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Performance Variation of Enhanced Vapor Injection System#

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ABSTRACT

The enhanced vapor injection system (EVIS) has received considerable attention for its energy efficiency. In this paper, the EVIS is in-depth studied theoretically and numerically. First, the EVIS is equated to two cascaded single-stage systems. Then, the expression of the EVIS COP as a function of the intermediate temperature is obtained and numerically analyzed based on a new assumption of thermal perfection. The results show that there is not always an optimal intermediate temperature for the EVIS. Under different compressor isentropic efficiencies, the EVIS performance shows four different trends with the intermediate temperature. This finding will be of great value for the design and maintenance of EVIS and all types of two-stage compression systems.

Keywords: intermediate temperature, enhanced vapor injection system, thermodynamic analysis

NONMENCLATURE

Abbreviations	
a , *b* , *c*	Fitting factors
COP	Coefficient of performance
EVIS	Enhanced vapor injection system
h	Enthalpy
m	Mass flow rate
Q	Cooling or heating capacity
T	Temperature
TCSS	Two cascaded single-stage systems
W	Power consumption
Greeks	
β	COP / COP_{max}
η	Thermal perfection
Subscripts	
Carnot	Carnot cycle
cond	Condensing
evap	Evaporating
H	High stage or Highest
int	Intermediate
is	Isentropic
L	Low stage or Lowest
liq	Liquid
p	Isobaric process

1. INTRODUCTION

As an energy-saving and environment-friendly technology, the heat pump has already played an important role in many fields, such as buildings and transportation. The energy efficiency of the heat pump is affected by the operating conditions, and the performance of the conventional single-stage compression heat pump decreases significantly under low temperatures. Comparatively, the enhanced vapor injection system (EVIS) has received more attention due to its better low-temperature adaptability [1].

In an EVIS, part of the refrigerant from the condenser outlet is injected into the mix chamber of the EVI compressor to increase the system refrigerant mass flow rate and reduce the compressor discharge temperature. Compared to a single-stage compression system, the compression process of the refrigerant in the EVIS is nearer to the isothermal process [2].

Many studies have demonstrated the energy saving benefits of the EVIS. Liu et al. [3] applied the enhanced vapor injection technology to an air source heat pump water heater and achieved an improvement in performance and a reduction in exhaust temperature. Li et al. [4] used an EVIS to improve low temperature heating performance of an electric vehicle heat pump system. Cheng et al. [5] proposed a heat pump system for closed-loop drying, which employed an enhanced vapor injection compressor.

The performance of the EVIS is related to the intermediate pressure or temperature (p.s. the intermediate temperature indicates the saturation temperature of the injected refrigerant). Ma & Li [6] analyzed the effect of the intermediate pressure on the performance by using the experimental data. They concluded that the optimal intermediate pressure is

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between 1.1 and 1.3 times of the geometric mean of the high and low pressures. Zeng et al. [7] concluded that the value of the optimal intermediate pressure depends on the optimization objective. Jiang [8] found that the optimal intermediate pressure is always biased towards making the high-efficiency compression process consume more power. Most of these studies are based on experiments to analyze the optimal operation of the EVIS, and there are relatively few theoretical studies.

In this paper, the relationship between the intermediate temperature and the performance of the EVIS will be investigated. Under different compressor isentropic efficiencies, the EVIS performance is found four different trends with the intermediate temperature. It is valuable for the design and maintenance of EVIS and even other two-stage compression systems.

2. MODEL SIMPLIFICATION AND ASSUMPTIONS

2.1 Model simplification with cycle separation method

There are different structures of EVIS. Here we only discuss the common EVIS with a flash tank as shown in Fig. 1 Based on the cycle separation method [9], the EVIS is equivalent to two cascaded single-stage systems.

The *p-h* diagram of EVIS is shown in Fig. 2 (a), and the p-h diagram of the two cascaded single-stage systems (TCSS) is shown in Fig. 2 (b). In the TCSS, the condensing pressure of the low-pressure system and the evaporating pressure of the high-pressure system are equal to the intermediate pressure. The same numbered state points in Fig. 2 (a) and Fig. 2 (b) correspond to the same refrigerant states. m_L denotes the refrigerant mass flow rate of the low-pressure system, and m_H denotes the refrigerant mass flow rate of the high-pressure system.

