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Simulation and Analysis of Biogas operated Double Effect GAX Absorption Refrigeration System

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Abstract : A thermodynamic simulation of a double effect generator heat exchanger absorption refrigeration cycle using biogas as source of energy has been carried out. The binary mixture considered in the present investigation was NH_3 – H_2O (Ammonia - Water). This simulation was performed in order to investigate the effect of the temperature and pressure of the high temperature generator and the pressure of evaporator have over the Coefficient of Performance (COP) for a constant condenser and absorber temperatures. The basic parameters at various state points of the cycle was computed using standard correlations. The solution circulation rates and volume of biogas required for operation of the cycle are analysed for the variations in operating parameters at the high temperature generator and evaporator.

Keywords: Absorption refrigeration, Double effect cycle, Binary mixture, Coefficient of performance, Solution circulation rate, Thermodynamic Simulation.

I. INTRODUCTION

The need to augment world's energy sources has recently stimulated those interested in renewable energy sources and waste heat utilization to reexamine the potential of absorption cycles for refrigeration applications. Though, the absorption cycles have low COP values, with increasing energy prices, Ozone layer depletion, Global warming effects, these units are going to become more competitive in the days to come.

Unlike mechanical vapor compression refrigerators, these systems cause no ozone depletion and reduce demand on electricity supply. Besides, heat powered systems could be superior to electricity powered systems because of the use of inexpensive waste heat, solar, biogas, biomass or geothermal energy sources for which the supplying cost is negligible in many cases. Despite using an economic energy source, the system is characterized by its low COP, for that reason it is necessary to perform a study in order to find the most efficient operation range. One of the main factors that have helped to develop this kind of systems is the thermodynamic simulation that can be carried out in order to study the different variables affecting the performance of the equipment.

In 1977 Shwarts and Shitzer[1] analyzed thermodynamically the possibility to operate the solar absorption refrigeration system for air conditioning. Their results showed that the system was suitable for domestic use. Van Passen[2] presented the thermodynamic simulation of a solar absorption refrigeration system. Whitlow[3] studied the absorption refrigeration cycle from the thermodynamic point of view. The use of heat exchangers and some other binary mixtures were recommended. Da Wen Sun[4], analyzed and performed an optimization of the water – ammonia cycle. As a result, he obtained a mathematical model that allowed the simulation of the process. Sun[5], presented a thermodynamic design and performed an optimization of the absorption refrigeration process in order to map the most common cycles for water – ammonia, and lithium bromide – water. The results can be used to select the operation conditions in order to obtain a maximum performance from the system. Sun[6], performed a thermodynamic analysis of different binary mixtures considered in the absorption refrigeration cycle.

A lot of work has been done in this area and the effect of the generator and evaporator temperatures have been considered extensively. M.A. Siddiqui et. al[5]. has studied the optimum generator temperatures for single effect ammonia water absorption systems at subfreezing evaporator temperatures.

In this work, a double effect vapour absorption refrigeration system operating on NH_3 – H_2O as refrigerant and absorbent pair is considered for analysis. Initially, the basic properties at various state points of the cycle was computed using standard correlations. The high temperature generator temperature and pressure and the evaporator pressure were varied and the effect of the variation over the Coefficient of Performance (COP) for a constant condenser and absorber temperatures have been studied. The solution circulation rates and volume of biogas required for operation of the cycle are analysed for the variations in operating parameters at the high temperature generator and at the evaporator.

II. SYSTEM DESCRIPTION : DOUBLE EFFECT VAPOUR ABSORPTION REFRIGERATION SYSTEMS

A number of different configurations can produce a double effect absorption chiller. The two basic types are the double-condenser double effect and the double-absorber double effect. Their principle is based on the fact that the cooling capacity depends primarily on the amount of refrigerant that is vaporized in the evaporator and that by reusing the waste heat from the condensation or the absorption stages, more refrigerant can be desorbed from solution. In the former case, a primary high temperature generator yields vapor whose latent heat is used to fire a secondary low temperature generator at a lower pressure. The latent heat of the vapor desorbed in the secondary generator is then rejected to the heat sink. In the latter case heat from one absorber is used to fire the secondary generator. The vapor from both generators is combined and enters a single condenser.

