SIMULATION ANALYSIS OF CO_2 HEAT PUMP WATER HEATERS : COMPARATIVE WITH OTHER NATURAL WORKING FLUIDS

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ABSTRACT: The natural working fluids are interested to replace CFCs, HCFCs and HFCs refrigerants because they are not harmful to the environment. The natural fluids considered are hydrocarbon substances, water, air, ammonia and carbon dioxide and they are used as refrigerants in a heat pump water heater system to produce high temperature water. The efficiency of R-744(CO_2) system is found to be lowest but it can produce highest water temperature at low evaporating temperature. For the other systems, they have higher coefficient of performance and produced the hot water temperature less than that of the R-744 system. Although, R-744 is the worse of efficiency but it is a safe working fluid and it is non-flammable and non-toxic compared with the other natural working fluids.

Key words: natural fluid, heat pump, water heater

1. INTRODUCTION

More than fifty year ago, the CFCs and HCFCs refrigerants are used as working fluids in refrigeration, air conditioning and heat pump systems. But CFCs and HCFCs destroy ozone layer and they impact global warming. For the short term replacement, the HCFCs and HFCs working fluids are used instead of the CFCs. The HCFCs have low ozone depletion potential (ODP) but they are still high global warming potential (GWP) and will phased out in the near future. The natural working fluids are the long term replacement for use. They are halogen-free natural working fluids and environmentally benign due to their very low or equal zero of ODP and GWP. The natural working fluids consist of ammonia, air, water, carbon dioxide and hydrocarbon substances as shown in Table 1.

Riffat et al. [1] presented a review of the application of the main natural refrigerants, for refrigeration and air-conditioning systems, consist of ammonia, hydrocarbons, carbon dioxide, water and air. Ammonia is a well-known refrigerant, widely used in medium and large refrigeration systems but it is toxic and flammable. It requires special safety precautions. Air is the safest and cheapest refrigerant. The air refrigeration cycle is attractive at lower temperatures. For water, it is one of the most acceptable natural refrigerants, because of its safety and cost. The refrigeration effect of water is roughly twenty times of R-12 and the system needs compressor that can handle almost two hundred times of volumetric flow, for a given refrigeration capacity. For the hydrocarbon substances, they have been used as refrigerants for many years, but these refrigerants are highly flammable. In the last substance, carbon dioxide is the natural fluids, which is neither flammable nor toxic and widely available for use.

Chaichana et al. [2] studied the natural working fluids for use in solar boosted heat pump system as a substitute of R-22. They found the R-717 or ammonia has an opportunity to be a prime candidate for systems over the other natural working fluids.

The focus of this paper is to study the natural working fluids which are hydrocarbon and non-hydrocarbon groups for use in heat pump water heater systems as a substitute of R-22 and R-134a. The hydrocarbon group is propane(R-290), butane(R-600), isobutene(R-600a) and propylene(R-1270). The non-hydrocarbon group comprises of ammonia(R-717) and carbon dioxide(R-744).

Table 1 Characteristics and properties of some refrigerants. [3-5]

Working fluid	**R-22**	**R-134a**	**R-290**	**R-600**	**R-600a**	**R-1270**	**R-717**	**R-744**
Chemical formula	$CHClF_2$	CF_3CH_2F	C_3H_8	C_4H_{10}	C_4H_{10}	C_3H_6	NH_3	CO_2
Molar mass (g mol^{-1})	86.46	102.03	44.10	58.12	58.12	42.08	17.03	44.01
Critical temperature (°C)	96.14	101.0	96.7	152.0	134.7	92.4	131.9	31.1
Critical pressure (kPa)	4990	4067	4247	3796	3640	4665	11333	7384
Critical density (kg m^{-3})	562.0	552.5	220.0	227.8	224.4	222.6	225.0	466.5
Boiling point[a] (°C)	-40.9	-26.1	-42.1	-0.5	-11.6	-47.7	-33.3	-78.4
Flammability	No	No	Yes	Yes	Yes	Yes	Yes	No
Toxicity	No	No	No	No	No	No	Yes	No
ODP	0.05	0	0	0	0	0	0	0
GWP[c]	1700	1300	3	3	3	3	0	1

