ISSN (Print) : 2320 – 3765
ISSN (Online): 2278 – 8875

International Journal of Advanced Research in Electrical, Electronics and Instrumentation Engineering
Vol. 2, Issue 4, April 2013

Performance Analysis of Solar Assisted Cascade Refrigeration System of Cold Storage System

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ABSTRACT: By preserving food products in the cold storage, good quality of food can be supplied throughout the year, minimizing the damage and enhancing the agricultural economy. Huge amount of electric energy is required to run a cold storage. Minimizing the use of electrical energy or proper utilization of electrical energy and thus by enhancing the COP would definitely be the major factor for sustainability of the cold storage and the decrease of the price of preserving the food in it. Theoretical modeling of solar-assisted cascade refrigeration system in cold storage is studied in this work. The system consists of electricity-driven vapor compression refrigeration system and solar-driven vapor absorption refrigeration system. The vapor compression refrigeration system is connected in series with vapor absorption refrigeration system. The results shows higher COP as compared with the conventional vapor compression refrigeration system. COP of this type refrigeration system increases as sunlight becomes intense.

Keywords: coefficient of performance (COP),Cold storage, solar-assisted cascaded refrigeration, vapor absorption refrigeration system,vapour compression refrigeration system.

I.INTRODUCTION

A. History of refrigeration:

I. Murthy et al. (1991) tested different ejector dimensions at the cooling capacity about 0.5 kW. R12 was used as the refrigerant. A COP in the range of 0.08-0.33 was obtained.
II. Bejan et al. (1995) designed a single-stage solar-driven ejector system with 3.5 kW of refrigeration capacity at an evaporating temperature of 4°C and a generating temperature of 90-105°C with R114.
III. A solar assisted ejector-vapor compression cascade system was proposed by Göktun (2000). The inter-cooler was installed serving as a condenser for the vapor compression system and an evaporator for the ejector system.
IV. Sözen and Özalp (2005) proposed a solar-driven ejector absorption system. The main focus of this study is to investigate the possibility of using this system in Turkey. As a result of the analysis, using the ejector, the COP improved by about 20%.
V. Performance variations of a solar-powered ejector cooling system (SECS) using an evacuated-tube collector has been presented by Ersoy et al. (2007) in different cities in Turkey.
VI. Varga et al. (2009) carried out theoretical study to assess system and refrigeration efficiencies of a solar-assisted ejector cycle using water as the operating fluid. The results indicated that in order to achieve an acceptable coefficient of performance, generator temperatures should not fall below 90°C. Evaporator temperatures below 10°C and condenser temperatures over 35°C.

II. VAPOR COMPRESSION REFRIGERATION CYCLE

In vapor compression refrigeration cycle (shown in Figure 1) the refrigerant is vaporized completely before it is compressed and the turbine is replaced with a throttling device. Path description is given in Table 1

Table 1: VCRS path description

Path	Processes
1 – 2	isentropic compression in a compressor
2 – 3	constant – pressure heat rejection in a condenser
3 – 4	throttling in an expansion device
4 – 1	constant – pressure heat absorption in an evaporator

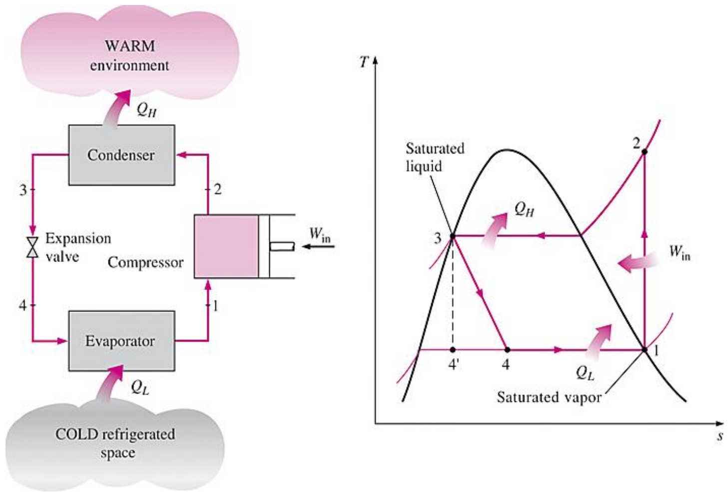


Figure 1: vapor compression refrigeration cycle

$$\mathbf{COP_R} = \frac{q_L}{w_{net\ ,in}} = \frac{h_1 - h_4}{h_2 - h_1} \quad \mathbf{.....(i)}$$

