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Thermodynamic Analysis of a Two Stage Vapour Compression Refrigeration System Utilizing the Waste Heat of the Intercooler for Water Heating

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Abstract

This article focuses on the various aspect of use of waste heat from intercooler of a multi-stage vapor compression refrigeration system using ammonia as a refrigerant. Energy and exergy analysis of the multi- stage refrigeration system having an intercooler is performed. The COP of such a system is found to be increased by 4 to 5%, and calculated to be approximately 3.24. Heat recovery through the intercooler proved to be beneficial as the COP of the system is improved along with heat recovery of 20 kJ/s. The analysis can represent a physical system where water can be heated through the heat extracted from the intercooler.

Keywords: Two-stage vapour compression system; Intercooler; Heat recovery; Coefficient of performance; Waste heat recovery

List of Symbols/Abbreviations:

GWP: Global Warming Potential;

ODP: Ozone Depletion Potential;

COP: Coefficient of Performance;

MMVCS: Modified Multistage Vapour Compression System;

EDR: Exergy Destruction Ratio;

HP: High Pressure;

LP: Low Pressure;

$\dot{Q}$: Rate of Heat Transfer (kW);

$\dot{W}$: Work Rate (kW);

X : Exergy Destruction Rate (kW);

$\dot{X}$: Exergy Rate (kW);

$\dot{m}_r$: Rate of Thermal Exergy Flow Rate (kW);

$\dot{m}_r$: Mass Flow Rate of Refrigerant (kg/s);

T: Temperature (K);

S: Specific Entropy (kJ/kg-K);

H: Specific Enthalpy (kJ/kg);

V: Velocity of Fluid (m/s);

Greek symbols

H: Efficiency;

E: Effectiveness;

X: Specific Exergy;

Δ: Efficiency defect;

Subscripts

E: Evaporator;

Comp: Compressor;

C: Condenser;

T: Throttle valve;

J: j[th] component of the system;

R: Region to be cooled or refrigerant;

I: Inlet to the control region;

E: Outlet to the control region;

R: Refrigerator;

O: Ambient state;

Rev: Reversible;

Sub: Sub-cooling;

Su: Superheat;

Vcr: Vapour compression refrigeration system

Introduction

Heat rejected from refrigeration air conditioning plants is of low grade quality. Due to the high costs related with the recovery of such heat and the availability of alternate means for meeting low grade heat requirements, low grade waste heat is generally rejected to the atmosphere.

Kaushik et al. [1] presented an investigation of the feasibility of the heat recovery from the condenser of a simple vapour compression refrigeration system through a Canopus heat exchanger which acts as an auxiliary condenser between the compressor and condenser components. Results were compared for different working fluids and found that heat recovery factor of the order of 2.0 and 40% of condenser heat can be removed through the canopus heat exchanger.

Zubair and Yakub [2] investigated and compared the results with the existing practices in the industry. Furthermore, the theoretical results of a two-stage refrigeration system performance were also compared with experimental values for the refrigerants mentioned above.

Nikolaidis et al. [3] observed the performance of two-stage compression refrigeration having flash-chamber and water-intercooler using refrigerants R22 has been demonstrated by the thermodynamic analysis. Any reduction in irreversibility rate of the condenser gives approximately 2.40 times greater reduction in the irreversibility rate for the whole plant, because the changes in the temperature in the condenser and the evaporator contributed significantly to the overall plant performance.

Rahman et al. [4] presented the performance of the recently developed integrated space condition system. A conventional split type

air conditioner is modified to reclaim the superheated portion of the heat leaving the compressor to be utilized for the space conditioning purposes. The end result was expected to be faster cooling and prolonged compressor life.

Cabello et al. [5] proposed the main operating variables on the energetic characteristics of the vapour compression plant. They concluded that the refrigerant mass flow rate is largely dependent on the suction specific volume and therefore on the suction conditions.

Apera et al. [6] performed of a vapour compression refrigeration plant using as working fluids R22 and R417a. The problem related to the replacement of the fully HCFC and of the partially HCFC have been only partially solved. Refrigerant fluid experimentally tested as a substitute for R22 is the non-azeotropic mixture R417a.

Ouadha et al. [7] calculated components exergetic loses by operating at constant evaporating temperature of -30°C and condensation temperatures of 30, 40, 50 and 60°C with two natural substitutes of HCFC22, namely propane (R290) and ammonia (R717) as working fluids. They found that the optimum interstage pressure for a two-stage refrigeration system is very close to the saturation pressure.