Cooling capacity and power consumption of the EVIS can be expressed as

$$Q_{\mathrm{EVIS,cooling}} = m_{\mathrm{L}}\left(h_1 - h_9\right) \tag{1}$$

$$W_{\mathrm{EVIS}} = m_{\mathrm{L}}\left(h_2 - h_1\right) + m_{\mathrm{H}}\left(h_5 - h_4\right) \tag{2}$$

Energy equation in mixing chamber of the EVI compressor is given by

$$m_{\mathrm{H}} h_4 = m_{\mathrm{L}} h_2 + \left(m_{\mathrm{H}} - m_{\mathrm{L}}\right) h_3 \tag{3}$$

Energy equation in flash tank is given by

$$m_{\mathrm{H}} h_7 = m_{\mathrm{L}} h_8 + \left(m_{\mathrm{H}} - m_{\mathrm{L}}\right) h_3 \tag{4}$$

COP of the EVIS can be expressed as

$$\mathrm{COP}_{\mathrm{EVIS,cooling}} = \frac{h_1 - h_9}{h_2 - h_1 + \dfrac{\left(h_2 - h_8\right)\left(h_5 - h_4\right)}{h_4 - h_7}} \tag{5}$$

Cooling capacity and power consumption of the low-pressure system can be expressed as

$$Q_{\mathrm{L,cooling}} = m_{\mathrm{L}}\left(h_1 - h_9\right) \tag{6}$$

$$W_{\mathrm{L}} = m_{\mathrm{L}}\left(h_2 - h_1\right) \tag{7}$$

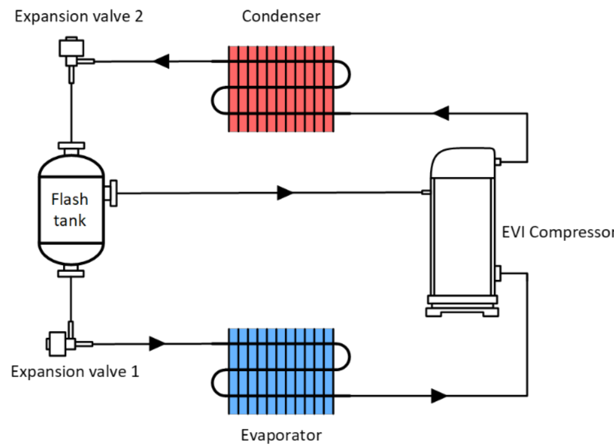

Fig. 1 Flash-tank enhanced vapor injection system

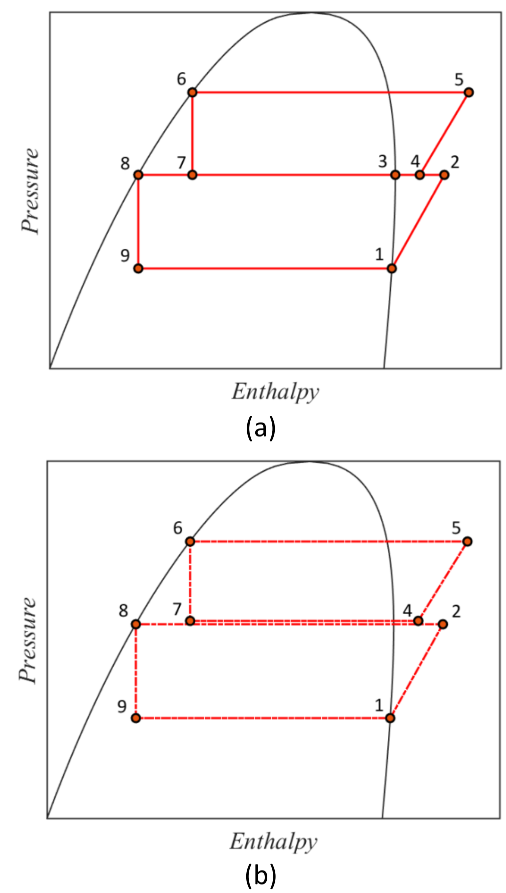

Fig. 2 Process of enhanced vapor injection system in p-h diagram

Cooling capacity and power consumption of the high-pressure system can be expressed as

