Thermodynamic Evaluation for Intermediate Temperature Optimization in Low Temperature Heat Source Cascade Heat Pump Technology

Ali Hussain Tarrad

Mechatronic Department, University of Southern Denmark
Denmark
Email: dr.alitarrad [AT] yahoo.com

ABSTRACT--- ***A thermodynamic analysis assessment for the intermediate temperature in the cascade heat exchanger of a Cascade system is outlined. R407C/R134a and R410A/R134a pairs are studied at low evaporating temperature range between (-10 and -2) °C and (70) °C and (75) °C high temperature condenser levels. Low temperature heat sources are considered as the external driving potential sources and brines as thermal fluid carriers of energy. The analysis was based on a target temperature of hot water at the range of (60-70) °C out of the heat pump. Cascade heat exchanger intermediate temperature range of (28.5 to 39) °C was studied. The high extreme intermediate temperature exhibited the best heating COP for both refrigerant systems regardless of LT evaporator or HT condenser temperatures. The minimum isentropic efficiency of commercially available compressors was used in this investigation, a value of (70 %) was chosen. R407C/R134a system achieved only (1 %) higher COP than that of R410A/R134a for the whole test range of intermediate, LT evaporator, and HT condenser temperatures. The results showed that the heat pump heating COP at LT evaporator of (-7) °C and HT condenser temperature of (70) °C showed similar characteristic values to that of LT evaporator (-2) and HT condenser of (75) °C for intermediate temperature range between (31-39) °C. The highest COP was achieved at (-2) LT evaporator temperature and (70) °C HT condenser temperature. It was ranged between (2.2) and (2.8) for the whole test range of intermediated temperature. The lowest heating COP was experienced when the LT cycle refrigerant evaporates at (-10) °C and the HT cycle refrigerant condenses at (75) °C for both refrigerant pair systems. It was ranged between (1.95) and (2.4) at (28.5 to 39) °C intermediate temperature. Increasing of compressors isentropic efficiency to (90 %) has improved the heating COP of both refrigerant systems potentially by more than (20 %) and minimized the power consumption by (24 %) under the same operating conditions.***

Keywords-- Cascade Heat Pump, Intermediate Temperature, Green Environment, Sustainable Energy, Optimum COP

1. INTRODUCTION

The international community has expressed their awareness and concerns towards the environment pollution, Ozone layer depletion and warming potential of earth for few decades ago. Montreal protocol was issued in (1987) and amended in (1990 and 1992); it has been agreed to phase-out of halocarbon refrigerants according to a firm and gradual transition plan. The first geothermal plant in Denmark based on deep wells was established in Thisted in 1984. In Sønderborg, the first heat from the geothermal water was produced early 2013 by implementing absorption heat pumps to produce up to 12 MW (Mahler et al., 2013). The choice of refrigerant pair to circulate in the Cascade heat pump mainly depends on the heat source temperature and the heat load output of the heat pump. Nanxi et al. (2005) developed a ternary mixture of R124/R142b/R600a, for moderately high temperature heat pumps. They demonstrated that the condenser outlet water temperature could reach and hold on about 90 °C with a high coefficient of performance. Zhang et al., (2010) carried out experimental investigations on non-azeotropic refrigerant mixtures composed of R152a and R245fa at different mass fraction composition. A water-to-water heat pump system in the condensing temperature range of (70–90) °C with a cycle temperature lift of (45) °C was implemented. The results showed that all of the mixtures deliver higher discharge temperature, higher heating capacity, and higher COP than R245fa.

Parka and Jung (2009) proposed a mixture composed of R170/R290 as alternative for R22 refrigerant in air-conditioners. Tarrad and Salim (2009); Tarrad and Abbas (2010); Tarrad et al. (2011); and Tarrad et al. (2013) focused on the performance of refrigerant alternatives for R22 circulated in air conditioners and water chillers. Tarrad and Al-Nadawi (2015); Tarrad and Al-Nadawi (2016); and Tarrad et al. (2016) concentrated on the thermal modeling of evaporators and condensers in vapor compression system circulating R407A, R407C and R404a as alternatives for R22. Kim et al. (2015) investigated experimentally R410A/ R134a cascade cycle for variable refrigerant flow heat pump systems. The higher COP condition was obtained when the intermediate temperature was in the range (40-41) °C and the ambient temperature at 7 °C regardless of the water inlet temperature to the high temperature condenser. Yrjölä and

Laaksonen (2015) investigated a ground source heat pump system performance with R407C/R134a. They pointed out that the optimum condensing temperature of the (LT) lies in the (35–37) °C range. This was true when the evaporating temperature of R407C and condensing temperature of R134a are at (−5) °C and (65) °C respectively.

Song et al. (2016) investigated the performance of combined R744/R134a and Cascade R744/ R134a systems for space heating and compared under specific operating conditions. The cascade system performed better at low ambient temperatures. Whereas, the combined system performed better under conditions of high ambient temperature and high hot water temperature differences between the system inlet and outlet. Tarrad (2017, "a") developed a Cascade compound system where both of sea water and ground heat sources were implemented. Intermediate temperature (35) °C showed a higher heating COP than that of (33) °C for both of refrigerant Cascade systems. R717/R134a system exhibited COP increase of about (3 %) higher than that of R410A/R134a system for both of the tested intermediate temperatures. More recently, Tarrad (2017, "b") investigated the thermal performance of eight refrigerant pairs, 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. The R744/R134a and R744/R290 systems revealed 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.

In the present study the effect of intermediate temperature in the cascade heat exchanger and high temperature condensing value on the steady state thermal performance in a heat pump will be outlined. Two refrigerant pairs, R407C/R134a and R410A/R134a were implemented in LT evaporator temperature of (-10 to -2) °C. Two HT condense temperature levels were examined, they were (70) °C and (75) °C when LT condensing temperature of (40) °C was implemented.

2. FUNDAMENTAL CONCEPT

Figure 1 shows the control volume of a heat exchanger and reveals the inlet and outlet parameter for both sides of fluid streams. The increment represents the flow streams for an evaporator where the refrigerant passes through the tube side and the thermal fluid energy carrier passes on the outside of the tube.

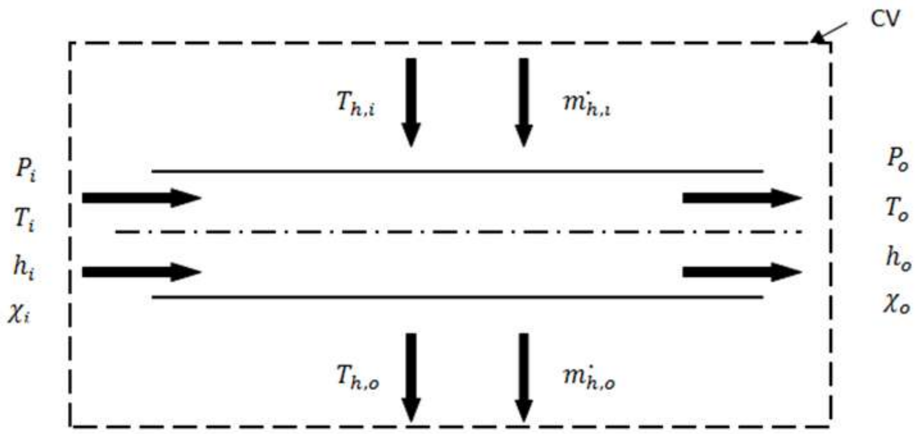

Figure 1: Control volume of an individual tube in a heat exchanger

The following analysis may be applied for evaporator, condenser and cascade heat exchanger regardless of the flow side of refrigerant and thermal fluid carrier.

