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Thermodynamic Performance Evaluation for Low Temperature Heat Source Cascade System Circulating Environment Friendly Refrigerants

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Abstract: The Cascade heat pump system is commonly used to overcome the high temperature lift problem of the system. In the present investigation eight refrigerant pairs were studied including R717/R134a, R410A/R134a, R407C/R134a, and R717/R600a, R744/R134a, R744/R290, R744/R600a, and R744/R717 at HT condenser of (70)°C and (75)°C. Hot water is to be produced at temperature range (60 to 65)°C with a proper flow demand. The evaporator temperature at the LT cycle side was ranged between (-10)°C and (-2)°C. The intermediate temperatures at the cascade heat exchanger were (20, 22.5, 33, and 35)°C depending on the refrigerant pairs implemented in the Cascade heat pump. Sea water at (7)°C was used as a sustainable low temperature heat source and 30% ethylene glycol-water brine as a thermal fluid carrier for heat extraction. The evaluation of the thermal performance of the refrigerant pairs was based on a fixed heat pump extraction load at the LT cycle evaporator. The R744/R134a and R744/R290 systems revealed the highest heat pump heating load production and highest compressors power consumption accompanied with the lowest COP at (20)°C intermediate temperature and HT condensation of (75)°C. R717/R600a showed the highest COP and lowest power consumption at (35)°C intermediate temperature and HT condensation of (70)°C. The results also revealed that a band of refrigerant pairs occupied the central zone of COP range with acceptable value; they are R410A/R134a, R407C/R134a and R744/R717.

Keywords: Sustainable Energy, Green Environment, Low Temperature Heat Source, Halogen Free Refrigerants

1. Introduction

Cascade system is widely selected for heat pumps when the temperature lift of the system is quite high. Two cycles are working independently and coupled by a cascade heat exchanger. The latter works as condenser and evaporator for the LT and HT cycles respectively. Similar or different refrigerants can be circulated separately for each cycle. The choice of refrigerants to circulate on each cycle depends on the heat source temperature and the heat load output of the heat pump. This is basically depends on the objectives of the heat pump, air conditioning, district hot water supply, and other technological application requirements. It also depends on the refrigerant Ozone Depletion Potential (ODP), Global Warming Potentials (GWP), toxicity and flammability in addition to the machine operating characteristics.

Bhattacharyya et al. (2006) [1] implemented (R744/R290) refrigerant system to study the performance and to determine the optimal evaporation temperature of R744 in a high-temperature cycle suitable for heating. Lee et al. (2005) [2] have theoretically analyzed the optimal condensation temperature of R744 in (R744/ R717) refrigerant pair at the low temperature cycle of the cascade system. Bansal and Jain et al. (2007) [3] evaluated the optimal condensation temperature of the R744 refrigerant by switching the refrigerant in the high-temperature cycle of a cascade system with various alternatives such as R717, R290, R1270 and R404A. Bingming et al. (2009) [4] and Dopazo and Fernández-Seara (2011) [5] provided experimental optimum intermediate temperature in the R717/R744 cascade heat pump system. They postulated that the COP is a load dependent; the optimized intermediate temperature should be obtained at a fixed heating capacity.

Kim et al. (2012) [6] studied experimentally the performance of a heat pump; it circulates R134a and R410A as refrigerants on the high and low cycle respectively. They concluded that the performances are deteriorated at high water inlet temperature and low ambient temperature. The optimal intermediate temperature is increased as ambient temperature increases and water inlet temperature increases. Kim et al. (2013) [7] performed a thermodynamic analysis of the optimal intermediate temperature by using R134a and R410A in high-temperature and low temperature cycle respectively. Baker and Schaefermeyer (2013) [8] presented a Cascade system to provide hot water in Alaska where sea water temperature around 10°C was considered. Kim et al. (2014) [9] studied the effect of the water temperature lift on the performance of cascade heat pump water heater. The cascade heat pump was operated to meet the constant heating capacity and target intermediate temperature by special control on compressors speed on both sides of the system.

Uhlmann et al. (2014) [10] investigated the performance of heat pump with hybrid heat sources such as buildings with a heat pump and a solar thermal collector or buildings with waste heat recovery. They concluded that the proposed cycle can use varying amounts of waste heat and increases the heat pump efficiency by up to 30% over a wide range of operating conditions. Chae and Choi (2015) [11] investigated the effects of high stage refrigerant charge amount on the performance in a steady state and heating mode operation. The variations of temperature difference at cascade heat exchanger and performance parameters with the high stage refrigerant charge amount were presented. Kim et al. (2015) [12] conducted experimental investigation on R134a/R410A cascade cycle for variable refrigerant flow heat pump systems. (*COP*) showed independence on the inlet water temperature, it increased up to 16% when water inlet temperature decreased. The higher (*COP*) condition was realized when the intermediate temperature was (40-41)°C and the ambient temperature was 7°C, regardless of the water inlet temperature to the high temperature condenser.

Yrjölä and Laaksonen (2015) [13] investigated a ground source heat pump system performance with R407C/R134a. They stated that the optimum condensing temperature of the (LT) to be (35–37)°C when the evaporating temperature of R407C and condensing temperature of R134a are constants at (−5)°C and (65)°C respectively. Minglu et al. (2016) [14] proposed a control strategy for Cascade air source heat pump water heater to adjust the load variation. They concluded that controller developed successfully helped realize the control of the intermediate pressure, the degree of superheat and the evaporating temperature in terms of control accuracy and sensitivity.

In the present work, the thermal performance of eight refrigerant pairs is to be investigated with low temperature heat source. These refrigerants include several categories of chemical structures; such as pure, blend, non-Chlorine and non-Halogen refrigerants. The sea water represents the low temperature heat source and 30% ethylene glycol-water mixture is the thermal fluid carrier to the heat pump system. The LT evaporation temperature is ranged between (-10 and -2)°C and the HT condensation temperatures are (70)°C and (75)°C to produce hot water at the range of (60 to 65)°C out of the heat pump.

2. Thermodynamic Analysis

2.1. Cascade System

This Cascade system consisted of two separate cycles connected through a cascade heat exchanger as shown in Figure 1. Both cycles composes of the principle components of any refrigeration cycle. The low temperature evaporator is provided with heat load from the sea water source by a thermal fluid carrier and the high temperature cycle condenser produces the required hot water demand.

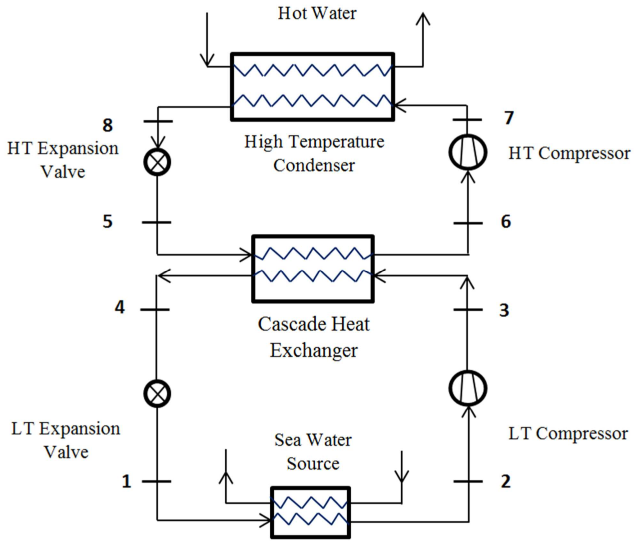

Figure 1. *A schematic diagram for a Cascade system.*

A schematic p-h diagram of this cycle is illustrated in Figure 2 for a typical Cascade heat pump system.