Fig. 1 illustrates the main components of the double effect absorption refrigeration cycle. High-pressure liquid refrigerant from the condenser passes into the evaporator through a precooler and an expansion valve (V-4) that reduces the pressure of the refrigerant to the low pressure existing in the evaporator. The liquid refrigerant vaporizes in the evaporator by absorbing heat from the material being cooled and the resulting low-pressure vapor passes to the absorber through a precooler, where it is absorbed by the strong solution coming from the low temperature generator through an expansion valve (V-1), and forms the weak solution.

The weak solution is pumped to the high temperature generator pressure through Preheater – I and Preheater - II, and the refrigerant is boiled off. The remaining solution flows first to the low temperature generator through Preheater – II. The refrigerant boiled off from the high temperature generator is passed through the low temperature generator inside a pipe where heat exchange takes place thereby further liberating refrigerant from the solution in the low temperature generator. The refrigerant generated from the low temperature generator and the high temperature generator enter the condenser.

The weak solution from the low temperature generator is passed through the Preheater – I to the absorber where it absorbs the refrigerant vapours coming from the evaporator thus completing the double effect cycle. By weak solution is meant that the ability of the solution to absorb the refrigerant vapor is weak, according to the ASHRAE definition. In order to improve system performance, the preheaters are included in the cycle. An analyzer and a rectifier need to be added to remove water vapor from the refrigerant mixture leaving the generator before reaching the condenser. For the current study, it is assumed that the refrigerant vapor is 100% ammonia and the analysis of analyser and rectifier have not been considered in this work.

III. SYSTEM MODELING AND SIMULATION

In order to analyze the system, mass and energy balance must be performed at each component. The mass, material and energy balance at each component as a control volume gives the following relations

At absorber

$$\dot{m}_1 = \dot{m}_{10} + \dot{m}_{19}$$
(1)

$$\dot{m}_1 x_1 = \dot{m}_{10} x_{10} + \dot{m}_{19} x_{19}$$
(2)

$$\dot{Q}_A = m_{19} h_{19} + m_{10} h_{10} - m_1 h_1$$
(3)

At high temperature generator

$$\dot{m}_4 = \dot{m}_5 + \dot{m}_{11}$$
(4)

$$\dot{m}_4 x_4 = \dot{m}_5 x_5 + \dot{m}_{11} x_{11}$$
(5)

$$\dot{Q}_G = m_{11} h_{11} + m_5 h_5 - m_4 h_4$$
(6)

At low temperature generator

$$\dot{m}_7 = \dot{m}_8 + \dot{m}_{14}$$
(7)

$$\dot{m}_7 x_7 = \dot{m}_8 x_8 + \dot{m}_{14} x_{14}$$
(8)

At condenser

$$\dot{Q}_C = m_{13} h_{13} + m_{14} h_{14} - m_{15} h_{15}$$
(9)

At evaporator

$$\dot{Q}_E = m_{18} h_{18} - m_{17} h_{17}$$
(10)