2.1 Continuity Equation

The conversation of mass principle of control volume can be expressed as:

$$\frac{d}{dt}\int_{CV} \rho dV + \int_{CS} \rho(\vec{V}.\vec{n})dA = 0 \qquad (1)$$

Splitting the surface integral in the above equation into two parts-one for the outgoing flow stream (positive) and one for the incoming streams (negative) then the general conversation of mass relation can be expressed as:

$$\frac{d}{dt}\int_{CV} \rho dV + \sum_{out} \rho V_n dA - \sum_{in} \rho V_n dA = 0 \qquad (2)$$

The parameter (A) represents the cross sectional area of the flow conduit. Equation (2) can be expressed as

$$\frac{d}{dt}\int_{CV}\rho dV = \sum_{in}\dot{m} - \sum_{out}\dot{m} \qquad (3)$$

Or

$$\frac{dm_{CV}}{dt} = \sum_{in}\dot{m} - \sum_{out}\dot{m} \qquad (4)$$

For steady flow process through the heat exchangers equation (4) would be expressed as:

$$\sum_{in}\dot{m} = \sum_{out}\dot{m} \qquad (5)$$

2.2 Energy Equation

The energy equation for a fixed control volume can be expressed by:

$$\dot{Q}_{net,in} + \dot{W}_{shaft,net\ in} = \frac{d}{dt}\int_{CV} e\rho d\mho + \sum_{out}\dot{m}\left(h + \frac{V^2}{2} + gz\right) - \sum_{in}\dot{m}\left(h + \frac{V^2}{2} + gz\right) \qquad (6)$$

For steady state flow, $\frac{d}{dt}\int_{CV} e\rho d\mho = 0$, and typically heat exchangers involve no work interaction and negligible kinetic and potential energy change$\left(\frac{V^2}{2} = gz = 0\right)$. The term $e = u + \frac{V^2}{2} + gz$, is the total energy per unit mass. Rearranging and substituting in (6) yields to:

$$\dot{Q}_{net,in} = \sum_{out}\dot{m}\,h - \sum_{in}\dot{m}\,h \qquad (7)$$

The final expressions of general formulae for the mass conservation and energy balance, eq. (5) and (7) respectively were implemented in the present work. When there is no accumulation of mass or energy in the control volume under steady state operation and negligible heat loss, then:

$$\dot{Q}_{H.Exch.} = \dot{m}_f \times \Delta h \qquad (8)$$

This heat exchanger represents any of the LT evaporator, HT condenser and cascade heat exchanger. Equation (8) is valid regardless of the location of refrigerant and heating medium in the heat exchanger.

3. THERMODYNAMIC ANALYSIS

Figure 2 illustrates a schematic diagram for a Cascade heat pump. The LT evaporator extracts heat from the thermal fluid carrier and the HT cycle condenser produces the required hot water demand. The analysis is basically depending on the implementation of the low temperature heat source such as sea water and geothermal energy.

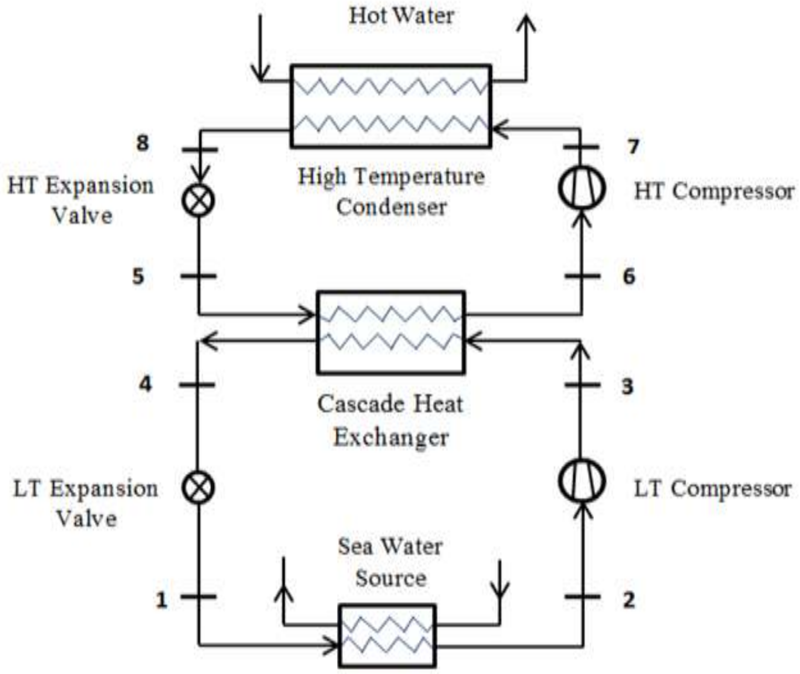

Figure 2: A schematic diagram for a Cascade system

A schematic p-h diagram of this cycle is illustrated in Figure 3 for a typical Cascade heat pump system.

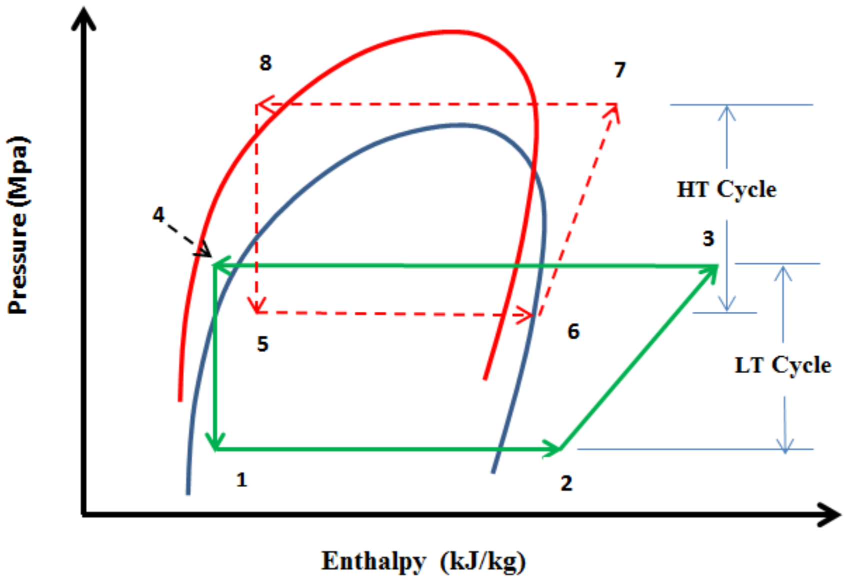

Figure 3: A schematic p-h diagram for a Cascade system

3.1 Low Temperature Cycle

The mass flow rate of refrigerant circulated through the LT evaporator is estimated from the heating load relation presented in eq. (8) as:

$$\dot{m}_{LT} = \frac{\dot{Q}_{LT,evap}}{(h_2 - h_1)} \tag{9}$$

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:

$$\dot{Q}_{cascade} = \dot{m}_{LT} \times (h_3 - h_4) \tag{10}$$

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

$$\dot{W}_{LT} = \dot{m}_{LT}(h_3 - h_2) \tag{11}$$

3.2 Cascade Heat Exchanger

The purpose of the present work is focused on the intermediate temperature through this heat exchanger. It is located between stream points (5-6) and (3-4) of the (HT) and (LT) sides of the Cascade system respectively. The intermediate temperature 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:

$$T_{int} = \frac{(T_{condensing})_{LT} + (T_{evaporating})_{HT}}{2} \tag{12}$$

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.