Arora and Kaushik [8] proposed a detailed exergy analysis of an actual vapour compression refrigeration (VCR) cycle. A computational model was developed for computing coefficient of performance (COP), exergy destruction, exergetic efficiency and efficiency defects for R502, R404A, and R507A. The results indicate that R507A is a better substitute to R502 than R404A. The efficiency defect in condenser is highest and lowest in liquid vapour heat exchanger for the refrigerants considered [9-12].

Santiago et al. [13] found that two stage cycle in case of high temperature difference between heat sink and heat source in order to overcome the high pressure ratios that deteriorate compressor volumetric and isentropic efficiency. Results showed that optimum intermediate pressure is close to the arithmetic mean in case of R-404a but there is a significant difference in case of ammonia.

Mishra et al. [14] worked on the replacement of R22 with several environment friendly refrigerants. They used hydrocarbon (HCF) refrigerants like R134a, R410a, R407C and M20. Out of the above stated refrigerants R407C proved to be a potential HFC refrigerant which can replace R22 with minimum investment and efforts. It proved to be a non-ozone depleting refrigerant giving high system efficiency [15-17].

Mishra [18] described a thermodynamic modeling of a vapour compression refrigeration system using R134a in primary circuit and Al_2O_3-water based nanofluids in the secondary circuit. Simulation showed that for the same geometric characteristics of the system performance increases from 17% to 20% by application of nanofluid as a secondary fluid in VCS [19-22].

Xiaoui She et al. [23] proposed a new sub-cooling method for vapor compression refrigeration system depending on expansion power recovery. Liquid refrigerant is sub-cooled by using evaporative cooler. This makes a hybrid refrigerant system. Analysis is to done by using different refrigerants and results shows that hybrid vapor compression refrigeration have more (C.O.P) than conventional vapor compression refrigeration system.

N Upadhya [24] presented a concept of effect of sub-cooling on performance of refrigeration system. In this a diffuser is used after condenser which converts kinetic energy in to the pressure energy of refrigerant, it results in reduction of power consumption and reduction

of condenser size. After studying of all above techniques, concludes that it will be helpful for future research.

M Yang et al. [25] Performed enhancement and exergy destruction analyses were conducted numerically for vapor-compression refrigeration systems using R22, R134a, R410A, and R717. The effects of cooling water in a subcooler, refrigerant pressure drop among heat exchangers, and superheating in an evaporator were also considered and compared with evaporation and condensation temperature. Optimal degrees of subcooling obtained according to the second law of thermodynamics were consistently higher than those obtained according to the first law of thermodynamics.

Going through the historical background of various research articles, the gap in the literature review is found that the thermodynamic analysis of a two stage vapor compression refrigeration system with intercooler and flash chamber both in the cycle using ammonia as a refrigerant yet to be performed and also to evaluate the amount of heat from the intercooler which can be recovered for various applications.

Thermodynamic modelling of a modified two stage vapour compression refrigeration system

The present study focuses on the waste heat recovery from a two-stage VCR system from intercooler. The purpose of incorporating heat recovery is to further decrease the temperature of refrigerant (ammonia) coming out of the intercooler exit thereby reducing the net work done on the system and hence increasing the overall coefficient of performance of the system. A computational model was developed for carrying out the analysis of the system using Engineering Equation Solver software EES [22].