[a] Boiling point at atmospheric pressure

[b] At 20 ° C

[c] Reference to CO_2 with base values of 1

2. SYSTEM MODELING

Fig.1 shown the P-h diagram of the heat pump system. For conventional system shown in Fig 1a, for all natural working fluids except CO_2 and Fig. 1b is for CO_2. The heat rejection of CO_2 system is operating in trans-critical process, over the critical point. So, the CO_2 in this process is not change the status of fluid from vapor to liquid as the other working fluids systems.

2.1 Compressor

In the compression process (1 – 2), we can calculate the work of compression from

$$W_{comp} = m_r (h_2 - h_1)\,, \tag{1}$$

$$W_{comp} = \frac{W_{isen}}{\eta_{isen}}\,. \tag{2}$$

W_{is} is isentropic compression work that can defined from

$$W_{isen} = m_r (h_{2isen} - h_1)\,. \tag{3}$$

For isentropic efficiency, Robinson and Groll [6], are a correlation as

$$\eta_{is} = 0.815 + 0.022(\frac{P_2}{P_1}) - 0.0041(\frac{P_2}{P_1})^2 + 0.0001(\frac{P_2}{P_1})^3\,. \tag{4}$$

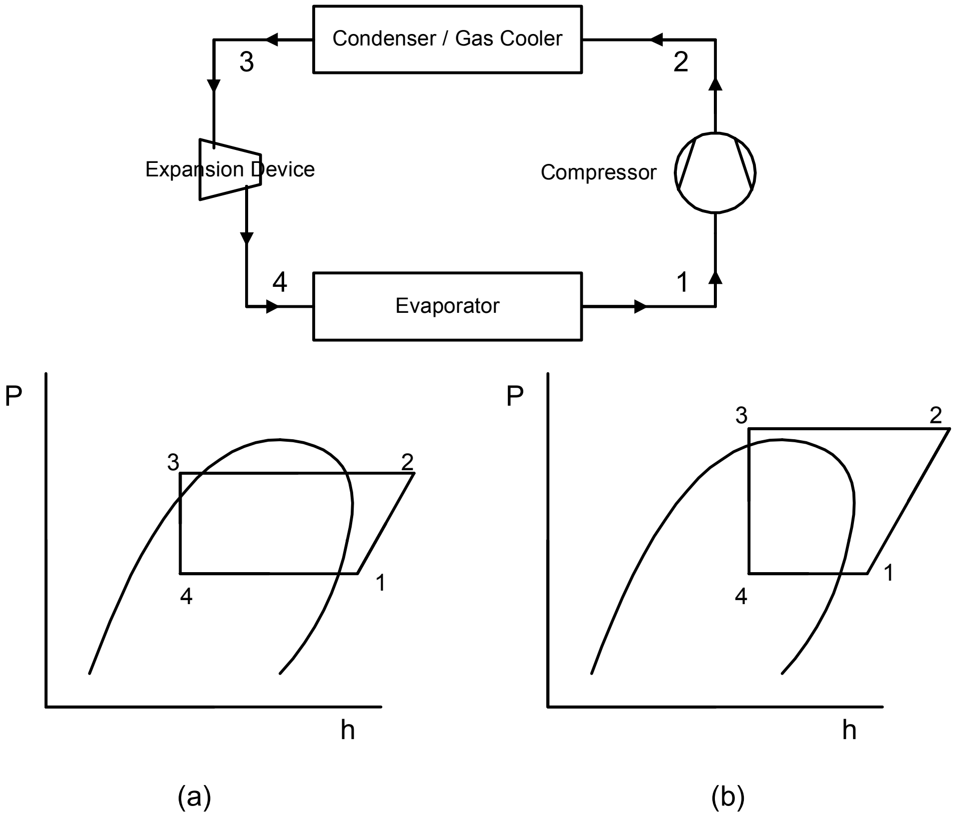

Fig. 1 P-h Chart for heat pump water heater system

a) For R-22, R-134a, R-290, R-600, R-600a, R-1270 and R-717

b) For R-744

2.2 Condenser

The heat capacity in the condenser (2 – 3) can be calculated from

$$Q_{cd} = m_r (h_2 - h_3), \tag{5}$$

$$Q_{cd} = m_w Cp_w (T_{wo} - T_{wi}). \tag{6}$$

In the condenser, the working fluid in vapor condenses and becomes liquid. We separated the condenser into 3 parts, sub-cooled, condensation and superheated fluids. The condenser in this model used the double pipe counter-flow heat exchanger. The calculations of heat transfer in the condenser use NTU-Effectiveness method as follows

