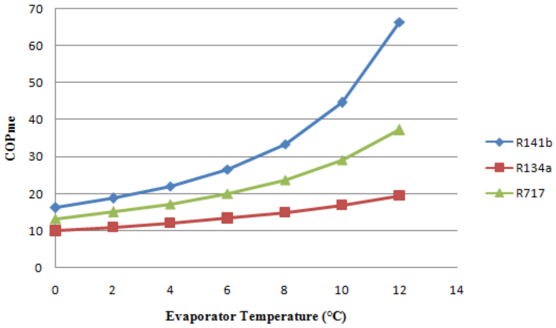

Fig. 9 Effect of the evaporator temperature on COP_{me}

Fig. 8 and Fig. 9 show that the generator temperature's effects are minimal on COP_{th}, yet significant on COP_{me}. When the generator temperature increases, COP_{me} tends to rise rapidly. Evaporator is a device that works totally within the vapor compression sub-cycle, thus it has significant effects on COP_{me}. When the evaporator temperature increases, evaporator pressure also increases, while the intercooler temperature and pressure are kept constant. As a result, compressor requires a smaller amount of work to complete the compression process. Because Q_e is assume to be constant, the increase of COP_{me} is logical.

5.4 Effects of intercooler temperature

Because intercooler is a transition device between the two sub-cycles, it has direct effects on both of them. Studies have pointed out that intercooler has an impact on the whole system. The design of optimal intercooler temperature plays a crucial role in the design of ejector combined system and compressor.

In this experiment, generator, condenser and evaporator temperatures are held constant respectively at 80°C, 40°C, 5°C; the intercooler temperature changes from 11°C to 35°C. The diagrams in Fig. 10 and Fig. 11 represents the relationship between the intercooler temperature and COP_{th}, COP_{me} with single refrigerants R141b, R134a and R717.

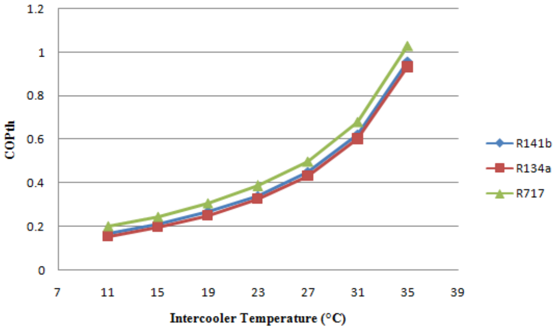

Fig. 10 Effect of the intercooler temperature on COP_{th}

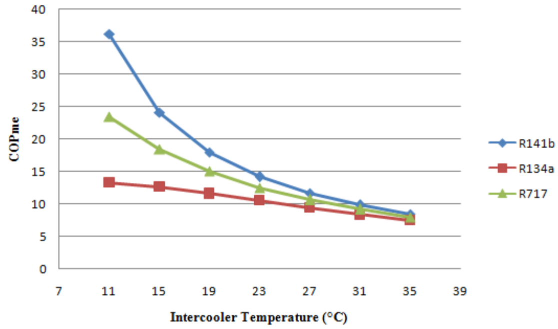

Fig. 11 Effect of the intercooler temperature on COP_{me}

COP_{th} and COP_{me} show two opposite trends when intercooler temperature increases. When the intercooler temperature rises, COP_{th} increases while COP_{me} decreases. The intercooler is both an evaporator for the ejector sub-cycle and a condenser for the vapor compression sub-cycle. When the intercooler temperature rises with other parameters held constant, primary mass flow rate in the ejector sub-cycle drops more rapidly than does the secondary mass flow rate. Therefore, the entrainment ratio increases. In the vapor compression sub-cycle, the compressor needs to provide a larger amount of work to acquire a higher pressure. The opposite also occurs if intercooler temperature decreases.