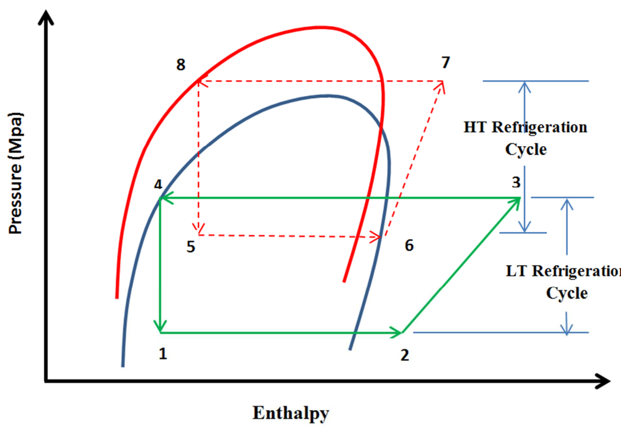

Figure 2. *A schematic p-h diagram for a Cascade system.*

In both evaporators discharge ports, it is assumed that gases leave as superheated condition prior to the compression process in points (2) and (6) for the LT and HT cycle s respectively. The basic thermodynamic relations are applied to each component of the Cascade heat pump system shown in Figure 1. The notations used in Figure 2 can be implemented for the thermal analysis with the aid of the known input heat load of each cycle.

2.2. Low Temperature Cycle

The sea water heat source provides heat load to the (LT) evaporator, stream path (1-2) in Figure (2), (30%) ethylene glycol/water mixtures enters at a temperature of (5)°C. The temperature variation through the evaporator is represented by Figure 3. The mass flow rate of refrigerant is estimated from the heating load, energy balance and adiabatic boundary condition as:

$$\dot{m}_{LT} = \frac{\dot{Q}_{LT,evap}}{(h_2-h_1)} \tag{1}$$

The cascade heat exchanger works as an evaporator for the high temperature (HT) cycle between (5) and (6) stream points. The condensation load of this heat exchanger is estimated as:

$$\dot{Q}_{cascade} = \dot{m}_{LT} \times (h_3 - h_4) \tag{2}$$

The power consumption by (LT) compressor may be calculated from:

$$\dot{W}_{LT} = \dot{m}_{LT}(h_3 - h_2) \tag{3}$$

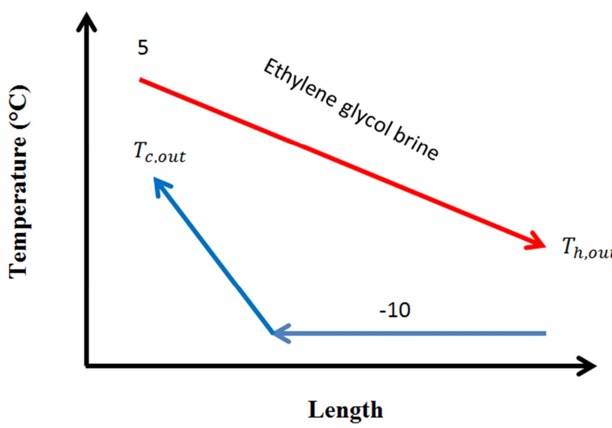

Figure 3. Ethylene glycol solution/low temperature cycle at low temperature evaporator for sea water source.

2.3. High Temperature Cycle

The refrigerant path of this cycle flows through the cycle (5-6-7 and 8) in Figures (1 & 2). The evaporator is represented by the path bounded by points (5) and (6) which extracts heat from the condensation of the (LT) cycle refrigerant in the cascade heat exchanger. The mass flow rate of the refrigerant is estimated from the energy balance through the cascade heat exchanger to give:

$$\dot{m}_{HT} = \frac{\dot{Q}_{cascade}}{(h_6-h_5)} \tag{4}$$

The power consumption of the (HT) compressor is expressed as:

$$\dot{W}_{HT} = \dot{m}_{HT}(h_7 - h_6) \tag{5}$$

The heating load output of the heat pump represents the condensation load at the (HT) condenser located between points (7) and (8) in Figures (1 & 2) is calculated from:

$$\dot{Q}_{HT,cond} = \dot{m}_{HT}\,(h_7 - h_8) \tag{6}$$

2.4. Coefficient of Performance (COP)

It represents the ability of the heat pump to move heat from the low temperature source to a higher level with a lower power input. The higher the value of heat output for a given power input the better thermal performance and economic feasibility will be attained. It is defined as:

$$COP_{System} = \frac{\dot{Q}_{HT,cond}}{\dot{W}_{total,comp}} \tag{7}$$

Where the total power consumption of the cascade system on the refrigerant side is calculated by:

$$\dot{W}_{total,comp} = \dot{W}_{LT} + \dot{W}_{HT} \tag{8}$$

2.5. Cascade Heat Exchanger

The intermediate temperature at the Cascade heat exchanger is an important parameter in the Cascade unit operation. This heat exchanger is shown in Figure 1, it is located between stream points (5-6) and (3-4) of the (HT) and (LT) sides of the Cascade system respectively. It represents the mean value of low temperature side condenser and the high temperature side evaporator. Hence, it maintains a proper temperature difference between both streams passing through the Cascade heat exchanger. It is defined as:

$$T_{int} = \frac{(T_{condensing})_{LT} + (T_{evaporating})_{HT}}{2} \tag{9}$$

The choice of the condensing pressure of the (LT) side and evaporating pressure on the (HT) side are determined from allowing a suitable saturation temperature difference between both streams. A value of (5-15)°C for the temperature difference between the two streams through the Cascade heat exchanger showed a good experience in the cascade system design.

3. Evaluation Methodology

3.1. Refrigerant Pairs

The Cascade system will be examined with the refrigerant pairs listed in table 1 for performance comparison.