$$Q_{cd} = UA_{cd} LMTD_{cd}, \tag{7}$$

$$NTU_{cd} = \frac{UA_{cd}}{C_{\min}}, \tag{8}$$

$$\varepsilon_{cd} = \frac{1 - \exp[-NTU_{cd}(1 - \frac{C_{\min}}{C_{\max}})]}{1 - (\frac{C_{\min}}{C_{\max}}) \exp[-NTU_{cd}(1 - \frac{C_{\min}}{C_{\max}})]}, \tag{9}$$

$$\varepsilon_{cd} = 1 - \exp^{-NTU_{cd}}. \tag{10}$$

For the effectiveness calculation, equation (9) is used for sub-cooled and superheated parts. Equation (10) is used for the condensation part.

$$Q_{\max cd} = C_{\min} (T_{hi} - T_{ci}). \tag{11}$$

2.3 Evaporator

The heat capacity at evaporator can define from:

$$Q_{ev} = m_r(h_1 - h_4). \tag{12}$$

2.4 System

The coefficient of performance can be calculated from

$$COP = \frac{Q_{cd}}{W_{comp}} = \frac{(h_2 - h_3)}{(h_2 - h_1)}, \tag{13}$$

2.4 Second law analysis

The property of the working fluid in each state can be modified into exergy. The exergy is defined as

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

The exergetic efficiency of the system can be calculated from

$$\eta_{ex} = \left|\frac{(e_2 - e_3)}{(e_2 - e_1)}\right|. \tag{15}$$

For CO_2 system, the process of heat rejection is trans-critical process. The pressure at the gas cooler is independent from the temperature. Thus, Liao et al. [7] presented an equation for estimate the pressure of gas cooler as

$$P_{gc} = (2.778 - 0.0157T_e)T_3 + (0.381T_e - 9.34). \tag{16}$$

3. SYSTEM SIMULATION

Figures 2 – 3 show the informative flow diagrams for simulating the system. Figure 2 is the diagram for CO_2 system and Figure 3 is the diagram for conventional system.