$$Q_{\mathrm{H,cooling}} = m_{\mathrm{H}}\left(h_4 - h_7\right) \tag{8}$$

$$W_{\mathrm{H}} = m_{\mathrm{H}}\left(h_5 - h_4\right) \tag{9}$$

Heating capacity of the low-pressure system is equal to cooling capacity of the high-pressure system

$$m_{\mathrm{L}}\left(h_1 - h_9\right) + m_{\mathrm{L}}\left(h_2 - h_1\right) = m_{\mathrm{H}}\left(h_4 - h_7\right) \tag{10}$$

COP of the TCSS can be expressed as

$$\mathrm{COP}_{\mathrm{TCSS,cooling}} = \frac{h_1 - h_9}{h_2 - h_1 + \dfrac{\left(h_2 - h_8\right)\left(h_5 - h_4\right)}{h_4 - h_7}} \tag{11}$$

Obviously, COPs of the EVIS and the TCSS are identical.

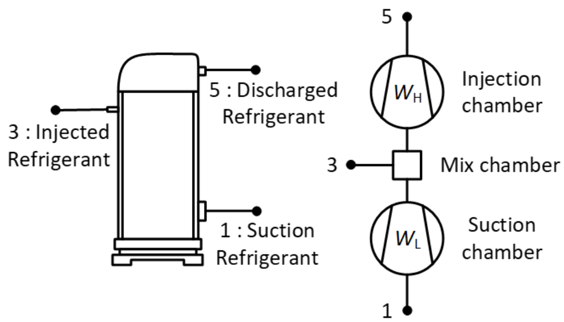

Fig. 3 Schematic of compressor

An EVI compressor can be considered as consisting of three parts[10]: suction chamber, mix chamber and injection chamber, as shown in Fig. 3. According to the previous derivation, the EVIS can be equated to the TCSS. Accordingly, the compression process in suction chamber of the EVI compressor can be replaced by the compression process of the low-pressure system, and the compression process in injection chamber of the EVI compressor can be replaced by the compression process of the high-pressure system. The equivalent simplified system is shown in Fig. 4 and will be further analyzed.

2.2 New assumption of thermal perfection

Thermal perfection [11,12] is used by as a modeling aid. The optimal intermediate temperature should be the geometric mean of the high and low temperatures. However, it is debatable that in these literatures the thermal perfection is set as a constant that doesn't vary with operating conditions.

Table. 1 Parameters for a single-stage system

Parameters	Value
Refrigerant	R410A
Evaporating temperature / °C	0 - 20
Condensing temperature / °C	40 - 60
Isentropic efficiency	1 , 0.7 , 0.4
Superheat / K	5
Subcooling / K	0

A single-stage compression system is constructed according to the parameters in Table. 1, and the thermal perfection is calculated at different working conditions. Calculation results are shown in Fig. 5.