3.3 High Temperature Cycle

The refrigerant path of this cycle flows through the cycle (5-6-7 and 8) in Figures 2 and 3. 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 HT cycle refrigerant flow rate is estimated from the energy balance through the cascade heat exchanger as:

$$\dot{m}_{HT} = \frac{\dot{Q}_{cascade}}{(h_6 - h_5)} \tag{13}$$

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

$$\dot{W}_{HT} = \dot{m}_{HT}(h_7 - h_6) \qquad (14)$$

The heating load output of the heat pump represents the condensation load at the (HT) condenser located between points (7) and (8) in Figures 2 and 3 is calculated from:

$$\dot{Q}_{HT,cond} = \dot{m}_{HT}\,(h_7 - h_8) \qquad (15)$$

3.4 Coefficient of Performance (*COP*)

The coefficient of performance of the heat pump is estimated from:

$$COP_{System} = \frac{\dot{Q}_{HT,cond}}{\dot{W}_{total,comp}} \qquad (16)$$

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

$$\dot{W}_{total,comp} = \dot{W}_{LT} + \dot{W}_{HT} \qquad (17)$$

4. METHODOLOGY

4.1 Case Study

The investigated refrigerant pairs are shown in Table 1 where some selected operating properties are listed. These refrigerants are widely used in the heat pump technology as friendly environment refrigerants with low global warming potential and zero Ozone depletion potential. This is in agreement with the general trend of heat pump technology to phase out the Chlorine and Halogen refrigerant compositions.

The work of (Kim et al., 2015) for a Cascade heat pump circulates R410A/R134a and (Yrjölä and Laaksonen, 2015) for R407C/R134a pair will be recalled. The first found that the best COP was revealed for intermediate temperature in the range (40-41) °C and the ambient temperature at 7 °C. The second pointed out that the optimum condensing temperature of the LT cycle in the (35–37) °C. This was for a ground heat source when the evaporating temperature of R407C and condensing temperature of R134a are at (−5) °C and (65) °C respectively. The previous work conducted by (Tarrad, 2017, “a”; Tarrad, 2017, “b”) for low temperature heat sources such as sea water and ground in Cascade heat pumps will also be considered as well. Hence, a wider range for both LT and HT evaporator temperatures were chosen.

Table 1: Some selected physical properties of the suggested refrigerants

Property	**R-407C**	**R-410A**	**R134a**
Composition and Refrigerant (Formula)	R32/125/134a (23/25/52) % by Weight	R32/125 (50/50) % by Weight	CF3CH2F (100) %
Molecular Weight (kg/kmol)	86.2	72.58	102.03
Normal Boiling Point (°C)	-43.4	-51.58	-26.06
Temperature Glide (°C)	7.4	<0.2	0
Critical Pressure (MPa)	4.62	4.926	4.0603
Critical Temperature (°C)	86.2	72.13	101.08
Ozone Depletion Potential	0	0	0.005
Global Warming Potential	1600	1725	1430

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

- Cascade system with useful superheat degree in evaporators of (6) °C and subcool degree of (2) °C in condensers for both cycles. Superheat degrees in piping system were (3) °C and (1) °C at the LT and HT cycles respectively were considered.
- The (HT) evaporation and condensation temperatures were set at (17 - 38) °C and (70 and 75) °C respectively with (LT) condenser at (40) °C, Table 2. 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.

- The sustainable heat load of (150) kW is extracted from sea water or ground, at (7) °C. Thermal fluid carrier entering temperature of (5) °C to the LT evaporator.

The available code (CoolPack), (DTU, 2001) for vapor compression refrigeration cycles was implemented wherever it is needed to collect the physical properties of the analyzed refrigerants and assessment verification objectives.

Table 2: Operating conditions of R-410A/R134a and R-407C/R134a Cascade systems.

LT Refrigerant	$(T_{condensing})_{LT}$ (°C)	$(T_{evaporating})_{HT}$ (°C)	$(\Delta T)_{Cascade,Exch}$ (°C)	$T_{intermediate}$ (°C)/(K)
R-410A & R-407C	40	17	23	28.5/301.65
	40	22	18	31/304.15
	40	26	14	33/306.15
	40	28	12	34/307.15
	40	30	10	35/308.15
	40	32	8	36/309.15
	40	34	6	37/310.15
	40	38	2	39/312.15

4.2 Evaluation Criteria

The assessment of test systems is basically dependent on a fixed load extracted from a sustainable energy source. The target evaluation criterion is represented by the heating coefficient of performance COP expressed in eq. (16) but not the heat load output of the heat pump represented by eq. (15). This is simply because the HT condensation load is a dependent variable on the power consumed and hence different heating output will be obtained in regards to operating condition.

5. RESULTS AND DISCUSSION

5.1 Compressors Power Consumption

Figure 4 illustrates a comparison for the power consumed by the test refrigerant pairs at both HT condensation temperature of (70 and 75) °C. Despite, the results showed that R410A/R134a system exhibited higher power consumption than that of R407C/R134a for the whole test range, but it was insignificant. The discrepancy between both systems was within (2 %) regardless of intermediate, LT evaporator and HT condenser temperature.

