Hittite Journal of Science and Engineering, 2014, (1) 1 27-36
ISSN NUMBER: 2149-2123
DOI: 10.17350/HJSE19030000005

Thermodynamic Analysis of A Heat Pump for Different Refrigerants

Saban Tamdemir and Ali Kilicarslan*
Hitit University, Department of Mechanical Engineering, Corum, TURKEY

ABSTRACT

In this study, the air to air heat pump that was installed to a room having dimensions of 6000x4000x3800 mm, was tested between the outdoor temperatures of -5°C and 5°C for different refrigerants , namely R134a, R404A, R407C and R410A. The electric power drawn by the compressor, indoor unit and outdoor unit fans, temperatures and pressures at the state points were measured. Experiments at the same outdoor temperatures were repeated three times under the controlled conditions. Analyses of the results obtained from the experiments were carried out by the computer code that was developed by means of Engineering Equation Solver (EES-V9.172-3D). The power consumed by the compressor, heating capacity of indoor unit, the absorbing heat capacity of outdoor unit, coefficient of performance of the heat pump, suction and discharge pressures of the compressor were investigated according to the outdoor temperatures. As the outdoor air temperature increased, the power consumed by the compressor increased whereas the indoor unit and outdoor unit capacities and coefficient of performance of the heat pump decreased on average. R134a is the most favourable refrigerants among the refrigerants under study due to consuming the least compressor power and having the highest coefficient of performance

Article History:
Received: 2014/11/22
Accepted: 2014/12/24
Online: 2014/12/31

***Correspondence to**: Ali Kilicarslan,*
Hitit University, Faculty of Engineering,
Department of Mechanical Engineering,
Corum, Turkey
Tel: +90 (364) 227-4533 (1236)
Fax: +90 (364) 227-4535
E-Mail: alikilicarslan@hitit.edu.tr

Key Words:
Refrigeration; Heat Pump; Performance; Energy Analysis

INTRODUCTION

Heat pumps are the systems delivering the heat from the low temperature source, to the high temperature source. According to the second law of thermodynamics, the heat does not spontaneously flow from a sink at a lower temperature to a source a at higher temperature source. The energy is required to provide this heat flow. Heat pump supplies this energy by using the electrical energy (mechanical heat pump) or heat energy (thermal heat pumps) [1].

Heat pumps are used in the areas of heating, refrigerating and air conditioning. Furthermore, they are used for meeting the hot water needs. Heat pumps differ from each other in terms of the source that they used (air, water, soil, etc.) and the heat that they conduct [2]. Air source heat pumps are employed with the applications of different heating systems such as a floor heating system, radiator heating system for homes, etc. and provide an optimum comfort conditions. The hot water needs are met throughout the year. In addition to the ease of installation, they also do not require a high-cost drilling. They are used in the limited areas because of having a small outdoor unit [3]. The performance of a heat pump was experimentally carried out by using R432a as a refrigerant instead of R22 and it was observed that the performance and heating capacity of the heat pump increased between 8.5% - 8.7% and 1.9% - 6.4%, respectively [4].

The performance of a heat pump using a near azeotropic refrigerant R431a that is a mixture of 71% propane and 29% R152a was experimentally compared with that of a heat pump using R22 as refrigerant. According to the experimental results, it was seen that the performance of the heat pump with R431a is higher by 3.5% - 3.8% than that of the heat pump with R22 at the same operating conditions [5]. The effects of two different airflow consisting of 75% and 100% of nominal air flow and refrigerant R410a charge, namely 75%, 100%, and 125% of regular value of an air on the performance of an source heat pump were experimentally carried out. It was observed that the increase in refrigerant charge while keeping the airflow constant at the rated value increased the COP by 5% at the heating season. However, the decrease in both air flow and refrigerant charge decreased the COP by 10% [6]. In a case study carried out in Beijing, China, a different prototype air source heat pump (ASHP) was designed and manufactured, it was observed that ASHP has a higher performance and heating capacity with the snowy days having high relative humidities and temperatures of around -12°C.

In this study, the performance of a air source heat pump was experimentally carried out for refrigerants R134a, R404a, R407c and R410a at various outdoor air temperatures. It was seen from the literature survey that there are few studies concerning with ASHP in which R407c and R410a are used as refrigerants. Furthermore, it was seen that the performance of heat pumps were compared in terms of refrigerants as the studies in the literature was investigated. But this study also aims to observe the coefficient of performance of ASHP at different outdoor temperatures.

THEORETICAL STUDY

The first law of thermodynamics for a control volume can be written as [8],

$$\dot{Q}_{cv}=(dE_{cv} / dt) + \Sigma\dot{m}_e (h+(v^2/2)+gz)_e - \Sigma\dot{m}_i (h+(v^2/2)+gz)_i + \dot{W}_{cv} \quad (1)$$

Where;

dE_{cv} / dt: change of energy with respect to time within a control volume

$\dot{W}_{cv}$: work done per unit time by a control volume

$\dot{Q}_{cv}$: heat rate delivered to a control volume

h_i : enthalpy of the refrigerant at a control volume inlet

h_e : enthalpy of the refrigerant at a control volume exit

The following assumptions are valid for the air source heat pump (ASHP) under consideration,

- The flow of refrigerant throughout ASHP occurs at steady-state and steady-flow process.
- The heat transfer from the compressor is negligible.
- The electrical energy rate driven by the compressor is equal to the compression work per unit time.
- The changes in kinetic and potential energies in the elements of ASHP are negligible.
- Sub cooling occurs in the indoor unit, the pressure losses are negligible.

