Performance Study of Cascade Refrigeration System using Natural Refrigerants (R290-R744)

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ABSTRACT: *In view of the concern over the use of synthetic refrigerants, CO_2, a natural refrigerant with excellent thermo-physical properties, low toxicity, non-flammability and good availability, is a promising option. The objective of the present study is to investigate the potential of cascade systems using natural refrigerants. An experimental two stage Cascade system was developed, with Carbon-Dioxide as the refrigerant in Low Temperature Circuit (LTC) and Propane in the High Temperature Circuit (HTC). LTC uses a tube-fin evaporator, a hermetic reciprocating compressor and an Electronic Expansion Valve. The Electronic expansion valve maintained a constant superheat by controlling the flow in response to the pressure sensor at compressor inlet. The HTC has a thermostatic expansion valve, a reciprocating hermetic compressor and a fin-tube condenser. Experiments and Simulations were carried with two different cascade condensers, brazed Plate and Shell & tube type. The operation of the system was observed at different ambient temperatures and load conditions. Simulations were carried out using MATLAB as a mathematical modelling tool to predict system performance under varying conditions of load, ambient and cascade temperatures. The analysis showed that the cascade temperature has a dominating effect on the efficiency of the system. Simulation results showed that better performance can be obtained by using a plate heat exchanger than a shell and tube heat exchanger.*

KEYWORDS: *Cascade refrigeration, Carbon-dioxide, Electronic expansion valve, Plate heat Exchanger, Cascade temperature*

1 INTRODUCTION

In an era with ever-increasing demand of low temperature refrigeration, cascade refrigeration systems have provided a major break-through due to their higher efficiency than the single and multi-stage refrigeration systems at ultra-low temperatures of the order of (-)50^{0}C and lower [1]. Single and multi-stage refrigeration systems are not feasible in these temperature ranges due to reasons like solidification temperature of refrigerant, extremely high pressures in condenser, very high discharge temperatures, which can result in compressor breakdown, very low volumetric efficiency and COP. In cascade refrigeration system series of single stage systems are thermally interlinked together by a cascade condenser and the refrigerants used in each circuit are different and are selected for optimum performance at the given evaporator and condensing temperatures of the circuit.

Furthermore, the environmental issues associated with the use of artificial refrigerants have compelled the scientific community to devise and develop solutions to these concerns. Efforts are focused on using natural refrigerants such as water, air, ammonia, hydrocarbons and Carbon-Dioxide because of their low or minimum contribution to direct greenhouse effect and also to ozone depletion. In this context, Carbon dioxide has emerged as a promising alternative owing to its low toxicity, non-flammable nature, and excellent thermo-physical properties facilitating smaller compressors, easy availability and economical price [1]. More over utilizing CO_2 reduces the amount released in atmosphere, thus it also reduces the greenhouse effect.

Studies to capitalize these advantages have been carried out by a number of researchers. Bhattacharyya et. al. [3] worked for simultaneous heating and cooling applications of CO_2-propane cascade system compared to that provided by CO_2-NH_3 cascade systems. Their study showed that COP of HTC and COP of LTC exhibit monotonically increasing and decreasing trends, respectively, with increase in intermediate temperature. The system COP attains the same value as that of the COP heating and COP cooling at the point of their intersection, which gives the optimum condition.

Lee et. al. [4] observed that in a CO_2/NH_3 cascade refrigeration system the optimal condensing temperature of a cascade condenser increases with condensing temperature, evaporator temperature and temperature difference in cascade condenser, whereas the maximum COP increases with only evaporator temperature.

Getu and Bansal [5] observed that maximum COP is highest for ethanol followed by R717 and lowest for R404A when CO_2 is same in LTC for two stage cascade system.. Silva et. al. [6] studied refrigeration systems consisting of a cascade cycle and observed that CO_2 presented a higher efficiency than the R404A and R22 system. However, when both the R404A and R22 systems used thermostatic expansion valves, CO_2 became even more efficient Hansaem et. al [17] studied the optimal intermediate temperature of a cascade refrigeration system with R134a and R410A.Theoretical analysis was carried out to establish a mathematical model for predicting optimal intermediate temperatures that make the system operate with its maximum efficiency. The comparison between the results of thermodynamic analysis and experiments provided good agreement.

Coloradoa et. Al [18] examined a cascade system for simultaneous refrigeration and heating. Simulations were carried out with different working fluids, ammonia, R134a, butane and propane in the low-temperature (LT) cycle with carbon dioxide (CO_2) high cycle. Results showed that the cascade system using butane in the LT cycle increased the COP up to 7.3% in comparison with those obtained with NH_3–CO_2. On the other hand, the cascade systems operating with the mixtures R134a–CO_2 and R290-CO_2 presented similar results reaching COP up to 5% higher.