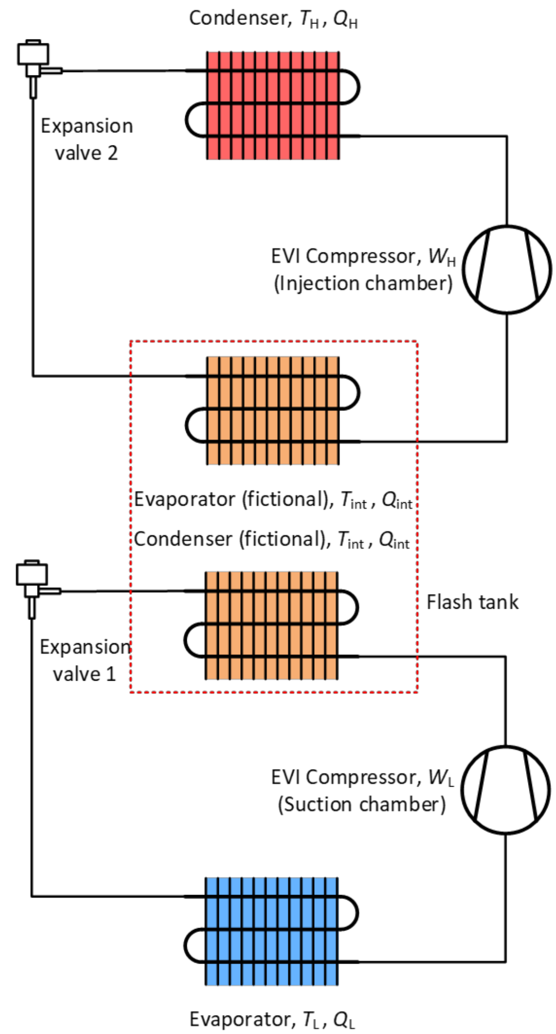

Fig. 4 Simplified system

It can be seen that the thermal perfection of the single-stage refrigeration compression system varies approximately linearly with the evaporating and condensing temperatures. We have also carried out calculations with a typical low-temperature refrigerant R23 and a typical high-temperature refrigerant R245fa as the working fluid in their respective temperature ranges. The results are consistent with the conclusion in Fig. 5.

The thermal perfection of single-stage refrigeration compression system can be interpreted as follows.

Power consumption per unit mass flow rate can be approximated as following [13]

$$W \approx \frac{1}{\eta_{\mathrm{is}}} i_{\mathrm{evap}} \frac{T_{\mathrm{cond}} - T_{\mathrm{evap}}}{T_{\mathrm{evap}}} \tag{12}$$

Cooling capacity per unit mass flow rate can be approximated as following

$$Q_{\text{cooling}} \approx i_{\text{evap}} - c_{\text{p,liq}}\left(T_{\text{cond}} - T_{\text{evap}}\right) \tag{13}$$

The thermal perfection can be expressed as

$$\begin{aligned} \eta_{\text{cooling}} &= \eta_{\text{is}} + \eta_{\text{is}} \frac{c_{\text{p,liq}}}{i_{\text{evap}}} T_{\text{evap}} - \eta_{\text{is}} \frac{c_{\text{p,liq}}}{i_{\text{evap}}} T_{\text{cond}} \\ &= aT_{\text{evap}} + bT_{\text{cond}} + c \end{aligned} \tag{14}$$

Eq.(14) represents an nearly linear relationship over a certain temperature range. In addition, the thermal perfection is influenced by the isentropic efficiency of the compressor and the properties of the refrigerant, which is also consistent with the general perception.

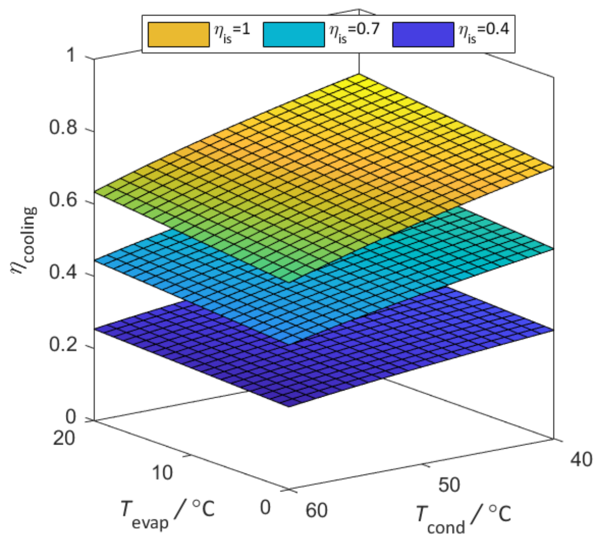

Fig. 5 η_{cooling} of a single-stage compression system

3. THERMODYNAMIC MODEL

Reversed Carnot cycle efficiency is the highest efficiency that can be achieved by a refrigeration system. Thermal perfection indicates the degree to which an actual system tends towards a reversed Carnot cycle under the current operating conditions.