As the assumptions mentioned above are taken into consideration, the energy required for the compressor can be derived from Equation 1,

$$\dot{W}_{comp}=\dot{m}_r(h_2-h_1) \quad (2)$$

Where, $\dot{W}_{comp}$ is the energy rate driven by the compressor and is measured by the energy analyser, is the mass flow rate of the refrigerant throughout the ASHP, is the specific enthalpy at the compressor inlet and is the specific enthalpy at the compressor exit.

The capacity of the indoor unit $\dot{Q}_{iu}$ can be expressed as,

$$\dot{Q}_{iu}=\dot{m}_r(h_3-h_4) \quad (3)$$

Where, h_3 and h_4 are the specific enthalpies of the refrigerants at the inlet of the indoor unit and exit, respectively.

The heat transfer rate absorbed by the outdoor unit of the ASHP $\dot{Q}_{ou}$ can be written as

$$\dot{Q}_{ou}=\dot{m}_r(h_5-h_1) \quad (4)$$

Where, h_5 is the specific enthalpy of the refrigerant at the inlet of the indoor unit.

Coefficient of performance of the heat pump (COP_{hp}), a measure of ASHP, can be written as

$$COP_{hp} = \dot{Q}_{iu} / (\dot{W}_{comp} + \dot{W}_{iu} + \dot{W}_{ou}) \quad (5)$$

Where, W_{iu} and W_{ou} are the energies consumed by the fans of indoor and outdoor units, respectively. These energies are also measured by the energy analyser as in the energy consumed by the compressor. The number of revolutions per minute is constant for these fans, so the consumed energies by these fans are constant and given below,

$\dot{W}_{iu}$ = 0.176 kW

$\dot{W}_{ou}$ = 0.088 kW

EXPERIMENTAL

Experimental studies were performed in the one of the laboratories of the Department of Mechanical Engineering of Hitit University, having dimensions of 6000x4000x3800 mm and facing North.

The heating load of this laboratory is 5472 kcal. The indoor unit of the ASHP is shown in Figure 1 while the outdoor unit is shown in Figure 2.

Figure 1 also shows the energy analyser, data logger, manometers and thermocouple connections

ASHP mainly consists of semi hermetic compressor, air forced indoor and outdoor units, capillary type expansion valve. The auxiliary elements of the ASHP include four-way valve, high and low pressure switches and filter drier.

Figure 2. Outdoor unit of the ASHP

Uncertainty analysis of the study

There is a certain degree of uncertainty and error in every experiment. If the experiments are organized very precisely, the uncertainties and errors can be minimized in the experimental results. Three errors causing the uncertainty in the experimental results are presented by [9] as the error caused by person during reading the result from the experimental apparatus, the errors related to the primary and secondary part of the measuring apparatus.

Figure 1. Indoor unit of the ASHP

According to the uncertainty analysis, R is the parameter that must be measured during the experiments, and x_1, x_2, x_3 ....,x_n are the independent parameters affecting R. R can be expressed as a function of the independent parameters [10],

$$R= R(x_1,x_2,x_3 \ldots\ldots\ldots\ldots x_n) \quad (6)$$

w_1, w_2, w_3,..........., w_n are the uncertainty values related to every independent parameters and W_R , the uncertainty value of measured parameter R, can be written as

$$W_R=[(\partial RW_1/\partial x_1)]^2+ [(\partial RW_2/\partial x_2)]^2+\ldots+[(\partial RW_n/\partial x_n)]^2 \quad (7)$$

During the experiments that were carried out, the uncertainty values associated with the measured values were inserted into Equation 7 and the total values of the uncertainties were obtained as it is shown in Table 1.

RESULTS AND DISCUSSION

Air source heat pump (ASHP) was tested for different refrigerants including R134a, R404a, R407c and R410a at different outdoor air temperatures between -5°C and 5°C. A computer code was developed to analyze results by using Engineering Equation Solver (EES-V9.729-3D) that is commonly used in thermal systems [11]. The heating capacity of indoor unit, the amount of heat absorbed by outdoor unit, the energy consumed by the compressor, coefficient of performance of the ASHP, compressor suction and discharge pressures were investigated as a function of outdoor air temperatures for different refrigerants.

Figure 3 shows the change of compressor suction (inlet) pressure of ASHP with respect to the outdoor air

Table 1. Total uncertainties occurring in the experiments

Total uncertainties related to the measured parameters	Total value of uncertainty
Total uncertainties associated with the mass flow rate (W_m)	3.1%
Total uncertainties associated with the capacity of indoor unit (W_{Qiu})	3.1%
Total uncertainties associated with the capacity of outdoor unit (W_{Qou})	3.1%
Total uncertainties associated with the compressor power (W_{Wcomp})	1.3%
Total uncertainties associated with the coefficient of performance (W_{CoPhp})	3.3%