The use of natural refrigerants is relatively new and unexplored in India. At present it is not commercially available in India because of the lack of awareness in this field. Also there is a dearth of data on the performance of systems employing natural refrigerants specially CO_2. High operating pressures of CO_2 and less availability of components designed for CO_2 is also a reason for its unpopularity. Thus it is very important to experimentally determine the various parameters to establish the performance and reliability of these units.

This study aims to analyse the performance of a cascade refrigeration system and its dependence on various critical parameters such as the load, ambient and cascade temperature. A two stage cascade system was developed using natural refrigerants CO_2 and Propane. Experiments were conducted with two different cascade condensers, Shell-tube and Brazed Plate Type. All the temperature and pressure readings were recorded when the system reached a steady state. Simulation of the system was carried out using MATLAB as a modelling tool; the results were verified with experimental data. The performance of the two cascade condensers were compared using simulations and a parametric study was undertaken to investigate the influence of factors like ambient temperature and condenser fan velocity. The simulated model can be used to estimate the dynamic behaviour and performance of the system.

2 EXPERIMENTATION

2.1 SETUP

The experimental setup consisted of a two stage cascade refrigeration system using propane as the refrigerant in the High Temperature Circuit (HTC) and Carbon-Dioxide as the refrigerant in the Low Temperature Circuit (LTC), with the condenser of LTC and evaporator of HTC thermally coupled using a Cascade Condenser.

Fig.1 shows the details of the major components of the setup and Fig.2 shows the line diagram of the experimental setup.In LTC, evaporator was placed inside a cooling chamber where variable artificial load was provided by a heater powered by a variable voltage transformer. Controller of electronic expansion valve (EEV) controls mass flow rate of CO_2 for maintaining constant superheat, taking feedback from the pressure transduces and temperature sensor on the evaporator outlet. RTD's and pressure gauges were installed at appropriate locations to provide the temperature and pressure values. Data from the RTDs was fed into the data logger which stores and displays the data on a computer using Benchlink software. Temperature measurements of the RTDs were scanned every 5 seconds. Both circuits had mass flow meter installed before the respective expansion valves.

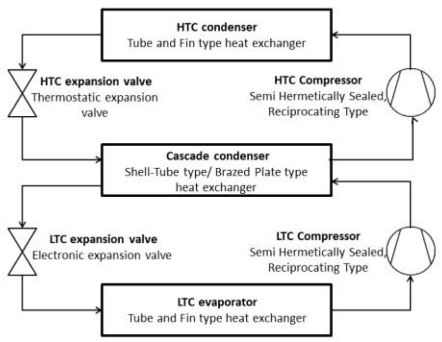

Fig. 1 Schematic of the experimental setup

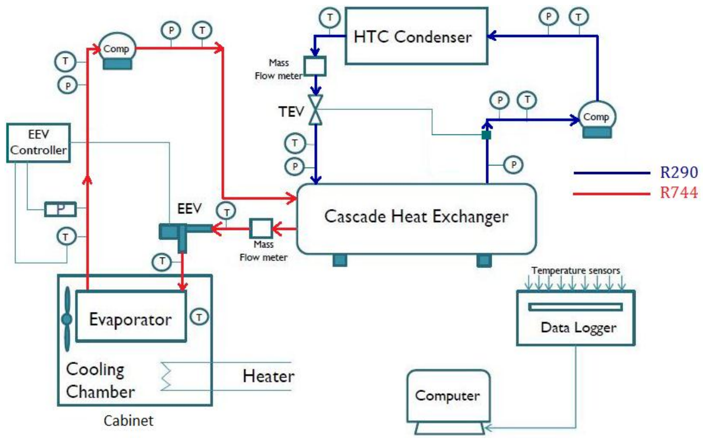

Fig. 2 Line Diagram of the Experimental Setup

2.2 METHODOLOGY

Experiments were carried out on two different types of cascade condensers, Brazed Plate type and Sell-Tube type heat exchangers. Effects of load, ambient temperature and cascade temperature were studied. External load to the evaporator was given using a heater which was connected with a transformer with variable input voltage.

To study the influence of ambient temperature, experiments were performed on different days with same load conditions. Flow through the thermostatic valve in the Propane circuit was adjusted to make changes in the cascade temperature. Pressure and temperatures values were noted when the system reached a steady state .The steady state was taken when the readings were constant for more than 20 min for a typical 3 hour experiment.

Data from all the experiments were plotted on the respective P-h diagrams and the values of enthalpy at each point calculated and used to do further calculations. Fig. 3 shows a P-h diagram of a typical experiment. REFPROP7 software was used for calculations of enthalpy values.

