Modelling the Performance of a Diffusion Absorption Refrigeration System

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Abstract

The diffusion absorption refrigeration cycle was first identified nearly a century ago but until relatively recently its application had been limited to niche cooling applications such as camping refrigerators. The diffusion absorption refrigerator consists of several main components: a generator and bubble pump (a thermal pump), a condenser, evaporator, and an absorber. Unlike a vapour-compression cycle, the cycle operates at a single pressure level and uses three working fluids: a refrigerant, an absorbant and an auxiliary gas that is used in the system to equalize the pressure. Furthermore, where the ubiquitous vapour-compression refrigeration cycle requires work input to drive the compressor, the diffusion absorption refrigeration cycle is a thermally driven process. This characteristic has seen the cycle begin to receive the renewed attention due to the potential for it to operate using solar thermal energy to drive it.

In this work the performance of an ammonia/water/hydrogen diffusion absorption refrigeration cycle is modelled for steady state operating conditions. The results show that the performance of the cycle is dependent on a number of variables including: the temperature and amount of heat added at the generator, the effectiveness of the heat recovery loops and the mass flow of the ammonia. Furthermore, it shows that the performance of the bubble pump plays a significant role in determining the performance of the system and is an area that requires further attention.

1.   Introduction

In order to cope with the growing demand of energy and environmental issues there has been an increase in interest of using thermally driven cooling devices. Among these technologies the diffusion absorption refrigeration (DAR) cycle is a particularly promising technology. Unlike the conventional vapour compression and absorption systems DAR devices do not utilize electrical energy for their operation and can be activated solely by thermal energy sources such as waste heat or by solar collectors.

Von Platen and Munters (1928) first introduced the DAR cycle nearly a century ago; however until recently it had received little attention. Since its invention the research on the DAR cycle has focussed on improving the design of the components (Starace and Pascalis, 2012 and 2013, Munoz and Flores, 2014) and identifying new working fluids (Bourseau and Bugarel, 1986, Ezzine et al, 2010, Long et al, 2013).

However, the performance of the DAR system and especially the cycle components is still to be fully examined. Therefore, the aim of this paper is to develop a model to numerically investigate the performance of a DAR cycle using ammonia (NH_3) as the refrigerant, water (H_2O) as the absorbent and hydrogen (H_2) as an auxiliary inert gas. A sensitivity analysis of the model was carried out in order to investigate the effect of different parameters on the performance of the system.

Nomenclature		abs	Absorber
COP	Coefficient of Performance	bp	Bubble Pump
Eff	Effectiveness	evp	Evaporator
f	Circulation Ratio	GHX	Gas Heat exchanger
h	Enthalpy, kJ/kg	gen	Generator
$\bar{M}$	Molar Mass	i	Inert
$\dot{m}$	Mass flow rate, g/s	l	Liquid
P	Pressure, kPa	$part$	Partial
$\dot{Q}$	Heat transfer, W	rec	Rectifier
T	Temperature, ℃	r	Refrigerant
x	Ammonia mass fraction in solution	SHX	Solution heat exchanger
y	Ammonia mass fraction in vapour	$syst$	System
		v	Vapour
Subscripts			
1, 2, 3…	System's point		

2.   Working Principle of Diffusion Absorption Refrigeration System:

The DAR cycle shown in Figure 1 consists of a generator, bubble pump, rectifier, condenser, evaporator, absorber, gas heat exchanger (GHX) and solution heat exchanger (SHX). In the generator, heat is supplied to the strong solution of ammonia and water in order to drive off the refrigerant and circulate the fluid. As heat is supplied, from a source such as solar thermal collectors, the solution boils and generates vapour bubbles of refrigerant. These bubbles coalesce forming gas plugs (Taylor bubbles) that act as pistons, lifting liquid slugs in the bubble pump and circulating the liquid in the DAR system to a separator. At the separator, the weak solution (low ammonia content solution) leaving the bubble pump flows back to the absorber via a SHX, allowing heat to be recovered from the solution. The vapour leaving the bubble pump continues to the rectifier where any residual water is condensed to join the weak solution. As such pure refrigerant (ammonia) leaves the rectifier and flows to the condenser where it condenses by releasing heat to the atmosphere. The condensed refrigerant leaves the condenser and flows into the evaporator. Since the evaporator is charged with hydrogen, the partial pressure of the ammonia decreases as it enters the evaporator and starts to boil into the hydrogen at low temperatures. The heat of vaporisation of the ammonia produces the cooling capacity of the whole system. The gaseous mixture of ammonia and hydrogen leaves the evaporator due to the density difference associated with the ammonia and hydrogen mixture and the pure hydrogen entering the evaporator. As the gaseous mixture flows to the absorber it passes through the GHX which, like the SHX, recovers heat from the hydrogen returning to the evaporator. When the mixture reaches the absorber, the ammonia is absorbed by the weak solution converting it to a strong solution. The hydrogen, which is non-absorbable, flows back to the evaporator via the GHX. This should reduce the temperature of the hydrogen going into the evaporator and increase the cooling capacity of the evaporator.