Table 1. *Some selected physical properties of the suggested refrigerants.*

Property	**R-407C**	**R-410A**	**R744**	**R134a**	**R717**	**R290**	**R600a**
Composition and Refrigerant (Formula)	R32/125/134a (23/25/52)% by Weight	R32/125 (50/50)% by Weight	CO_2 (100)%	CF_3CH_2F (100)%	NH_3 (100)%	C_3H_8 (100)%	C_4H_{10} (100)%
Molecular Weight (kg/kmol)	86.2	72.58	44.01	102.03	17.03	44.1	58.12
Normal Boiling Point (°C)	-43.4	-51.58	-78.5	-26.06	-33.3	-42.09	-11.7
Temperature Glide (°C)	7.4	<0.2	0	0	0	0	0
Critical Pressure (MPa)	4.62	4.926	7.284	4.0603	11.3	4.25	3.64
Critical Temperature (°C)	86.2	72.13	31	101.08	132.4	96.70	135
Ozone Depletion Potential	0	0	0	0.005	0	0	0
Global Warming Potential	1600	1725	1	1430	0	about 20	about 20

The following conditions were considered for the purpose of comparison between the postulated systems:

- Cascade system with useful superheat degree in evaporators of (3-6)°C and subcool degree of (2)°C in condensers for both cycles. Superheat and subcool degrees in piping system were (3)°C and (1)°C on both cycles.
- The (HT) evaporation and condensation were set at (26 and 30)°C and (70 and 75)°C respectively with (LT) condenser at (40)°C. These conditions produced intermediate temperatures at the cascade heat exchanger of (33)°C and (35)°C.
- The CO2 system was investigated at HT evaporator temperature of (15 and 20)°C with LT condensation at (25)°C, it is a subcritical system. These conditions produced intermediate temperatures at the cascade heat exchanger of (20)°C and (22.5)°C.
- Evaporation temperature at the low side evaporator was ranged between (-10) and (-2)°C
- The compressors are operating at (70%) and (80%) isentropic and volumetric efficiencies respectively with (10%) heat loss.
- Heat load input of (305) kW from the sustainable heat source, sea water, at entering temperature of (5)°C for 30% ethylene glycol –water mixture.
- This extracted load produces output loads approximately (465-550) kW and (480-580) at HT condensation of (70)°C and (75)°C respectively for compressors isentropic efficiency of (70%).

The eight investigated refrigerant pairs as cascade systems with the above conditions were divided into two main categories, they are:

- Systems compose of R717/R134a, R410A/R134a, R407C/R134a, and R717/R600a were tested at (33)°C and (35)°C intermediate temperatures.
- Systems compose of R744/R134a, R744/R290, R744/R600a, and R744/R717 were investigated at (20)°C and (22.5)°C intermediate temperatures.

3.2. Source Temperature Level

A detailed analysis was conducted to compare the thermal performance of the Cascade system at various operating conditions. The performance analysis was based on a fixed heat input through the evaporator of the low temperature cycle. The sea water temperature magnitude was set at (7)°C and to be discharged back to the sea at (4)°C. The 30% ethylene glycol/water brine freezing temperature is found to be (-14)°C. The temperature to which the brine is heated from sea water can be estimated from the energy balance through the sea water/brine heat exchanger. The discharge temperature of the brine from the heat exchanger may be set lower or higher than the exit temperature of the sea water. This is controlled by the size of the heat exchanger and its available surface area. Hence, the discharge temperature of (5)°C was considered, this value is close enough to the exit sea water temperature as shown in Figure 4. Its exit temperature depends on the LT evaporation temperature operating conditions and usually is well above the freezing point.

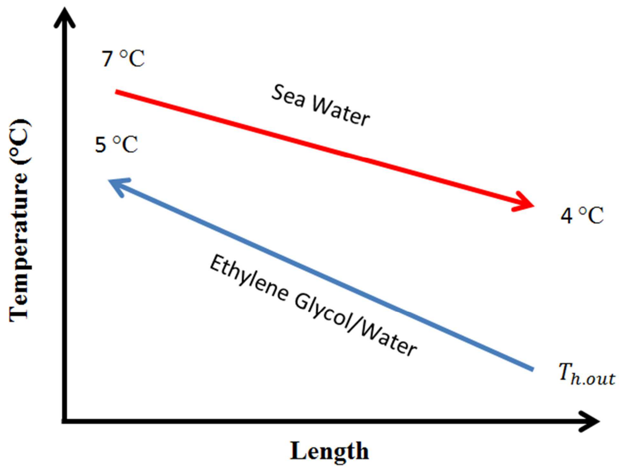

Figure 4. *Sea water/Ethylene Glycol solution heat exchanger.*

3.3. Hot Water Discharge Temperature

The hot water discharge temperature determines the highest temperature magnitude for the cascade system. It represents the limit for the operating pressure and condensing temperatures of the high temperature cycle. This is because the output of the unit represents the target temperature of water to be supplied to the consumers. Hence, two temperature levels were investigated to find out the proper high temperature of the unit. These were (70)°C and (75)°C for the condensing of the high temperature refrigerant side. This range will provide flexibility for the discharge water temperature out of the Cascade unit and a proper temperature driving force.

4. Results and Discussion

The proposed systems were tested and compared in two different intermediate temperature domains. The first includes the high temperature range for the first category in two specified values, are (33)°C and (35)°C. The second covers the domain which includes the (20)°C and (22.5)°C intermediate temperatures.

4.1. Heating Load Output

The heating load obtained from the Cascade system represents the HT condenser load. Hot water at the range of (60-65)°C is to be achieved from the cascade unit which implements the sea water as a heat source. The first category implements (R134a) on the HT cycle and circulates R407C, R410A, and R717 on the LT cycle. The base system for the second category implements (R744) to be circulated in the low temperature cycle and R134a, R290, R600a and R717 are circulated in the high temperature cycle. Figure 5 shows a comparison for the load output of both system categories at HT condensation of (70)°C and different intermediate temperatures.

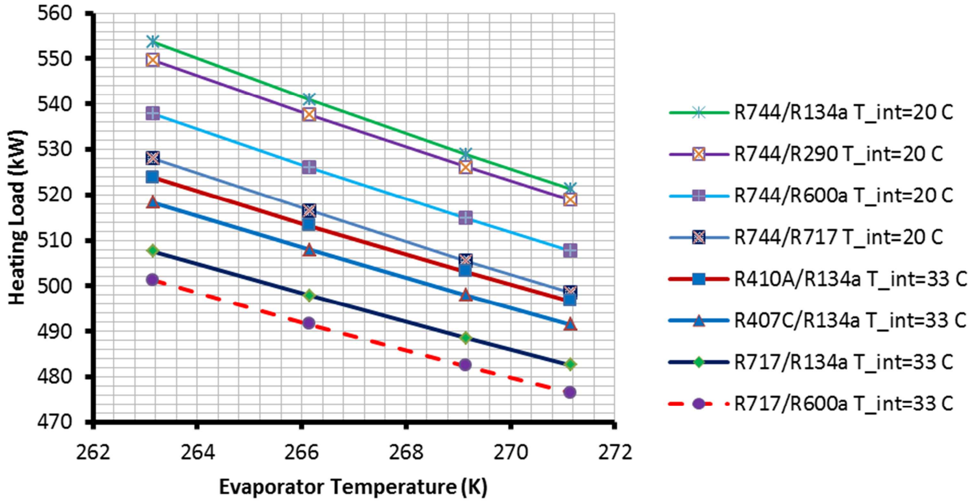

Figure 5a. *Heat pump heating load comparison of different systems at HT condensation of (70)°C with (20)°C and (33)°C intermediate temperatures.*