Heat taken in the LTC evaporator:

$$\dot{Q}_{evap1} = \dot{m}_{CO_2}(h_2 - h_1) \tag{1}$$

Work done by LTC compressor:

$$\dot{W}_{comp1} = \dot{m}_{CO_2}(h_4 - h_3) \tag{2}$$

Work done by HTC Compressor:

$$\dot{W}_{comp2} = \dot{m}_{Propane}(h_{10} - h_9) \tag{3}$$

$$COP = \frac{\dot{Q}_{evap1}}{\dot{W}_{comp1} + \dot{W}_{comp2}} \tag{4}$$

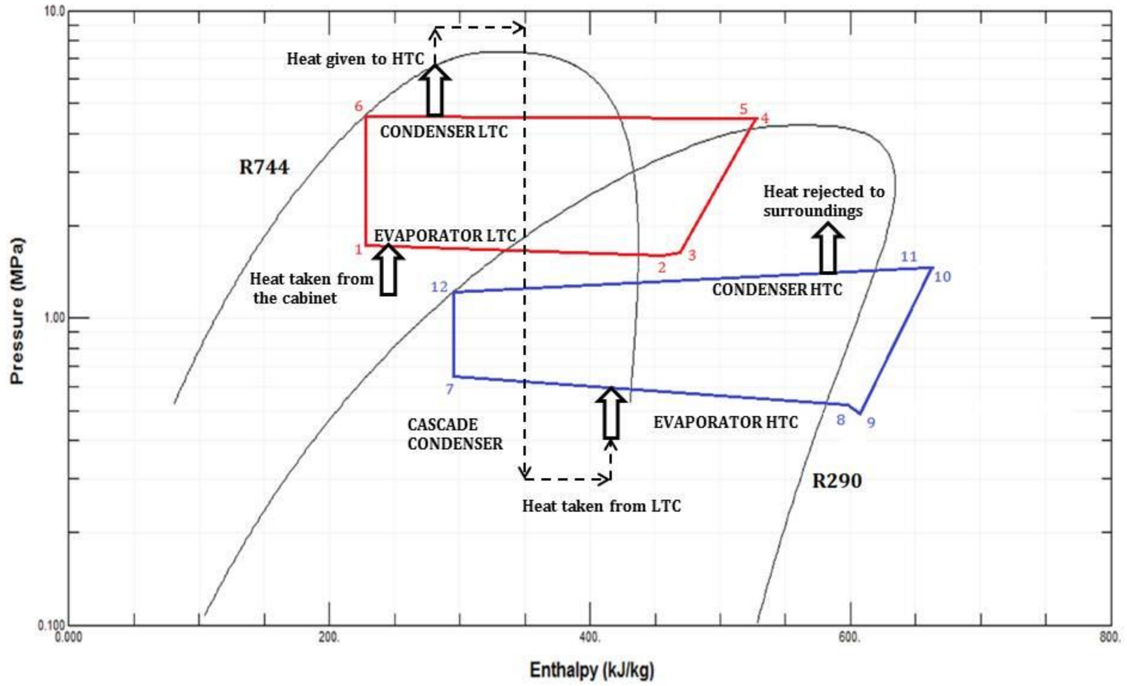

Fig. 3: P-h diagram of a typical experimental result

2.2 EXPERIMENTAL UNCERTAINITY

$$COP = \frac{\dot{Q}_{evap1}}{\dot{W}_{comp1} + \dot{W}_{comp2}} = \frac{\dot{m}_{CO_2}(h_2 - h_1)}{\dot{m}_{CO_2}(h_4 - h_3) + \dot{m}_{Propane}(h_{10} - h_9)} = \frac{\dot{m}_{CO_2} h_{2,1}}{\dot{m}_{CO_2} h_{4,3} + \dot{m}_{Propane} h_{10,9}}$$

Uncertainty in enthalpy is due to error in the pressure and temperature measuring instruments. Uncertainty value for one set of experimental result has been shown.

$$COP = \frac{2.166*(456.08-226)}{2.166*(529.60-469.35)+13.8*(665.84-607.16)} = 0.529$$

$$\Delta h_{2,1} = \sqrt{(0.35)^2 + (0)^2} = 0.35$$

$$\Delta h_{4,3} = \sqrt{(2.75)^2 + (0.77)^2} = 2.855$$

$$\Delta h_{10,9} = \sqrt{(2.54)^2 + (3.23)^2} = 4.11$$

$$\Delta\, COP = \left(\sqrt{\left(\frac{\Delta\, \dot{m}_{CO_2}}{\dot{m}_{CO_2}}\right)^2 + \left(\frac{\Delta\, \dot{m}_{Propane}}{\dot{m}_{Propane}}\right)^2 + \left(\frac{\Delta h_{2,1}}{h_{2,1}}\right)^2 + \left(\frac{\Delta h_{4,3}}{h_{4,3}}\right)^2 + \left(\frac{\Delta h_{10,9}}{h_{10,9}}\right)^2}\right) * COP = 0.047 * 0.529$$

$$COP \pm \Delta\, COP = 0.529 \pm 0.025$$

3 SIMULATION

Modelling of different components like compressor, evaporator, condenser, cascade condenser and expansion valve etc. was carried out in the mathematical tool MATLAB. These models were used to carry out the simulation of various components. Finally they were connected using an iterative algorithm for the simulation of complete system.