According to the second law of thermodynamics for the system shown in Fig. 4

$$W_{\text{Carnot,L}} = \frac{T_{\text{int}} - T_{\text{L}}}{T_{\text{L}}} Q_{\text{L}} \tag{15}$$

$$W_{\text{Carnot,H}} = \frac{T_{\text{H}} - T_{\text{int}}}{T_{\text{int}}} Q_{\text{int}} \tag{16}$$

where Q_{L} represents the cooling capacity of the low-pressure system, Q_{int} represents the heating capacity of the low-pressure system or the cooling capacity of the high-pressure system, T_{L} represents the lowest temperature and T_{H} represents the highest temperature.

According to the definition of thermal perfection

$$\eta_{\text{L,cooling}} = W_{\text{Carnot,L}} / W_{\text{L}} \tag{17}$$

$$\eta_{\text{H,cooling}} = W_{\text{Carnot,H}} / W_{\text{H}} \tag{18}$$

The thermal perfection can also be expressed as

$$\eta_{\text{L,cooling}} = a_{\text{L}}\left(T_{\text{int}} + b_{\text{L}}\right) \tag{19}$$

$$\eta_{\text{H,cooling}} = a_{\text{H}}\left(T_{\text{int}} + b_{\text{H}}\right) \tag{20}$$

where a_{L} and a_{H} are influenced by the isentropic efficiency of the compressor. That is, they are influenced by the isentropic efficiency of the compression process in suction and injection chambers of the EVI compressor.

Power consumption can be expressed as

$$W_{\text{L}} = \frac{1}{a_{\text{L}}\left(T_{\text{int}} + b_{\text{L}}\right)} \frac{T_{\text{int}} - T_{\text{L}}}{T_{\text{L}}} Q_{\text{L}} \tag{21}$$

$$W_{\text{H}} = \frac{1}{a_{\text{H}}\left(T_{\text{int}} + b_{\text{H}}\right)} \frac{T_{\text{H}} - T_{\text{int}}}{T_{\text{int}}} Q_{\text{int}} \tag{22}$$

By the first law of thermodynamics

$$Q_{\text{int}} = Q_{\text{L}} + W_{\text{L}} \tag{23}$$

$\text{COP}_{\text{cooling}}$ can be expressed as

$$\begin{aligned} \frac{1}{\text{COP}_{\text{cooling}}} &= \frac{T_{\text{int}} - T_{\text{L}}}{a_{\text{L}} T_{\text{L}}\left(T_{\text{int}} + b_{\text{L}}\right)} + \frac{T_{\text{H}} - T_{\text{int}}}{a_{\text{H}} T_{\text{int}}\left(T_{\text{int}} + b_{\text{H}}\right)} \\ &+ \frac{\left(T_{\text{int}} - T_{\text{L}}\right)\left(T_{\text{H}} - T_{\text{int}}\right)}{a_{\text{L}} a_{\text{H}} T_{\text{L}} T_{\text{int}}\left(T_{\text{int}} + b_{\text{L}}\right)\left(T_{\text{int}} + b_{\text{H}}\right)} \end{aligned} \tag{24}$$

Eq.(24) shows the curve of COP variation with the intermediate temperature. According to the previous opinion, there must be an optimal intermediate temperature. However, different values of the isentropic efficiency of the compression process in suction and injection chambers of the EVI compressor cause a_{L} and a_{H} to change, which in turn causes different trends between the EVIS COP and the intermediate temperature.

4. NUMERICAL ANALYSIS

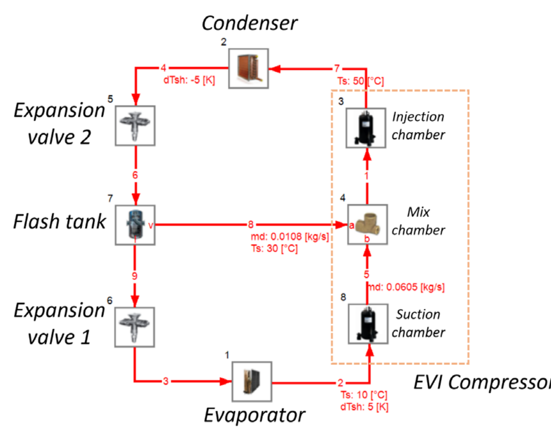

Fig. 6 EVIS model in GREATLAB

To verify the four different trends between the EVIS COP and the intermediate temperature, numerical analysis was carried out by the simulation software GREATLAB [14,15]. The interface of GREATLAB is shown in Fig. 6. The conditions are shown in Table. 2.