System description: The present work is on this modified multistage vapour compression refrigeration (MMVCS) system. The compressed refrigerant is passed through a water intercooler where the temperature is reduced down to the state 3' is directed into the flash chamber (process 3-4), where saturated liquid and saturated vapour refrigerant at intermediate pressure is separated out, and then it passes through high stage compressor (process 4-5) adiabatically, followed by condenser where it condenses (process 5-6) at constant pressure as a result of the removal of heat of condenser Q_{cond} and undergoes adiabatic expansion through high stage expansion valve, accompanied by a drop in pressure at constant enthalpy (process 6-7), it passes through again to flash chamber at intermediate pressure (process 7-8). The liquid refrigerant coming from the flash chamber undergoes adiabatic expansion through the low stage expansion valve, accompanied by a drop in pressure at constant enthalpy (process 8-9), low pressure vapour from the evaporator is compressed by the compressor and the cycle is repeated. In order to recover heat from the intercooler, an external fluid (water) is used to remove heat as shown in (Figures 1 and 2).
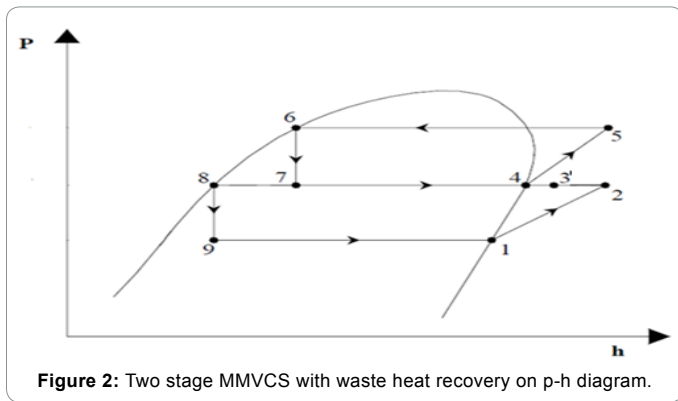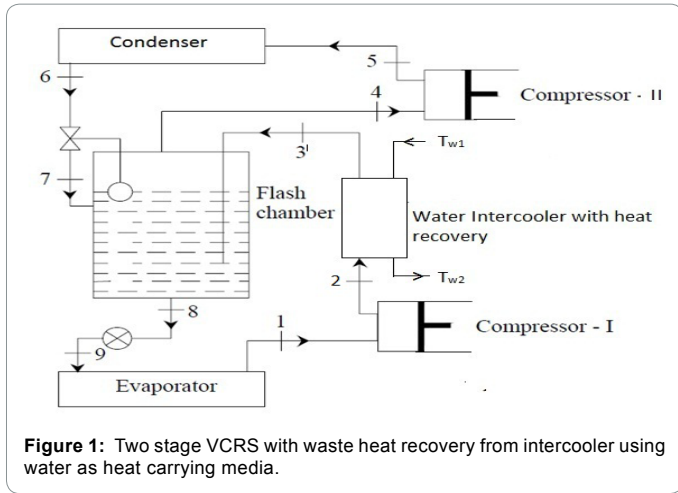
Thermodynamic modelling

Energy analysis:

The first law of thermodynamics or energy balance for the steady flow process of an open system is given by:

$$\dot{Q}_i + W_i + \sum \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) = \dot{Q}_e + W_e + \sum m_e \left(h_e + \frac{v_e^2}{2} + gz_e \right) \quad (1)$$

Evaporator:

Heat extracted in the evaporator:

Figure 1: Two stage VCRS with waste heat recovery from intercooler using water as heat carrying media.

Figure 2: Two stage MMVCS with waste heat recovery on p-h diagram.

$$\dot{Q}_{9-1} = \dot{Q}_e = \dot{m}_{LP}(h_1 - h_9) \tag{2}$$

Where m_{lp} = mass flow rate of refrigerant in low stage compressor

$$\dot{m}_{LP} = TR\left[\frac{3.5167}{h_1 - h_9}\right] \tag{3}$$

Low stage compressor:

Ideal work input to compressor:

$$\dot{W}_{LP} = W_1 = \dot{m}_{LP}\left(h_2 - h_1\right) \tag{4}$$

$$\eta_{comp,lp} = \frac{W_{isentropic}}{W_{actual}} = \left(\frac{h_3 - h_1}{h_2 - h_1}\right) \tag{5}$$

$$W_{LP} = \dot{m}_{LP}\left(h_2 - h_1\right) \tag{6}$$

$$W_{HP} = \dot{m}_{HP}\left(h_5 - h_4\right) \tag{7}$$

$$W_{net} = W_{LP} + W_{HP} \tag{8}$$

$$COP = \frac{Q_e}{W_{net}} \tag{9}$$

$$W_{HP,1} = \dot{m}_{HP,1}\left(h_5 - h_4\right) \tag{10}$$

$$W_{net,1} = W_{LP} + W_{HP,1} \tag{11}$$

$$COP_{new} = \frac{Q_e}{W_{net,1}} \tag{12}$$

Condenser":