3.1 MATHEMATICAL MODELLING

Each component of the cascade refrigeration system, compressor, expansion valve and the condenser was modelled [13]. These individual models were then connected using an iterative algorithm to simulate the complete system.

Cascade Condenser-Brazed Plate Heat Exchanger

Besides modelling the Shell and Tube heat exchanger [7, 13], Plate Heat exchanger was also included into the algorithm. Based on the temperature and the quality (x) of refrigerant entering into the condenser from LTC and HTC, the model provided the net heat exchange between two sides. Appropriate correlations [16], for average heat transfer coefficient on condensing and evaporating side in the two phase region were taken.

Pressure drop estimation

To make the simulation model more accurate, pressure drop inside the components was also taken into account. To gauge the order of the pressure loss, estimations were first made for PHE, being most susceptible. For the two-phase frictional pressure drop in the plate heat exchanger, correlation proposed by Huang et al [15] were used following the homogeneous model. Using correlations pressure drop was estimated for typical flow rates and temperature values in the Plate heat exchanger. For HTC, for a flow rate of 5.9 g/s and evaporating temperature of 262 K pressure differential across the PHE came out to be 3798 N/m^2. Similarly for CO_2, for a flow rate of 4.4g/s and 266K temperature gave a pressure drop of 812.8 N/m2. It was realized that the pressure loss in the PHE is much low as compared to the mean inline pressures, which for Propane is around 3.5 bar and goes as high as 35 bar for the CO_2 circuit, thus could be comfortably neglected. In conclusion of this exercise, pressure drop in other components were also found negligible.

3.2 ALGORITHM

An iterative procedure was used to calculate all the system parameters. The algorithm is composed of the Primary function 'Main', with LTC, HTC and Cascade functions as the subroutines. The inputs were given to main function, which are forwarded to the subroutines in order to evaluate the steady state conditions iteratively. A detailed diagram has been shown in Fig. 4.

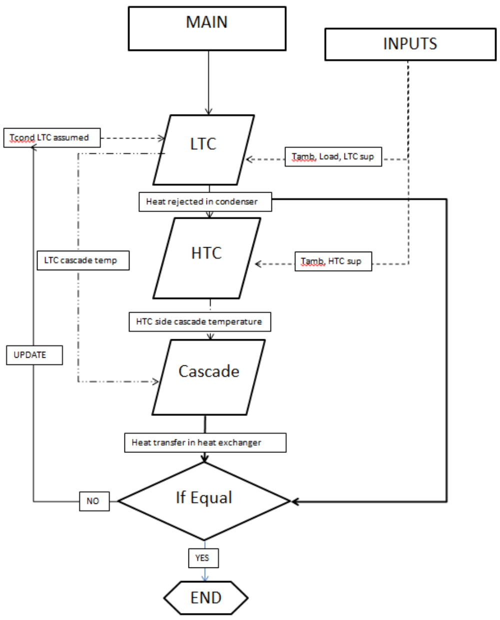

Fig. 4: Flow Chart for Main Program Algorithm

Main Program

Initially LTC condenser temperature was assumed to start the simulation. Inputs for the LTC subroutine were load in the cooling chamber, condenser temperature LTC, T_{amb} & Superheat. Heat rejected by CO_2 stream in the cascade condenser was calculated by the LTC. This rejected heat in cascade condenser along with superheat in HTC and atmospheric conditions were fed into HTC. Accordingly outputs from HTC include Cascade temperature, compressor work, and condenser temperature. Finally cascade temperatures of both circuits along with geometrical parameters of the heat exchanger were fed to the Cascade subroutine, which calculates the amount of heat transfer. Now it is checked whether heat transfer is equal to the heat rejected by LTC. If these heats are equal then simulation stops and all the parameters are recorded otherwise LTC condenser temperature is changed.

LTC Subroutine

Input parameters for the LTC are condenser temperature, load in the evaporator & superheat. First, evaporator temperature is assumed. The temperatures are fed into the compressor model which calculates the work required by the compressor, mass flow rate of refrigerant and condenser inlet temperature. By using the conservation of energy condenser outlet temperature is found. Now the evaporator temperature and the condenser outlet temperatures is fed into expansion valve. By using the model of the thermostatic expansion valve the mass flow rate of the refrigerant is obtained. Now both the mass flow rates (from compressor and expansion valve) are compared. The assumed evaporator temperature is stabilized by updating it till the mass flow rates from the two methods become equal. At that condition the heat rejected from the LTC condenser is calculated and fed to the HTC circuit.

HTC Subroutine

Input for the HTC is heat rejected from LTC, superheat and ambient temperature. To start the simulation both condenser temperature and evaporator temperature are assumed. By comparing the mass flow rate through compressor and expansion valve (similar to LTC), evaporator temperature is stabilized. Then the heat rejected in the condenser is calculated by using the ambient conditions and condenser inlet temperatures. This heat rejection is then compared with work done by the HTC compressor and the heat received from the LTC. If they are equal the HTC simulation is stabilized else the condenser temperature is modified accordingly till the balanced is reached.