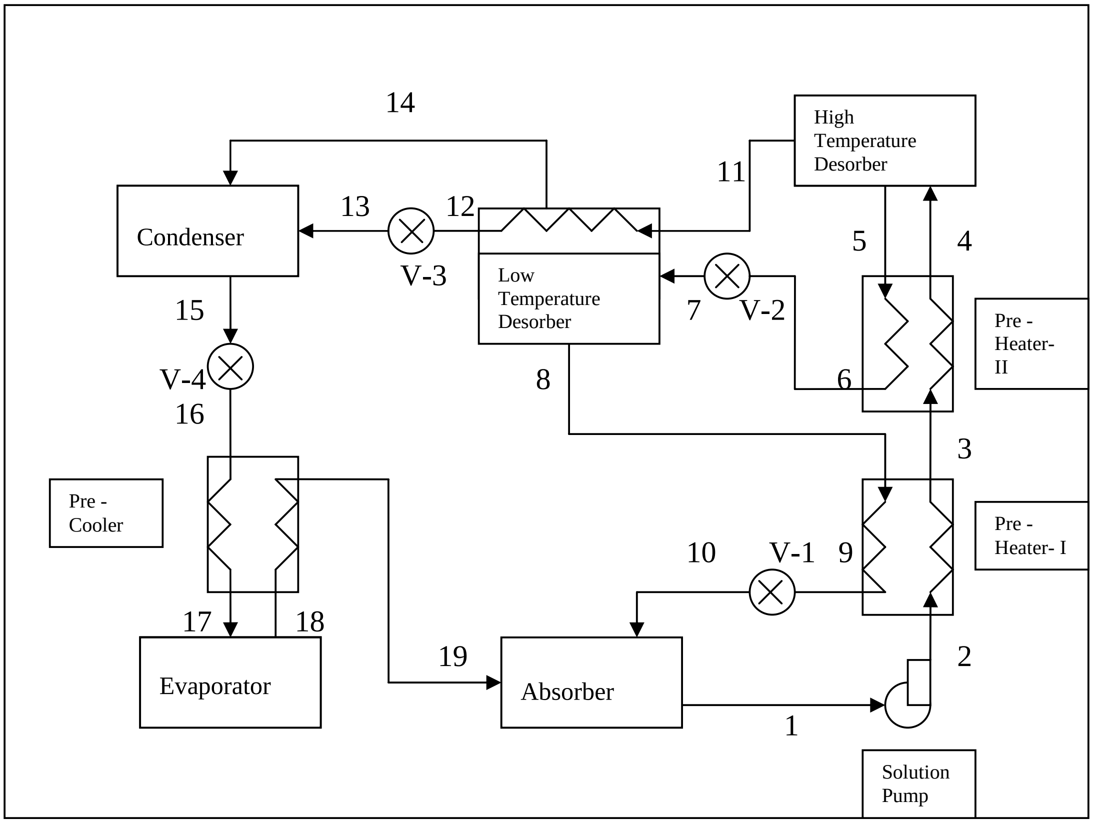

Fig.1: Double Effect GAX Vapour Absorption Refrigeration System

General

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4$$
(11)

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_7$$
(12)

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{11}$$
(13)

$$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{13}$$
(14)

$$\dot{m}_{14} = \dot{m}_{15} = \dot{m}_{16} = \dot{m}_{17} = \dot{m}_{18} = \dot{m}_{19}$$
(15)

$$x_1 = x_2 = x_3 = x_4$$
(16)

$$x_8 = x_9 = x_{10}$$
(17)

$$x_5 = x_6 = x_7$$
(18)

It is assumed that pure Ammonia is entering the condenser i.e ., (x=1) yields

$$x_{11} = x_{12} = x_{13} = x_{14} = x_{15} = x_{16} = x_{17}$$
$$= x_{18} = x_{19} = 1.0$$
….(19)

$$COP = \dot{Q}_E / \dot{Q}_G$$
(20)

Work input to solution pump is computed and included in the calculations.

Thermodynamic Properties

Equilibrium pressure data of pure Ammonia, Liquid enthalpy for NH_3 - H_2O mixture are from Zeigler & Trepp. Liquid and vapour enthalpy for NH_3, H_2O and NH_3 - H_2O mixture are from Infante Ferriera. Equilibrium pressure data of NH_3, H_2O mixture is from Perry and Clinton. The correlations for Superheated vapour enthalpy for NH_3 mixture was taken from C.P.Arora. The heating values of the biogas and the volume of the biogas required per TR of refrigeration were calculated using the relations developed by M.A. Siddiqui et. al.

