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Performance and Irreversibility analysis of two stage cascade refrigeration system for different refrigerant pairs

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ABSTRACT

The performance of two stage cascade refrigeration system and irreversibility of the system is analysed for different refrigerant pairs. Refrigerants R-134a, R-404A, R-152a are used in high temperature circuit and R508B & R23 are used in low temperature circuit. The various working parameters are evaporating temperature (-50°C to -70°C), condenser temperature (30°C to 50°C) and temperature difference in cascade condenser (4°C to 10°C). The degree of condenser sub-cooling and evaporator superheat are assumed as 5°C and 7°C respectively, for all cases. The isentropic efficiencies of the compressors are expected to be equal. The analysis shows that an increase in evaporating temperature (T_E), results increase in COP and decrease in irreversibility of the system. An increase in condenser temperature (T_C), results decrease in COP and increase in irreversibility of the system. An increase in temperature difference in cascade condenser (ΔT), results in decrease in COP and increase in irreversibility of the system. It is found that R134a – R23 has the maximum COP and low irreversibility of the system followed by R134a – R508B among all the considered refrigerants. R134a – R23 pair has maximum COP which is 7.8% more than R134a – R508B. R134a – R23 pair has less irreversibility which is 17% less than R134a – R508B.

KEY WORDS: Cascade, Refrigerants, COP, Irreversibility.

1. INTRODUCTION

The studies on vapour compression cascade refrigeration systems are usually based on energy analysis. Gupta and Prasad investigated the effects of sub-cooling, super-heating and the cascade overlap temperature on the performance of cascade refrigeration systems to find the best refrigerant couple among the couples R12–R23, R22–R23 and R717–R23. Agrawal obtained the optimum inter-stage temperature of a cascade refrigeration system for the refrigerant couples mentioned above using a simple graphical method for evaporator and condenser temperatures. Gupta presented a study aiming optimization of a multi-stage cascaded refrigeration–heat pump system in terms of overall coefficient of performance (COP) and total operating costs using R12 for the heat pump and R23for the refrigeration system, and found that COP between 2.5 and 7.4.

Kilicarslan experimentally inspected the effects of the refrigeration load and water mass flow rate on the performance of cascade and single stage refrigeration systems both using R134a and compared the COP, work of compression, discharge pressure and refrigerant mass flow rate of the cascade system with those of the single stage refrigeration system. The study performed by Dopazo, was based on the analyses of the parameters such as evaporator and condensing temperatures, temperature difference and isentropic efficiencies of the compressors and their effects on the coefficient of the performance (COP) and exergetic efficiency by using CO_2 for low temperature and NH_3 for high temperature sides of the vapour compression cascade refrigeration cycle as natural working fluids. For some industrial applications which require moderately low temperatures, single stage vapour compression refrigeration cycles become impractical. Cascade refrigeration cycle can be used to achieve low temperatures, where series of single-stage units are used that are thermally coupled through evaporator/condenser cascades, for a two-circuit cascade unit. Each circuit has a different refrigerant suitable for that temperature, the lower temperature units progressively using lower boiling point refrigerants. Generally, two-circuit and rarely three-circuit cascade systems are used. In general, if the desired temperature can easily be achieved in a single-stage machine, it will be more efficient than a cascade system due to irreversibility and losses associated with a large number of components.