CASCADE Subroutine

After all the parameters of LTC and HTC have been stabilized, cascade subroutine receives the condenser temperature and mass flow rate from LTC while HTC provides the evaporator temperature and its refrigerant flow rate. On the basis of correlations and the cascade condensers' parameters cascade subroutine gives the theoretical heat transfer that occurs between the HTC and LTC streams. This heat transfer is compared with the heat rejected from the LTC circuit. If the two are equal the simulation is stopped, else our first assumption, the LTC condenser temperature is updated according to the inequality.

4 RESULTS AND DISCUSSION

4.1 EXPERIMENTS

Results of the experiments performed with the Shell & tube cascade condenser

Effect of ambient temperature:

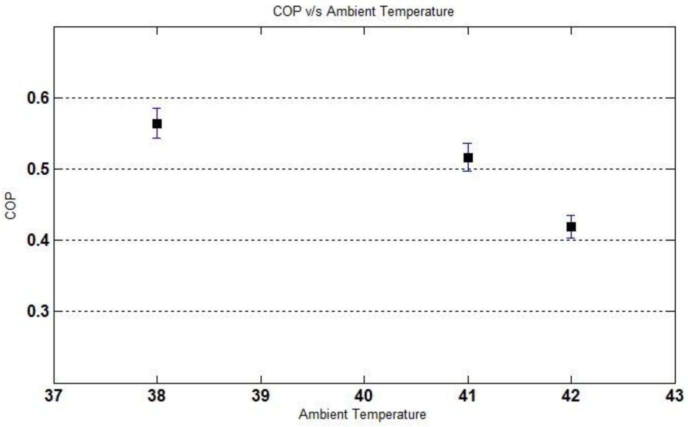

Fig. 5: COP v/s Ambient Temperature

Effect of Cascade Temperature

Cascade temperature is the temperature of the refrigerant in HTC, at the entrance of cascade condenser. It is controlled by changing the flow through thermo-static valve. The valve allows a minimum pressure drop which sets the upper limit of the cascade temperature whereas the maximum pressure cut-off limit of the compressor sets the lower limit for the cascade temperature. Fig. 6 plots the variation in performance of the system at various cascade temperatures.

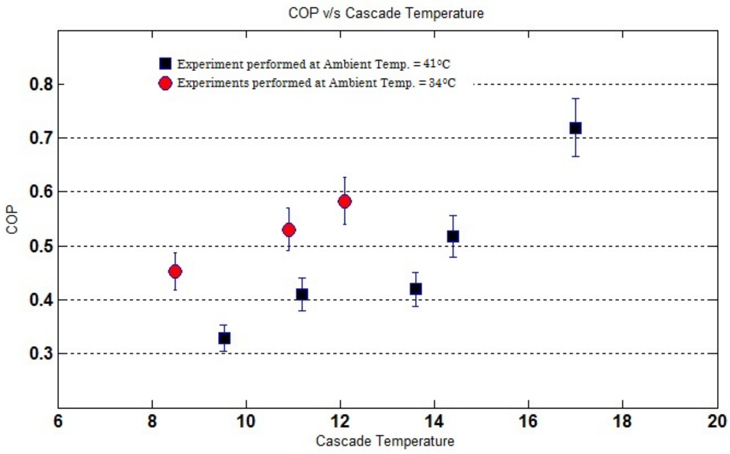

Fig. 6: COP v/s Cascade Temperature

The simulation results [13] which stated that there is an optimum value of cascade temperature where COP of the system is maximum and predicted the value of optimum temperature as approximately the square root of the product of the ambient and the cabinet temperature.

$$T_{optimum} = \sqrt{(314 * 271.5)} = 292\text{ K} = 19°\text{C (approx.)}$$

$$T_{optimum} = \sqrt{(307 * 264.0)} = 284.68\text{ K} = 11.7°\text{C (approx.)}$$

The range of variation of cascade temperature was below the optimum cascade temperature because of the upper limit of the expansion valve. In the range in which experiments have been performed, the results are in accordance with the simulation results. The COP of the system increases with increase in cascade temperature.

Effect of Cabinet Temperature

Fig. 7 plots the variation in the performance of the system with various T_{cab}. With lowering of the T_{cab} , there is an increase in work done by the LTC compressor. Since work done by LTC compressor is relatively small compared to work done by the HTC compressor, effect of change in T_{cab} on COP is small. The reduction in T_{cab} also lowers the theoretical value of the optimum T_{cas} which has an increasing effect on the COP . The variation in COP of the system is small and is in inverse relation with the T_{cab} .