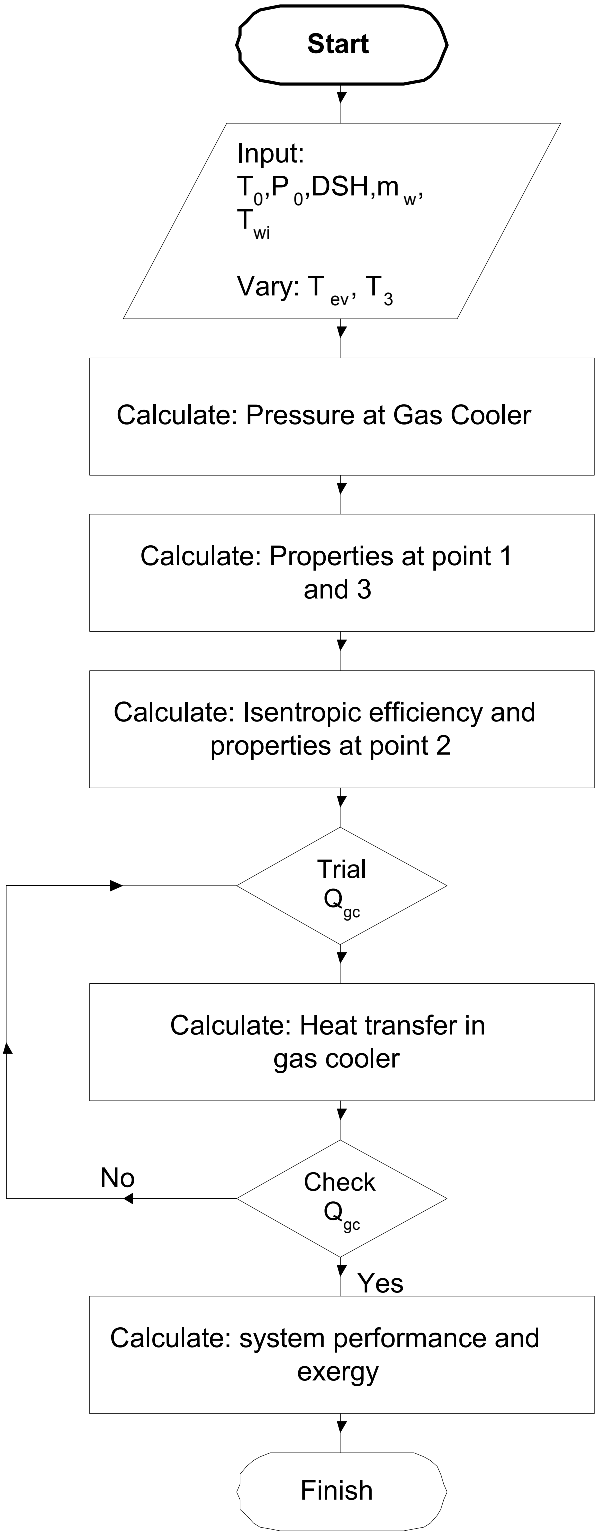

Fig. 2 Flow diagram of R-744 system

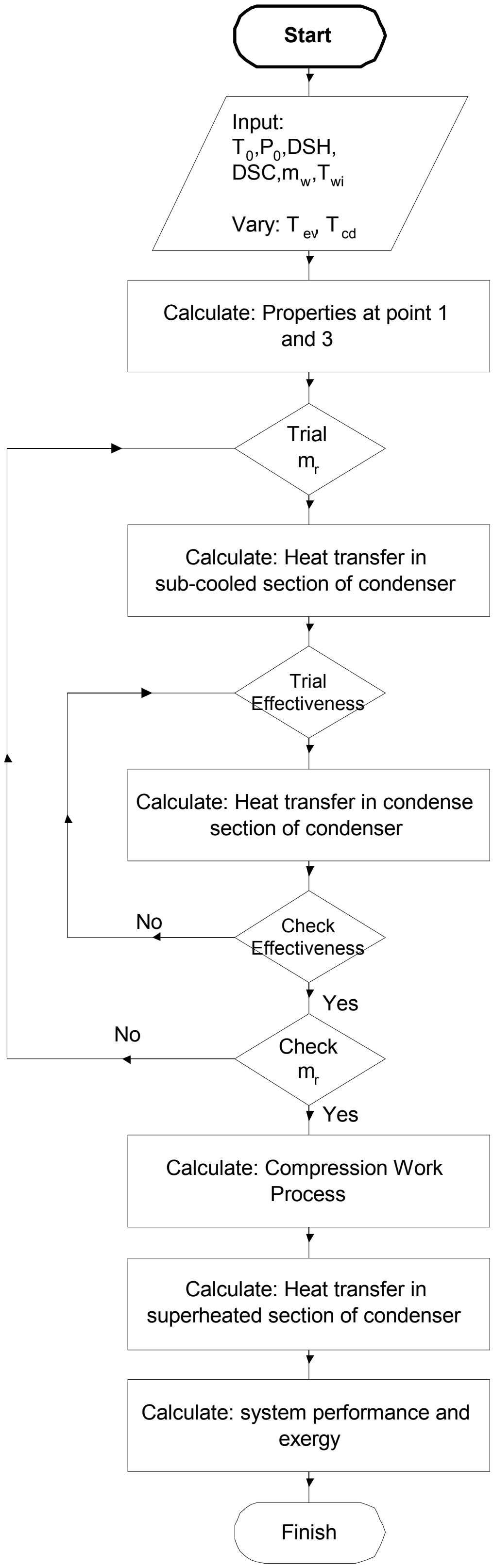

Fig 3 Flow diagram for working fluids except R-744

The parameters required by the model are given as follows:

- Degree of superheated = 10 C
- Degree of sub-cooled = 5 C
- Flow rate of water = 3 liters/min.
- Temperature of water inlet = 20 C
- Pressure and temperature at dead state = 30 C and 101.325 kPa
- Evaporating temperature (T_{ev}) = 5 – 25 C
- Condensing temperature (T_{cd}) = 45 – 65 C
- Temperature of gas cooler for CO_2 system (T_3) = 40 – 60 C.

The outputs obtained are refrigerant mass flow rate, work of compression, capacity of evaporator and condenser or gas cooler, water temperature outlet, coefficient of performance and exergetic efficiency.