Characteristics of VARS:

1. HIGHER ENERGY CONSUMPTION.

2. MODERATE COP.

3. LOWER INITIAL INVESTMENT.

4. EMITS GREEN-HOUSE GAS.

II. ABSORPTION REFRIGERATION SYSTEMS

There is a source of inexpensive thermal energy at a temperature of 100 to 200°C is absorption refrigeration.
Characteristics of AVRS:-

1. A liquid is compressed instead of a vapor and as a result the work input is very small.
2. The unit cost of thermal energy is low and is projected to remain low relative to electricity.
3. The COP of actual absorption refrigeration systems is usually less than unity.

$$\mathbf{COP}_{\mathbf{absorption}} = \frac{Desired\ \ output}{Required\ \ output} = \frac{Q_L}{Q_{gen} + W_{punp\ \ ,in}} \cong \frac{Q_L}{Q_{gen}} \quad \mathbf{....\ (ii)}$$

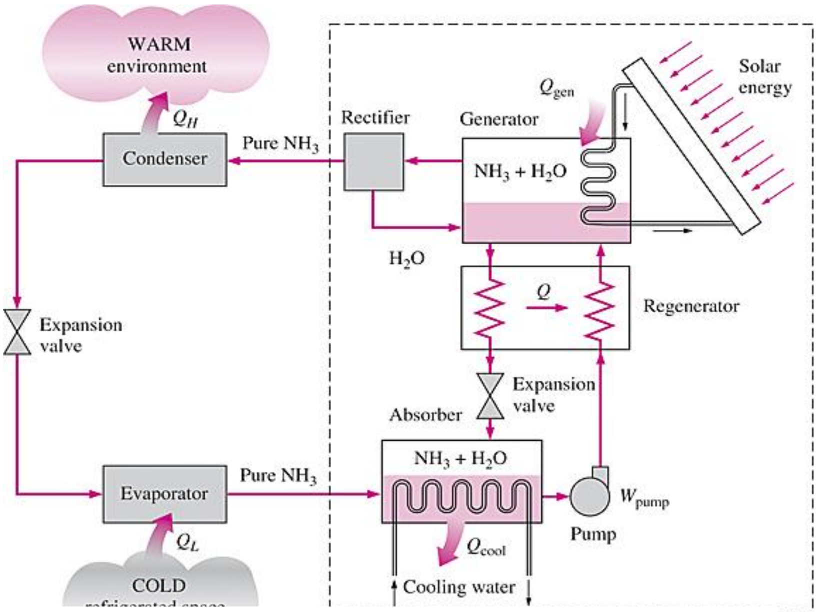


Figure 2: Absorption refrigeration system process cycle

III. PROPOSED SYSTEM FOR FUTURE USE

A major application area of thermodynamics is refrigeration, which is the transfer of heat from a lower temperature region to a higher temperature one. Devices that produce refrigeration are called refrigerators, and the cycles on which they operate are called refrigeration cycles.
The reversed Carnot cycle is the most efficient refrigeration cycle operating between TL and TH.

$$COP_{R,Carnot} = \frac{1}{\frac{T_H}{T_L} - 1} \qquad \text{..... (iii)}$$

Where, T_H = condenser temperature
T_L = evaporator temperature

Our main objective of this project is to increase the efficiency of refrigeration cycle. So we can increase the COPs as the difference between the two temperatures decreases, that is, as T_L rises or T_H falls.

But there is a certain limit to increase the evaporator temperature due to refrigerant limitation. So there is a possibility to increase the COP of refrigeration cycle by decreasing the condenser temperature.