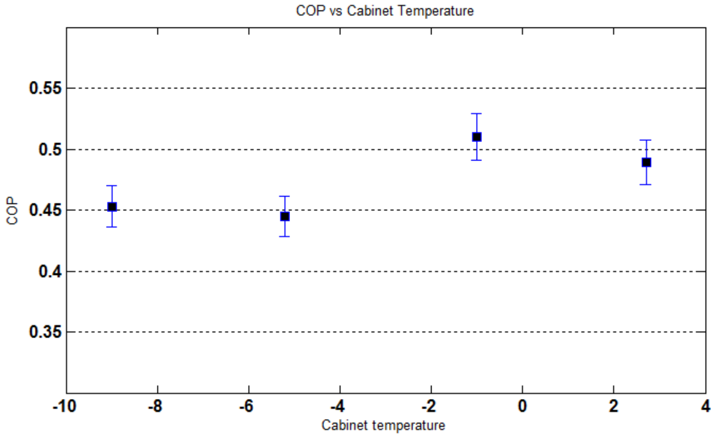

Figure 7: COP v/s Cabinet Temperature

4.2 SIMULATION

Comparison between Shell & tube and Brazed PHE performance

Simulations were carried out for the brazed plate and the shell-tube heat exchangers for both the circuits at different value of load and other parameters.

Effect of change in load on the cabinet temperature

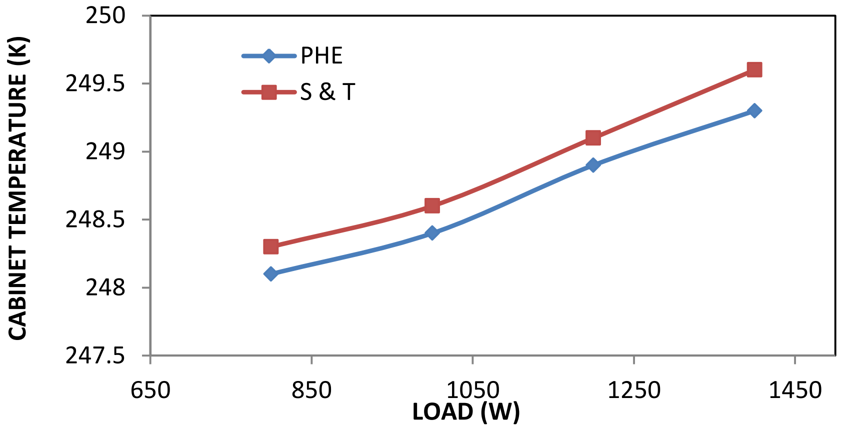

Figure 8: Graph showing variation of Cabinet Temperature with respect to load for two different cascade condensers

Fig. 8 plots the variation of Cabinet Temperature at various values of load. The ambient temperature was fixed at 300K and superheat was maintained constant at 5K in both circuits. It was observed that lower temperature is achieved in case of Brazed Plate heat exchanger as compared to the Shell and Tube type heat exchanger. Also as the load is increased the cabinet temperature increases for both types of heat exchangers. Unlike the Shell-tube HE, the slope of rise in case of PHE tends to get mild with greater load. This suggests better performance of Shell-tube HE as compared to the PHE at higher load conditions.

Effect of change in ambient on cabinet temperature:

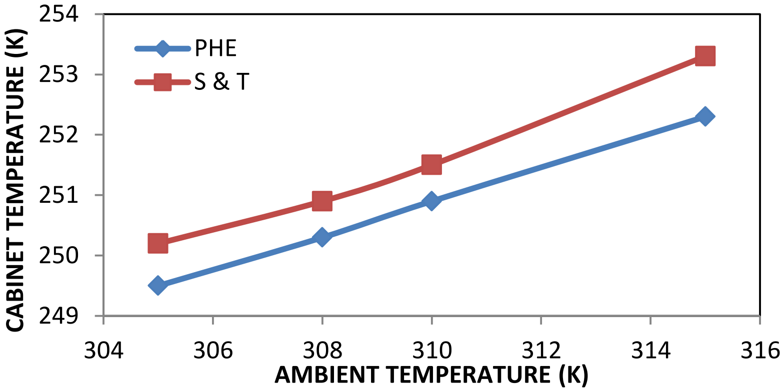

Figure 9: Variation of Cabinet Temperature with the ambient temperature.

Fig. 9 plots the variation of the cabinet temperature at various ambient temperatures. The cabinet temperature rises with the increase in ambient temperature. However on the higher side of the ambient the separation between the two tends to widen. The influence of ambient temperature in case of Shell-tube HE progressively gets more pronounced as compared to PHE suggesting an advantage of PHE. This can be attributed to the fact that with Shell-tube heat exchangers have much larger volume as compared to the plate type heat exchangers. This behaviour makes them more susceptible to heat ingression as compared to the plate type heat exchangers for the same heat transfer surface area.