In this study, the code is in FORTRAN 90 and linked to REFPROP 7.0 for calculation the properties of the working fluids.

This simulation is calculated for mass flow rate of water at 3 liters/min and water inlet temperature of 20 C.

4. RESULTS AND DISCUSSION

Fig. 4 shows the mass flow rate of natural working fluids in the systems at constant of evaporating temperature. The R-744 system gives the highest mass flow rate and the R-717 gives the lowest mass flow rate, because the density of the R-744 is larger than the other working fluids.

Fig. 5 shows the compression work of the heat pump system. The compression work of R-744 is highest because the mass flow rate of R-744 is so high when it compared with other fluids.

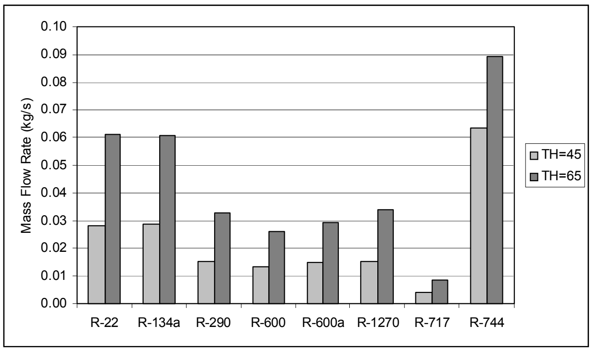

Fig. 4 Mass flow rate of the working fluids.

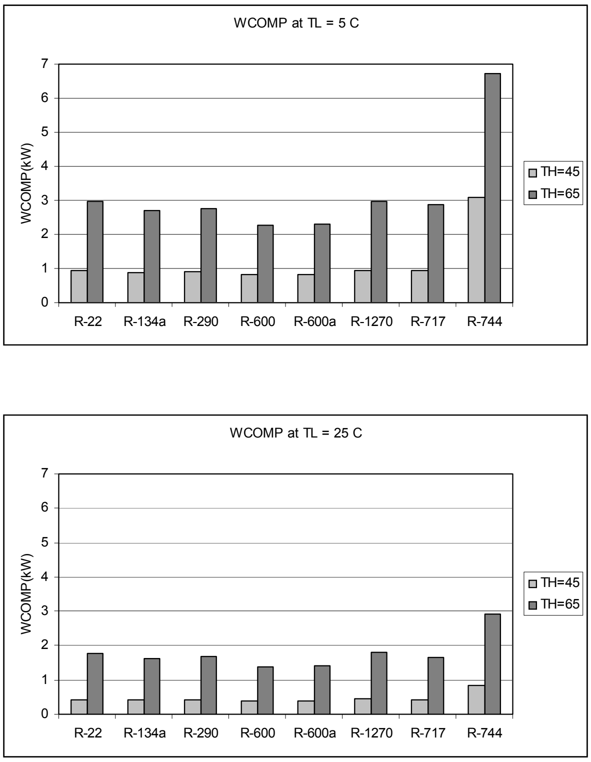

Fig. 5 Compression work of the heat pump systems.

