NUMERICAL SIMULATION OF TWO STAGE CASCADE REFRIGERATION SYSTEM USING REFRIGERANT R 12 – R170, R 502 – R 1150 AND R 717- R 1270

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ABSTRACT

The single stage vapour compression refrigeration system using various refrigerants is limited to an evaporator temperature of -40^oC. Below temperature of -40^oC the either cascade system or multi stage vapour compression system is employed. Capacity and Efficiency of any refrigerating system diminish rapidly as the difference between the evaporating and condensing temperature is increased by reduction in evaporator temperature. Present work describes the variation of Heat extracted by evaporator, mass flow rate required at High Temperature state and low temperature state and work input of individual compressors. Three different combinations of refrigerant R 12 – R170, R 502 – R 1150 and R 717 R - 1270 are taken in high temperature state and low temperature state respectively in two stage cascade refrigeration system. It has been observed that condensing and evaporating temperatures have strong effect on coefficient of performance of cycle. Most of exergy losses occur in compressor due to heat generation. The second law efficiency and the COP increases, and the total exergy loss decreases with decreasing temperature difference between the evaporator and refrigerated space and between the condenser and outside air.

Keywords: *Refrigeration, Simulation, Heat transfer, simulink.*

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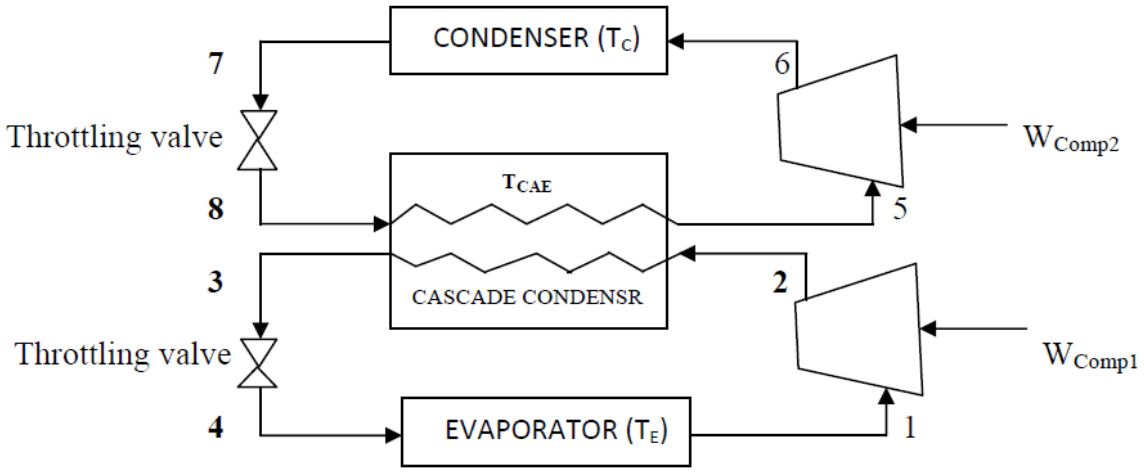
INTRODUCTION

Cascade refrigeration system is a low temperature refrigeration system and is used for very low temperature range about (-40C to -130C).At such low temperature simple Vapor Compression Refrigeration Cycle (VCRS) is not efficient due to very high compression ratio that further leads to high discharge problem and low volumetric efficiencies Whereas, cascade refrigeration is much efficient for such conditions. Cascade refrigeration cycle is nothing but simply a combination of two VCRS cycles named as low and high temperature circuit that are combined together by a cascade condenser. This cascade condenser unit act as evaporator for low temperature circuit and condenser for high temperature circuit, the low temperature circuit uses low boiling refrigerants such as R12, R502 etc and high temperature uses high boiling point refrigerants such as R170, R290, R404A, R1270, R507A etc. Various refrigerants have been analyzed in cascade refrigeration system among which few are R 12/ R170, R 502/R1150 and R717R/R1270. In the present study theoretical analysis is done on cascade refrigeration system by using blends of Hfc refrigerants and analysis results the effect of design and operating parameters on system performance.

CASCADE REFRIGERATION (SYSTEM DESCRIPTION):

The cascade refrigeration cycle is a combination of two vapor compression cycles which utilizes two different refrigerants. The primary refrigerant flows from low temperature circuit evaporator to low stage compressor and condensed in cascade condenser which also acts as evaporator for high temperature circuit. The heat rejected from condenser of low temperature circuit is extracted by evaporator of high temperature circuit containing secondary refrigerant then, this secondary refrigerant gets compressed in high stage compressor and finally condensed to outer atmosphere. The desired refrigerating effect is occurred from evaporator of low temperature circuit. The temperature difference in cascade condenser is an important design parameter that decides the COP of the entire refrigeration system.