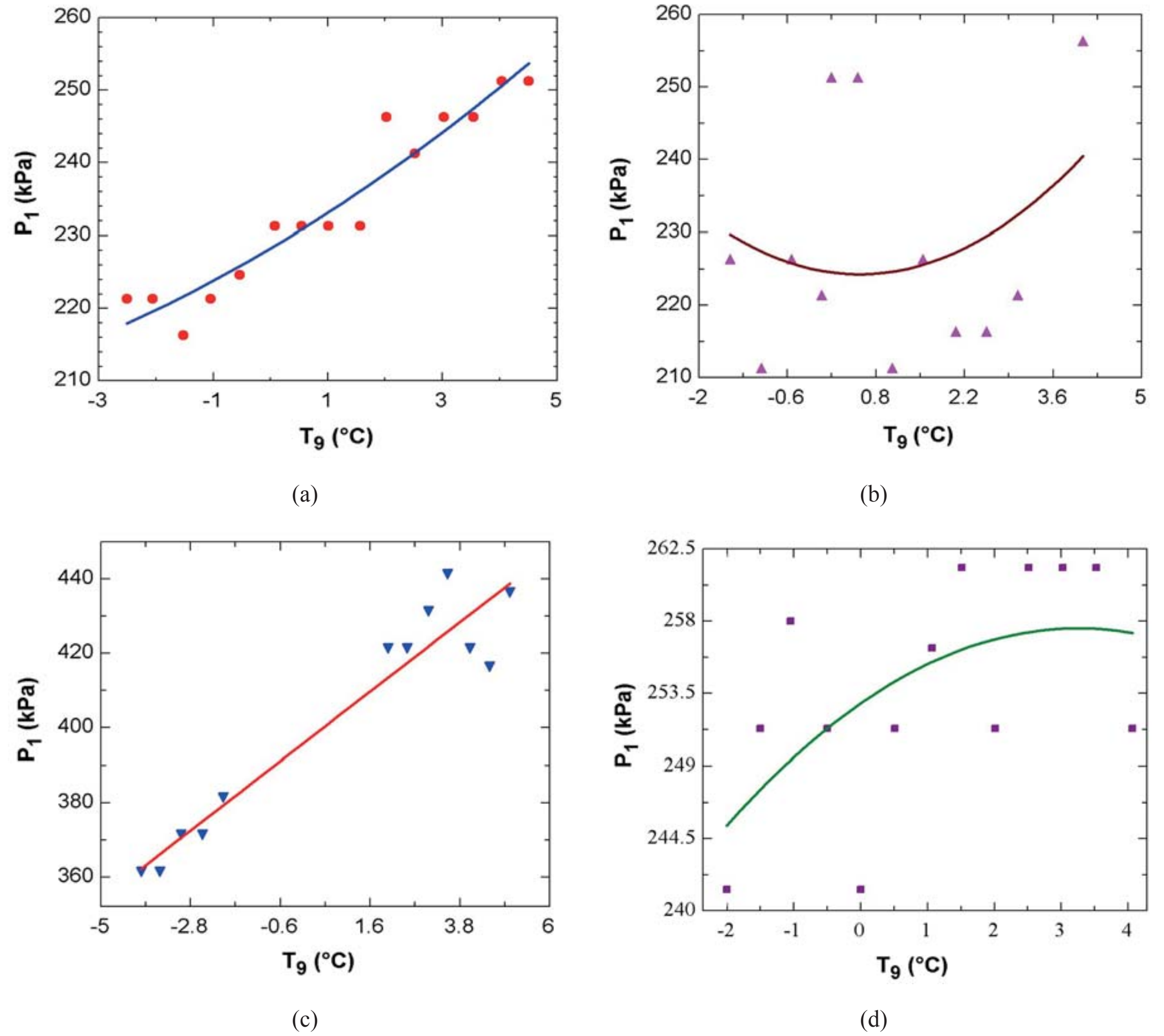

Figure 3. Compressor suction pressure as a function of outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

temperature for various refrigerants. As it is shown in Figure 3 that as the outdoor air temperature increases, the suction pressure of ASHP increases for all of the refrigerants except for refrigerant R404a under the study. The suction pressure firstly decreases and reaches to a minimum value increase, then increases for refrigerant R404a. As the outdoor air temperature increases, the amount of heat transfer to the outdoor coil increases, thereby increasing the saturation temperature of the refrigerant. This causes the saturation pressure, in other words, compressor inlet pressure corresponding to this temperature to increase. As shown in Figure 3a for refrigerant 134a, as the outdoor air temperature changes between -2.5°C and 4.5°C, the suction pressure of ASHP changes between 221 kPa and 251 kPa. In the case of R404a, when the outdoor air changes from -1.5°C to 4.0°C, the compressor suction pressure changes from 225 kPa to 235 kPa. The increase in suction pressure for refrigerant R134a is 13.5% while it is 4.44% for R404a as it is shown in Figure 3b.

Figure 3c shows that the experimental measurements were not obtained at the outdoor air temperatures between -2.0°C and 2.0°C because of weather conditions. As shown in Figure 3c, at the temperatures between -4.0°C and 5.0°C, the compressor suction pressure varies from 355 kPa to 430 kPa for refrigerant R407c and the increase in the pressure is 20% while the increase in the suction pressure is 5.52% for refrigerant R410a for the temperatures between -2.0°C and 4.0°C (Figure 3d). Figure 3 also shows that the maximum values of compressor suction pressures are obtained for R407c while the minimum ones are obtained for R404a.