For the heat capacity in the condenser and gas cooler as shown in Fig. 6, the heat capacity of gas cooler in R-744 system is largest at low evaporating temperature. When the evaporating temperature is increase, the heat capacity of gas cooler in R-744 system is decreased because the enthalpy difference is not much when compared with the other fluids.

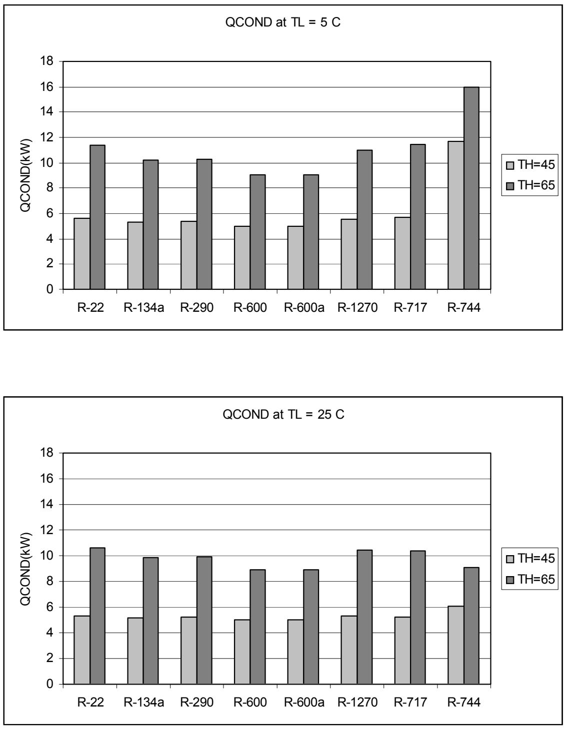

Fig. 6 Heat capacity at condenser / gas cooler of the heat pump systems.

Fig 7 shows the value of effectiveness of condenser and gas cooler. The value of effectiveness is between 0.50 – 0.75 depend on the properties and the conditions of the working fluids.

Fig. 8 shows the heat capacity of evaporator, the heat capacity of evaporator in R-744 system is higher than the other fluids at low evaporating temperature. When the evaporating temperature increases, the heat capacity of evaporator in R-744 system is lower than the others and as the evaporating temperature is near the critical temperature in case of R-744, the enthalpy difference in the evaporation process is rather low.

Fig. 9 shows the water temperature outlet from the systems. At low evaporating temperature, R-744 system is gives highest water temperature outlet compared with the other fluids. When the evaporating temperature increases, the heat capacity of gas cooler will decrease as shown in Fig. 6 and the performance to produce hot water will be less.

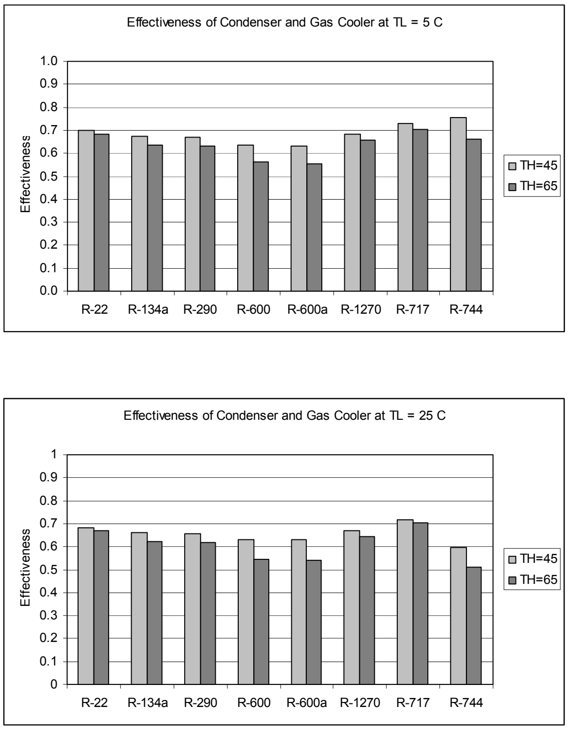

Fig. 7 Effectiveness of condenser and gas cooler in the heat pump system.