THEORETICAL ANALYSIS:

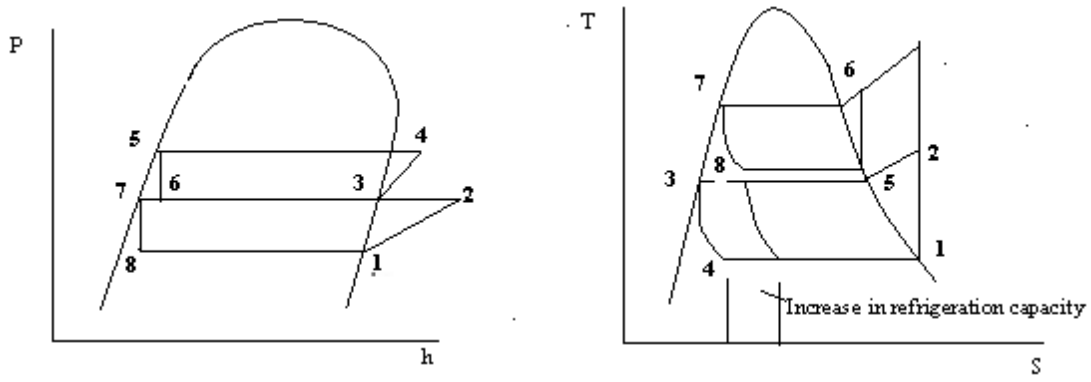
In present work a parametric study with fixed mass flow rate in low temperature circuit and varying different parameters such as evaporator temperature, condenser temperature and temperature difference in cascade condenser have conducted to determine effects of these parameters on system performance. The analysis is done by making general assumptions so as to simplify the analytical procedure these are as follows:

a) Negligible change in kinetic and potential energy.

b) Isenthalpic expansion of refrigerant in expansion valve.

c) Negligible pressure and heat loss/gain in pipe or other components

d) Compressor process is irreversible and adiabatic.



The range of parameters used for system analysis are the evaporating temperature (Te) varies from (-100°C to - 70°C), Condenser temperature (Tc) varied from (30°C to 40°C) cascade condenser temperature (TCAC) varies from (-45°C to -20°C), temperature difference in cascade condenser (Dt) varies from (2°C to 10°C).Also the parameters which are kept fixed by varying other parameters are TE= -95°C, TC= 35°C, DT=5°C,mr1=0.2kg/min. Based on

the assumptions mentioned above the equations for mass and energy balance are described for different components. Also each component is considered as a control volume. Therefore the following sequence of equations was applied for the analysis of cascade refrigeration cycle:

MATHEMATICAL MODELING

Mass balance

$$\Sigma_{in}\ mr = \Sigma_{out}\ mr \quad (1)$$

Energy balance

$$Q - W + \Sigma_{in} mrh - \Sigma_{out}\ mrh = 0 \quad (2)$$

Heat transfer rate to the cycle in the evaporator:

$$Q_E = m_{r1}\ (h_1 - h_4) \quad (3)$$

Total Work input to both compressor:

$$Wcomp_1 + Wcomp_2 = mr_1\ (h_2 - h_1) + mr_2\ (h_6 - h_5) \quad (4)$$

Rate of heat transfer in cascade condenser:

$$Q_{CAS} = mr_1\ (h_2 - h_3) = mr_2\ (h_5 - h_8) \quad (5)$$

Heat rejected by condenser of high temperature circuit:

$$Q_{COND} = mr_2\ (h_6 - h_7) \quad (6)$$

Mass flow ratio is given by equation:

$$\frac{mr1}{mr2} = \frac{h2-h3}{h5-h8} \quad (7)$$

Overall COP of the system:

$$COP = \frac{QE}{WCOMP1+WCOMP2} \quad (8)$$

COP Of LTC and HTC:

$$(COP)L = \frac{QE}{WCOMP1} \quad (9)$$

$$(COP)H = \frac{QCAS}{WCOMP\ 2} \quad (10)$$

Exergetic Efficiency of the whole system is given

$$\eta II = \frac{COP}{COP\ carnot} \quad (11)$$

Where, COP Carnot $= \frac{TE}{TE+TC}$ (12)

RESULT AND DISCUSSION

To demonstrate the influence of operating parameters on system COP and Exergy efficiency different range of variables as discussed above are varied and there effect are studied.