Effect of change in ambient on the COP:

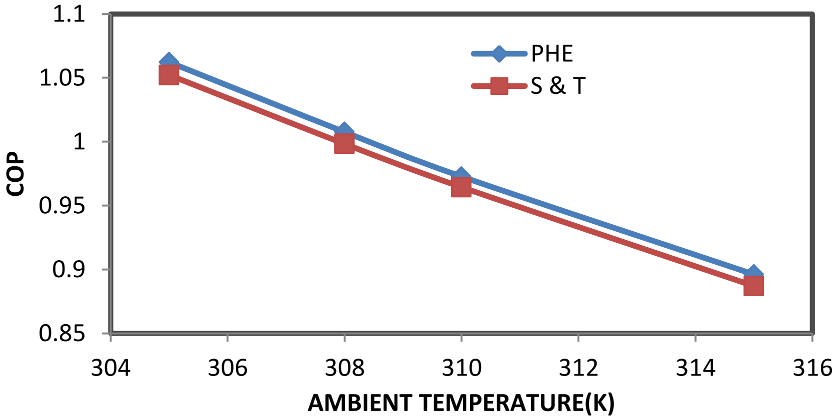

Fig. 10: Variation of COP with the ambient temperature

Fig. 10 depicts the variation of COP with change in T_{amb}. Load was fixed at 1000W and a fixed degree of superheat was given to both HTC and LTC. There is a similar decrease in COP in both types of cascade condensers. This happens because of the increase in compressor work with increasing T_{amb} .

Effect of change in condenser fan velocity on COP:

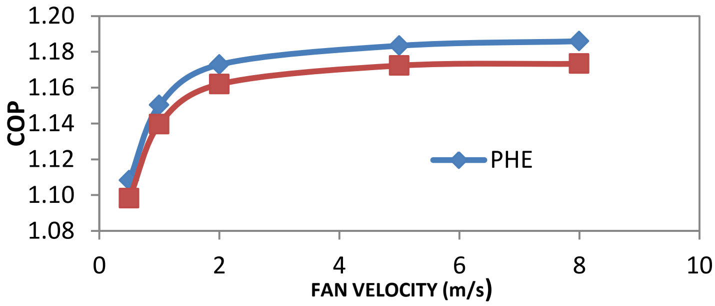

Fig. 11: Variation of COP with fan-air velocity inside the condenser.

Fig 11 plots the variation of the COP at various Fan velocities. Load was kept constant at 1000W and T_{amb} at 300K. The influence air velocity over the condensing pipes is quite significant in lower ranges of air speed and saturation is reached as the speed is increased. At lower air velocities the convective heat transfer coefficient is small and hence higher compressor work is required. With increase in air velocity, the convective heat transfer coefficient increases significantly reduces the requirement of compressor work. At high velocities the outer surface temperature of the Heat Exchanger tends to the ambient temperature and hence very small change in COP which increase in fan velocity thereafter.

Effect of change in condenser fan velocity on cabinet temperature:

In Fig. 12 the variation of cabinet temperature has been plotted for a fixed load of 1000W, ambient temperature of 300K and fixed superheat of 5K. It can be noticed that the cabinet temperature decreases as the air velocity is increased, but up to a limit and then it saturates. The reason can be related to the difficulty in rejecting heat in the condenser. More the resistance higher the condenser temperature and lower the mass flow rate of the refrigerant pumped through the compressor. For a fixed load this results in warmer cabinet temperature.

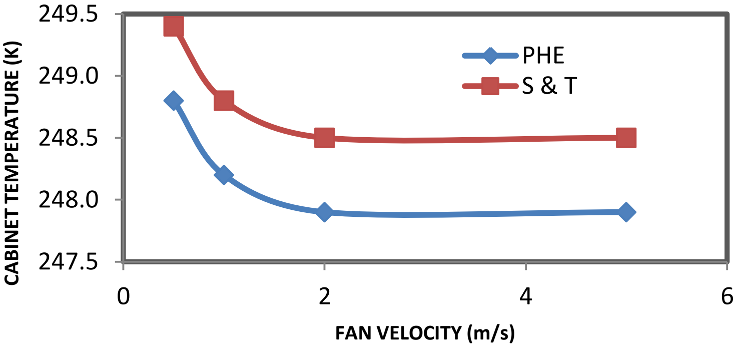

Fig. 12: Variation of cabinet temperature with fan-air velocity inside the condenser