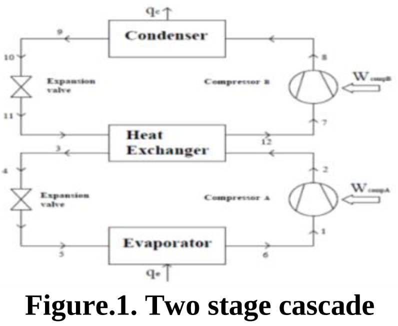

Figure.1. Two stage cascade refrigeration cycle

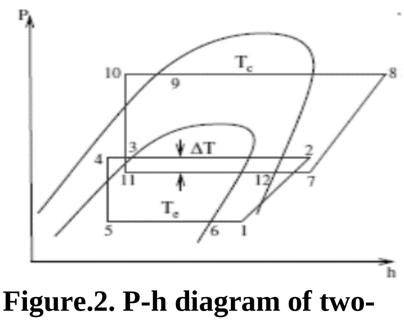

Figure.2. P-h diagram of two-stage cascade refrigeration cycle

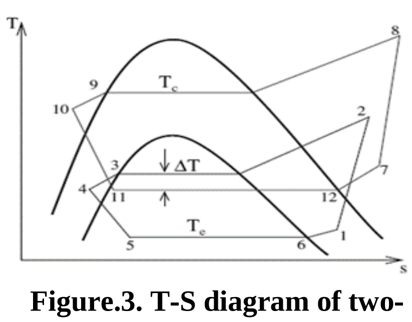

Figure.3. T-S diagram of two-stage cascade refrigeration cycle

2. METHODOLOGY

The thermodynamic analysis of this two stage cascade refrigeration system was performed based on the various assumptions like Adiabatic compression with an isentropic efficiency of 0.9 for both high and low temperature compressors, Negligible pressure drop, heat gain/loss in system components, Isenthalpic expansion of refrigerants in expansion valves, Negligible changes in kinetic and potential energy.

Performance Analysis: The capacity of the evaporator,

$$Q_E = m_L (h_1 - h_5) \quad (1)$$

Compressor power consumption for low temperature circuit,

$$W_L = m_L (h_2 - h_1) \quad (2)$$

Compressor power consumption for high temperature circuit,

$$W_H = m_H (h_8 - h_7) \quad (3)$$

Where, m_L = mass flow rate of low temperature system refrigerant; m_H = mass flow rate of high temperature system refrigerant.

The heat transfer rate in cascade heat exchanger,

$$m_L(h_2 - h_4) = m_H (h_7 - h_{11}) \quad (4)$$

The enthalpies at various state points; $h_1 = h_6 + C_{pv} (T_1 - T_6)$; $h_4 = h_3 - C_{pl} (T_3 - T_4)$; $h_4 = h_5$; $h_{10} = h_{11}$; $h_{10} = h_9 - C_{pl} (T_{10} - T_9)$; $h_7 = h_{12} + C_{pv} (T_7 - T_{12})$

The enthalpies at various state points;

$S_1 = S_6 + C_{pv} \ln \frac{T_1}{T_6}$; $S_1 = S_2$; $S_2 = S_{2'} + C_{pv} \ln \frac{T_{sup}}{T_{CC}}$; $S_7 = S_{12} + C_{pv} \ln \frac{T_7}{T_{12}}$; $S_7 = S_8$; $S_8 = S_{8'} + C_{pv} \ln \frac{T_{sup}}{T_C}$

The overall work of compression,

$$W_C = \frac{W_L + W_H}{\eta_C} \quad (5)$$

The overall COP of the system,

$$COP = \frac{Q_E}{W_C} \quad (6)$$

Irreversibility Analysis: Irreversibility (I) is the measure of difference from the reversible processes. In other words, it is the difference between the reversible power and useful power.

$$I = W_{rev} - W_{use} \quad (1)$$

$$I = T_O . S_{gen} \quad (2)$$

The irreversibility of the cascade system consists of the summation of the irreversibilities in the low temperature system (system A), high temperature system (system B) and cascade heat exchanger.

$$I_{cas} = I_A + I_B + I_{HX} \quad (3)$$

Irreversibility's in system are caused by the compressor, expansion valve A and evaporator. Then, irreversibility of system A,

$$I_A = I_{compA} + I_{TXVA} + I_{evap} \quad (4)$$

Irreversibility of a compressor increases with reducing isentropic efficiency. It also increases by increasing compressor outlet temperature by keeping the pressure ratio constant. The irreversibility of the compressor A,

$$I_{compA} = T_O . m_A (S_2 - S_1) \quad (5)$$

Where T_0 is the environment temperature, S_1 and S_2 are the specific entropies across compressor A.