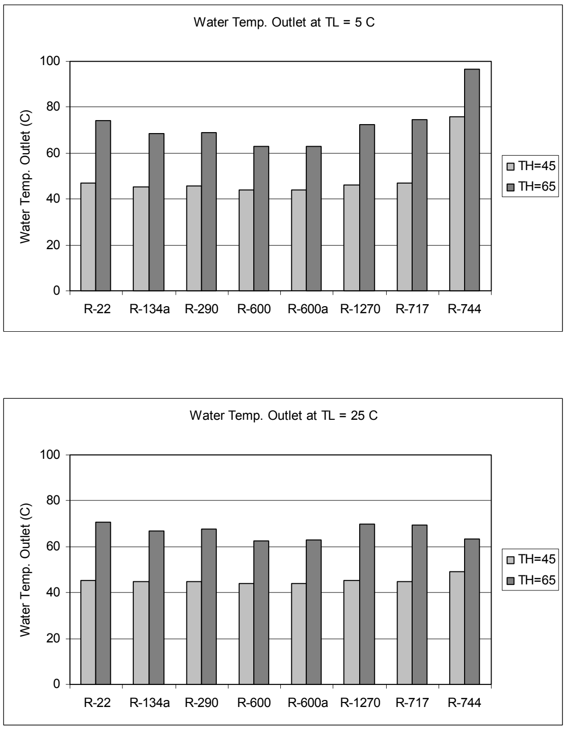

Fig. 9 Water temperature outlet of the heat pump system.

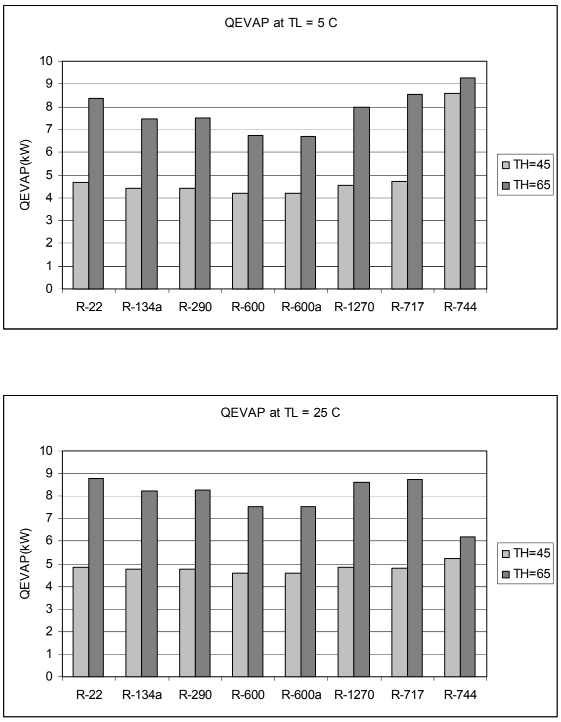

Fig 8 Heat capacity of evaporator in the heat pump system.

The effect of mass flow rate of R-744 in Fig. 4 on the compression work and the coefficient of performance in Fig. 10. The COP of R-744 is lowest compared with the others. For exergetic efficiency as shown in Fig 11, the exergy efficiency of the R-744 system is similar to the others. The efficiency will decrease when the evaporating temperature is increase.

