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Effect of Operation Conditions on the Second Law Analysis of a desiccant cooling system

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ABSTRACT: *In order to guage the potential of a desiccant cooling based evaporative air conditioning system, a second law analysis of the systems was examined. Throughout this work, a wide range of working parameters that are regeneration temperature from 70 ℃ to 90℃, volume rate of flow from 1000m^3 to 2000m^3 and ambient air conditions that are relative humidity, close air temperature and wet-bulb temperature is administrated for performance calculation of the system. It's ascertained from results that the physicist COP and so the thermal COP of the system varies the other way around throughout the day. The very best worth of physicist COP and also the thermal COP is concerning 11.98 and 0.074, respectively. Also, the cooling capacity has important impact on the second law potency. The importance of this study shows that the second law analysis will give useful information with relevance the theoretical higher limit of the system performance that cannot be obtained from the first law of thermodynamic analysis alone.*

1. INTRODUCTION

Due to the negative impact of vapor-compression refrigeration systems, novel alternative cooling systems are being tried to develop recently. The negative effects of these systems have more difficulty due to energy spending and using of Chloro-Fluoro-Carbons caused greenhouse effect and creating ozone layer destruction. One of the significant novel cooling systems is desiccant based cooling systems that are environmentally safe, affordable, energy efficient. The operating range of these systems is quite extensive and they can work alone systems, or may work in conjunction with another chiller to increase indoor air quality of all types of buildings [1-4].

A general desiccant assisted cooling system consists of filters, fans, heat exchanger rotary desiccant dehumidifier, indirect/direct evaporative cooler, pumps, controls and ducts [5-9].

In these types of systems, the most used equipment is an evaporative cooling unit which is especially used for dry and hot ambient air conditions, but this component is not efficient for humid air conditions. To eliminate the disadvantages of humid air conditions, desiccants are used to apply evaporative cooling in such ambient air condition. One of the other important units for desiccant cooling system is a desiccant dehumidifier that is crucial component of these type systems [10-12].

Due to the formation of a low vapor pressure on the desiccant surface area, they draw moisture from the air. Because of the high partial pressure of the water in humid air, the water molecules move into the surface of the desiccant that has a low partial pressure. Therefore, the ambient air turns into dry air form with losing his moisture [13]. To understand the performance of air conditioning systems, one of significant thermodynamic analysis method is second law as is evaluated as an efficient method to achieve high efficiency of the system [14-17].

In order to show the real capacity of desiccant assisted cooling system, a number of thermodynamic analyses on the second law are performed in last years. A review of previous work carried out in the field of desiccant assisted cooling system and the methodology adopted to reduce their limitation is summarized as follows:-

Lavan et al. [18] evaluated a desiccant air conditioning system depend on the second law analysis and observed the performance of its individual components. In step with investigation, they evaluated the reversible COP of desiccant system and present equal Sadi Carnot temperatures plan for evaluating reversible COP. As result, they found that the reversible COP and analyses are related to the operative parameters. Once the idea of Lavan et al. [18], the thermodynamic properties of these type of cooling systems have been studied in more depth.

Kodama et al. [19], developed a second law analysis and defined the entropy equilibrium of the system. The results of their study illustrate that the total of the entire take into accounted entropy generation completely expressed the difference between the Carnot COP and the actual COP of the unit.

Van den Bulck et al. [20] presented a second law evaluation on rotary dehumidifiers. They developed the equations for entropy generation for the adiabatic flow of humid air over a solid desiccant dehumidifier. They found that the efficiency of second law of rotary desiccant wheel is a function of working parameters.

Moreover, it stressed the importance of operating conditions supported first and second law of thermodynamic in another study [22]. In this study, the researchers investigated an open desiccant air conditioning system which operates in two modes. In their study, they investigated thermal coefficient of performance (COP) and the cooling capacity according the temperature and relative humidity of ambient air. They additionally evaluated the reversible COP of the system with second law analysis at AIR conditions and located that the reversible COP of the system in recirculation mode is over that of the ventilation mode with 11.98 and 11.15 respectively.

Kanoglu et al. [23] Investigated a desiccant based mostly cooling system that is operated in recirculation mode. They used natural zeolite material for solid rotary dehumidifier. Researchers investigated also energy and exergy approach for evaluating performance of this system. Based on this approach for their system, they derived exergy destruction and exergy efficiency for reversible COP.

La et al. [8] thoroughly investigated for every component of a desiccant cooling system. According to their investigation, they also derived convenient indicator according to thermodynamic laws. The COP concept is very important indicator, illustrating a cooling system performance and this concept mainly is the ratio of useful cooling capacity to needed energy.