Many researchers believe that because COP_{th} and COP_{me} have opposite trends when intercooler temperature is raised, it is possible to find an optimal temperature value corresponding to the given condition. In reality, however, the assessment of different kinds of energies faces many challenges. In some cases mentioned above, thermal energy could be provided from certain free sources; thus COP_{th} is for reference only. The calculation and design of intercooler temperature is a economic – technical matter that asks for consideration of other related factors.

4.5 Comparison between COP_{th}, COP_{me} of combined cycle and COP of individual cycles

Fig. 12 demonstates the comparison between COP_{me} of the combined cycle and COP of the corresponding single vapor compression cycle. Fig. 13 demonstates the comparison between COP_{th} of the combined cycle and COP of the corresponding single ejector cycle. The generator, condenser, evaporator and intercooler temperatures are 80°C, 40°C, 0°C và 20°C respectively.

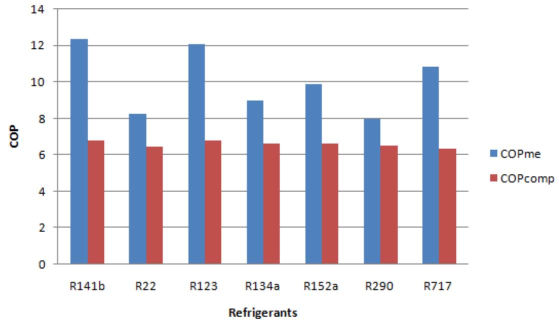

Fig. 12 Comparison between COP_{me} and COP_{comp}

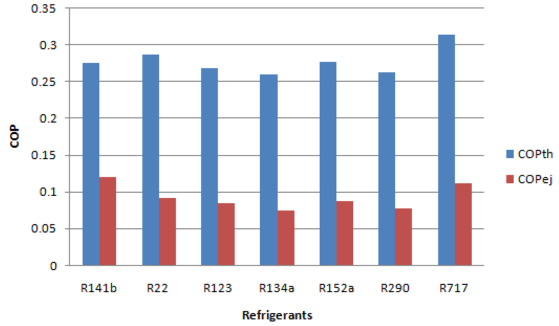

Fig. 13 Comparison between COP_{th} and COP_{ej}

In Fig. 12 and Fig. 13, both COP_{me} and COP_{th} are higher than COP of corresponding individual cycles. With the given input, COP_{me} is 1.3 to 1.8 times higher compared to COP of the traditional single vapor compression cycle; while COP_{th} is 2.3 to 3.4 times higher than that of the traditional single ejector cycle. However, as pointed out in 4.4, COP_{me} and COP_{th} have opposite trends, leading to the considerable ratio changes with different intercooler temperatures.

When intercooler temperature approaches condenser temperature, the COP_{me} of the combined system is close to the COP of the single vapor compression cycle; when intercooler temperature approaches evaporator temperature, the COP_{th} of the combined system is close to the COP of the traditional ejector cycle (with the same input temperatures). We can observe that the combined cycle raised both COP_{me} and COP_{th} compared with the individual cycles.

4.6 Refrigerant selection

As mentioned above, refrigerant selection meets with many difficulties, because it depends not only on the analytical results. With the perspective that there is no perfect dual refrigerant, the selected is the one that satisfies as many criteria as possible.

Because the vapor compression cycle is a popular cycle, common refrigerants on the market such as R22, R134a and R32 are acceptable in this case. With the ejector cycle, the different refrigerants in Table 1 are assessed one by one. The results are shown in Fig. 14, Fig. 15 and Fig. 16.

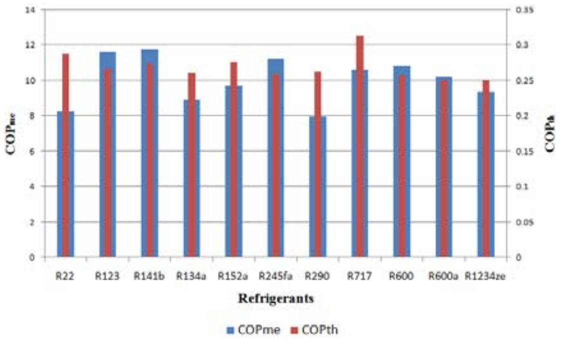

Fig. 14 Performance of various refrigerants selected for ejector sub-cycle while R22 is selected for Vapour Compression sub-cycle