5. CONCLUSION

The R-744 (CO_2) system has the potential to generate at water high temperature but the performance and efficiency of the system is worse compared with the other working fluids. From the simulation results, the R-744 system will set high performance and high water temperature at low evaporating temperature. Although, R-744 is the worse in term of efficiency but it is a safe working fluids. It is non-flammable and non-toxic compared with other natural working fluids.

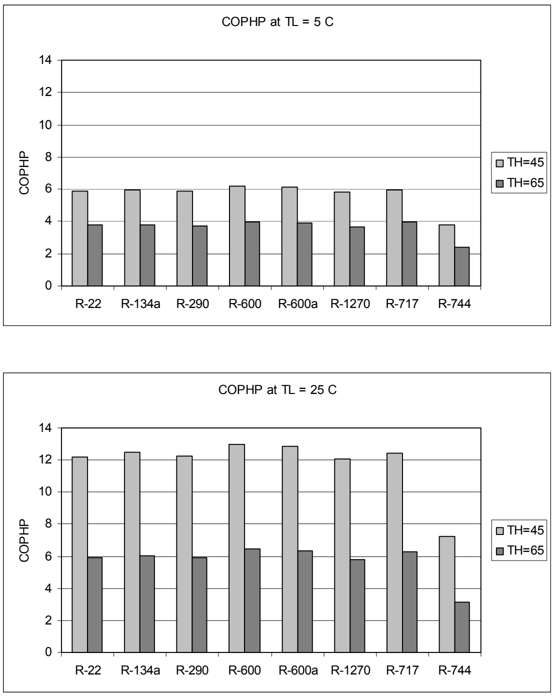

Fig. 10 Coefficient of performance of the heat pump system

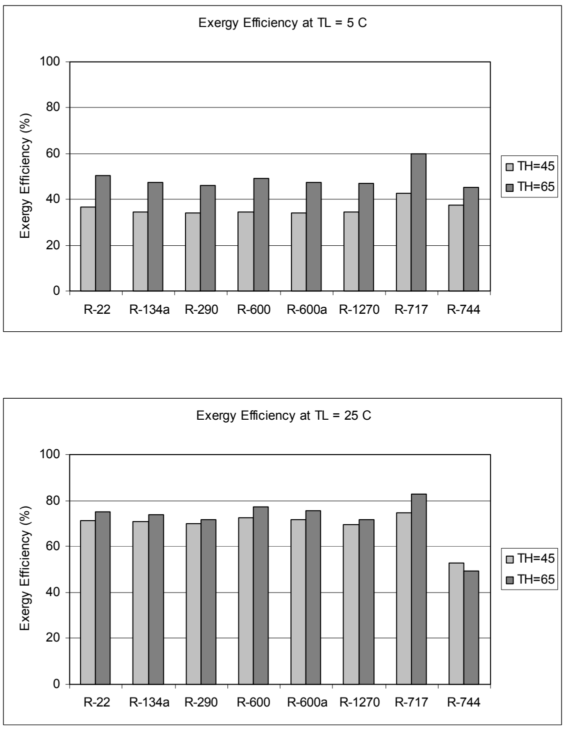

Fig. 11 Exergy efficiency of the heat pump system

6. ACKNOWLEDGEMENT

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7. NOMENCLATURE

COP	Coefficient of Performance
Cp	Specific heat (kJ/kg.K)
e	Exergy (kJ/kg)
h	Enthalpy (kJ/kg)
LMTD	Log mean temperature difference
m	Mass flow rate(kg/s)
P	Pressure (bar)
Q	Heat capacity (kW)
s	Entropy(kJ/kg.K)
T	Temperature (°C)
W	Work(kW)
ε	Effectiveness
η	Efficiency

Subscript

cd	Condenser
comp	Compression process
e	Evaporating
ev	Evaporator
ex	Exergy
gc	Gas Cooler
H	Heating
isen	Isentropic process
max	Maximum
min	Minimum
w	Water
wi	Water inlet
wo	Water outlet
0	Dead state
1	Outlet from evaporator
2	Outlet from compressor
3	Outlet from condenser or gas cooler
4	Outlet from expansion device

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