The main cause of the irreversibility during expansion process is the sudden expansion of the refrigerant. The loss in specific refrigeration capacity also increases the irreversibility of an expansion valve. The irreversibility of thermostatic expansion valve,

$$I_{TXVA} = T_O . m_A (S_5 - S_4) \quad (6)$$

Where S_4 is the specific entropy at the thermostatic expansion valve inlet and S_5 is the specific entropy at the outlet. The irreversibility in an evaporator is mainly caused by the heat transfer through a finite temperature difference between the evaporator and refrigerated space. Degree of superheating inside refrigeration space also affects the irreversibility of the evaporator. The irreversibility of the evaporator is given by,

$$I_{evap} = T_O . m_A [(S_1 - S_5) - \frac{h_1 - h_5}{T_{rs}}] \quad (7)$$

Irreversibility analysis of the refrigeration system B can be performed by adopting the same assumptions used in the analysis of the refrigeration system A. The irreversibility of system B can be given by,

$$I_B = I_{compB} + I_{TXVB} + I_{cond} \quad (8)$$

Where, I_{compB}, I_C and I_{TXVB} are irreversibility of the compressor, condenser and thermostatic expansion valve in system B, respectively.

Assuming that there is no heat transfer to the environment through the surface of the compressor in system B, the irreversibility of the compressor can be written as,

$$I_{compB} = T_O . m_B(S_8 - S_7) \qquad (9)$$

Where, S_7 and S_8 are the specific entropies across compressor B.

The main cause for the irreversibility in a condenser is the heat transfer between the condenser and its condensing medium at a finite temperature difference. The irreversibility of the condenser is given by,

$$I_{cond} = T_O . m_B[(S_{10} - S_8) - \frac{h_{10}-h_8}{T_O}] \qquad (10)$$

Where, S_{10} is the specific entropy of refrigerant at expansion valve B inlet.

The irreversibility of thermostatic expansion valve A can be expressed as,

$$I_{TXVB} = T_O . m_B(S_{11} - S_{10}) \qquad (11)$$

Where, S_{11} is the specific entropy of refrigerant at the evaporator inlet.

The irreversibility for the heat exchanger can be determined from,

$$I_{HX} = T_O[m_A(S_4 - S_2) - m_B(S_7 - S_{11})] \; (12)$$

The irreversibility's associated with the heat exchanger are because of heat transfer through the fluid streams and phase change of the fluids in the heat exchanger. Note that the irreversibility caused by fluid friction is not the case for the heat exchanger because the pressure losses throughout the cascade system are neglected.

3. RESULTS AND DISCUSSION

By carrying out the thermodynamic analysis and irreversibility of the system for the conditions (Evaporator temperature T_E (°C), Condenser temperature T_C (°C), Cascade heat exchanger temperature difference ΔT(°C)), values at various state points of the cascade refrigeration cycle have been obtained. In the present work, Coefficient of performance of two stage cascade refrigeration cycle and irreversibility has been calculated.

The performance of cascade refrigeration system with 1 kW capacity is analysed under various conditions using MATLAB software. The various refrigerant thermo-physical properties are taken using REFROP software. By changing the evaporator temperature, condenser temperature, temperature difference in cascade heat exchanger, various results like COP of the cascade system and Irreversibility of the system are obtained.