In this study a computer code has been developed to compute the first law analysis of the absorption chiller. A detailed analysis of the absorption chiller requires a knowledge of main system pressures which are maximum, intermediate and minimum pressures, temperatures and flow rates at strategic points in the system. In the simulation the following operating parameters – i) HT generator temperature ii) HT generator pressure and iii) Evaporator pressure were varied and the effect of the variations are studied to arrive at the optimum operating conditions.

A number of model runs have been performed and compared in order to investigate the interactions of different operating conditions on the performance of the absorption unit. The following assumptions were made during the analysis:

1. The condenser temperature is kept equal to the absorber temperature.
2. Heat losses and gains between the system and its environment are neglected.
3. Friction and pressure losses in pipes and components are neglected.

For the design values assumed initially
Absorber = 25 ^{0} C, High temp. Generator = 190 ^{0} C, Evaporator = -10 ^{0} C, Condenser = 25 ^{0} C,
t_B = biogas source temperature at HTG inlet = 300 ^{0} C;
Capacity = 1 kW;

The component analysis is done as follows:

Evaporator : Assuming, t_{20} = 6 ^{0} C, t_{21} = 12 ^{0} C,
$Q_{E\text{-}I} = m_{17*}(h_{18} - h_{17}) = 1.0$ kW;
$\Delta T_{E\text{-}I}=[(t_{21} - t_{E\text{-}I})-(t_{20} - t_{E\text{-}I})] / \ln [(t_{21} - t_{E\text{-}I}) / (t_{20} - t_{E\text{-}I})]$
$\Delta T_{E\text{-}I} = 3.348663759307$;
Evaporator Area = $A_{E\text{-}I} = Q_{E\text{-}I} / K_{E\text{-}I}\Delta T_{E\text{-}I}$
$= 1.325772243311\ m^2$;

High temperature Generator :
$Q_{HTG} = m_{11*}h_{11} + m_{5*}h_5 - m_{4*}h_4 = 1.654536617663$ kW;
$\Delta T_{HTG} = [(t_B - t_{HTG}) - (t_B - t_{11})] / \ln [(t_B - t_{HTG}) / (t_B - t_{11})]$
$\Delta T_{HTG} = 4.342944819033$;
High temperature Genarator Area = A_{HTG}
$= Q_{HTG} / K_{HTG*}\Delta T_{HTG} = 1.691346294560\ m^2$;

Condenser : Assuming t_{22} = 34 ^{0} C, t_{23} = 38 ^{0} C,
$Q_C = m_{14*}h_{14} + m_{13*}h_{13} - m_{15*}h_{15} = 0.6948565091813$ kW;
$\Delta T_C = [(t_C - t_{22}) - (t_C - t_{23})] / \ln [(t_C - t_{22}) / (t_C - t_{23})]$
$= 2.885390081778$;
Condenser area = $A_C = Q_C / K_C\Delta T_C = 1.069131338694\ m^2$;

Absorber : Assuming, t_{24} = 32 ^{0} C, t_{25} = 38 ^{0} C,
$Q_A = m_{19*}h_{19} + m_{10*}h_{10} - m_{1*}h_1 = 2.091597538768$ kW;
$\Delta T_A = [(t_A - t_{24}) - (t_A - t_{25})] / \ln [(t_A - t_{24}) / (t_A - t_{25})]$
$= 3.348663759307$;
Absorber area = $A_A = Q_A / K_A\Delta T_A = 2.77298196107\ m^2$;

Preheater–I :
$Q_{PH\text{-}I} = m_{2*}(h_3 - h_2) = 0.3772252620176$ kW;
$\Delta T_{LTHE\text{-}I}=[(t_8 - t_3) - (t_9 - t_2)] / \ln [(t_8 - t_3) / (t_9 - t_2)]$
$= 35.99530549732$;
Preheater – I area = $A_{PH\text{-}I} = Q_{PH\text{-}I} / K_{PH\text{-}I}\Delta T_{PH\text{-}I}$
$= 0.04652596282671\ m^2$;