Fig.1 depicts, while varying the low circuit evaporator temperature (T_E) and keeping other parameters constant overall COP of the system increases from (0.88 to 2.11) as there is decrease in pressure ratio. Hence refrigerating effect is increased for all three refrigerant with decrease in compressor work. Also Second Law efficiency increases with very small amount from (0.68 to 0.89) with increase in evaporator temperature.

Fig.2 shows on decreasing evaporator temperature Te from -100 ^{o}C to -70 ^{o}C mass flow rate decreases . Where it is maximum in case of refrigerant R12 and minimum for R717.

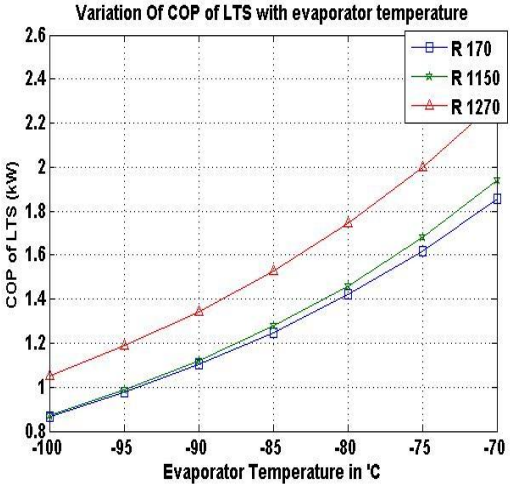


Fig 1

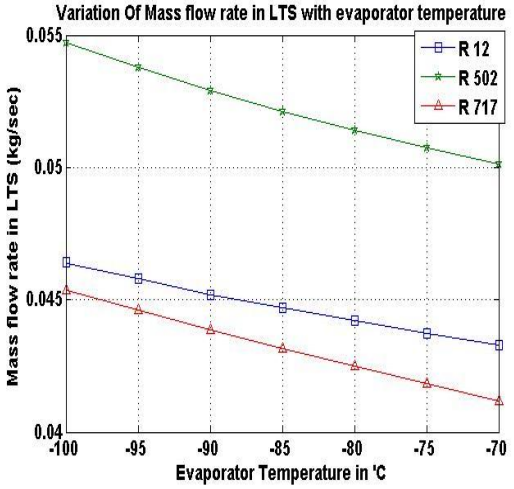


Fig 2

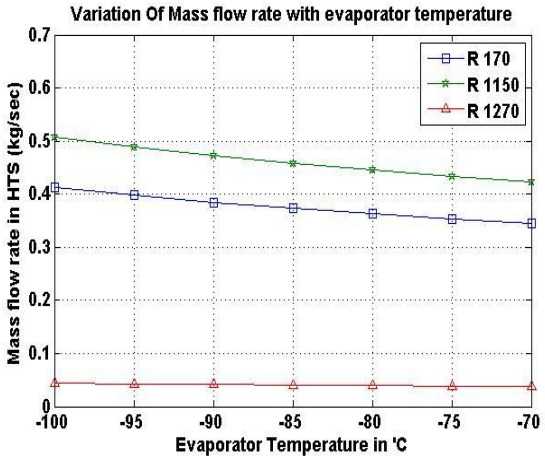


Fig 3

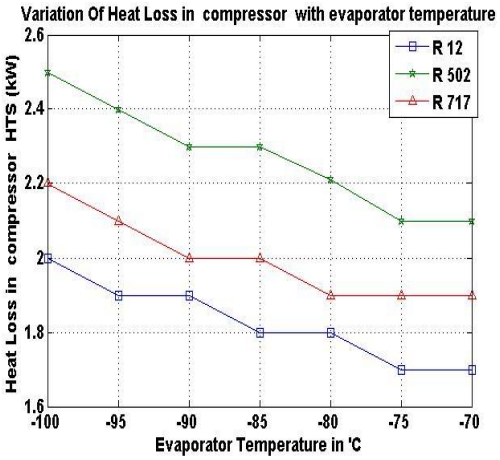


Fig 4

Fig 3 shows variation of evaporator temperature with mass flow rate of refrigerant in high temperature stage for three different working fluids. It is found that mass flow rate required for achieving desire temperature is minimum for R 1270 whereas it maximum for refrigerant R1150.