Finally, the strong solution exiting the absorber flows back to the generator via the SHX. In the SHX the strong solution gets preheated by the weak solution flowing towards the absorber. The preheating of the strong solution should improve the performance of the system, as less heat is required to regenerate the strong solution.
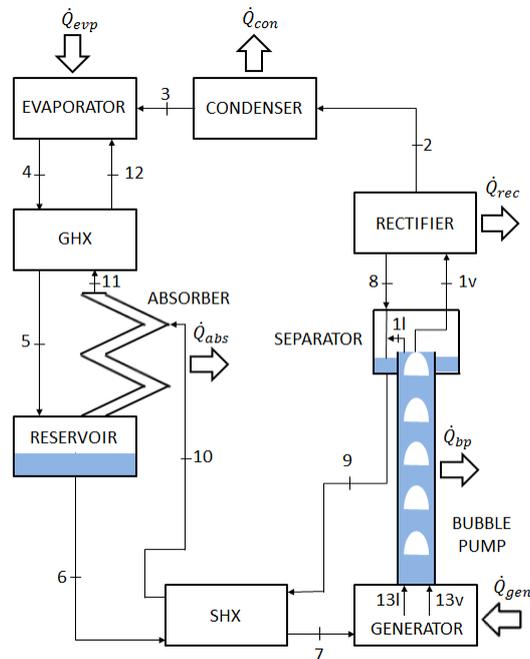
Figure 1. Schematic of the DAR system

2.1. *Thermodynamic Analysis of Diffusion Absorption Refrigeration system:*

In order to understand the factors affecting the performance of the DAR system (Figure 1), a thermodynamic analysis was undertaken. In this analysis the mass and the energy balance equations for each component of the system were described under the following assumptions:

1. The system is assumed to be at steady state.
2. Hydrostatic pressure is neglected and there is no pressure and heat loss in the pipes.
3. The temperature of the liquid and vapour leaving the bubble pump are equal.
4. The vapour leaving the rectifier from point (2) and evaporator from point (4) are saturated.
5. In the case of no temperature drop in the bubble pump, the temperature of the vapour and the liquid leaving the bubble pump are equal, i.e. $T_{gen} = T_{13} = T_1$
6. The condenser and the absorber use the same cooling medium, i.e. $T_3 = T_6$
7. The temperature of the weak solution entering the absorber and the vapours leaving the absorber are equal, i.e. $T_{10} = T_{11}$.

2.1.1. Generator

In the generator the strong solution entering from point (7) is regenerated. The strong solution starts to boil and generates vapour bubbles which, along with the liquid, leaves the generator from point (13). The mass and energy balances for the generator are given by Equations 1-3:

$$\dot{m}_7 = \dot{m}_{13v} + \dot{m}_{13l} \tag{1}$$

$$x_7\dot{m}_7 = y_{13}\dot{m}_{13v} + x_{13}\dot{m}_{13l} \tag{2}$$

$$h_7\dot{m}_7 + \dot{Q}_{gen} = h_{13v}\dot{m}_{13v} + h_{13l}\dot{m}_{13l} \tag{3}$$

2.1.2. Bubble Pump

The vapour bubbles along with the liquid enter the bubble pump from point (13) and leaves from point (1). The mass and energy balances for the bubble pump are given by Equations 4-6:

$$\dot{m}_{13v} + \dot{m}_{13l} = \dot{m}_{1v} + \dot{m}_{1l} \tag{4}$$

$$y_{13}\dot{m}_{13v} + x_{13}\dot{m}_{13l} = y_1\dot{m}_{1v} + x_1\dot{m}_{1l} \tag{5}$$

$$h_{13v}\dot{m}_{13v} + h_{13l}\dot{m}_{13l} = h_{1v}\dot{m}_{1v} + h_{1l}\dot{m}_{1l} + \dot{Q}_{bp} \tag{6}$$