Heat rejected by the condenser to the condenser to the surrounding is given by:

$$Q_c = Q_e + W_{net} \tag{13}$$

$$Q_{wic} = \dot{m}w * C_w *\left(T_{w2} - T_{w1}\right) \tag{14}$$

The effectiveness ε_{wic} of the water intercooler with heat recovery can be written as

$$T_3 = T_{3,1} + \varepsilon_{wic} *\left(T_3 - T_{w1}\right) \tag{15}$$

Exergy analysis

Evaporator:

$$\dot{E}D_E = \dot{X}_9 + \dot{Q}_E\left(1 - \frac{T_o}{T_e}\right) - \dot{X}_1 = \dot{m}_{LP}\left[(h_9 - h_1) - T_o(s_9 - s_1)\right] + \dot{Q}_E\left(1 - \frac{T_o}{T_e}\right) \tag{16}$$

Compressor-1:

$$\dot{E}D_{comp1} = \dot{X}_1 + \dot{W}_{comp1} - \dot{X}_3 = \dot{m}_{LP}\left[T_o\left(s_1 - s_2\right)\right] \tag{17}$$

Condenser:

$$\dot{E}D_c = \dot{X}_5 - \dot{Q}_c\left(1 - \frac{T_o}{T_c}\right) - \dot{X}_6 = \dot{m}_{HP,1}\left[(h_5 - h_6) - T_o(s_5 - s_6)\right] - \dot{Q}_c\left(1 - \frac{T_o}{T_c}\right) \tag{18}$$

Water Intercooler with heat recovery:

$$\dot{E}D_{wic} = \left(\dot{X}_3 + \dot{X}_{w1} - \dot{X}_{31} - \dot{X}_{w2}\right) = \dot{m}_{LP}\left[(h_3 - h_{31}) - T_o(s_3 - s_{31})\right] + \dot{m}_w\left[(h_{w1} - h_{w2}) - T_o(s_{w1} - s_{w2})\right] \tag{19}$$

Flash chamber:

$$\dot{E}D_{flchm} = \left(\dot{X}_{3,1} + \dot{X}_7 - \dot{X}_8 - \dot{X}_4\right) = \dot{m}_{LP}\left[(h_{3,1} - h_8) - T_o(s_{3,1} - s_8)\right] + \dot{m}_{HP,1}\left[(h_7 - h_4) - T_o(s_7 - s_4)\right] \tag{20}$$

Throttle valve-1:

$$\dot{E}D_{TV1} = \dot{X}_6 - \dot{X}_7 = \dot{m}_{hp,1}\left[(h_6 - h_7) - T_o(s_6 - s_7)\right] \tag{21}$$

Throttle valve-2:

$$\dot{E}D_{TV2} = \dot{X}_8 - \dot{X}_9 = \dot{m}_{LP}\left[(h_8 - h_9) - T_o(s_8 - s_9)\right] \tag{22}$$

Compressor-2:

$$\dot{E}D_{comp2} = \dot{X}_4 + \dot{W}_{comp2} - \dot{X}_5 = \dot{m}_{HP,1}\left[T_o(s_4 - s_5)\right] \tag{23}$$

Total exergy destruction

It is the sum of exergy destruction in different components of the system.

$$\dot{E}D_{Total} = \dot{E}D_E + \dot{E}D_{comp1} + \dot{E}D_{comp2} + \dot{E}D_{wic} + \dot{E}D_c + \dot{E}D_{TV1} + \dot{E}D_{TV2} + \dot{E}D_{flchm} \tag{24}$$

Exergetic efficiency

$$\eta_{exergetic} = \frac{\dot{Q}_E\left|\left(1 - \frac{T_o}{T_r}\right)\right|}{\dot{W}_{comp}} = \frac{COP_{vcr}}{COP_{rev}} \tag{25}$$

Exergy destruction ratio (EDR)

$$EDR = \frac{\dot{E}D_{total}}{\dot{X}_E^Q} \tag{26}$$

$$EDR = \frac{1}{\eta_{exergetic}} - 1 \qquad (27)$$

Efficiency defect (δ_j):

$$\delta_j = \frac{\dot{ED}_j}{\dot{w}_{comp}} \qquad (28)$$

Selection criteria of the refrigerant, input parameters and their values used for analysis

Ammonia has better heat transfer properties than most of chemical refrigerants and therefore it allows for the use of equipment with a smaller heat transfer area, it has highest refrigerating capacity per pound of any refrigerant, specific volume of ammonia is high, and the compressor displacement required per ton of refrigeration is quite small because wet state ammonia is compressed when an intercooler is used along with flash chamber.