This can be done by using a vapour compression refrigeration system is connected in series with vapor absorption refrigeration system. Solar-driven vapour absorption refrigeration has advantages in energy conservation and environmental protection.However, conventional solar-driven vapour compression refrigeration has many disadvantages, such as

1. Low efficiency
2. Intermittent operation
3. Too large unit cost
4. High capital cost

Besides, the popularity and application of electricity-powered vapour compression refrigeration have been one of the main reasons of summer and winter electricity demand peak in recent years. In particular, 60% of the total electricity capacity is generated by coal. Not only does it exacerbate the depletion of fossil fuel, but also it can produce dangerous gases such as carbon dioxide, nitrogen oxides and sulphur oxides, which cause the greenhouse effect and deteriorate the global environment. In order to solve the above problems in the conventional electricity-powered vapour compression refrigeration system and solar-driven one, a solar assisted cascade refrigeration system is analysed.

The main objective of this paper is to design, fabricate and evaluate a new solar-assisted cascaded refrigeration system which was the combination of both vapour absorption refrigeration system and vapour compression refrigeration system.

IV. MODEL DEVELOPMENT

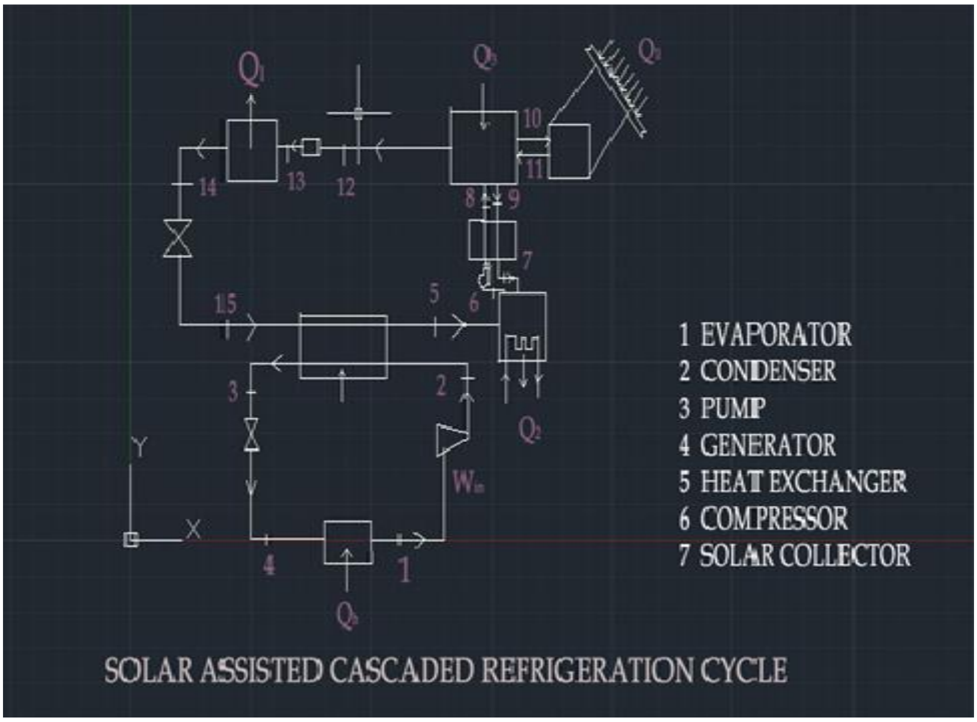


Figure 3: Line diagram of Solar Assisted Cascaded Refrigeration Cycle

Characteristics features of SACR cycle:-

1. It is very efficient and environmental friendly.
2. Power consumption is reduced greatly.
3. Coefficient of performance is nearly 25-30 % greater than VCRS.
4. It uses environment friendly energy resources.

$$\textbf{COP} = \frac{Q_H}{W_{compressor} + Q_{generator} + W_{pump}} \quad \textbf{.... (iv)}$$