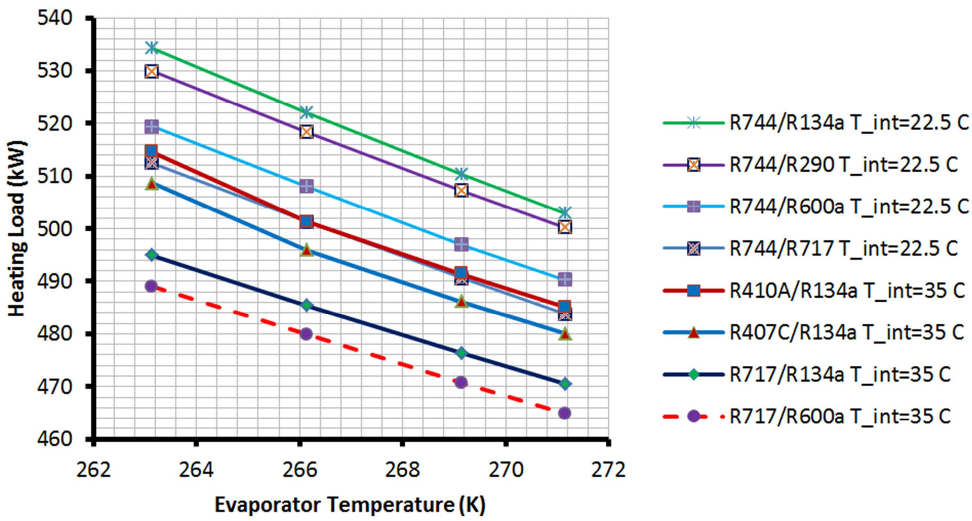

Figure 5b. *Heat pump heating load comparison of different systems at HT condensation of (70)°C with (22.5)°C and (35)°C intermediate temperatures.*

It is obvious from these curves that a higher heating load level is achieved with the lower intermediate temperatures and lower evaporation temperature at the LT evaporator zones. The data revealed that R744 system category exhibited a higher heating load than that obtained by the other category. This is partly due to the higher pressure level at the cascade heat exchanger and power consumed to achieve the temperature lift on the HT cycle.

The highest load output is obtained with R744/R134a system at intermediate temperature of (20)°C ranged between (521) kW and (554) kW achieved at (-2)°C and (-10)°C respectively. These values are higher than those obtained by the same system at (22.5)°C by (3.6%) for the whole range of LT evaporator temperature. For the R134a system, the highest heating load was exhibited with R410A/R134a system at (33)°C intermediate temperature to be within the range of (497) kW to (524) kW at (-2)°C and (-10)°C respectively. These loads are higher than those achieved at

(35)°C by (2)% and (2.5)% at (-2)°C and (-10)°C respectively.

Figure 6 illustrates a comparison of all systems at HT condensation temperature of (75)°C for the investigated intermediate temperatures. The produced heating load values here are higher than those obtained at (70)°C and exhibiting the same trend at the various temperature intermediates.

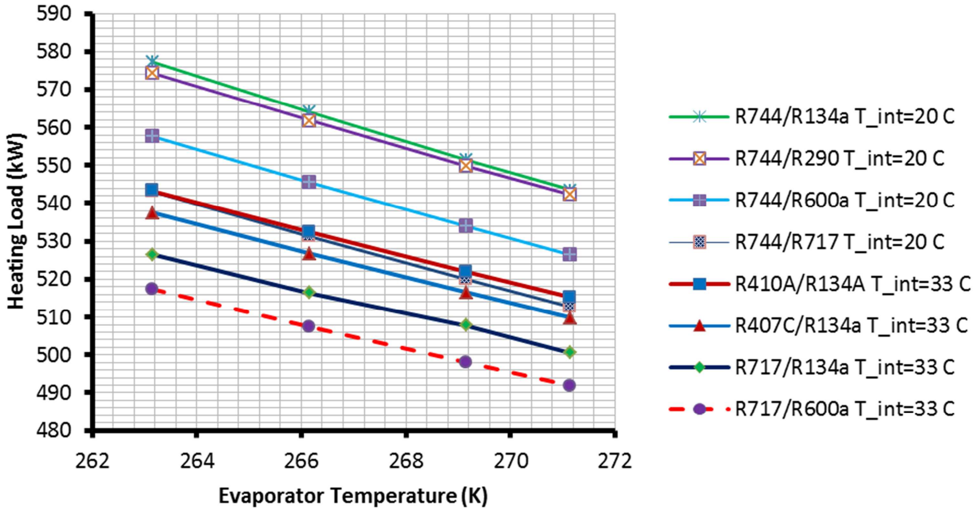

Figure 6a. *Heat pump heating load comparison of different systems at HT condensation of (75)°C with (20)°C and (33)°C intermediate temperatures.*

R717/R600a showed the lower heating output at (70)°C and (35)°C for the HT condensation and intermediate temperatures respectively. The heating load is ranged between (465) kW and (490) kW at (-2)°C and (-10)°C respectively. R744/R134a and R744/R290 revealed almost the same heating loads due to the close pressure ratio on the HT cycle and temperature-pressure variation of the HT refrigerants. This behavior will be reflected on the thermal characteristics such as power consumed and discharge temperature of compressors which produces similar behavior.

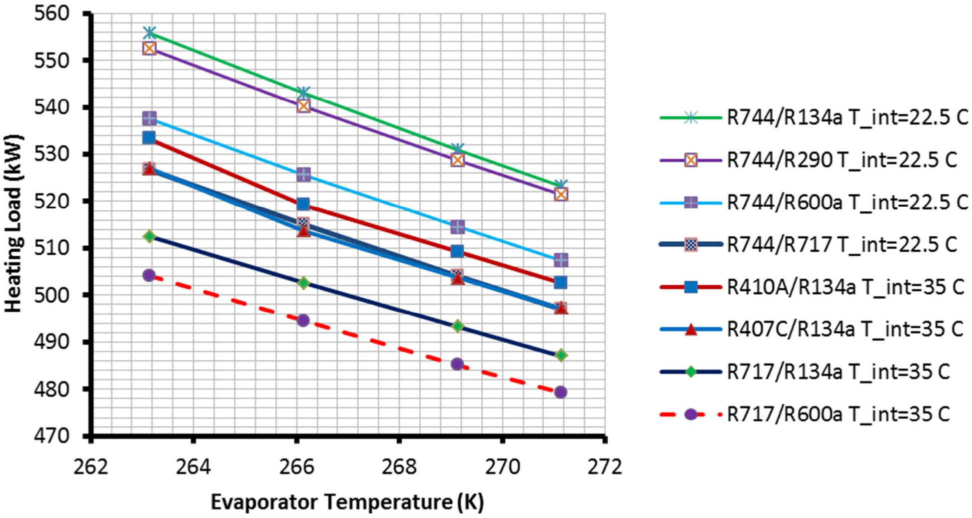

Figure 6b. *Heat pump heating load comparison of different systems at HT condensation of (75)°C with (22.5)°C and (35)°C intermediate temperatures.*

The middle zone of heating load output composes of (R744/R717, R410A/R134a and R407C/R134a) systems. These Cascade system arrangements provide close values regardless of the high temperature condensation, intermediate temperature and LT evaporator temperature, Figures (5 & 6).