A mathematical computational model is developed for performing the energy and exergy analysis of the Integrated Refrigeration System as shown in (Figure 1) using EES software [22].

(a)   Refrigerant: Ammonia

(b)   Effectiveness of the water intercooler ε_{wic} : 0.85

(c)   Condenser temperature T_c : 40°C-52°C

(d)   Evaporator temperature T_E : -40°C

(e)   Isentropic efficiency of compressor: (ηcomp,lp)=0.8

(f)   Ambient state temperature (To): 298 K

(g)   Water inlet temperature flowing through the condenser (Tw1):290 K

(h)   Ambient atmospheric pressure (Patm): 101.325 Kpa

(i)   Tons of refrigeration (TR): 10

Mass flow rate of cooling fluid (water) is varied from 0.18 to 0.85, while discussing its effect on the system performance.

Results and Discussion

An extensive exergy analysis has been performed and the results obtained are arranged in the tabular form. Also the comparison between various parameters has been done e.g. after varying evaporator temperature, ambient state temperature, and mass flow rate of water has been represented in (Figures 3-9).

The effect of evaporator temperature and condenser temperature on COP, heat recovery by water intercooler, exergetic efficiency, exergy destruction ratio, efficiency defect, rise in temperature and mass flow rate with various heat recovery parameters are presented in (Figures 3-9), it reveals that:

•   With the increase in condenser temperature the total exergy destruction increases at a fixed evaporator temperature.

•   Exergetic efficiency decreases with the increase in condenser temperature. Exergetic efficiency has a maximum value of 0.6852 at T_c=313 K, T_e=223 K.

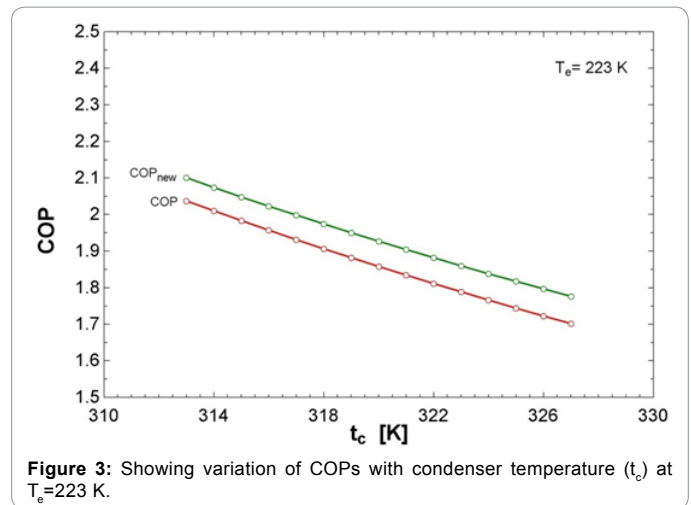•   Evaporator is the most efficient component in the system whereas

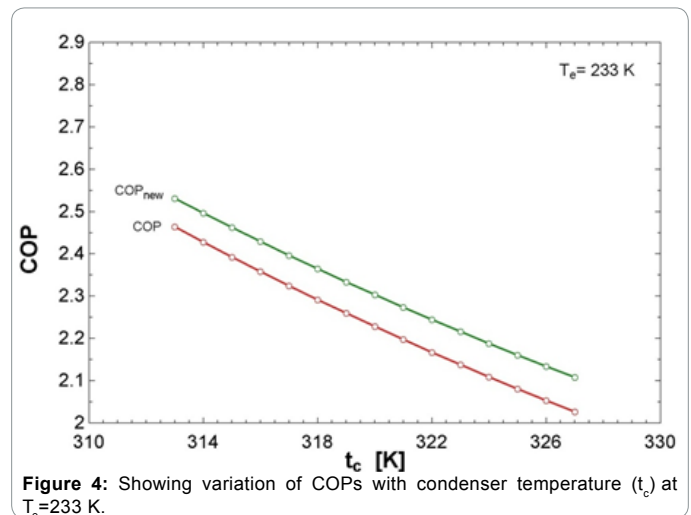**Figure 3:** Showing variation of COPs with condenser temperature (t_c) at T_e=223 K.