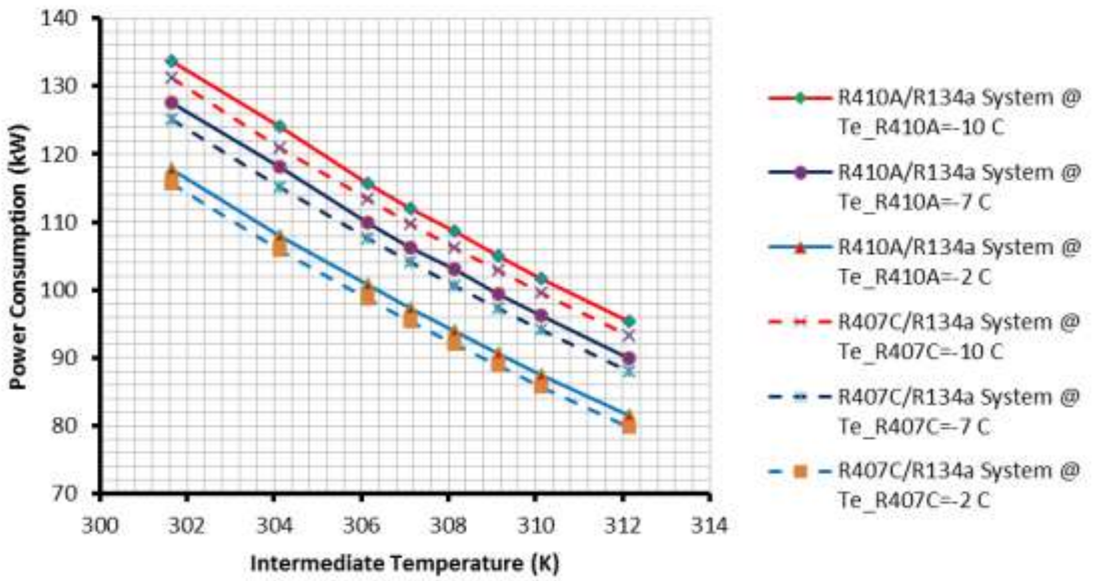

Figure 4a: High temperature cycle condensation at (70) °C

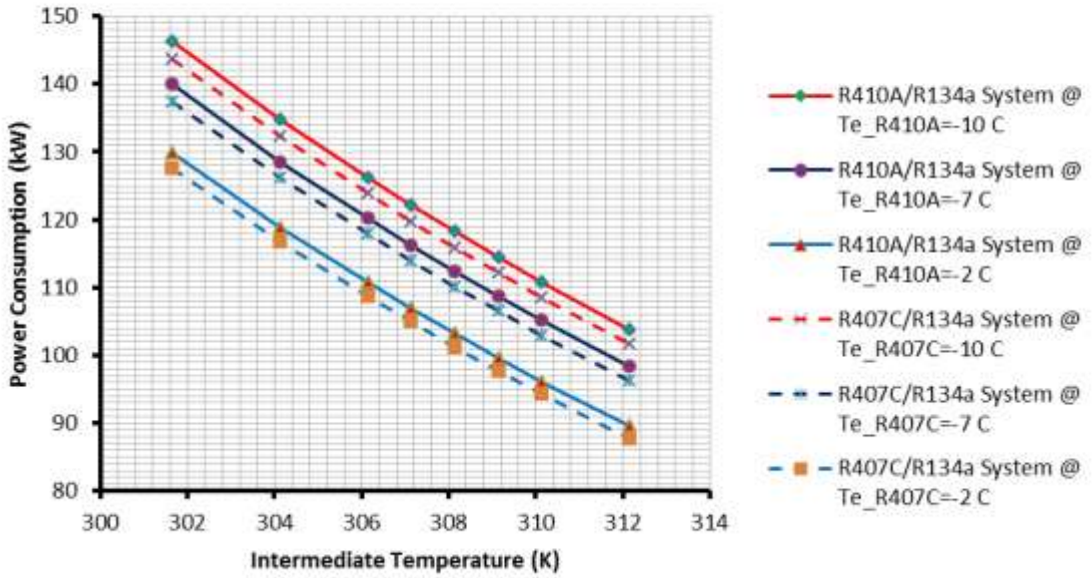

Figure 4b: High temperature cycle condensation at (75) °C

Figure 4: Power consumption comparison of both systems at LT condensation of (40) °C

The power consumed by compressors of the Cascade system revealed that it was highest at the lower LT cycle evaporator of (-10) °C for both systems regardless of the HT condensation temperature level, Figure 5. This is mainly due to the higher power consumed at the LT cycle compressor to achieve the condensation at (40) °C. In other words, higher pressure ratio is required on the LT cycle for (-10) °C than that required for (-2) °C evaporation temperature.

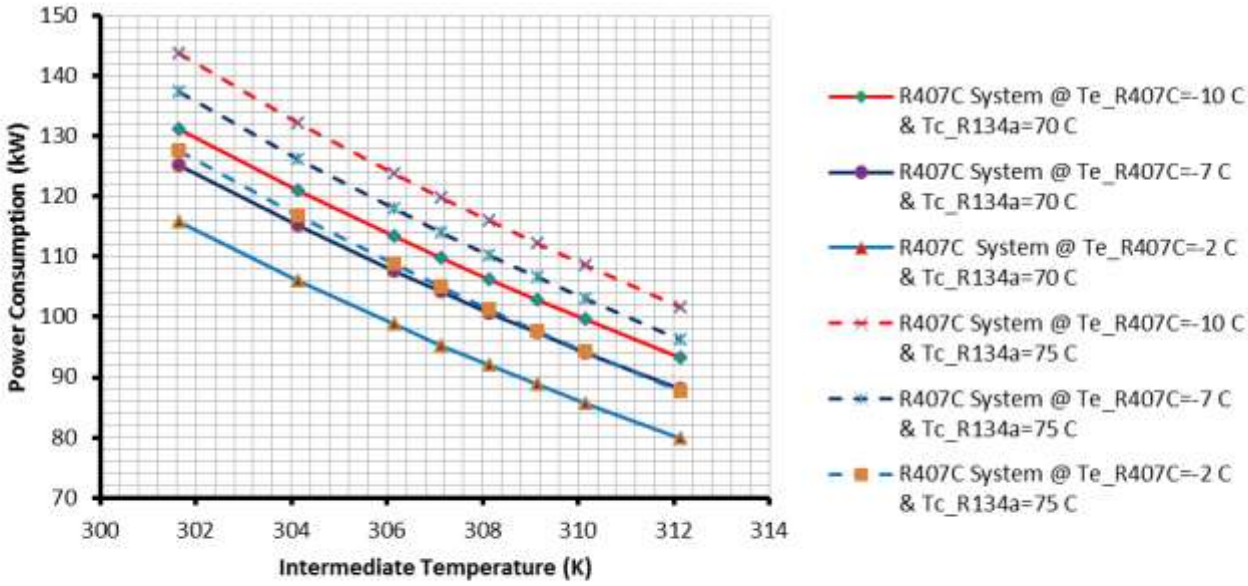

Figure 5a: R407C/R134a system

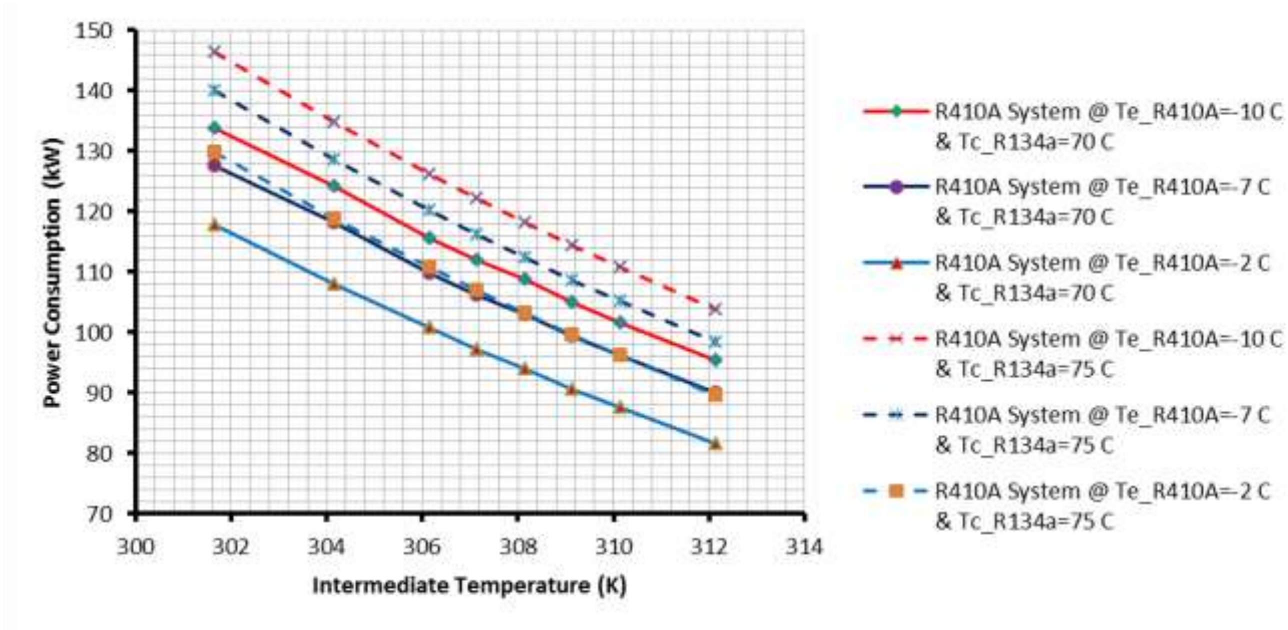

Figure 5b: R410A/R134a system

Figure 5: Power consumption comparisons at various LT evaporator and HT condenser temperatures for the test refrigerant pairs

For R407C/R134a system at HT cycle condensation (70) °C, compressors consumed power at (-10) °C higher than those at (-2) °C by (17 %) and (13%) for (39) °C and (28.5) °C intermediate temperatures respectively, Figure 5a. Almost similar values were experienced at (75) °C HT condensation, the respective values were (16 %) and (13 %) at the same intermediate and LT evaporator temperatures. The same argument is true for the R410A/R134a system but with a higher margin of about (2 %) than those of R407C/R34a ones as presented in Figure 5b.