Fig 4 shows variation of evaporator temperature with heat loss of compressor in high temperature stage. Heat loss is minimum when working refrigerant in HTS cycle is R 12 and it is maximum when the same is R502.

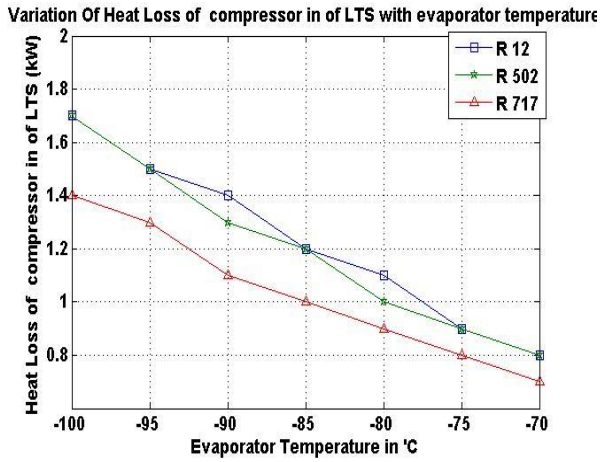


Fig 5

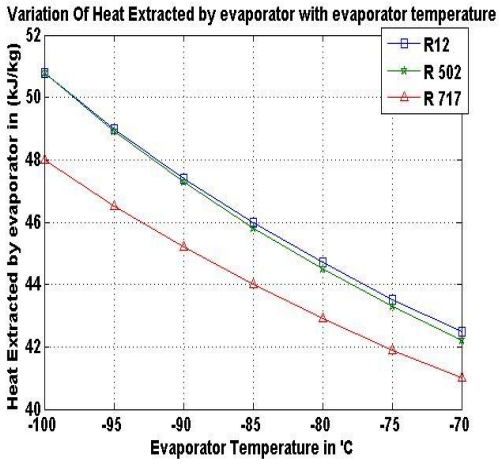


Fig 6

Fig 5 shows variation of evaporator temperature in low temperature stage with Heat loss in Compressor, it is being found that heat loss is minimum in for R717 and it is almost same for refrigerant R 12 and R502 .

Fig 6 represent the variation of evaporator temperature with amount of heat extract in evaporator and while analysis it is found that heat extracted in evaporator in kJ/kg is minimum for refrigerant R717 whereas refrigerant R12 and R502 shows same performance characteristics.

CONCLUSION

As it is clear from fig1 as we are decreasing evaporator temperature from -100^{o}C to -70^{o}C Cop of the cycle decreases, this is due to reason COP is directly proportional to refrigerating effect and inversely proportionally to Pressure ratio (compressor power), and. ECOP for Refrigeration Cycle is the ratio of exergy flow rate across evaporator to the Summation of exergy flow rate across each compressor. Fig 2 shows variation of mass flow rate in LTS cycle on decreasing evaporator temperature from -100^{o}C to -70 ^{o}C . Exergy efficiency is the ratio of reversible work to actual work, actual work is always greater than reversible work due to some physical losses in system. It is clear from Fig 5 and Fig 6 on decreasing evaporator temperature from -100^{o}C to -70 ^{o}C heat loss in compressor and heat extracted by evaporator both decreases gradually and finally we can conclude that efficiency is obtained at higher pressure ratio on further increasing pressure ratio there is sudden fall in second law efficiency.

NOMENCLATURE

T_e	Evaporator Temperature in ^{o}C
T_c	Condenser Temperature in ^{o}C
T_o	Ambient Temperature in ^{o}C

h_o	Enthalpy at datum in kJ/kg
s_o	Entropy at datum in kJ/kg-K
P	Pressure in bar
X	Dryness fraction
m	Mass flow in kg/sec
h	Specific Enthalpy in kJ/kg
s	Specific entropy in kJ/kg-K
h_v	Specific Enthalpy in kJ/kg
s_v	Specific entropy in kJ/kg-K
h_f	Specific Enthalpy in kJ/kg
s_f	Specific entropy in kJ/kg-K
h_{fg}	Specific Enthalpy in kJ/kg
s_{fg}	Specific entropy in kJ/kg-K
Q_e	Heat extracted by evaporator in kJ/kg
Q_c	Heat rejected by condenser in kJ/kg
$Wcomp_1$	Work done by compressor 1 in kW
$Wcomp_2$	Work done by compressor 2 in kW
$Wcomp_3$	Work done by compressor 3 in kW
W_{total}	Total Work input in kW

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