The cycle can be broken into different flows, one comprising of the ammonia-water mixture and the other comprising of the R-134a alone. Points (1-4) are the cycle of the R-134a, and the rest of the points constitute the ammonia vapor cycle. In the bottom cycle R-134a taking heat vaporizes in the evaporator (1) then the it is compressed by the compressor (2) and goes to the condenser (the evaporative condenser heat exchanger) at point (3) and then expanded in the expansion valve (4) and comes back to the evaporator again. Now in the top cycle has solution rich in refrigerant at point (7) is pumped to higher pressure through the solution heat exchanger (5) into the generator where heat is added and an ammonia-water vapor mixture is sent to the rectifier, and the solution poor refrigerant is sent back through the solution heat exchanger to the absorber. The ammonia-water vapor is purified in the rectifier by condensing the water vapor in the mixture into liquid. The pure ammonia vapor is sent to the condenser and the water liquid is sent back to the generator. The ammonia vapor loses heat to the surrounding by convection as it goes through the condenser and is cooled into liquid ammonia. The ammonia liquid is passed through a flow restrictor where it experiences a sudden drop in pressure and evaporates because this new pressure is less than its saturation pressure. The ammonia is now a saturated vapor at a temperature that corresponds to this new pressure. This temperature is always lower than the desired compartment temperature. The saturated ammonia vapor is sent to the evaporator where heat from the condenser of bottom cycle is absorbed. The ammonia vapor then absorbs heat before returning to the absorberwhere it is absorbed into the water and the process repeats again.

Now, from energy balance equation:

$$m_a h_2 + m_b h_5 = m_a h_3 + m_b h_6$$
$$\Rightarrow m_a (h_2 - h_3) = m_b (h_6 - h_5)$$
$$\Rightarrow \frac{m_a}{m_b} = \frac{h_6 - h_5}{h_2 - h_3}$$

Now,

$$COP = \frac{cooling\ \ effect}{net\ \ work\ \ input} = \frac{Q_H}{W_{compressor} + Q_{generator} + W_{pump}} \text{.... (v)}$$

[For reversible device, $W_{1-2} = -\int_1^2 vdp$,here pump is a reversible device and working fluid is in liquid phase, so *v* is very small and hence W_{pump} is negligible]

$$\Rightarrow COP = \frac{m_a (h_1 - h_4)}{m_b (h_2 - h_1) + Q_{generator}}$$

From the top cycle, using the first law

$$\delta Q - \delta W = 0$$

$$\Longrightarrow Q_{generator} + Q_c = Q_{absorber} + Q_k \quad [\text{As} \ \delta W = W_{pump} \approx 0]$$
$$\Longrightarrow Q_{generator} = Q_{absorber} + Q_k - Q_c$$

Now, $Q_u = m_b (h_{15} - h_{14})$

$$Q_{absorber} = m\, c_p (t_{in} - t_{out})$$
$$Q_c = m_b (h_6 - h_5)$$
$$\therefore Q_{generator} = m\, c_p (t_{in} - t_{out}) + m_b (h_{15} - h_{14}) - m_b (h_6 - h_5)$$
$$\therefore COP = \frac{m_a (h_1 - h_4)}{m_a (h_2 - h_1) + m\, c_p (t_{in} - t_{out}) + m_b (h_{15} + h_5 - h_6 - h_{14})}$$

V. RESULTS AND DISCUSSIONS

Data obtained experimentally

$m_a = 0.28\ kg/s$

$m_b = 0.023\ kg/s$

$t_{absorber\ \ out} = 40.6\ ℃$

$t_{generator\ \ out} = 95\ ℃$

$Q_l = m_b (h_1 - h_4) = 0.023\ (239.16 - 55.16) = 4.232\ kj/kg$

$W_{compressor} = m_b (h_2 - h_1) = 0.023\ (255.93 - 239.16) = 0.389\ kj/kg$

$Q_{generator} = m_a c_p \left(t_{generator\ \ out} - t_{absorber\ \ out}\right) = 0.28 * 0.037\ (95 - 40.6) = 0.457$

$$\therefore COP = \frac{Q_l}{W_{compressor} + Q_{generator}} = \frac{4.232}{0.3857 + 0.457} = 5.022$$