2.1.3. Separator

In the separator the two phases of the working fluid are separated from each other. The vapour moves towards the rectifier from point (1v) while the liquid phase mixes at point (1l) with the liquid coming from the rectifier. As such, the mass and energy balances for the separator are given by Equations 7-9:

$$\dot{m}_8 + \dot{m}_{1l} + \dot{m}_{1v} = \dot{m}_9 + \dot{m}_{1v} \tag{7}$$

$$x_8\dot{m}_8 + x_1\dot{m}_{1l} = x_9\dot{m}_9 \tag{8}$$

$$h_8\dot{m}_8 + h_{1l}\dot{m}_{1l} = h_9\dot{m}_9 \tag{9}$$

2.1.4. Rectifier

In the rectifier the vapour removes the absorbent from the refrigerant while releasing heat. The absorbent removed from the vapour leaves the rectifier from point (8) meaning that the mass and energy balances for the rectifier are given by Equations 10-12:

$$\dot{m}_{1v} = \dot{m}_2 + \dot{m}_8 \tag{10}$$

$$y_1\dot{m}_{1v} = y_2\dot{m}_2 + x_8\dot{m}_8 \tag{11}$$

$$h_{1v}\dot{m}_{1v} = h_2\dot{m}_2 + h_8\dot{m}_8 + \dot{Q}_{rec} \tag{12}$$

2.1.5. Condenser

The refrigerant vapour leaves the rectifier from point (2) and flows to the condenser where it condenses and exits from point (3). The mass fraction and the mass flow rate of the refrigerant at the inlet and exit of the condenser remains the same, i.e. $y_2 = x_3$ and $\dot{m}_2 = \dot{m}_3$. The energy balance across the condenser is given by Equation 13:

$$h_2\dot{m}_2 = h_3\dot{m}_3 + \dot{Q}_{con} \tag{13}$$

2.1.6. Evaporator

In the evaporator the liquid refrigerant from the condenser enters from point (3), and the auxiliary inert gas enters from point (12). The vapour mixture of refrigerant and auxiliary inert gas leaves the evaporator from point (4) meaning that the mass and energy balances for the evaporator can be given by Equations 14-16:

$$\dot{m}_3 + \dot{m}_{12} + \dot{m}_{12i} = \dot{m}_4 + \dot{m}_{4i} \tag{14}$$

$$x_3\dot{m}_3 + y_{12}\dot{m}_{12} = y_4\dot{m}_4 \tag{15}$$

$$h_3\dot{m}_3 + h_{12}\dot{m}_{12} + h_{12i}\dot{m}_{12i} + \dot{Q}_{evp} = h_4\dot{m}_4 + h_{4i}\dot{m}_{4i} \tag{16}$$

Since, in the evaporator the refrigerant continuously evaporates and hence its partial pressure changes meaning that this needs to be accounted for when undertaking the mass and energy balance. The partial pressure at point (4) and (12) can be calculated using Equation 17:

$$P_{part} = \frac{\dot{m}_r/\bar{M}_r}{\dot{m}_r/\bar{M}_r + \dot{m}_i/\bar{M}_i} \, P_{syst} \tag{17}$$

In Equation 17, $\bar{M}$ is the molar mass of the refrigerant-water mixture and can be calculated using Equation 18:

$$\bar{M} = \frac{1}{y/\bar{M}_{NH_3} + (1-y)/\bar{M}_{H_2O}} \tag{18}$$

2.1.7. Gas Heat Exchanger

The mass flow rate and the mass fraction of the constituents at the inlet and exit of the GHX remain the same, i.e. $\dot{m}_4 = \dot{m}_5$, $\dot{m}_{11} = \dot{m}_{12}$, $\dot{m}_{4i} = \dot{m}_{5i}$, $\dot{m}_{11i} = \dot{m}_{12i}$, $y_4 = y_5$ and $y_{11} = y_{12}$ and so the energy balance for the gas heat exchanger is given by Equation 19:

$$h_4\dot{m}_4 + h_{4i}\dot{m}_{4i} + h_{11}\dot{m}_{11} + h_{11i}\dot{m}_{11i} = h_5\dot{m}_5 + h_{5i}\dot{m}_{5i} + h_{12}\dot{m}_{12} + h_{12i}\dot{m}_{12i} \tag{19}$$