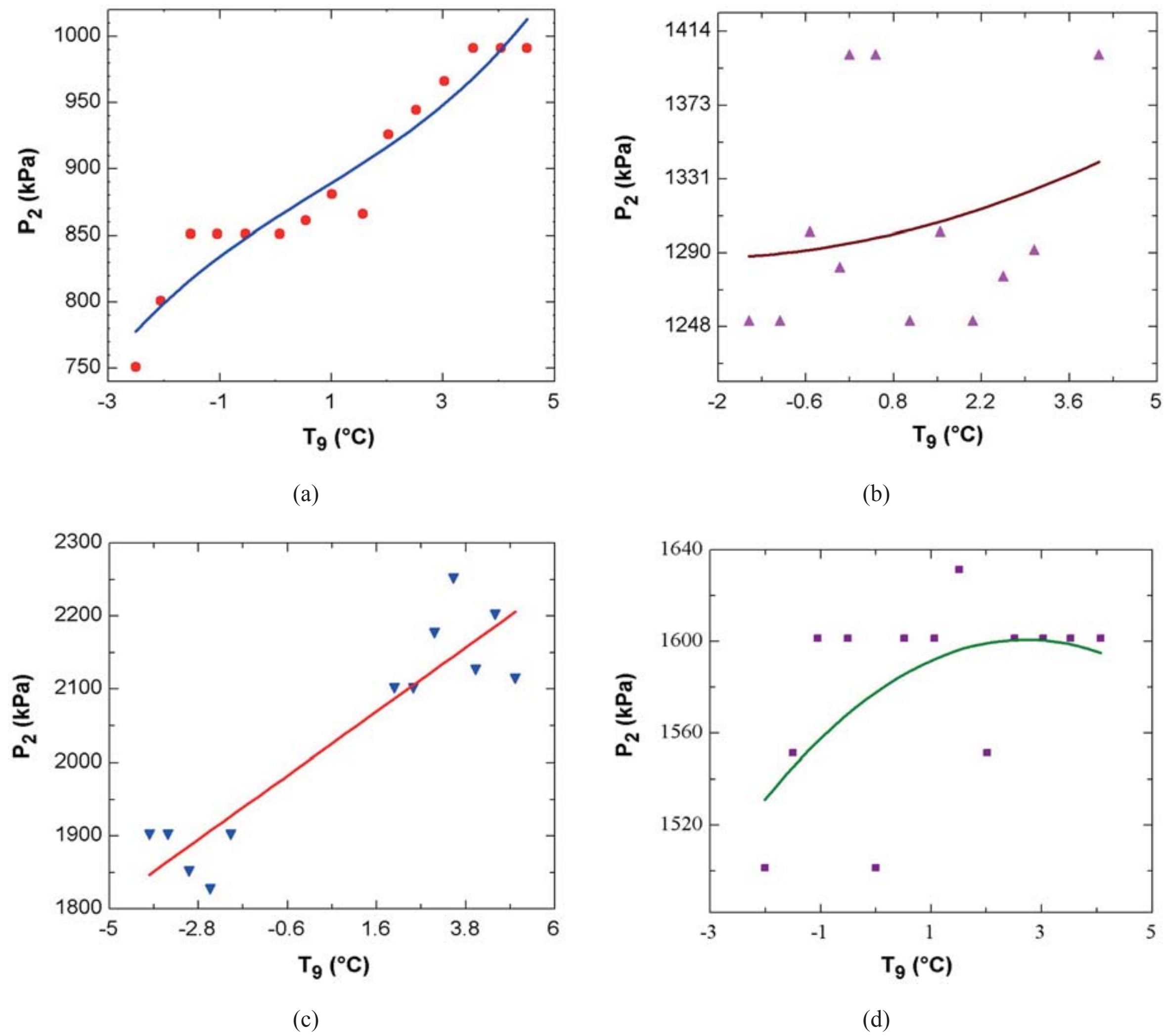

Figure 4. Compressor discharge pressure as a function of outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

The change of compressor discharge pressure of the ASHP with respect to the outdoor air temperature for the refrigerants including R134a, R404a, R407c and R410a is depicted in Figure 4. The heat transfer to the refrigerant because of increasing outdoor air temperature also causes the compressor discharge pressure to increase for all the refrigerants as in the case of the compressor suction pressure as it is shown in Figure 4. The maximum compressor discharge pressures, changing between 1850 kPa and 2200 kPa, are obtained as refrigerant 407c is used in the ASHP cycle while the minimum ones, changing between 7750 kPa and 1025 kPa, are obtained by using R134a as refrigerant in the cycle.

Figure 4 also shows that the discharge pressures of the ASHP lie in the middle range for the refrigerants, namely R404a and R410a. The discharge pressure values change between 1290–1345kPa for refrigerant R404a and 1520 –1610 kPa for refrigerant R410a. The maximum rate of increase in compressor discharge pressure occurs by using R134a as refrigerant in the cycle at the outdoor temperatures between -2.5°C and 4.5°C while the minimum occurs by using R410a at the temperatures between -1.5°C and 4°C.