Here COP increases by $\left(\frac{5.022 - 3.97}{3.97}\right) = \mathbf{26.49}\ \%$

VI. FIGURES AND TABLES

Graphical plot of cop:

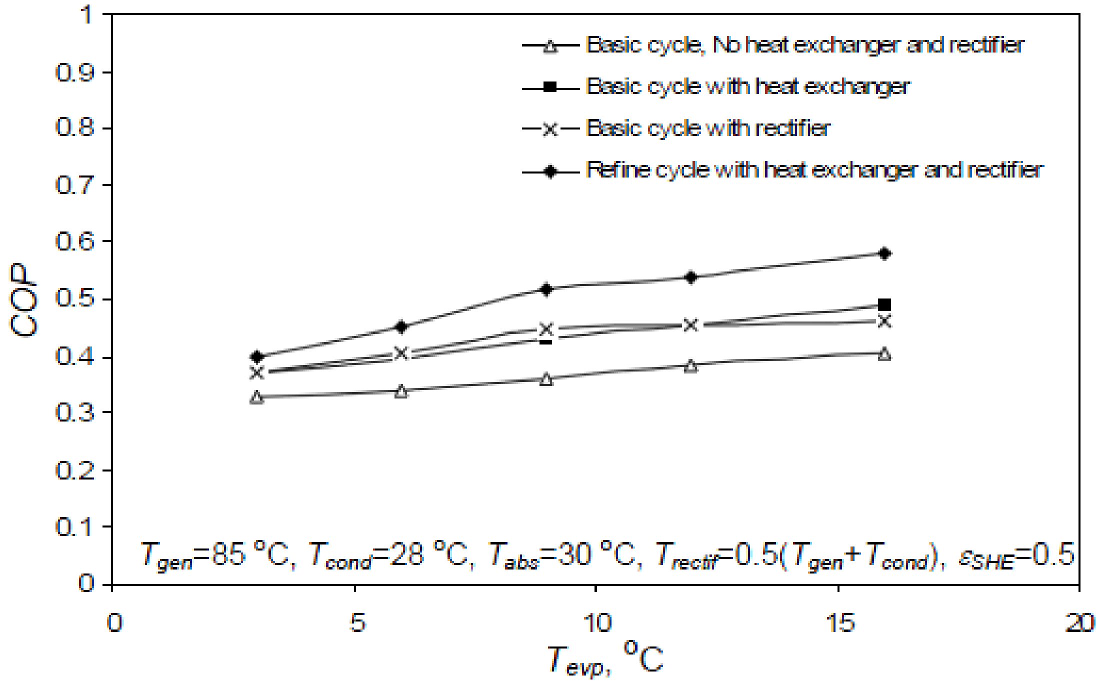


Figure 4: Variation of COP with respect to evaporator temperature

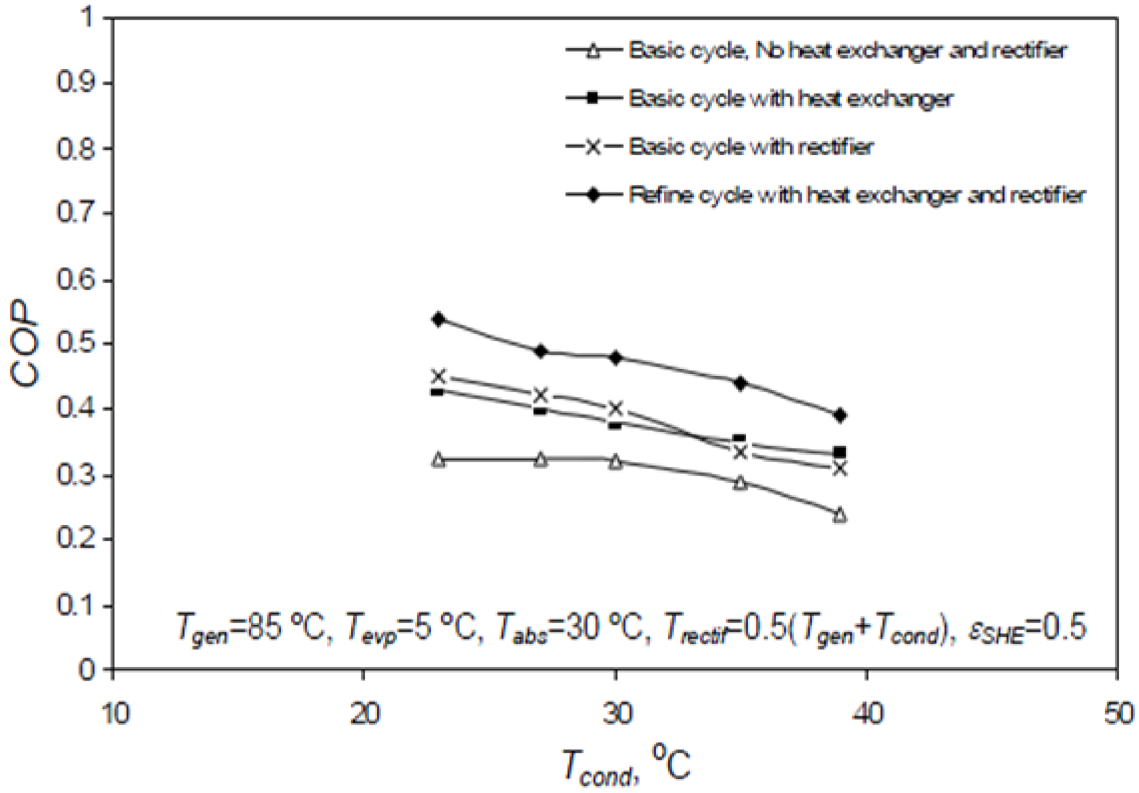


Figure 5: Variation of COP with respect to condenser temperature

V. CONCLUSION

The solar-assisted cascade refrigeration system includes the solar-driven vapour absorption refrigeration unit and electricity-powered vapour compression refrigeration unit. COP of the solar cascade refrigeration system is up to 6.1 with the solar intensity of 700W/m2, outdoor air temperature of 35°Cand chilled water supply temperature of 7°C. Power consumption of the cascade refrigeration system is 50% lower than that of the conventional vapour compression refrigeration (CVCS) in the cooling mode. The COP of the new vapour compression refrigeration system (VCRS) increases as sunlight becomes intense. COP of the cascade refrigeration system is up to the maximum when COP of the conventional vapour compression refrigeration system is minimal.

REFERENCES