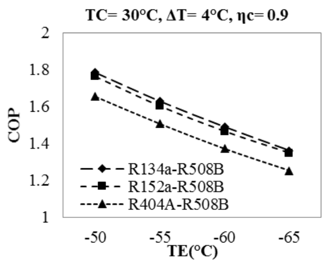

Figure.4. COP variations with evaporator temperature

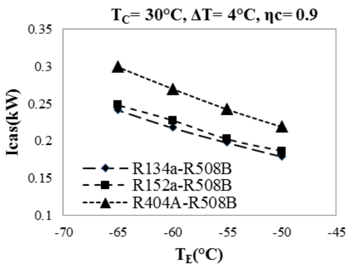

Figure.5. Irreversibility variations with evaporator temperature

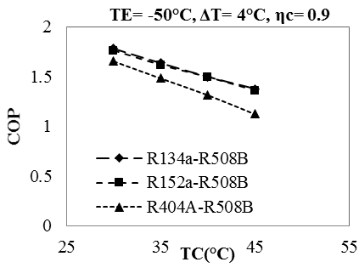

Figure.6. COP variations with condenser temperature

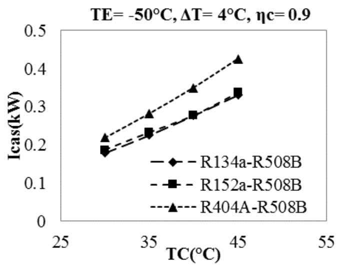

Figure.7. Irreversibility variations with condenser temperature

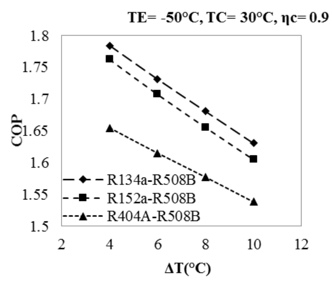

Figure.8. COP variations with temperature difference

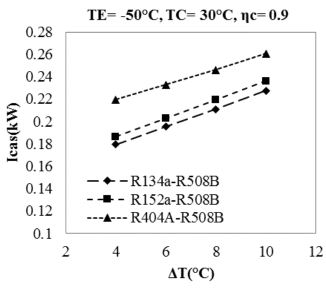

Figure.9. Irreversibility variations with temperature difference

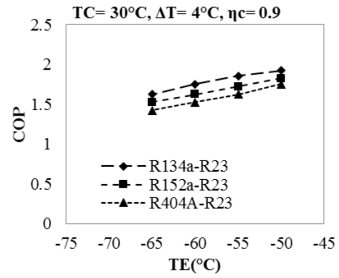

Figure.10. COP variations with evaporator temperature

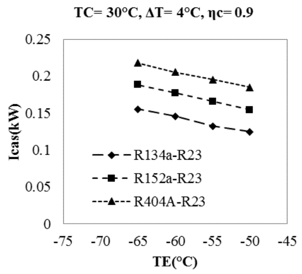

Figure.11. Irreversibility variations with evaporator temperature

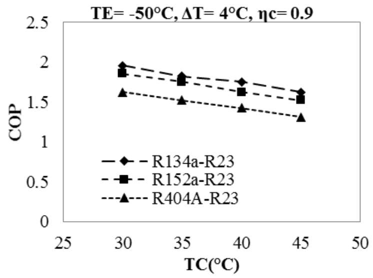

Figure.12. COP variations with condenser temperature

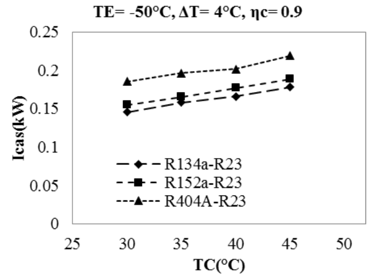

Figure.13. Irreversibility variations with condenser temperature

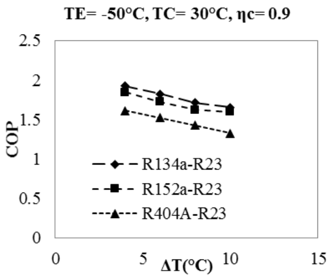

Figure.14. COP variations with temperature difference

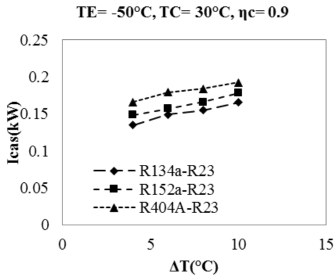

Figure.15. Irreversibility variations with temperature difference

4. CONCLUSION

The performance analysis of two stage cascade refrigeration system has been performed and results were obtained and plotted graphically. It is found that R134a – R23 has the maximum COP under all conditions followed by R134a-R508B. And also R134a – R23 has less irreversibility followed by R134a-R508B when compared with all other refrigerants.

The analysis shows that an increase in evaporating temperature (T_E), results increase in COP and decrease in irreversibility of the cascade refrigeration system. An increase in condenser temperature (T_C), results decrease in COP and increase in irreversibility. An increase in temperature difference in cascade condenser (ΔT), results decrease in COP and increase in irreversibility.

And also R134a refrigerant has got maximum COP and less irreversibility when compared with all refrigerants for HTC. R23 refrigerant has got maximum COP and less irreversibility when compared with all refrigerants for LTC. R134a – R23 pair has got maximum COP which is 7.8% more than R134a – R508B. R134a – R23 pair has got less irreversibility which is 17% less than R134a – R508B.

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