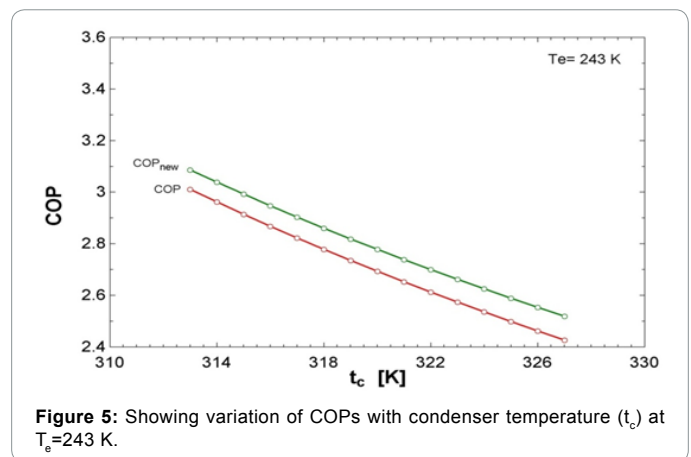**Figure 4:** Showing variation of COPs with condenser temperature (t_c) at T_e=233 K.

Figure 5: Showing variation of COPs with condenser temperature (t_c) at T_e=243 K.

throttling valve (corresponding to the HP side) is the worst component followed by the throttling vale of LP side, condenser, compressor, water intercooler and flash chamber.

•   With the increase in ε_{wic} there is an increase in the COP_{new} for a given evaporator and condenser temperature. The maximum value of COP_{new} obtained is approx. 3.24 at T_C = 303 K, T_e = 303
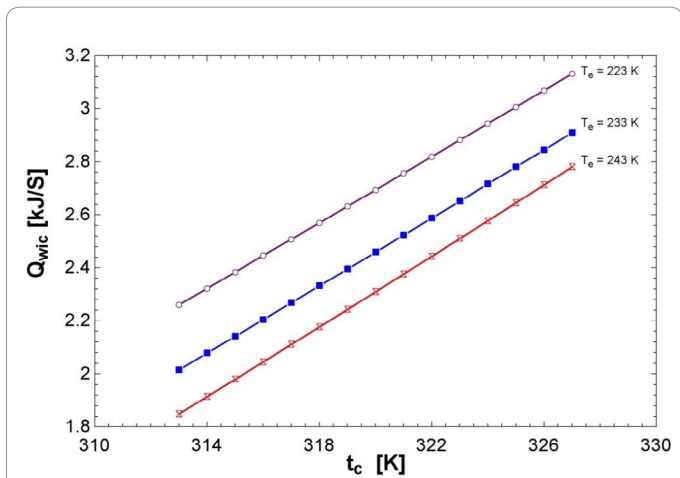
Figure 6: Showing variation of Q_{wic} with condenser temperature (t_c) at different evaporator temperatures.
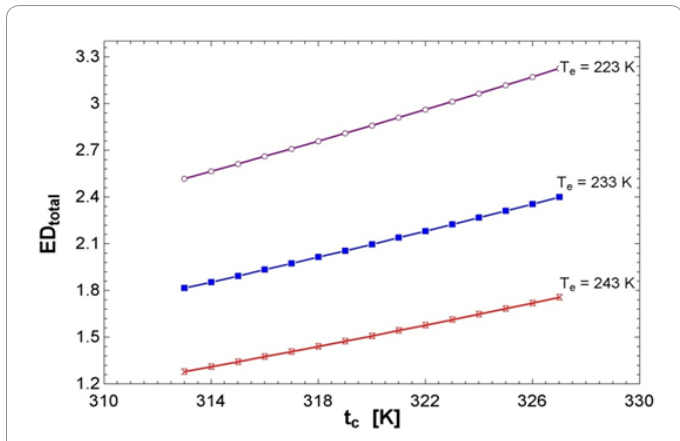

Figure 7: Showing variation of ED_{total} with condenser temperature (t_c) at different evaporator temperatures.
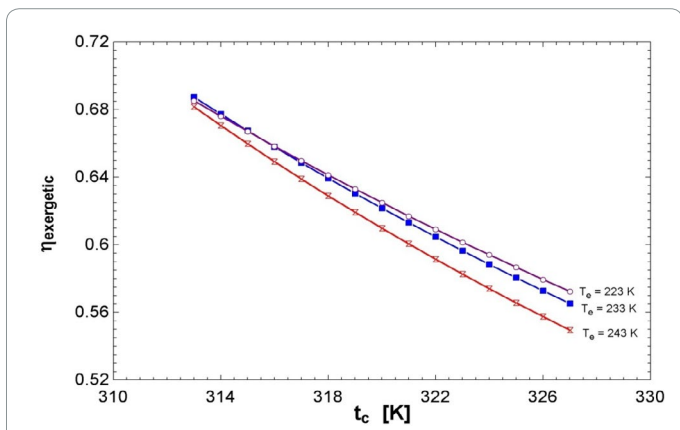

Figure 8: Showing variation of $\eta_{exergetic}$ with condenser temperature (t_c) at different evaporator temperatures.