Higher values of power consumed by compressor were experienced at the higher HT cycle condensation temperature for both systems. This is mainly due to the higher power to be used to raise the temperature to (75) °C than that to raise it to (70) °C, higher pressure ratios to be expected. At (75) °C, R410A/R134a system consumed (10 %) higher than that at (70) °C for LT evaporator of (-2) °C and test range of intermediate temperatures. It consumed a higher power in the range of (9-9.4) % when the LT evaporator temperature was at (-10) °C. Similarly, the power consumed at (75) °C for R410A/R134a system was higher than that at (70) °C. It was higher by (10-11) % and (9-10) % at (-2) °C and (-10) °C respectively for the whole range of intermediate temperatures.

Figure 5 revealed important operating condition characteristics for these refrigerant pairs at the test intermediate, LT evaporator and HT condenser temperatures. The power consumed at LT evaporator of (-7) °C and HT condenser temperature of (70) °C showed almost similar behavior and equal values to that of LT evaporator (-2) and HT condenser of (75) °C for intermediate temperature range between (31-39) °C. This is due to the fact that at LT cycle when the evaporator temperature is (-7) °C consumes higher power than that at (-2) °C case. Whereas, at HT condensation of (75) °C consumes higher power than that at (70) °C with the same percentage of the reduction at (-2) °C. Hence the total power consumed is almost the same for both cases regardless of refrigerants couple and operating lower and higher temperature levels.

5.2 Heating Load Output

The heating load of both systems R407C/R134a and R410A/R134a are presents in Figure 6 when the HT cycle condenser operates at (70 and 75) °C.

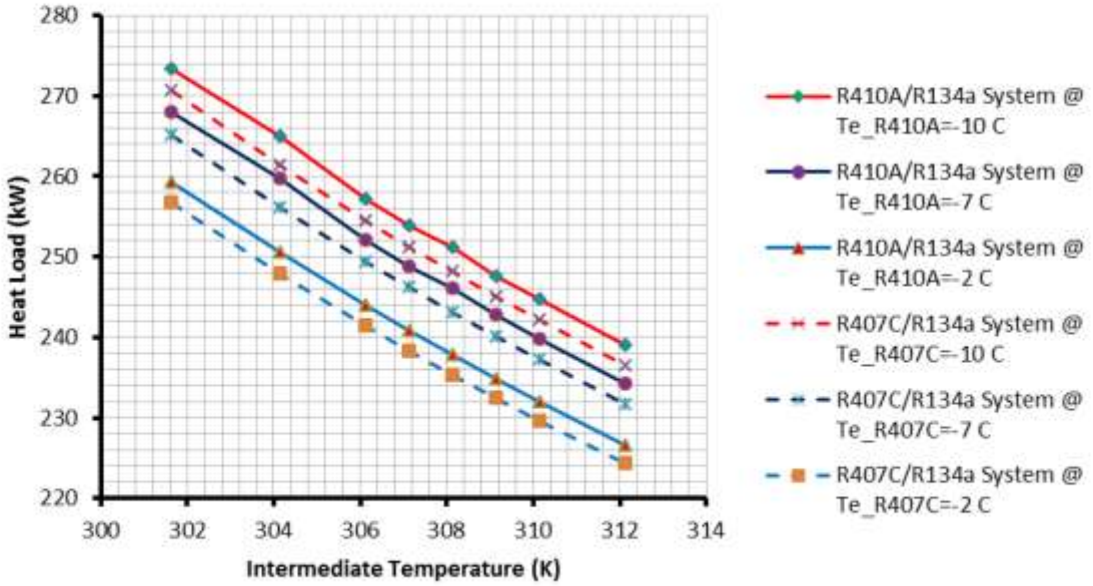

Figure 6a: High temperature cycle condensation at (70) °C

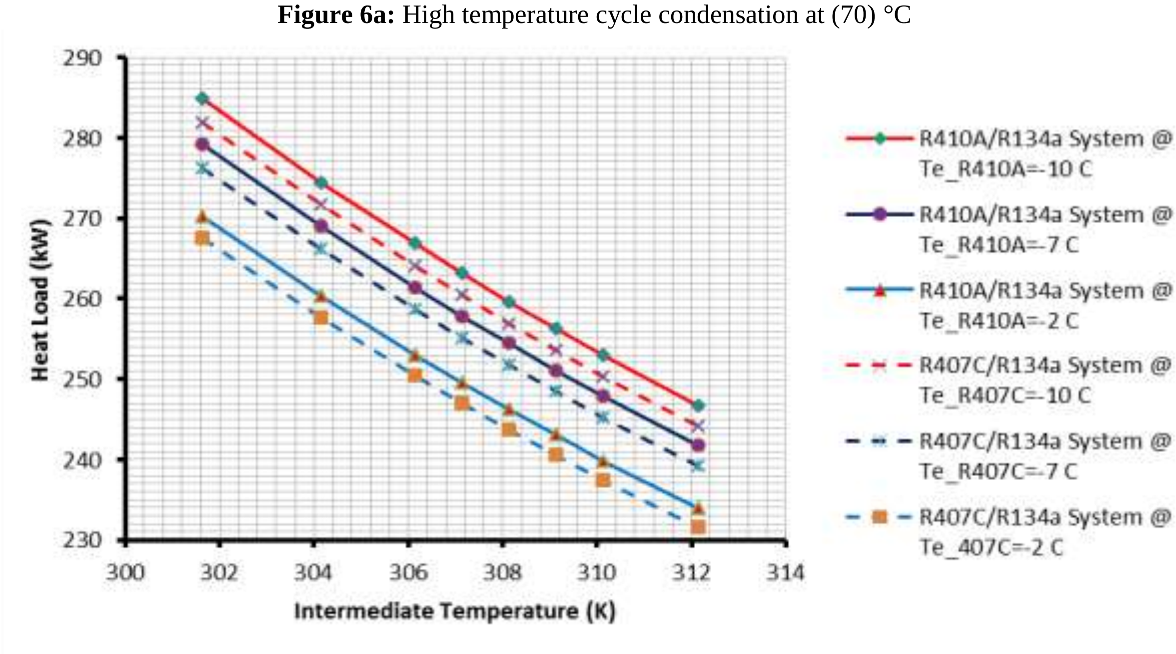

Figure 6b: High temperature cycle condensation at (75) °C

Figure 6: Heat pump heating load output comparison of both systems at LT condensation of (40) °C

The results showed that the heating load output for R410A/R134a pair is higher than that of the R407C/R134a one throughout the whole range of test conditions. For ideal conditions, the total load dissipated at the condenser of HT cycle is equal to the summation of power consumed by compressors and the extracted load on the LT evaporator, it is expressed as:

$$\dot{Q}_{HT,cond} = \dot{Q}_{cascade} + \dot{W}_{HT} \qquad (18)$$

$$\dot{Q}_{cascade} = \dot{Q}_{LT,evap} + \dot{W}_{LT} \qquad (19)$$

Hence, the total heat load at the HT condenser is related to the LT evaporator by:

$$\dot{Q}_{HT,cond} = \dot{Q}_{LT,evap} + \dot{W}_{total,comp} \qquad (20)$$

Equation (20) explains why the R410A/R134a system exhibited higher heating output for the same extracted heat at the LT evaporator and similar operating conditions. The trend of the data showed that the lowest LT evaporator temperature of (-10) °C is resulted in a higher heat output and vice versa for LT evaporator of (-2) °C for the entire test range of temperatures. Figure 6 showed that both systems at LT evaporator of (-10) °C exhibited higher heating load output than that at (-2) °C by (5.5 %) for the whole range of intermediate and HT condenser operating temperatures.