S. Tamdemir and A. Kilicarslan / Hittite J Sci Eng, 2014, 1 (1) 27–36

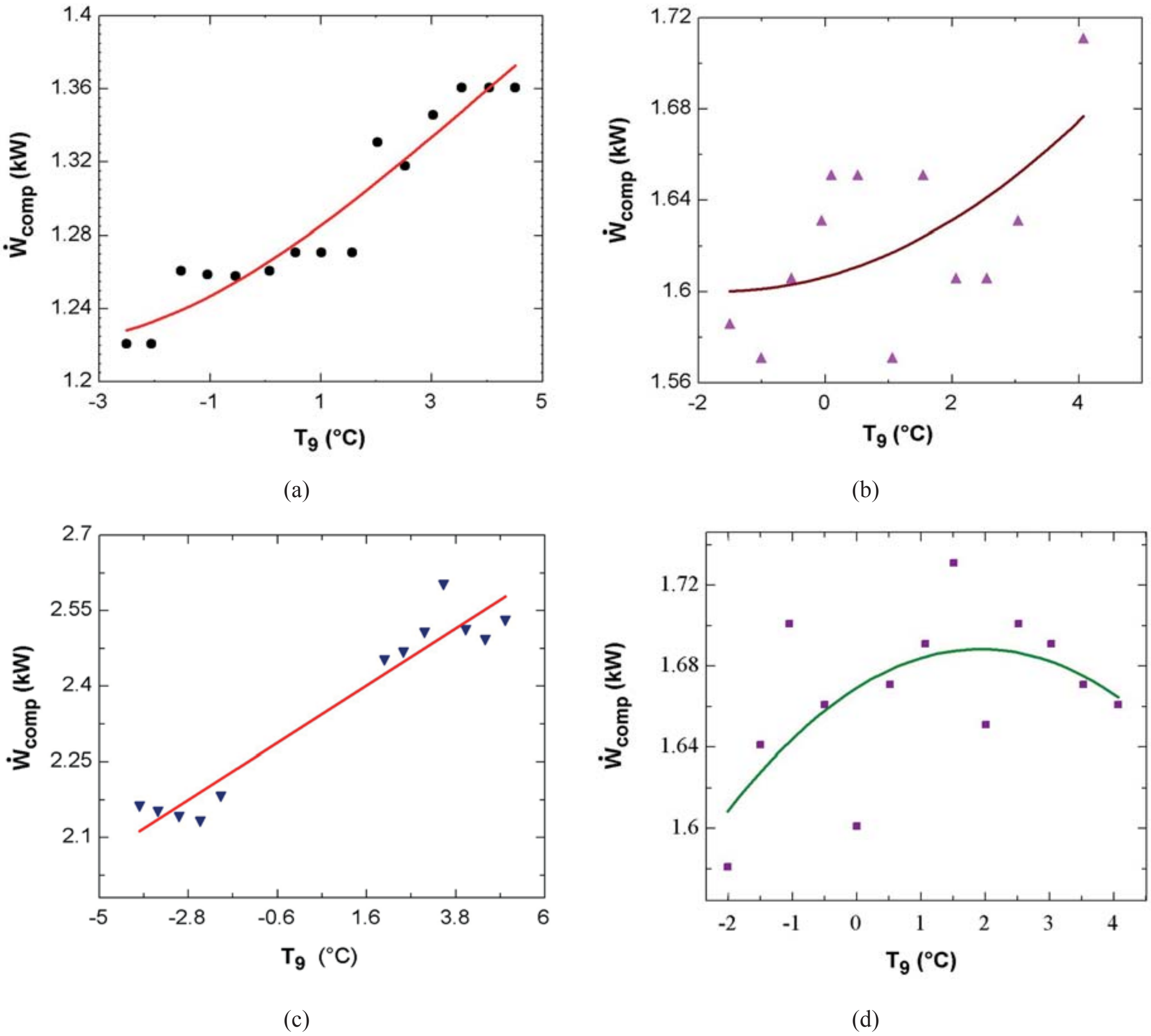

Figure 5. The power required for the compressor versus outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

The variation of the power required for the compressor as a function of outdoor air temperature for refrigerants R134a, R404a, R407c and R410a is depicted in Figure 5. As the outdoor air temperature increases, the pressure ratio across the compressor also increases. This causes the power required for the compressor to increase as shown in Figure 5. At the outdoor air temperatures between -2.5°C and 4.5°C, the power consumed by the compressor increases by 11.4% as refrigerant R134a is used in the ASHP cycle as shown in Figure 5a whereas the power increases by 5% for refrigerant R404a as depicted in Figure 5b. The maximum amount of power, namely approximately 2.5 kW is consumed in the ASHP cycle as refrigerant 407c is used as refrigerant at an outdoor temperature of 5°C as it is shown in Figure 5c while the minimum power, namely approximately 1.23 kW is consumed in the cycle for refrigerant R134a at an outdoor temperature of -2.5°C.