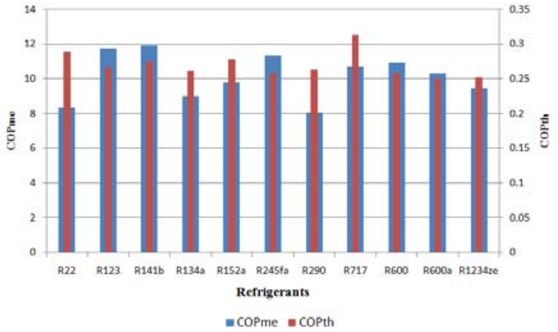

Fig. 15 Performance of various refrigerants selected for ejector sub-cycle while R134a is selected for Vapour Compression sub-cycle

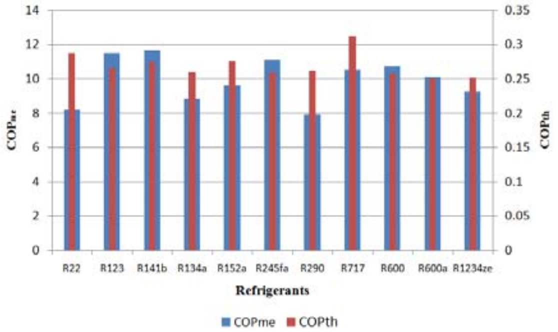

Fig. 16 Performance of various refrigerants selected for ejector sub-cycle while R32 is selected for Vapour Compression sub-cycle

The most popular refrigerant in Vietnam nowadays is R22. As one of the HCFC refrigerants, it will soon be replaced in the near future. The outcomes indicate that with R22 on the vapor compression side, R141b and R123 on the ejector side give the highest performance.

In case that the HCFC group is totally forbid, the selection of R134a and R32 on the vapor compression is a fair solution. While both of these two have zero ODP, R134a has higher GWP than does R32. On the ejector side, R22, R123, R141b are thus not allowed. The rest of the refrigerants, including R245fa, R717, R600, R600a, R1234ze result in relatively high COP. They have their own advantages and disadvantages. R245fa with high GWP is uncommon and unsafe. The natural refrigerant R717 has foul smell and high toxicity when leaked. R1234ze is a new refrigerant with such advantages as low ODP, low GWP, no flammability and no toxicity. It is expected to be applied in various refrigeration techonolgies in the future. However, R1234ze has yet to become popular and fully researched. R600 and R600a are considered to be the best choices, since they are natural and dry refrigerants, which give high performance and belong to the safety group A. However, their flammability must be noted.

5. Conclusion

The article presents the theoretical analysis of the combined system between ejector cycle and vapor compression cycle. A program, used to assess a system's performance, was developed based on balance equations using EES. The following conclusions were drawn from the theoretical analysis above:

1. When generator temperature increased, COP_{th} rose rapidly while COP_{me} dropped slightly.
2. When condenser temperature increased, both COP_{th} and COP_{me} appeared to drop; however, the effect of condenser temperature on COP_{th} was larger than that on COP_{me}.
3. Evaporator temperature had a large effect on COP_{me} but not on COP_{th}. COP_{me} increased rapidly as evaporator temperature increased rapidly.
4. Intercooler temperature is the most important value in this combined system. Intercooler temperature has large effects on both COP_{th} and COP_{me}. The two parameters had opposite trends when intercooler temperature changed: when intercooler temperature increased, COP_{th} rose while COP_{me} dropped.
5. COP_{th} and COP_{me} were higher than COP of single ejector cycle (under the same operating condition). The result indicated that the integrated cycle both raised COP for ejector cycle and saved energy for vapour compression cycle.
6. With the given criteria, R134a or R32 on the vapor compression side, and R600 or R600a on the ejector side were the most satisfactory options.

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