K and ε_{wic} =0.95.

- Q_{wic} decreases with the increase in T_{w1}. Q_{wic} is maximum at T_{w1} = 280 K, T_c= 313 K, T_e = 223 K with a value of 2.83 kJ/s.
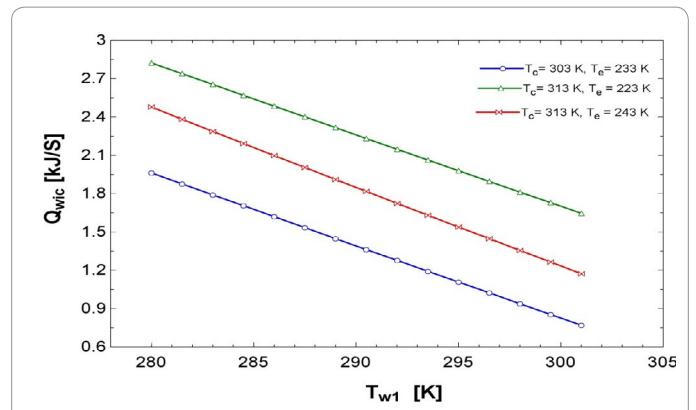

Figure 9: Showing variation of Q_{wic} with inlet water temperature (T_{w1}) at different evaporator & condenser temperatures.

- With the increase in $\eta_{comp,lp}$, $\eta_{exergetic}$ decreases at a given evaporator and condenser temperature.

- T_{w2} increases with increase in TR from 10 TR to 100 TR. At TR =100 heat recovery of 20.15 Kj/s is achieved with T_{w2}=337.2 K. 100TR.

Model validation

Thermodynamic modeling of similar system done by Ouadha et al. predicted components exergetic loses by operating at constant evaporating temperature of -30°C and condensation temperatures of 30, 40°C, with two natural substitutes of HCFC22, namely propane (R290) and ammonia (R717) as working fluids, which can be validated for the same as calculated in the present study to be constant evaporating temperature of -30°C and condensation temperatures of 40°C respectively. The error may be due to approximation but in agreement to be accepted. Similarly, optimal degrees of sub-cooling obtained according to the second law of thermodynamics, Yang et al. enhancement and exergy destruction analyses were conducted numerically for vapor-compression refrigeration systems using R22, R134a, R410A, and R717. The effects of cooling water in a sub-cooler, refrigerant pressure drop among heat exchangers, and superheating in an evaporator were also considered and compared with evaporation and condensation temperature. In the present study the same has been computed as -30°C to -50°C evaporator temperature and 30°C, 40°C condenser temperature for ammonia and with intercooler COP is improved. The error may be due to approximation and could be agreed upon.

Conclusion

During extensive energy and exergy analysis with refrigerant discussed in results in a modified two stage vapour compression cycle, following conclusions are summarized below:

The COP of the system is improved by 4-5% with incorporation of heat recovery through intercooler. It is observed that COP decreases with increase in condenser temperature at a fixed evaporator temperature. Also, as the evaporator temperature increases COP increases and it is maximum (3.087) at evaporator temperature of 243 K and condenser temperature of 313 K. With the increase in condenser temperature the total exergy destruction increases at a fixed evaporator temperature.

This particular result is significant from the point of view of the

application of hot water coming from the intercooler. At lower tonnage of the system, this hot water is useful for household works in cold climate condition. At high tonnage (100TR) of the system, it could be useful for process industries also as difference of the outlet and inlet temperature obtained is approximately 50°C.

Hence, it can be accomplished that the available waste heat of the intercooler of a two stage modified vapor compression system could be utilized and also results in overall increase of COP of the system. Hence, heat recovery through a water intercooler of a two stage vapor compression refrigeration system is found to be feasible and can be maximized by selecting optimum water flow rate, inlet water temperature, suitable operating conditions, and working fluid.

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