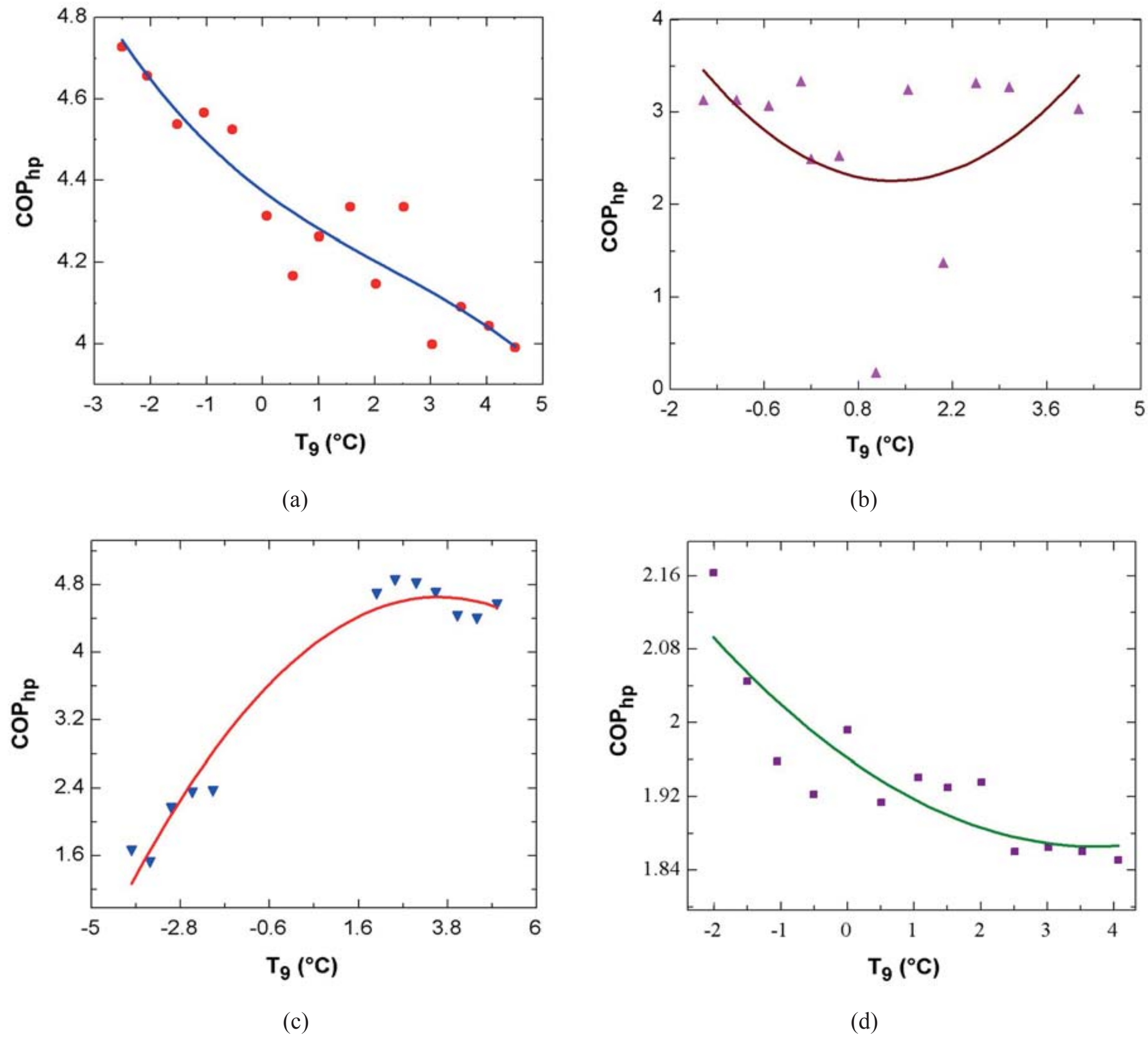

Figure 6. The coefficient of performance of the ASHP versus outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

Figure 6 shows the coefficient of performance of the ASHP as a function of outdoor air temperature for refrigerants R134a, R404a, R407c and R410a. The coefficient of performance of the ASHP increases for all the refrigerants except for refrigerant R407c as the outdoor air temperature increases. At the increasing values of outdoor temperatures, the energy required for the compressor increases, but the capacity of indoor unit decreases, thereby decreasing the coefficient of performance of the ASHP as it is shown in Figures 6a, 6c and partly 6b. But, for refrigerant R407c, the coefficient of performance of ASHP decreases because of increasing the capacity of indoor unit at higher outdoor temperatures.

The coefficient of performance firstly decreases between until the outdoor temperature reaches 1°C and then increases between 1.0°C and 4.0°C for refrigerant R404a. The coefficients of performance of the ASHP ranging from 4 to 4.8 are obtained by using R404a as a refrigerant in the cycle while those ranging from 1.85 to 2.1 are obtained by using R410a.