4.3 COMPARISON OF THE EXPERIMENTAL AND SIMULATION RESULTS

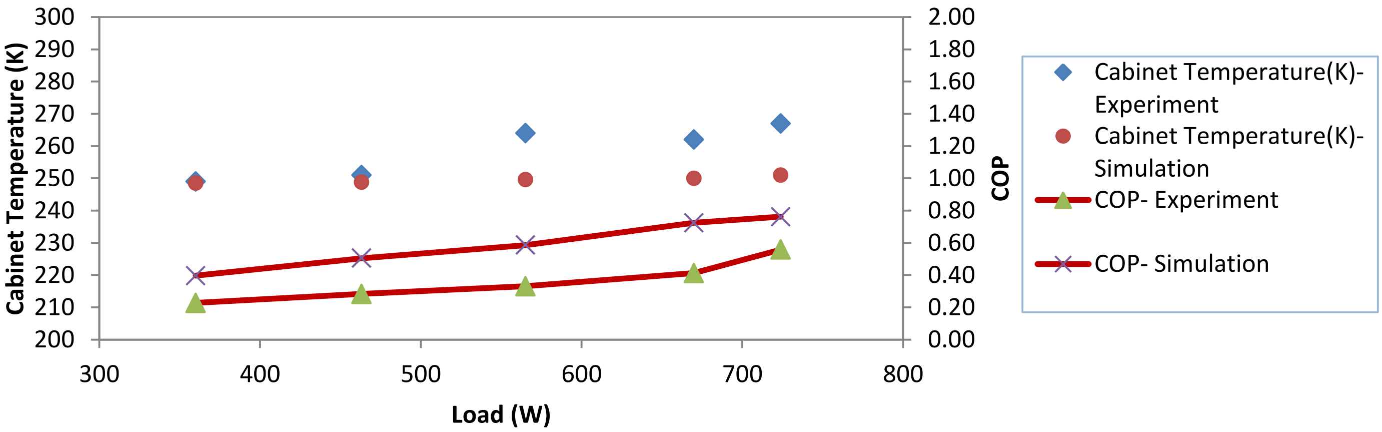

Figure 13: Variation of COP with Load

Simulation received 4 input parameters from the experiment which are load, superheat, air velocity (HTC condenser) and the ambient temperature. A plot of the variation of COP and cabinet temperature with load has been shown in Fig. 13. COP achieved in case of simulation is higher as compared to the experiments as the simulation model does not account for the heat ingress in the pipes, which reduces the overall performance of the system. A general noticeable factor is that there is an increase in COP with load. But with increase in load the cabinet temperature also increases. This implies that we can achieve better performance with higher load, but the cabinet temperature needs to be sacrificed.

5 CONCLUSIONS

The aim of the study was to analyse the performance of a cascade refrigeration system using natural refrigerants with two different cascade condensers, Plate heat exchanger as compared to a Shell and Tube heat exchanger through experimentation and simulation. The following inferences can be noted:-

1) On observing the experimental results it can be concluded that Cascade temperature has a significant effect in the performance of the system.
2) On comparing the experimental results with that of the simulation, it was found that better COP is predicted by the theoretical model by virtue of its limitations to account for all its heat ingression, none the less experimental performance closely follows the simulated results. This gave us a validated simulation model, to predict dependence of performance on various parameters.
3) After studying the results obtained from the simulations for PHE and Shell and Tube heat exchanger one can observe that it is possible to achieve lower cabinet temperature with Plate heat exchanger as compared Shell and Tube heat exchanger for the same heat transfer area for different load and ambient temperature conditions.
4) The variation of COP and cabinet temperature with load suggests that as COP increase with increasing load values, the cabinet temperature also increases. Hence higher COP could not be sustained at lower cabinet temperature values.
5) The COP of the system was quite low, ranging from 0.3 to 0.78, which is because of the high ambient temperature and also due to losses in the system.

Nomenclature

β	Chevron angle (radian)
COP	Coefficient of performance
D_h	Hydraulic diameter (m)
EEV	Electronic Expansion Valve
F_{Rf}	Geometrical factor
f_{tp}	Friction factor
G	Mass flux through the plate heat exchanger ($kg/cm^2/sec$)
HTC	High temperature Circuit
HE	Heat Exchanger
h	Specific enthalpy ($kJ\ kg^{-1}$)
i	Enthalpy (J/kg)
L	Equivalent length of the channels (m)
LTC	Low temperature circuit
$\dot{m}$	Mass flow rate ($kg\ s^{-1}$)
Nu	Nusselt number
P	Pressure (Pa)
P_{co}	Heat exchanger pitch
Pr	Prandtl number
PHE	Plate Heat Exchanger
$\dot{Q}$	Heat transfer rate (kW)
q	Heat flux (W/m^2)
Re_{tp}	Two phase Reynolds number
RTD	Resistance Temperature Detectors
S&T	Shell and Tube Heat Exchanger
T	Temperature (^0C or K)
t	Time (s)
v	Specific volume ($m^3\ kg^{-1}$)
v/s	Versus
$\dot{W}$	Work (kW)
x	Vapour quality
x_m	Mean vapour fraction for the two phase flow

Greek Symbols

ρ_f	Density of saturated liquid (kg/m^3)
ρ_g	Density of saturated vapour(kg/m^3)
ρ_m	Mean density for two phase flow (kg/m^3)
μ_{tp}	Equivalent viscosity for the two phase flow
ϑ_m	Mean specific gravity (m^3/kg)
$\Delta\ COP$	Uncertainty in the value of COP
ΔP_f	Pressure drop (N/m^2)

Subscripts

evap1	LTC evaporator
comp1	LTC compressor
comp2	HTC compressor
Amb	Ambient
Cab	Cabinet
Cas	Cascade

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