4.2. Power Consumption

The power consumption by compressors of the test refrigerant systems depends mainly on their efficiency and technology. The compressors manufacturers state that the isentropic efficiency of their products is laid in the range of (70%) to (90%). The choice of isentropic efficiency is of a vital importance in the thermal performance assessment of the Cascade system (*COP*) and hence its thermal efficiency. The evaluation of performance of the studied systems was based on the implementation of the lower isentropic efficiency. Any increase in this factor will of course enhance the (*COP*) of the system considerably.

Figure 7 shows a comparison of the power consumption by compressors for the investigated refrigerant pairs at (70)°C and different intermediate temperatures. R744/R134a system showed the highest power consumption of all of the refrigerant pairs of this study regardless of the intermediate, HT condenser and LT evaporator temperature. The power consumed by the R744/R134a and R744/R290 systems exhibited close values for the whole range of test temperatures due to the similar temperature-pressure behavior of both systems, Figure 7.

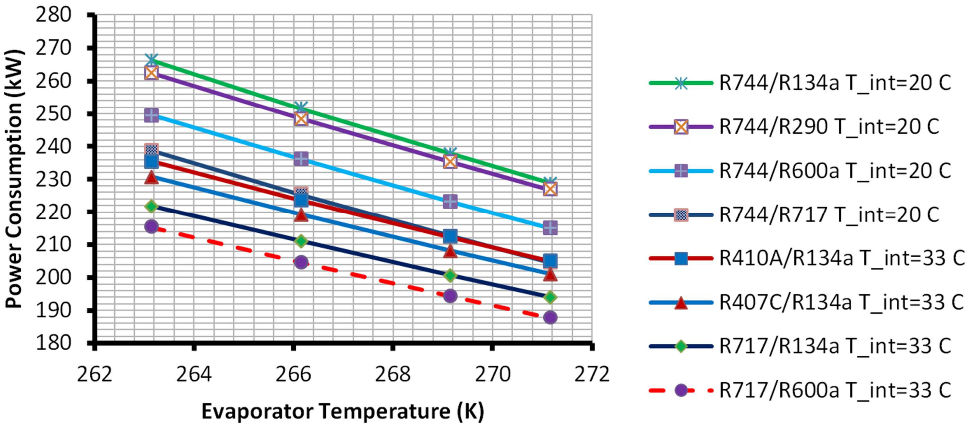

Figure 7a. *Heat pump power consumption comparison of different systems at HT condensation of (70)°C with (20)°C and (33)°C intermediate temperatures.*

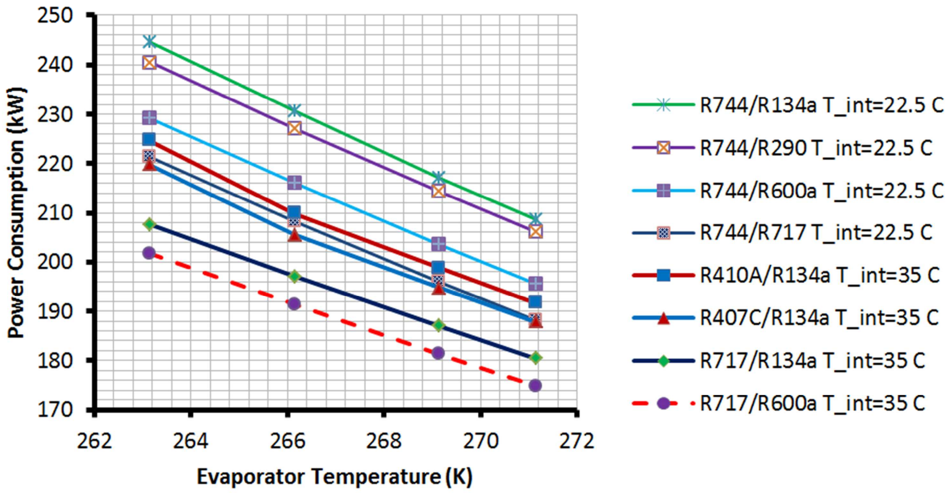

Figure 7b. *Heat pump power consumption comparison of different systems at HT condensation of (70)°C with (22.5)°C and (35)°C intermediate temperatures.*

Low intermediate temperature showed a higher power consumption band than that of the low one for the studied LT evaporator temperature. This is mainly due to the higher temperature lift existed at the low intermediate temperature compared to that at the higher value at the cascade heat exchanger.

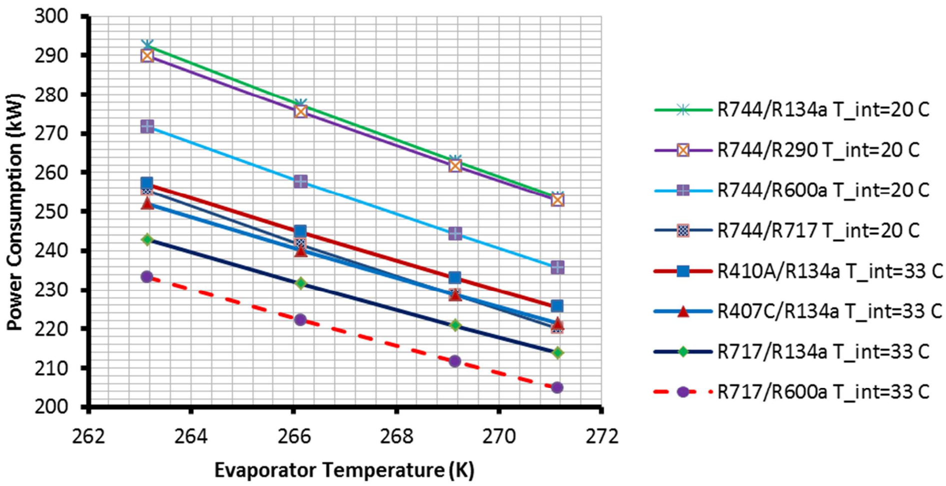

Figure 8a. *Heat pump power consumption comparison of different systems at HT condensation of (75)°C with (20)°C and (33)°C intermediate temperatures.*