Preheater – II :
$Q_{PH\text{-}II} = m_{3*}(h_4 - h_3) = 0.4364708999052$ kW;
$\Delta T_{PH\text{-}II} = [(t_5 - t_4) - (t_6 - t_3)] / \ln [(t_5 - t_4) / (t_6 - t_3)]$
$= 31.10903497992$;
$A_{PH\text{-}II} = Q_{PH\text{-}II} / K_{PH\text{-}II}\Delta T_{PH\text{-}II} = 0.0622886931282\ m^2$;

Precooler :
$Q_{PC} = m_{16*}(h_{16} - h_{17}) = 0.08105933170947$ kW;
$\Delta T_{PC} = [(t_{16} - t_{19}) - (t_{17} - t_{18})] / \ln [(t_{16} - t_{19}) / (t_{17} - t_{18})] = 6.000254105701$;
$A_{PC} = Q_{PC} / K_{PC}\Delta T_{PC} = 0.05997549491652\ m^2$;

IV. RESULTS AND DISCUSSION

Figs. 2. & 3. show the variation of COP against the HT generator pressures for different temperatures keeping the evaporator pressure constant at 210 kPa, condenser and absorber temperatures constant at 40^0 C. Marked improvement in COP is observed when the system is operated at generator pressures between 2800 and 4000 kPa. It is also observed that moderate HT generator pressures and lower temperatures show better operating performance and yields a higher COP. It is also observed that at higher HT generator pressures require higher temperatures and thus the COP curves become almost flat as it tends to increase the average temperatures in the condenser and absorber.

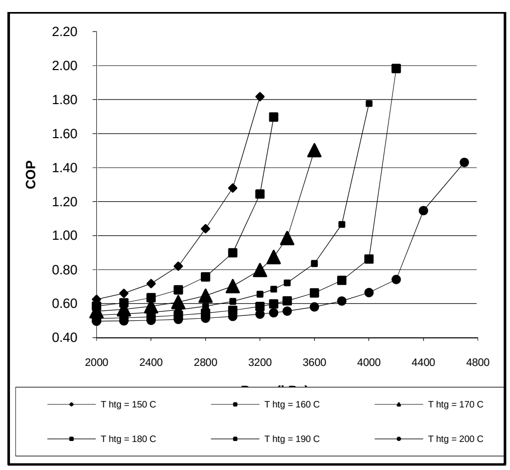

Fig. 2. COP vs. variation of pressure at high temperature Generator

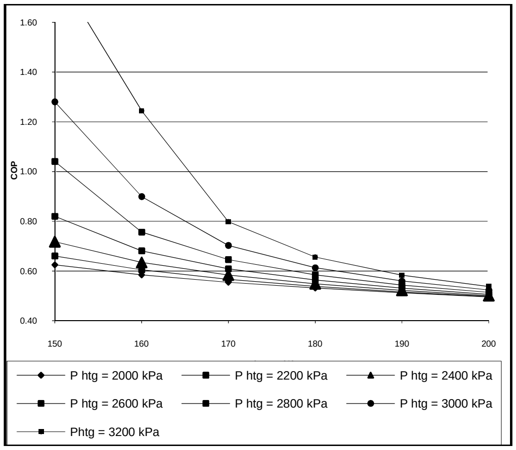

Fig. 3. COP vs. variation of temperature at high temperature Generator

Fig. 4. shows the minimum volume of biogas required for running the cycle for variations in HT generator pressure. The requirement of biogas reduces as the generator pressure increases due to the fact that as the COP increases the quantity of heat input required decreases accordingly. It can also be noted that at low HT generator temperatures requirement of biogas is low indicating generation of low temperatures at the desorber need reduced energy inputs.