Figure 7 describes the variation of the heating load with intermediate temperature and HT condenser level. Obviously, the higher HT condenser level of (75) °C showed higher heating load than that at (70) °C for the whole range of test conditions. At intermediate temperature of (39) °C, Both of R407C/R134a and R410A/R134a systems revealed higher heating load output at (75) °C than that at (70) °C by (3.2 %) regardless of the LT evaporator temperature. The corresponding value at (28.5) °C intermediate temperature was (4.2 %) higher for the (75) °C than that of the (70) °C. The heat pump heating load at LT evaporator of (-7) °C and HT condenser temperature of (70) °C exhibited similar characteristic values to that of LT evaporator (-2) and HT condenser of (75) °C for intermediate temperature range between (31-39) °C. This behavior is similar for both refrigerant pair systems and having the same trend. This is due to the fact that both systems showed the same power consumption as illustrated in Figure 5. Hence the heating load output is nearly equal according to eq. (20) regardless of refrigerants couple and operating lower and higher temperature levels.

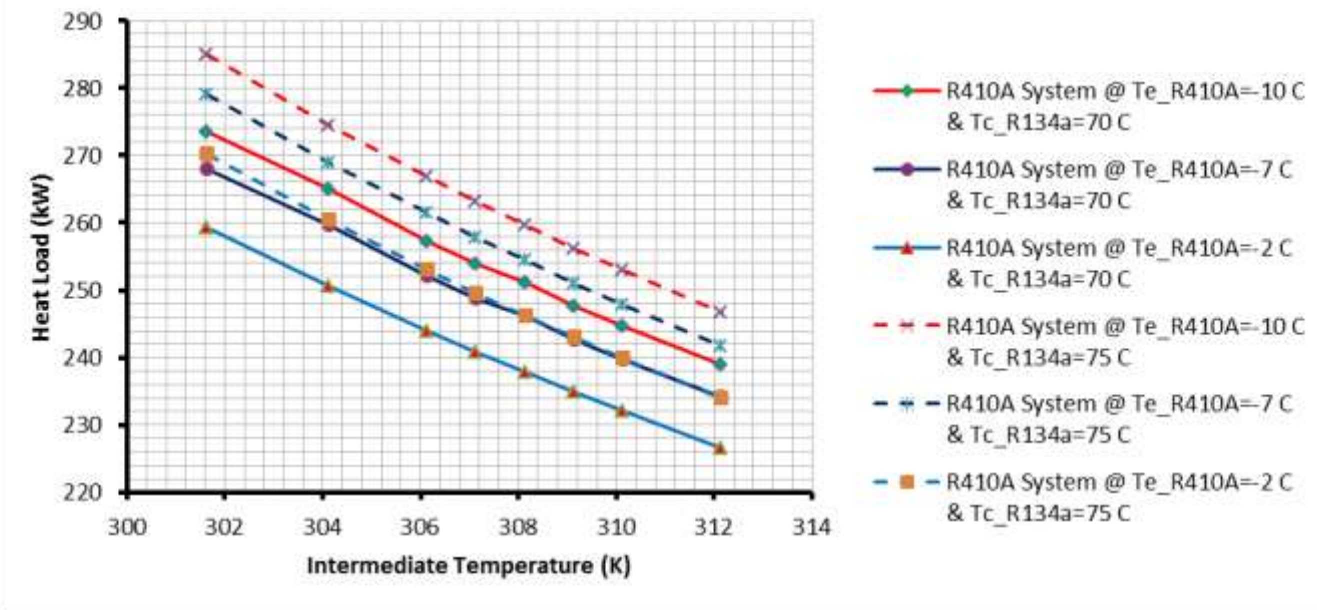

Figure 7a: R410A/R134a system

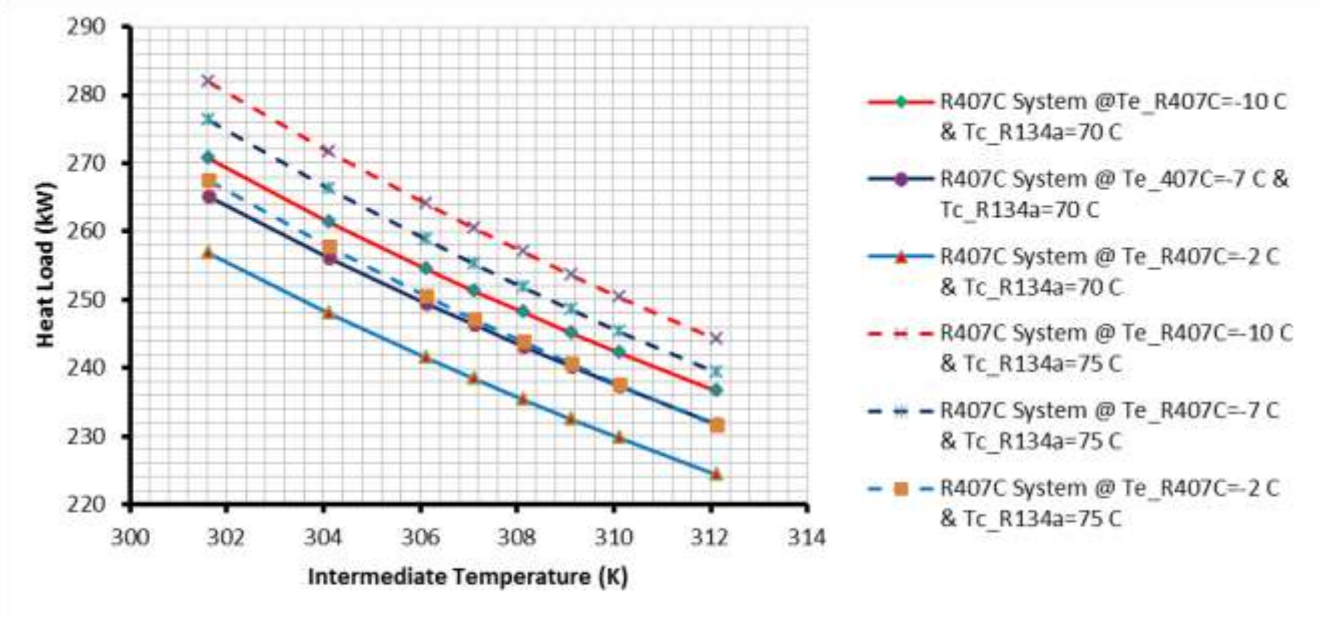

Figure 7b: R407C/R134a system

Figure 7: Heat pump heating load output comparison at various LT evaporator and HT condenser temperatures for the test refrigerant pairs

5.3 Heating Coefficient of Performance

The heating coefficient of performance for both systems are compared for the whole range of intermediate test temperature between (28.5) °C and (39) °C using eq. (16). Figure 8 illustrates a comparison for the achieved heating COP of both systems at LT evaporator (-10 and -2) °C when the HT condenser temperature of (70 and 75) °C.