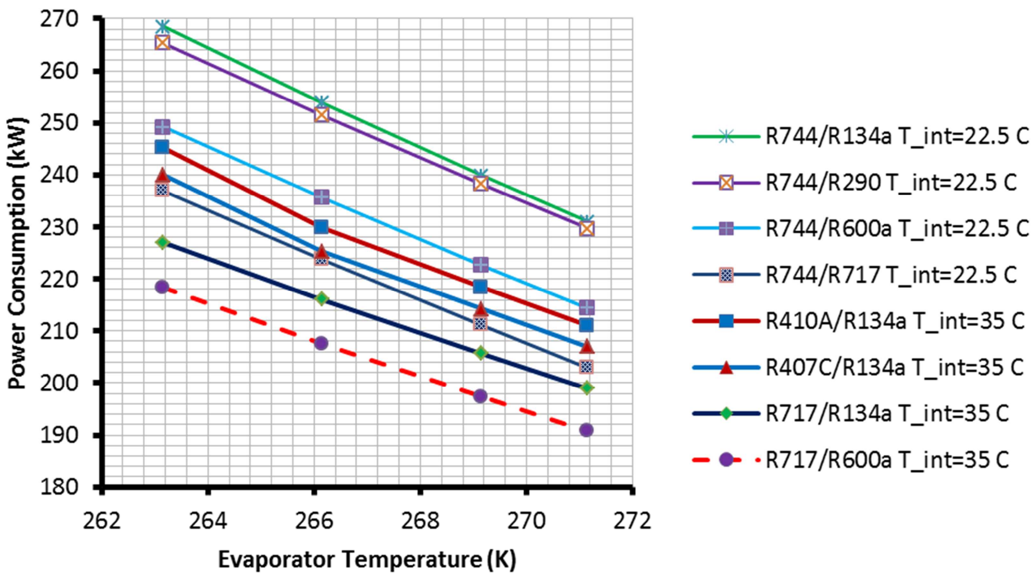

Figure 8b. *Heat pump power consumption comparison of different systems at HT condensation of (75)°C with (22.5)°C and (35)°C intermediate temperatures.*

Increase of the HT condensation temperature to (75)°C raised the power consumption by compressor for all of the refrigerant pair systems with various magnitudes. The lower intermediate temperatures exhibited higher power consumption than those at higher values zone as shown in Figure 8. The lower power consumption was obtained for the R717/R600a system at HT condenser of (70)°C and (35)°C intermediate temperature. It is ranged between (175) kW and (202) kW for LT evaporator temperature of (-2)°C and (-10)°C respectively. This is mainly due to the low temperature lift and pressure ratio required in order to attain the specified conditions at the HT cycle. The R717/R600a and R717/R134a systems showed the highest compressor discharge temperature of R717 at the LT cycle among the rest of test refrigerant pairs, it was about (156)°C. The R744 systems exhibited the highest compressor discharge pressure at the LT cycle, it was about (64) bar.

The compressors specific power consumption is defined as:

$$\acute{W}_{comp} = \frac{3.5}{COP} \tag{10}$$

It represents the power consumed by the compressors in kW per (3.5) kW heating load produced from the Cascade heat pump system. Figure 9 shows a comparison of $\acute{W}_{comp}$ between the test refrigerant pair systems at intermediate temperature of (33)°C and HT condensation temperatures of (70)°C and (75)°C.

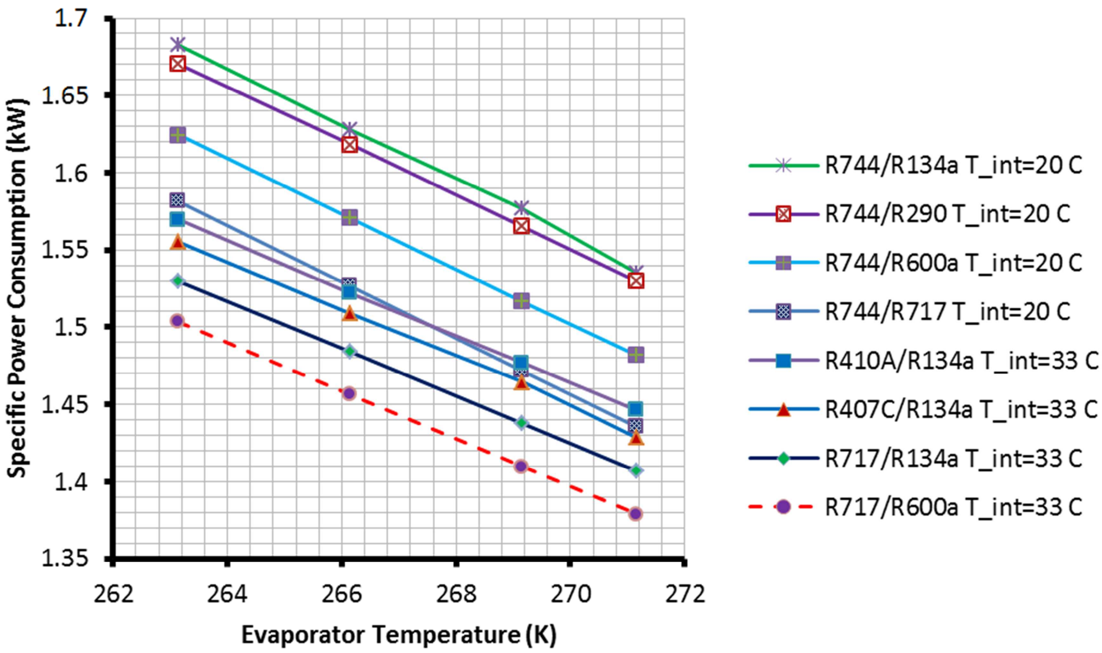

Figure 9a. *Heat pump specific power consumption comparison of different systems at HT condensation of (70) C with (20)°C and (33)°C intermediate temperatures.*

The variation of the specific power consumption showed an increase as the LT evaporation temperature decreases. The specific power consumed by the R744/R134a and R744/R290 systems exhibited close values and were the higher values among the test refrigerants. It is ranged between 1.6 kW and 1.8 kW at (-2)°C and (-10)°C respectively when the HT condensation temperature was (75)°C. The corresponding values at (70)°C were (1.5) kW and (1.7) kW at (-2)°C and (-10)°C.