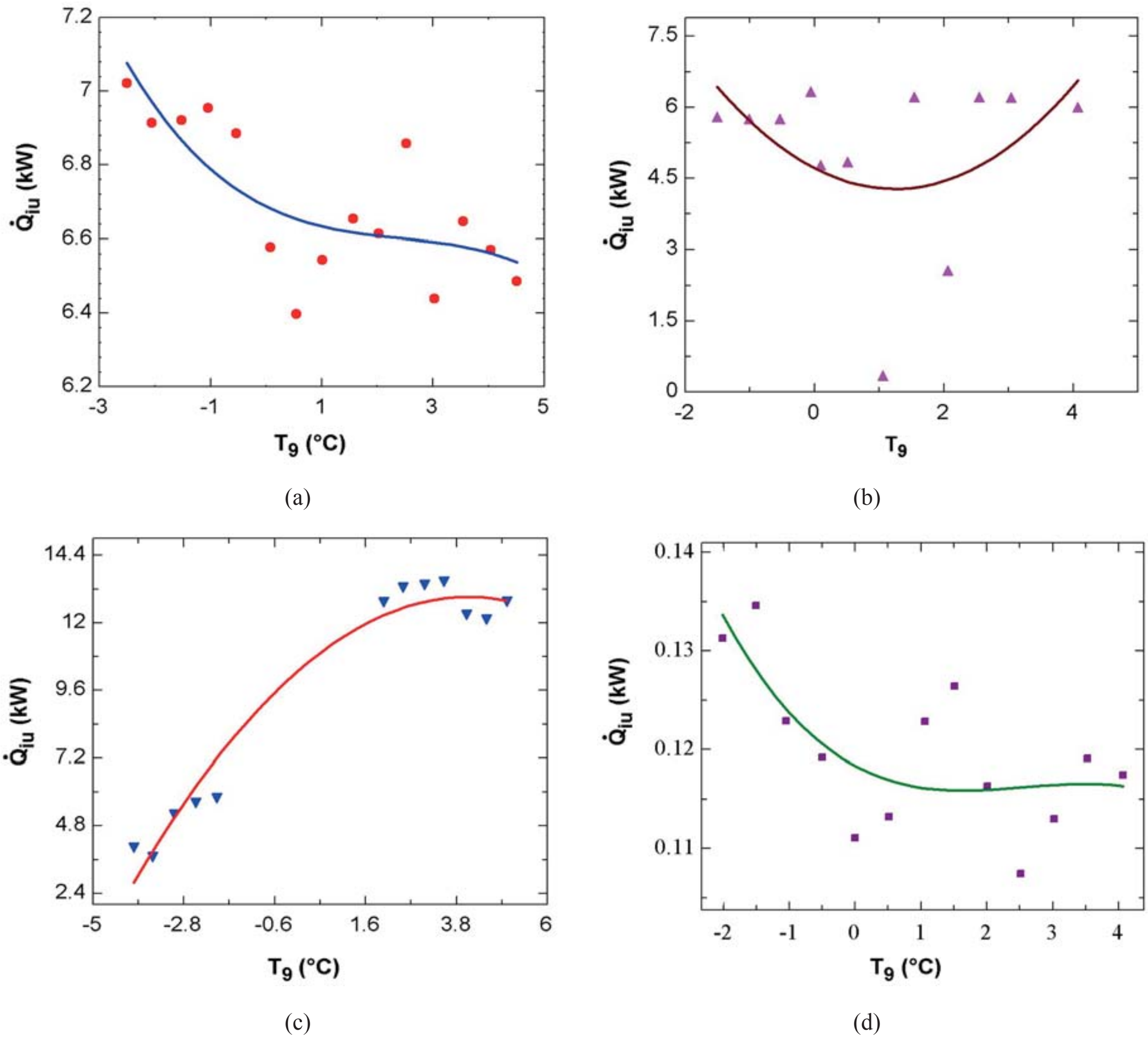

Figure 7. The capacity of indoor unit of the ASHP versus outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

The change of the capacity of the indoor unit of the ASHP as a function of outdoor air temperature for refrigerants R134a, R404a, R407c and R410a is depicted in Figure 7. The capacity of the indoor unit decreases at the outdoor air temperatures between -2.5°C and 4.5°C for refrigerant R134a as it is depicted in Figure 7a. In the mentioned temperature range above, the capacity decreases on average by 7.60%. For refrigerant R404a, the capacity firstly decreases between the temperatures of -1.5°C and 1.0°C and it has a minimum value of 4kW corresponding of 1.0°C and then increases between 1.0°C and 4.0°C as shown in Figure 7b. Figure 7c shows that the capacity of indoor unit ranges from 3 and 13 kW for refrigerant R407c at the outdoor air temperatures between -4.0°C and 5°C. At the outdoor air temperatures between -2.0°C and 1.5°C, the capacity of indoor unit decreases on average by 14.81 and there is a small increase in the capacity after a temperature of 2.0 °C for refrigerant R410a as it is shown in Figure 7d.

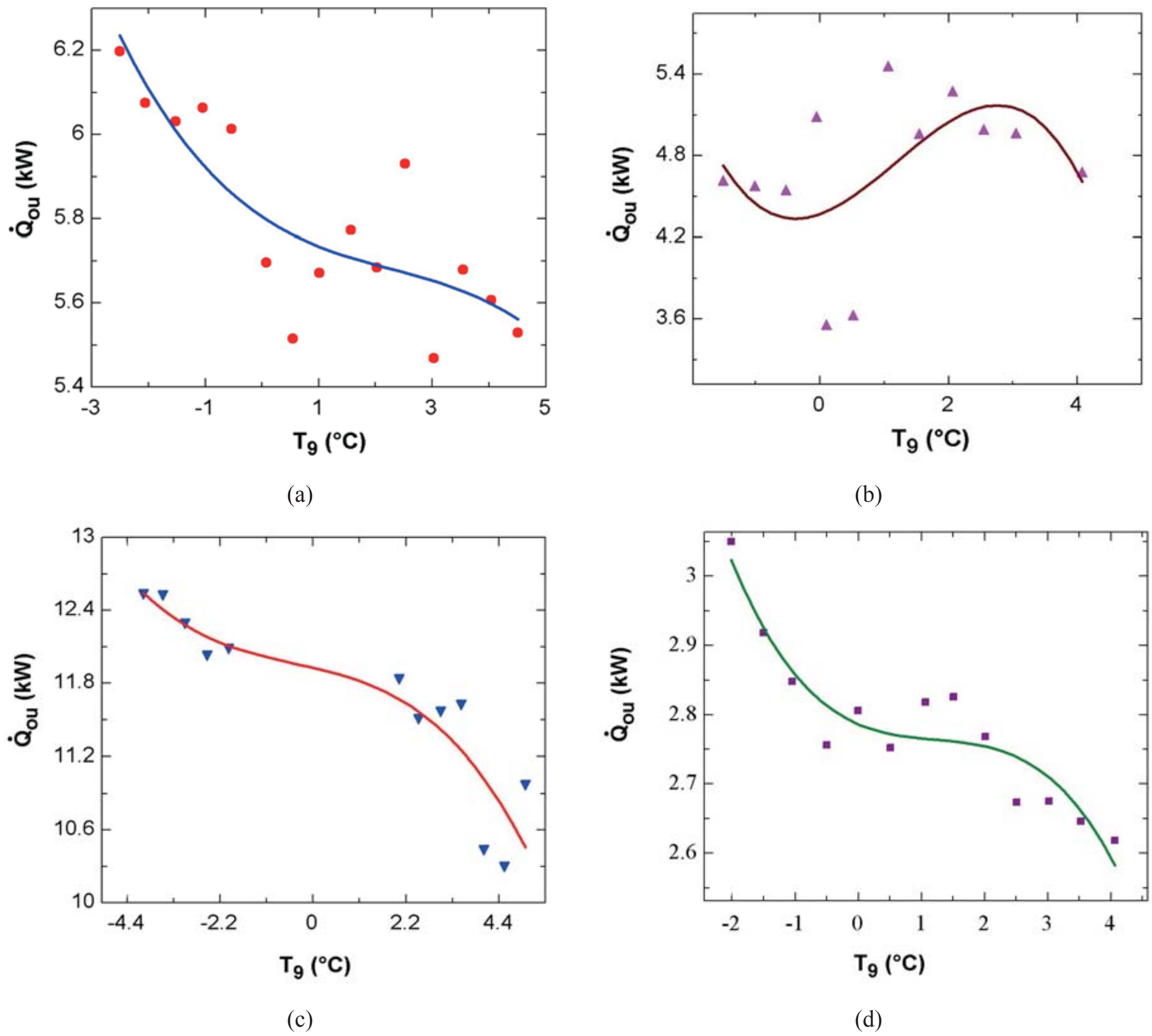

Figure 8. The capacity of outdoor unit of the ASHP versus outdoor air temperature a) R134a, b) R404a, c) R407c, d) R410a