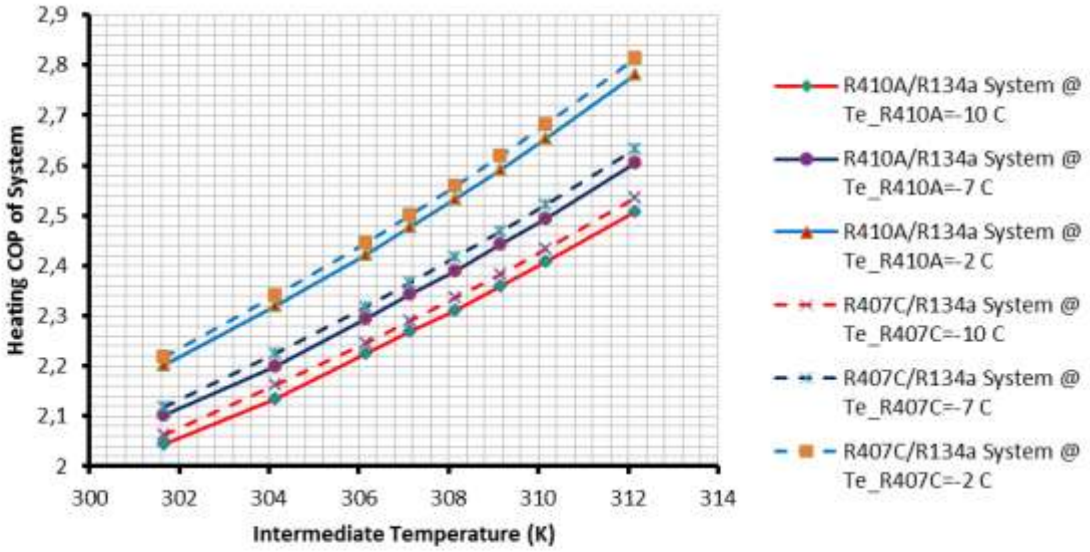

Figure 8a: High temperature cycle condensation at (70) °C

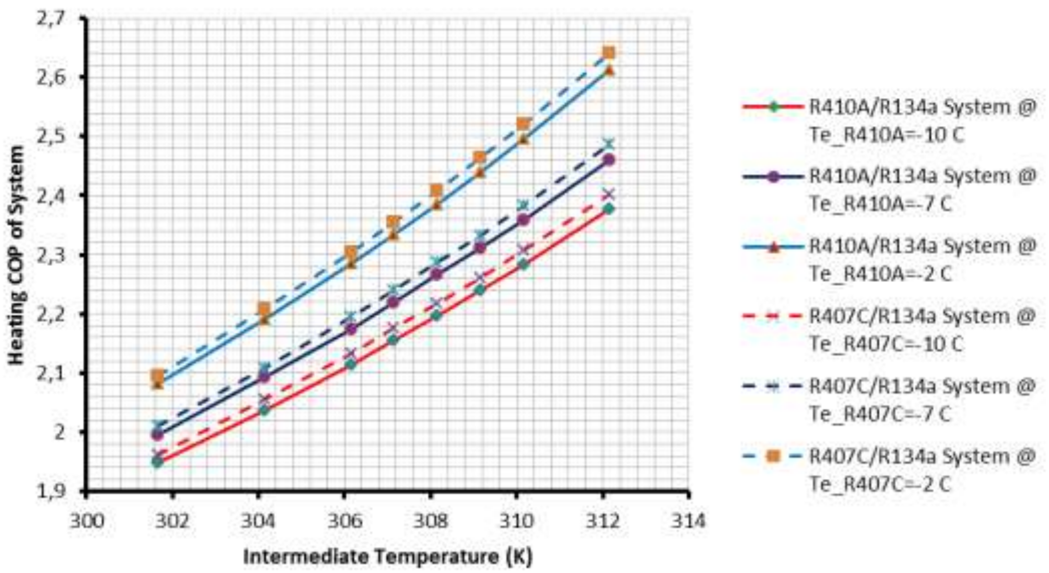

Figure 8b: High temperature cycle condensation at (75) °C
Figure 8: Heat pump heating COP comparison of both systems at LT condensation of (40) °C

It is obvious that R407C/R134a system produced higher COP than that of R410A/R134a when compared at the same operating conditions for both HT condensing temperatures. However, the discrepancy between both systems is insignificant. At HT condenser temperature of (70) °C and (75) °C, the percentage of increase is within (1 %) calculated for (28.5 to 39) °C intermediate temperatures. These values are experienced regardless of the tested LT evaporator temperature range.

Figure 9 shows a comparison for the COP of both systems at different HT condenser operating temperatures. Both test systems showed that the highest COP was achieved at the highest LT evaporator temperature and (70) °C HT condenser temperature. It ranged between (2.2) and (2.8) for intermediated temperature in the range of (28.5 to 39) °C. The lowest heating COP was experienced when the LT cycle refrigerant evaporates at (-10) °C and the HT cycle refrigerant condenses at (75) °C for both refrigerant pair systems. It was ranged between (1.95) and (2.4) at (28.5 to 39) °C intermediate temperature.

The results showed that the heat pump heating COP at LT evaporator of (-7) °C and HT condenser temperature of (70) °C showed similar characteristic values to that of LT evaporator (-2) and HT condenser of (75) °C for intermediate temperature range between (31-39) °C. This behavior is similar for both refrigerant pair systems and having the same trend. This is due to the fact that both systems showed the same power consumption as illustrated in Figure 5 and similar values for the heating load output under these conditions. Hence the heating COP is nearly equal according to eq. (16) regardless of refrigerants couple and operating LT and HT cycle temperature levels.