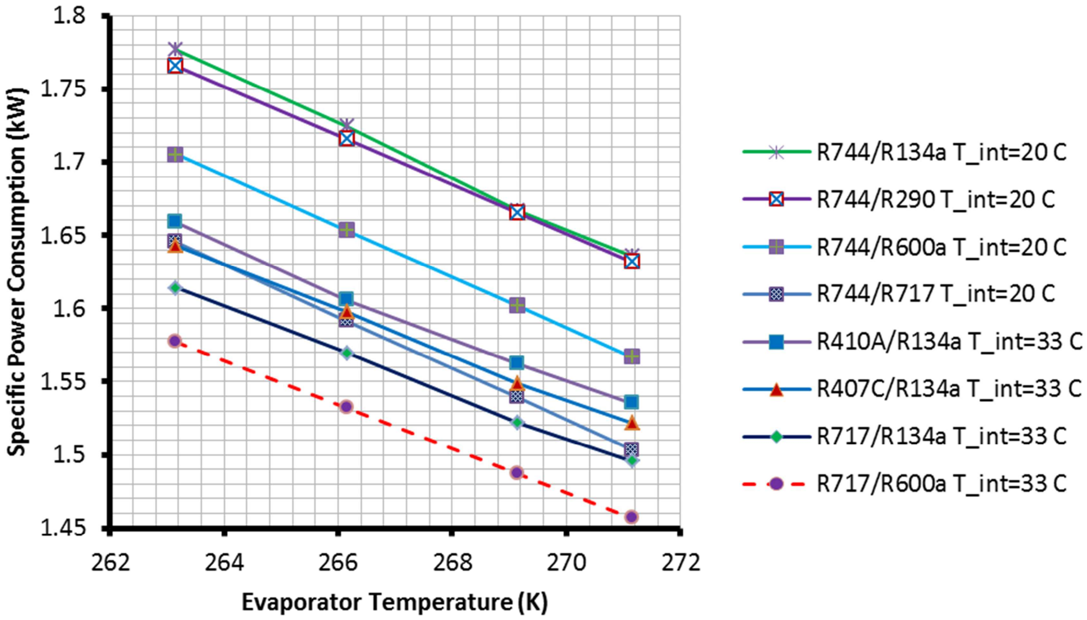

Figure 9b. *Heat pump specific power consumption comparison of different systems at HT condensation of (75) C with (20)°C and (33)°C intermediate temperatures.*

R717/R600a exhibited the lower $\acute{W}_{comp}$ among the test systems showing (1.5) kW to (1.6) kW for the test LT evaporator temperature range at HT condensation of (75)°C. The rest of test systems lay in between R717/R600a and R744/R134a boundary limits of power consumed.

4.3. Coefficient of Performance (COP)

The thermal performance of a heat pump is usually determined by the coefficient of performance COP criterion. Figure 10 shows a comparison for the eight refrigerant pair systems at (70)°C at the test conditions. R717/R600a system revealed the highest COP among the studied refrigerant pair systems for the whole range of intermediate, HT condenser and evaporator temperature. It showed a value of (2.4-2.7) at (35)°C and (2.3-2.5) at (33)°C for LT evaporating temperature range between (-2 and -10)°C.

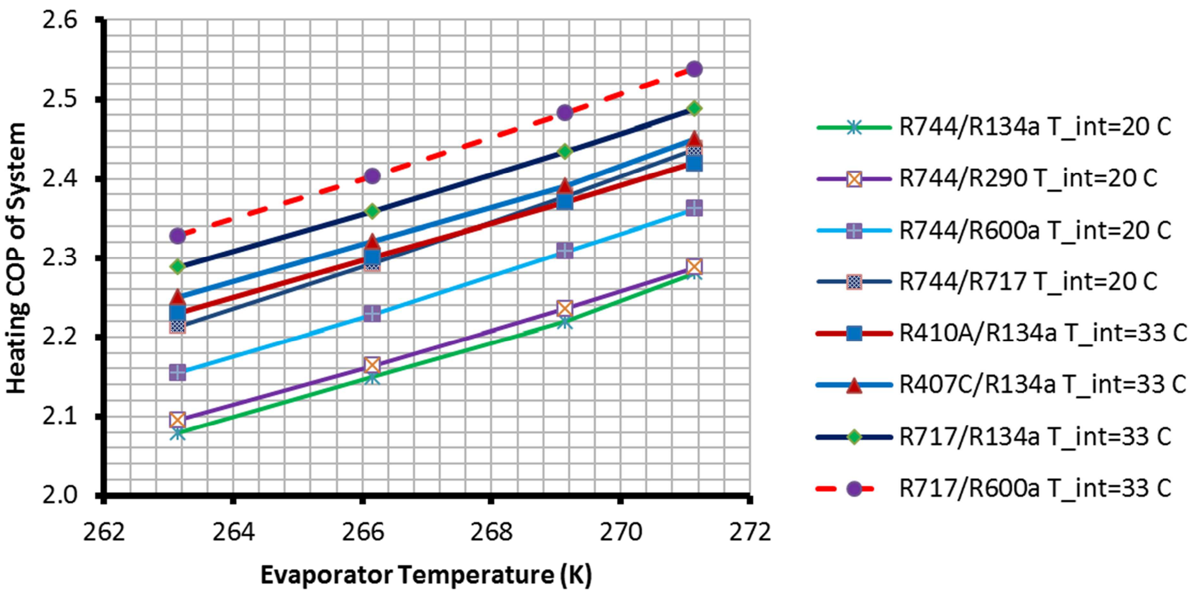

Figure 10a. *Heat pump heating COP comparison of different systems at HT condensation of (70)°C with (20)°C and (33)°C intermediate temperatures.*

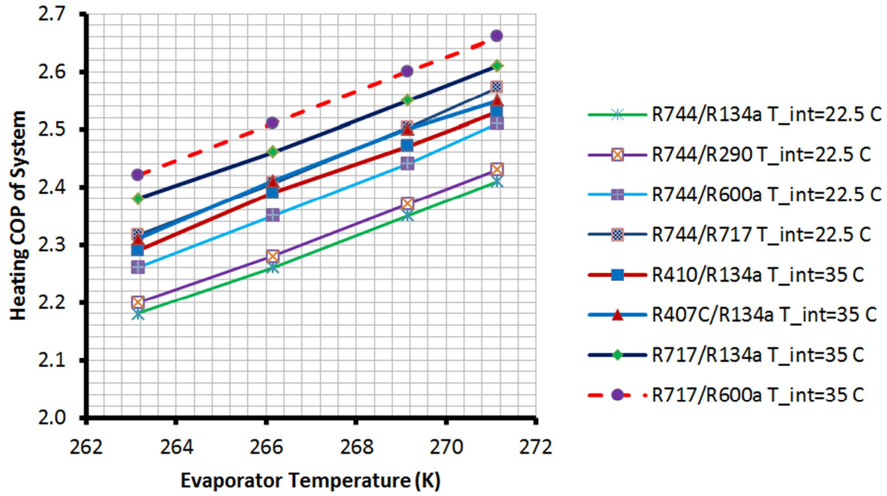

Figure 10b. *Heat pump heating COP comparison of different systems at HT condensation of (70)°C with (22.5)°C and (35)°C intermediate temperatures.*

R744/R134a and R744/R290 achieved the lowest COP among the test refrigerant pairs for Cascade systems regardless of the temperature operating conditions. It was between (2.2-2.4) at (22.5) C and (2.1-2.3) at (20) C for LT evaporator temperature range (-10 to -2) C. Figure 11 represents a comparison for the COP of the studied system operating at HT condensation temperature of (75)°C. Again the same trend has been shown by the behavior of COP variation with the operating conditions.