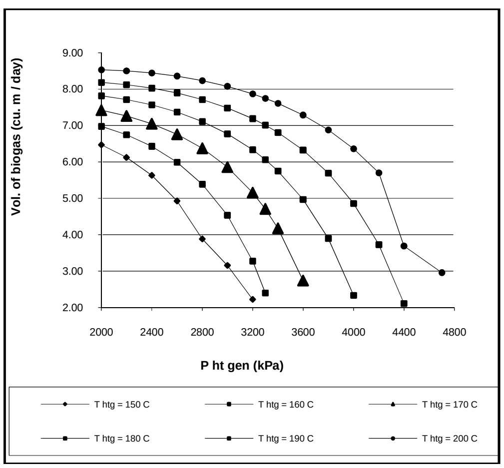

Fig. 4. Variation of pressure at high temperature Generator vs. Vol. of Biogas requirement

Fig. 5. indicates increased solution circulation rates for increase in HT generator pressure and it decreases with the increase in HT generator temperatures. With the increase in solution circulation rates either the cooling capacity increases or the energy input required for the same cooling capacity reduces. Fig. 6. gives information about the pressure variations at the evaporator. The COP reduces with the increase in evaporator pressure drastically and thereafter remains almost constant for any further increase.

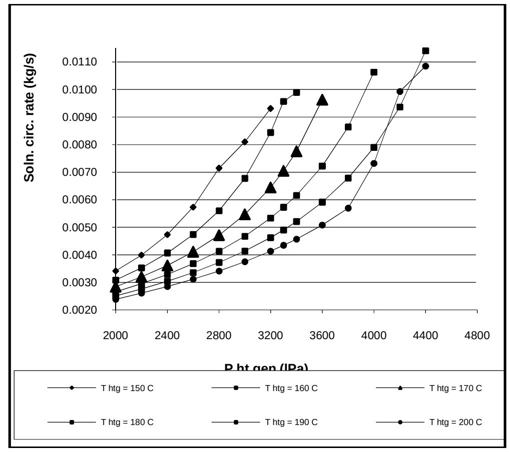

Fig. 5. Variation of pressure at high temperature Generator vs. Solution Circulation rate

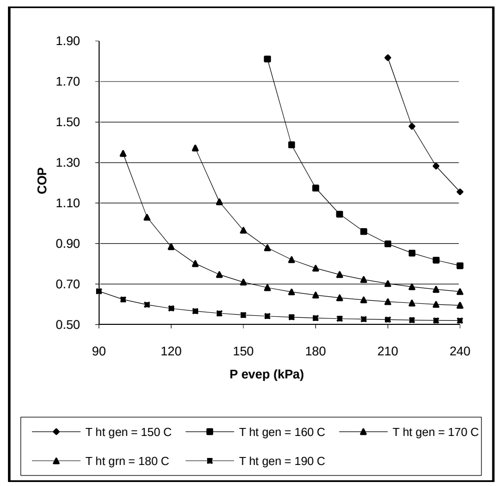

Fig. 6. COP vs. variation of pressure at evaporator

V. CONCLUSION

Ammonia water absorption refrigeration cycle was analyzed, with the thermodynamic properties at strategic points being calculated from the established correlations. The coefficient of performance (COP) of this cycle versus high temperature generator pressure and temperature and evaporator pressure was analyzed and it was noticed that these parameters are important in determining the optimum operating conditions for the system. Moderate HT generator pressures and lower temperatures yield good results and better performance of the system. Similarly lower HT generator temperatures result in reduced requirements of energy inputs and hence low quantities of biogas is sufficient to power the absorption cycle. Better performance of the system is observed when the generator temperatures are kept low for any fixed evaporator pressure. With the variations in pressures and temperatures at the HT generator and evaporator the solution circulation rates are greatly influenced.

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