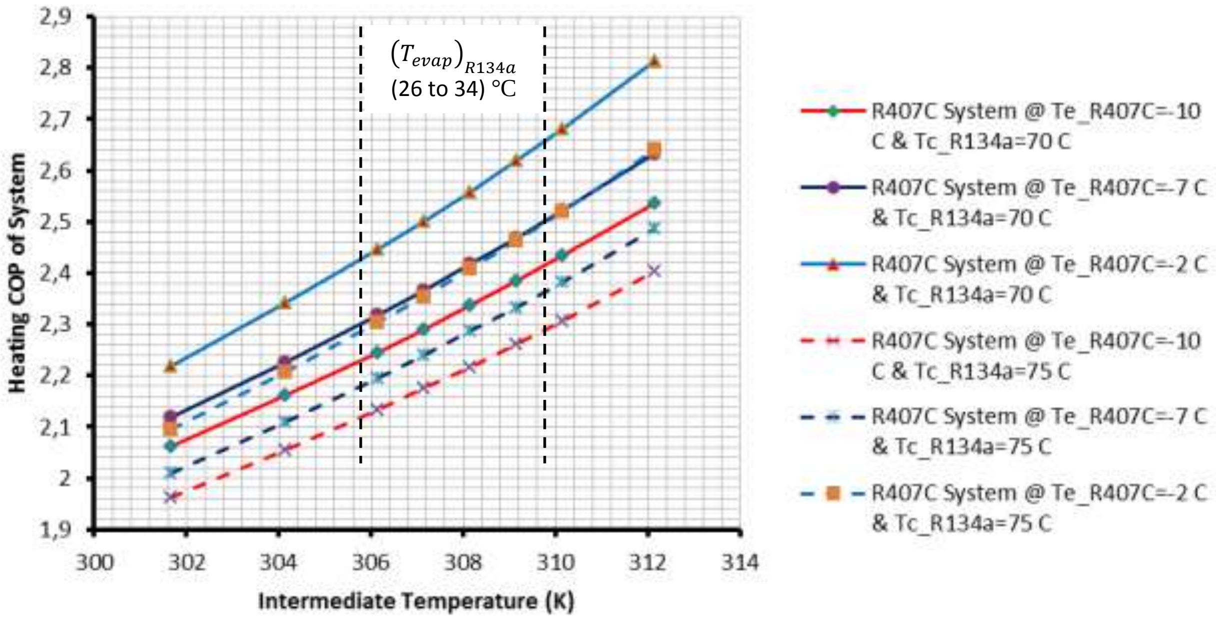

Figure 9a: R410A/R134a system

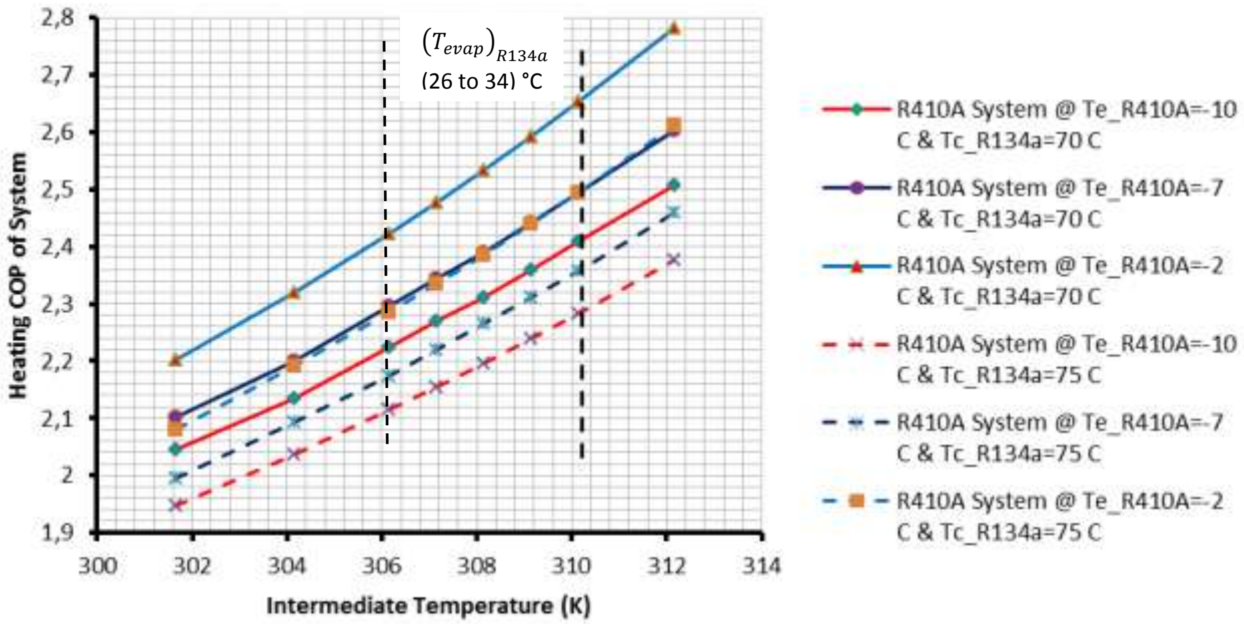

Figure 9b: R407C/R134a system

Figure 9: Heat pump heating coefficient of performance comparison at various LT evaporator and HT condenser temperatures for the test refrigerant pairs

6. CLOSURE STATEMENT

Three zero Ozone depletion potential refrigerants are implement where two of them are blend refrigerants circulated on the LT cycle. Although R410A possesses high operating pressure, but it is in favor to be selected on the LT side than R407C. This is due to its excellent evaporation and condensation heat transfer coefficient characteristic which in turn minimizes evaporator and cascade heat exchanger surface area and overall size.

Despite, the extreme high intermediate temperature (39) °C produced the highest heating COP, but still it doesn't represent the best choice for operating conditions of the Cascade system. This intermediate temperature produced only (2) °C as a temperature driving force between both sides of the cascade heat exchanger. Hence, a higher surface area and size will be required to achieve a given heat duty which will be a cost effective.

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 in the cascade heat exchanger. 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 (Tarrad, 2017, "b"). It has been confirmed that the best performance can be attained in the range nearby (33 to 37) °C intermediate temperature which gives temperature difference range of (6 to 14) in the cascade heat exchanger, Figure 9.

The assessment of the R410A/R134a and R407C/R134a systems was based on the maximum possible power to be consumed by the Cascade unit. Increasing the isentropic efficiency of the compressors to (90 %) revealed a significant reduction in the power consumed by more than (24 %) and an increase of the COP by about (20 %). For R407C/R134a system at (-2) °C LT evaporator, (70) °C HT condenser and (39) °C intermediate temperatures, showed that the COP was increased from (2.8) to (3.4). The corresponding numerical values for (28.5) °C intermediate temperature were (2.2) and (2.6) for isentropic efficiency of (70 %) and (90 %) respectively.

7. CONCLUSIONS

The present investigation outcomes can be implemented towards the heat pump technology operating with low temperature heat sources. Taking into consideration the thermal performance characteristic of R410A revealed its preference in the application of heat pump in spite of its high operating pressure. The heating COP of heat pumps is a dependent variable on the LT evaporator, intermediate and HT condenser temperature. In the range of test operating conditions, it has been found that the best thermal performance was obtained at (-2) °C and (70) °C for LT evaporator and HT condenser temperatures respectively for the whole range of intermediate temperature. It is ranged between (2.2) and (2.8) for intermediated temperature in the range of (28.5 to 39) °C. The best performance can be achieved in the intermediate temperature range close to (33 to 37) °C. This gives temperature difference between both streams in the cascade heat exchanger at the range of (6 to 14) °C. The heating COP at LT evaporator of (-7) °C and HT condenser temperature of (70) °C showed similar characteristic values to that of LT evaporator (-2) and HT condenser of (75) °C for intermediate temperature range between (31-39) °C.

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Nomenclature

Parameter	**Definition**
A	Area (m^2)
COP	Coefficient of performance (----)
g	Gravitational acceleration (m/s^2)
h	Fluid enthalpy, (kJ/kg)
$\dot{m}$	Fluid mass flow rate (kg/s)
P	Pressure (bar)
$\dot{Q}$	Heating load (kW)
t	Time (s)
T	Temperature (°C)
u	Specific internal energy of fluid (kJ/kg)
V	Fluid velocity (m/s)
$\dot{W}$	Compressor power consumption (kW)
χ	Vapor quality
z	Port elevation from ground datum (m)

Subscripts

cond	Condenser
CS	Control surface
CV	Control volume
evap	Evaporator
f	Fluid
h,o	Exit value on the hot side
h,i	Exit value on the hot side
H.Exch.	Heat exchanger
HT	High temperature side
i	Inlet side
LT	Low temperature side
net	Net value
o	Outlet side
total	Total value

Greek Letters

ρ	Fluid density (kg/m^3)

cascade	Cascade heat exchanger	Δ	Difference
comp	Compressor		

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