Figure 8 depicts the change of the capacity of outdoor unit of the ASHP with respect to the outdoor air temperature for refrigerants R134a, R404a, R407c and R410a. The capacity of the outdoor unit decreases for all the refrigerants except for refrigerant R404a as the outdoor air temperature increases. The capacity of outdoor unit firstly decreases and then increases on average for refrigerant R410a as the outdoor air temperature increases. As it is shown in Figure 8a that the outdoor unit capacity has a maximum value, namely 6.2 kW at an outdoor temperature of -2.5°C for refrigerant R134a and then the capacity decreases, and approximately reaches a minimum value of 5.5 kW at an outdoor temperature of 5°C. Among the refrigerants under study, the maximum outdoor unit capacities ranging from 10.5 kW to 12.5 kW are obtained by employing R407c as a refrigerant in the ASHP cycle as depicted in Figure 8c while the minimum values ranging from 2.6 kW to 3 kW are obtained with refrigerant R410a (Figure 8d).

CONCLUSIONS

An air source heat pump was tested for refrigerants R134a, R404a, R407c and R410a at outdoor air temperatures ranging from -5°C to 5°C. The main conclusions are summarized below.

Compressor suction pressure of the ASHP increases as the outdoor air temperature increases. The maximum pressure increase in the compressor suction pressure, namely 21.4% is obtained by employing R407c as a refrigerant in the ASHP cycle while the minimum increase, namely 4.44% is obtained with refrigerant R404a. The values of increase in the compressor suction pressure for refrigerants R134a and R410a are 13.5% and 5.52%, respectively. The compressor suction pressure ranges from 355 to 430 kPa for refrigerant R407c and from 355 to 430 kPa for refrigerant R404a.

- Compressor discharge pressure of the ASHP also increases at increasing values of outdoor air temperatures. The maximum pressure increase in the compressor suction pressure, namely 32% is obtained by employing R134a as a refrigerant in the ASHP cycle while the minimum increase, namely 4.74% is obtained with refrigerant R404a. The values of increase in the compressor suction pressure for refrigerants R407c and R410a are 18.91% and 5.92%, respectively.
- The electrical power delivered to the compressor (compressor power) increases parallel to the pressure ratio of the compressor as the outdoor air temperature increases. The compressor power has the maximum value of increase ratio, namely 21.4% for refrigerant R407c while the minimum ratio, 3.65% is obtained for R410a and it ranges from 2.1 to 2.55 kW for R407c and from 1.63 to 1.69 kW for R410a.

  The increase ratio in the compressor power is 11.4% for refrigerant R134a and 5% for refrigerant R404a, respectively.
- The heating capacity of indoor unit, in other words, the heat rejected to the room to be heated, has a maximum value of 15 kW for refrigerant R407c and a minimum value of 0.115 kW for refrigerant R410a.
- The heat absorbed from the outdoor air (the outdoor unit capacity) is maximum, namely 12.5 kW for refrigerant R407c and is minimum, 2.65kW, for refrigerant R410a.

ACKNOWLEDGMENTS

We are grateful to Hitit University Scientific Research Projects Unit for supporting this project numbered (MUH03.12.001)

REFERENCES

1. Dursun B. Soğuk iklim koşullarında ısı pompalarının kullanımı, Yüksek Lisans Tezi, Atatürk Üniversitesi Fen Bilimleri Enstitüsü Makine Mühendisliği Ana Bilim Dalı, Erzurum, 2006.
2. Ceylan T. İki katlı bir binanın güneş enerjisi destekli hava kaynaklı ısı pompası ile ısıtılması ve sıcak su eldesinin analizi. Yüksek Lisans Tezi, Gazi Üniversitesi Fen Bilimleri Enstitüsü, Ankara, 2010.
3. Özyirmidokuz MH. Hava kaynaklı ısı pompası ve ekserji analizi. Yüksek Lisans Tezi, Erciyes Üniversitesi Fen Bilimleri Enstitüsü, Kayseri, 2010.
4. Park KJ, Shim YB, Jung D. Experimental performance of R432A to replace R22 in residential air–conditioners and heat pumps. Applied Thermal Engineering, 29 (2009) 597-600.
5. Park KJ, Shim YB, Jung D. A 'drop–in' refrigerant R431A for replacing HCFC22 in residential air–conditioners and heat pumps. Energy Conversion and Management, 50 (2009) 1671-1675.
6. Palmiter L, Kim JH, Larson B, Francisco PW, Groll E. A.I, Braun J.E. Measured effect of airflow and refrigerant charge on the seasonal performance of an air–source heat pump using R–410A, Energy and Building, 43 (2011) 1802–1810.
7. Wang R, Xie G, Wu Q, Wu Y, Yuan J. An air source heat pump with an advanced cycle for heating buildings in Beijing, Energy Conversion and Management, 52 (2011), 1493-1500.
8. Wylen GJV, Soontag RE. Fundamentals of Classical Thermodynamics, John Wiley&Sons, New York, 1985.
9. Schenck Jr H. Theories of Engineering Experimentation, McGraw–Hill Book Company, USA, 1968.
10. Kline SJ, McClintock FA. Describing uncertainties in single–sample experiments. Mechanical Engineering, 75 (1953) 3–8.
11. Klein SA. Engineering Equation Solver (EES), Professional version V9.723–3D, F–Chart Software (2010).