The temperature of the inert gas at evaporator entrance i.e. T_{12} is calculated using Equation 20:

$$Eff_{GHX} = (T_{11} - T_{12})/(T_{11} - T_4) \tag{20}$$

2.1.8. Absorber

In the absorber the gaseous mixture of refrigerant and auxiliary inert gas enters from point (5) where the refrigerant is absorbed by the weak solution coming from point (10) to form a strong solution. The auxiliary inert gas flows back to the evaporator from point (11) while the strong solution flows towards the generator from point (6). The mass and energy balances for absorber and reservoir are given by Equations 21-23:

$$\dot{m}_5 + \dot{m}_{5i} + \dot{m}_{10} = \dot{m}_6 + \dot{m}_{11} + \dot{m}_{11i} \tag{21}$$

$$y_5\dot{m}_5 + x_{10}\dot{m}_{10} = x_6\dot{m}_6 + y_{11}\dot{m}_{11} \tag{22}$$

$$h_5\dot{m}_5 + h_{5i}\dot{m}_{5i} + h_{10}\dot{m}_{10} = h_6\dot{m}_6 + h_{11}\dot{m}_{11} + h_{11i}\dot{m}_{11i} + \dot{Q}_{abs} \tag{23}$$

2.1.9. Solution Heat Exchanger

The mass flow rate and the mass fraction of the constituents at the inlet and exit of the GHX remains the same, i.e. $\dot{m}_6 = \dot{m}_7$, $\dot{m}_9 = \dot{m}_{10}$, $x_6 = x_7$ and $x_9 = x_{10}$ so that the energy balance for SHX is given by Equations 24:

$$h_6\dot{m}_6 + h_9\dot{m}_9 = h_7\dot{m}_7 + h_{10}\dot{m}_{10} \tag{24}$$

The weak solution temperature at absorber entrance i.e. T_{10} is calculated using Equation 25:

$$Eff_{SHX} = (T_9 - T_{10})/(T_9 - T_6) \tag{25}$$

2.1.10. Performance parameters

For this work, the performance of the DAR system is evaluated in terms of the Coefficient of Performance (COP) and the circulation ratio, which can be determined by Equations 26 & 27 respectively:

$$COP = \dot{Q}_{evp}/\dot{Q}_{gen} \tag{26}$$

$$f = \dot{m}_7/\dot{m}_2 \tag{27}$$

3.   Results & Discussion:

Now to determine the performance of the system, the model was solved using the Engineering Equation Solver (EES) software package (F-Chart Software, 2014) which contains the thermodynamic properties of various working fluids. In this respect the properties of ammonia vapour and water vapour were calculated from the EES database whereas the properties of the ammonia-water solution at various points of the system were calculated by using the functions given by Patek and Klomfar (1995) as well as by Bourseau and Bugarel (1986). The initial input reference conditions for the simulation model are shown in Table 1.

**Table 1. Input reference conditions for simulation model**

Generator temperature, T_{13} =180 ºC	Strong solution concentration, $x_7 = 0.3$
Condenser temperature, $T_3 = 50$ ºC	Refrigerant vapour concentration, $y_2 = y_{11} = 1$
Evaporator temperature, $T_4 = 10$ ºC	Heat Exchanger Effectiveness, $Eff_{SHX} = Eff_{GHX} = 1$
Heat input, $\dot{Q}_{gen}$=2500 W	Temperature loss in rectifier, $\Delta T_{rec} = 1$ ºC

Figure 2 shows the effect of the generator temperature on COP and f. At high generating temperature the COP tends to decrease. This is due to fact that at high temperatures the enthalphy of the working fluid increases. However to ensure continuity, the mass flow rate decreases resulting in lower flow of refrigerant towards the evaporator and eventually a decrease in the performance. The circulation ratio is high at low generator temperature due to the increase in mass flow rate of the strong solution into the generator. At low temperatures less refrigerant is removed from the strong solution, hence, in order to remove more refrigerant from the strong solution, a larger amount of strong solution is required in the generator.