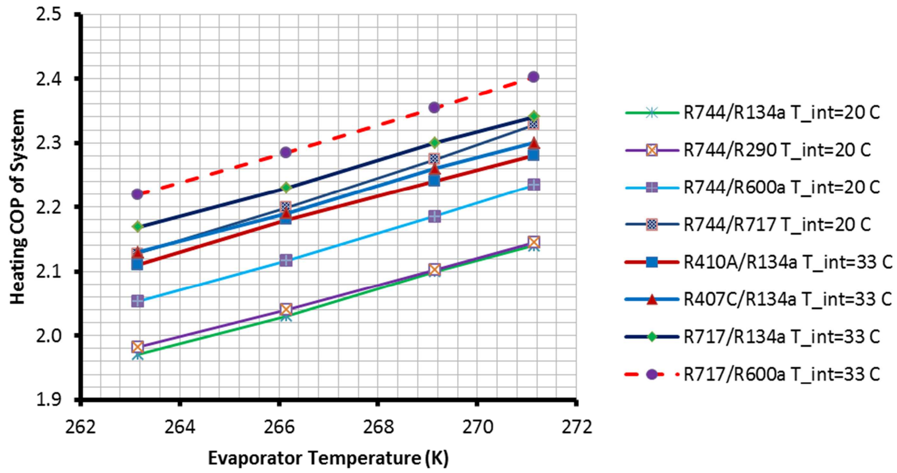

Figure 11a. *Heat pump heating COP comparison of different systems at HT condensation of (75)°C with (20)°C and (33)°C intermediate temperatures.*

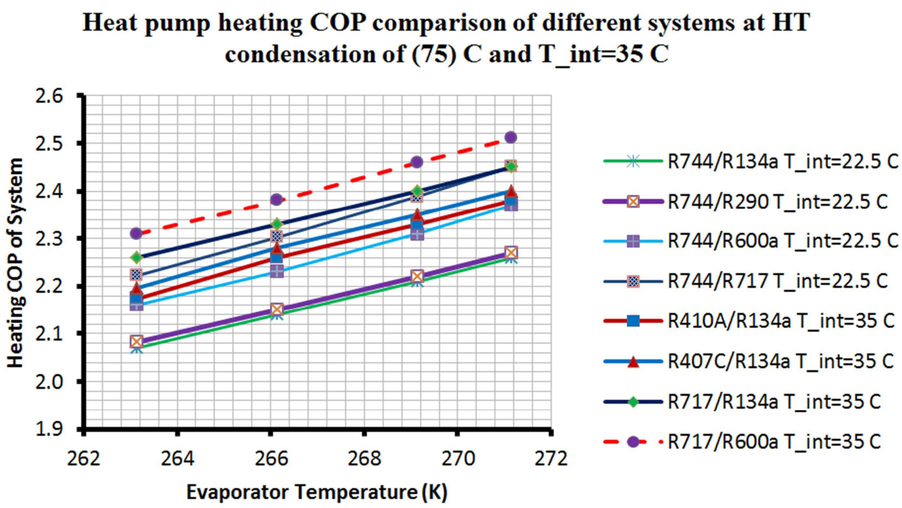

Figure 11b. *Heat pump heating COP comparison of different systems at HT condensation of (75)°C with (22.5)°C and (35)°C intermediate temperatures.*

The R744/R134a and R744/R290 revealed the lowest COP among the test refrigerant pairs for Cascade systems regardless of the temperature operating conditions. It was between (2.1-2.3) at (22.5) C and (2.0-2.1) at (20)°C for LT evaporator temperature range (-10 to -2)°C. R717/R600a system revealed the highest COP, it showed a value of (2.3-2.5) at (35)°C and (2.2-2.4) at (33)°C for LT evaporating temperature range between (-2 and -10)°C.

The results showed that COP of the test systems varied between (2) to (2.7) for the whole range of the selected operating temperature at the LT and HT cycles. The rest of the test systems lay in between these values with different magnitudes. R407C/R134a and R410A/R134a systems occupy the central domain for the band of COP range and could be selected for the heat pump design with sea water heat source. The performance of both systems was also confirmed by Tarrad (2017) [15]. They showed close COP at the same LT evaporator and intermediate temperature ranges. The COP values could be increased considerably when the power consumption by compressors was minimized. Increasing the isentropic efficiency of the compressor to (90)% might increase the COP values by about (20)% to reach a value of (3.2) at the highest range.

The results presented in Figures (10 & 11) revealed that the highest COP was achieved when the LT evaporator works at (-2)°C evaporation temperature. This is the criterion for all the investigated refrigerant pairs and operating conditions mainly due to the low power consumption by compressors. Table 2 represents a comparison between three of the test systems which exhibited the best coefficient of performance COP at LT evaporation of (-2)°C. The data showed that R717/R600a achieved the highest numerical values of (2.7) and (174.8) kW for COP and compressor power consumption respectively at (35)°C intermediate temperature and (70)°C HT cycle condenser temperature.

Table 2. *Performance comparison at the low side evaporation temperature of (-2)°C.*

Refrigerant Pair	T_int (°C)	Tc_LT (°C)	Tc_HT (°C)	Heating Load (kW)	Power Input (kW)	COP (---)
R717/R134a	33	40	70	482.6	194.0	2.5
			75	500.6	213.9	2.3
	35	40	70	470.5	180.5	2.6
			75	487.1	199.0	2.5
R717/R600a	33	40	70	476.5	187.7	2.5
			75	491.9	204.8	2.4
	35	40	70	464.9	174.8	2.7
			75	479.2	190.7	2.5
R744/R717	20	25	70	498.5	204.5	2.4
			75	512.7	220.2	2.3
	22.5	25	70	483.8	188.1	2.6
			75	497.1	202.9	2.5

5. Conclusions

Refrigerant blends, non-Chlorine and non-Halogen refrigerants were tested in eight refrigerant pair in Cascade system with LT range of (-10 to -2)°C and HT condensation of (70 and 75)°C. The highest power and specific power consumptions were exhibited by R744/R134a and R744/R290 systems and minimum was showed by R717/R600a one. The latter system revealed the best coefficient of performance among the test refrigerants. It was ranged between (2.4-2.7) at (35)°C and (2.3-2.5) at (33)°C intermediate temperature for LT evaporating temperature range between (-2 and -10)°C and HT condensation at (70)°C. R410A/R134a, R407C/R134a and R744/R717 systems occupied the middle zone of the COP band with acceptable operating range.

The highest COP was achieved at the LT evaporation of (-2)°C for all of refrigerant pairs and the highest was achieved by R717/R600a one. The test range of LT evaporator temperature was suitable for low temperature heat source such as sea water. The isentropic efficiency of compressors is important factor in the COP evaluation of a heat pump. Increasing of the compressor isentropic efficiency to (90%) improved the COP to reach a value of (3.2) as a peak value for R717/R600a cascade system.

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Nomenclature

Parameter	Definition
COP	Coefficient of performance (Dimensionless)
h	Fluid enthalpy, (kJ/kg)
$\dot{m}$	Fluid mass flow rate (kg/s)
$\dot{Q}$	Heating load (kW)
T	Temperature (°C)
$\dot{W}$	Compressor power consumption (kW)
$\acute{W}$	Compressor specific power consumption (kW), defined by eq. (10)

Subscripts

cascade	Cascade heat exchanger
comp	Compressor
cond	Condenser
c,out	Exit value on the cold side
evap	Evaporator
h,out	Exit value on the hot side
HT	High temperature side
LT	Low temperature side
total	Total value

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