Table. 2 Simulation parameters

Parameters	Value
Refrigerant	R410A
Evaporating temperature / °C	10
Condensing temperature / °C	50
Intermediate vapor injection temperature	15 - 45
Isentropic efficiency of suction chamber	0.85 , 0.5 , 0.15
Isentropic efficiency of injection chamber	0.85 , 0.5 , 0.15
Superheat / K	5
Subcooling / K	5

Define the coefficient β as

$$\beta = \frac{\mathrm{COP_{cooling}}}{\mathrm{COP_{cooling,max}}}, \beta \in \left(0,1\right] \tag{25}$$

The variation curve of β with the intermediate temperature is plotted as shown in Fig. 7. $\eta_{is,SC}$ represents the isentropic efficiency of the compression process in suction chamber of the EVI compressor. $\eta_{is,IC}$ represents the isentropic efficiency of the compression process in injection chamber of the EVI compressor.

As shown in Fig. 7, there are four different trends between the COP and the intermediate temperature that accompany the change in the isentropic efficiency of the compression process in suction and injection chambers of the EVI compressor.

(1) As the intermediate temperature increases, the COP first increases and then decreases. There is an optimal intermediate temperature. This trend occurs when $\eta_{is,SC}$ and $\eta_{is,IC}$ are similar and neither of them is low.

(2) As the intermediate temperature increases, the COP increases. Optimal performance is achieved when the intermediate temperature is equal to the condensing temperature. This feature occurs when $\eta_{is,SC}$ is significantly higher than $\eta_{is,IC}$.

(3) As the intermediate temperature increases, the COP decreases. Optimal performance is achieved when the intermediate temperature is equal to the evaporating temperature. This feature occurs when $\eta_{is,IC}$ is significantly higher than $\eta_{is,SC}$.

(4) As the intermediate temperature increases, the COP first decreases and then increases. There is a worst intermediate temperature. This trend occurs when $\eta_{is,SC}$ and $\eta_{is,IC}$ are similar and both are low.

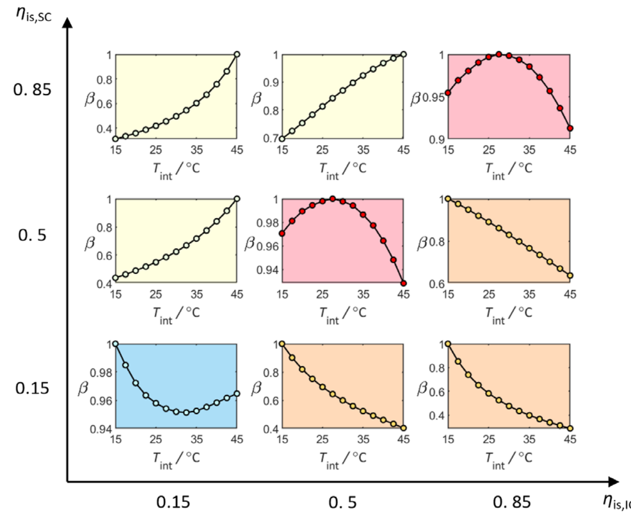

Fig. 7 Variation of COP to intermediate temperature

5. CONCLUSIONS

Based on theoretical and numerical analysis, the relationship between the performance and the intermediate temperature of the EVIS was investigated in this paper. The main conclusions are as follows.

The refrigeration thermal perfection of the single-stage compression system varies approximately linearly with the evaporating and condensing temperatures.

There is not always an optimal intermediate temperature for EVIS. There are four different trends between the performance and the intermediate temperature that accompany the change in the isentropic efficiency of the compression process in suction and injection chambers of the EVI compressor.

When the efficiency of the EVI compressor changes significantly, the performance of the EVIS may be lower than that of a single-stage compression system under the same operating conditions. It's worth noting that this phenomenon also occurs in a two-stage compression system. Therefore, when designing an EVIS or a two-stage compression system, it is necessary to have the ability to switch to a single-stage compression system.

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