Now in order to have better performance from the DAR system the value of the circulation ratio should be low, as less strong solution has to be circulated to generate more refrigerant (Long et al., 2013). At a fixed generator temperature an increase in heat input increases the cooling effect as shown in Figure 3. This is due to the fact that higher heat inputs increase the mass flow rate of the refrigerant towards the condenser.
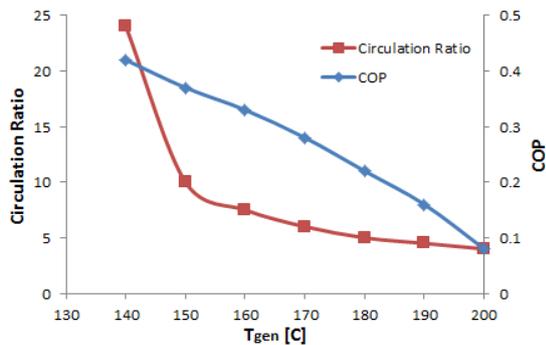
Figure 2. Effect of T_{gen} on COP and f
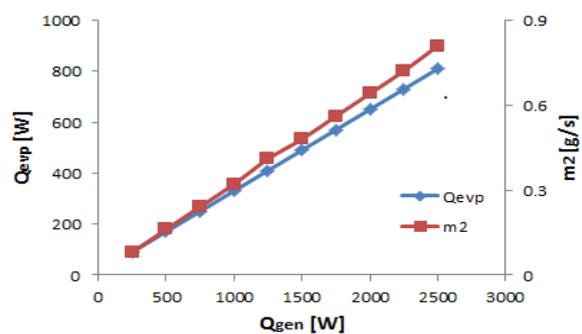
Figure 3. Effect of $\dot{Q}_{gen}$ on $\dot{Q}_{evp}$ and $\dot{m}_2$

Now, in the analysis of the DAR system the pressure of the system was set correspond to the saturation pressure at the condensation temperature, therefore an increase in condensation temperature would increase the system pressure. However as shown in Figure 4, an increase in condensation temperature increases the required generator temperature which results in a drop in COP. This finding agrees with the results reported by Ezzine et al (2010). A change in evaporator temperature slightly influences the performance of the system as shown in Figure 5. However, in order to achieve a low evaporator temperature more inert gas is needed in the evaporator to maintain the mass flow balance, as shown in Figure 6.
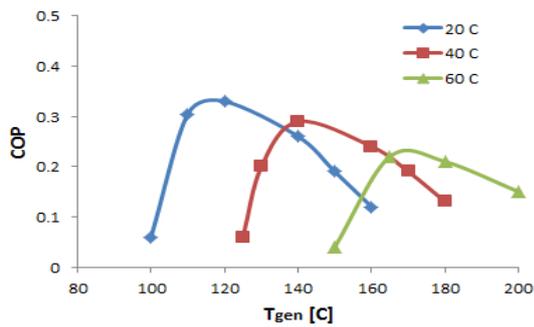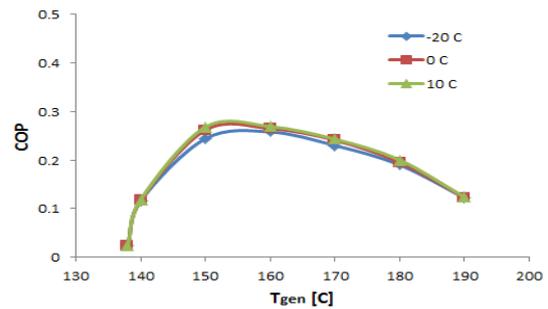
Figure 4. Effect of T_{cond} on COP

Figure 5. Effect of T_{evp} on COP

Counter to this, Figure 7 shows the effect of the ammonia mass faction on the COP of the system. At a fixed generator temperature, the COP rises with an increase in ammonia mass fraction, this is because more ammonia vapour is produced resulting in an increase in $\dot{Q}_{evp}$. Furthermore, with the increase in the ammonia mass fraction, the range of the generator temperature increases. However, the activation temperature in the generator reduces which makes it suitable for operating the DAR system with lower temperature input.
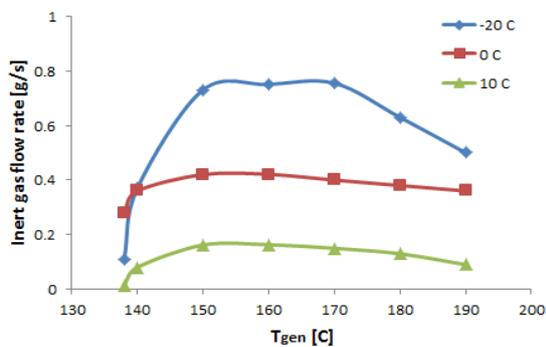