[1]. Thermodynamics, an engineering approach; *Yunus A Cengel, Michael A Boles*.**Journal paper:**

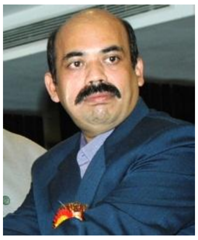
[2]. S. Rosiek, F.J. Batlles, "Integration of the solar thermal energy in the construction: Analysis of the solar-assisted air conditioning system installed in CIESOL building," *Renewable Energy, vol.34, 2009, pp1423-1431pp2820-2832.*

[3]. R.J. Romero, W. Rivera, R. Best, "Comparison of the theoretical performance of a solar air conditioning system operating with water/lithium bromide and an queousternary hydroxide," *Solar Energy Materials & Solar Cells, vol.63, 2000,pp387-399*

[4]. Lin Wang, Aihua Ma, Xiwen Zhou, Yingying Tan, Xiaona Yan and Wang Yu, "Environment and energy challenge of airconditioner in China," *The proceedings of the International Conference of Environmental Pollution and Public Health, China, vol.2,No. 5, 2008, pp4413-4416.*

[5]. B.J. Huang, J.M. Chang, V.A. Petrenko, K. B. Zhuk, "A solar ejector cooling system using refrigeration R141b," *SolarEnergy, vol.64, No.4-6, 1998, pp223-226.*

BIOGRAPHY



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