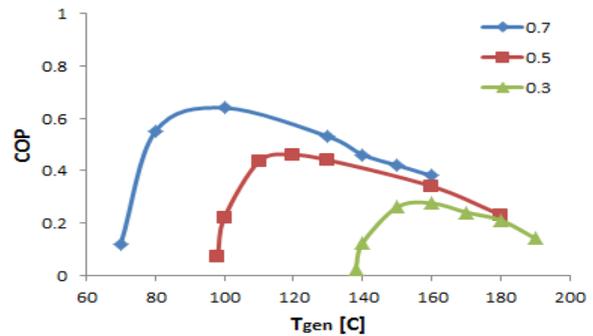**Figure 6. Effect of T_{evp} on mass flow rate of inert gas**

Figure 7. Effect of NH_3 mass fraction on COP

Finally, the cycle includes two heat exchangers for recovering heat within the cycle, in Figures 8 and 9 it can be seen that an increase in the effectiveness of the heat exchangers leads to an improvement in the system performance.
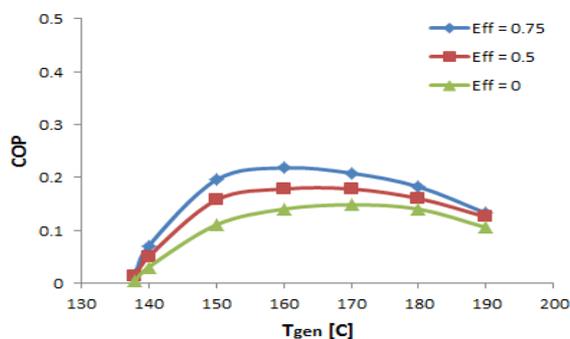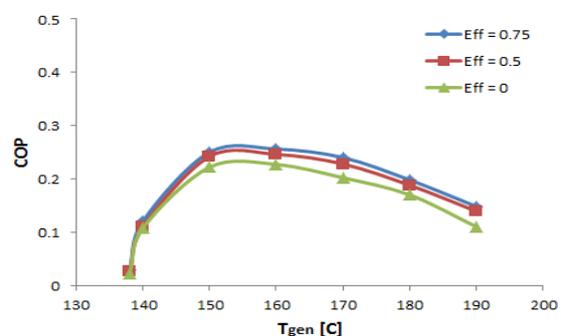

Figure 8. Effect of Eff_{SHX} on COP

Figure 9. Effect of Eff_{GHX} on COP

In the GHX the temperature of the inert gas flowing towards the evaporator is reduced thus improving the performance of the system, because the evaporator does not have to deal with the thermal load of the inert gas. In the SHX the strong solution before entering the generator is pre-heated by the weak solution leaving the separator, hence an increase in the effectiveness of SHX improves the performance of the system as less energy is required to regenerate the strong solution.

The results of the simulation model provide the knowledge for evaluating the performance of the system. It was analysed that the generator temperature has a strong influence on the performance of the system as well as the condenser temperature. Other important influences on the performance of the system are the mass flow rate of the inert gas as well as the ammonia concentration in the strong solution.

4.   Conclusion:

A thermodynamic model for a DAR was developed and a computer simulation was used for the parametric analysis of the system. The simulation showed that the DAR system gives good performance at generator temperatures of 70-160 $^{\circ}$C, which indicates that it could suit solar heat applications. However, at the low generator temperatures, such as may be experienced in solar thermal systems, it may be possible to improve performance by increasing the ammonia mass fraction in the strong solution. Moreover, the use of solution and gas heat exchangers improves the performance of the system and the performance of the system decreases at high condensation temperatures. Finally, if the condenser temperature is low, the performance of the system will drop at high ambient temperatures and, at low evaporator temperatures the inert gas flow in the evaporator needs to be increased.

5.   References:

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