In this study, we concentrated on the second law of thermodynamics, which put forward that processes occur in a certain direction and energy has quality as well as quantity. The first law of thermodynamics which assert that energy can be conserve during a process is evaluated also. According to thermodynamic rules, reversible cycles are the most efficient cycles and they consist entirely of reversible processes. The best known reversible cycle is the Carnot cycle and the Carnot COP define the upper limit of this cycle. But reversible cycles cannot be achieved in practice because the irreversibilities associated with each process cannot be eliminated. The thermal efficiency and the thermal coefficient of performance (COP) for any device determine their performances and these parameters are specified on the basis of the first law only. A cycle that is composed entirely of reversible processes served as the model cycle to which the actual cycles can be compared. This idealized model cycle enabled us to determine the theoretical limits of performance for cyclic devices under specified conditions and to examine how the performance of actual devices suffered as a result of irreversibility's.

In this work, an experimental desiccant based evaporative air conditioning system is investigated based on the second law of thermodynamic to determine upper limit. This system has a novel configuration and firstly operation of it is described. The objective of the study is to evaluate thermal COP, reversible COP, second law efficiency and cooling capacity based on operation and ambient air condition.

2. SYSTEM OVERVIEW

A schematic view of the experimentally investigated system is shown in Fig. 1. The configuration presented in this figure uses 100% fresh air. The evaluated system consists mainly of the following components: one solid rotary dehumidifier, three fans, two evaporative coolers that are used as direct, one electrical heater unit, three heat exchangers, two pumps and three air ducts. Further information about working of this system can be obtained in [24]. Three air ducts are used for fresh air regeneration air and waste air. The fresh air duct is used to take fresh air from outside environment and supply it into the investigated room. The waste air channel is designed to discharge the exhaust air from the room by a waste air fan. The main purpose of the third air duct is to remove moisture from fresh air. As described above, one of the most important components is rotary desiccant wheel in this system, due to remove moisture from fresh air.

In Fig. 1, the moisture removal process performed by dehumidifier occurs between state 2 and state 3. For the outlet of desiccant dehumidifier, the air is dehumidified but the temperature of fresh air increases also. In this system, the main cooling process is carried out by evaporative cooler 1 (EC1). For further cooling of the fresh air, the heat exchangers are installed before EC1. According to this installation, heat exchangers have a very important task that is precooling process of the fresh air.

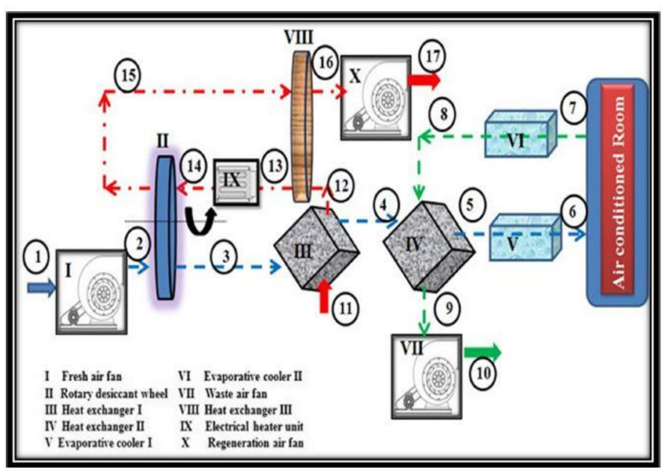

Fig -1: Schematic view of the desiccant cooling system studied.

I. DESIGN METHODOLOGY

The following simplifying assumptions underline the design procedure:

- All the components are perfectly insulated.
- The evaporative cooling processes are accepted isenthalpic.
- The desiccant dehumidification processes are accepted isenthalpic.

The temperature of the dry and warm fresh air is decreased first by the regeneration air in the heat exchanger 1 (HEX1) (process 3–4). After that the second precooling process is fulfilled by heat exchanger 2 (HEX2) in process 4 and process 5 by using exhaust air. Finally, the fresh air is entering into the EC1 (process 5–6) where last cooling process occurs. The fresh air cooled in these processes is supplied into the air conditioning room (state 6). In this system, the evaporative cooler 2 (EC2) is used for precooling of the fresh air by using of HEX2 (process 7–8). In this process, the exhaust air is again cooled until reaches to the saturation point. The increasing of waste air humidity of the exhaust air is not considered a trouble because this humidity is not transmitted to the fresh air. It is only used for precooling fresh air as an indirect evaporative cooling process (process 8–9). After that, it is discharged to atmosphere (process 9–10).

The regeneration air channel uses 100% fresh air, and is installed to regenerate desiccant wheel and to decrease regeneration heat by using two types of heat exchangers that are regenerative type heat exchanger and recuperative type heat exchanger.

The regeneration air stream firstly begins by the heat exchanger 1 in which preheating process occurs, to achieve the suitable temperature for the regeneration of the rotary dehumidifier (process 11–12). For further preheating of regeneration air a rotary heat exchanger is used (process 12–13). Moisture removal is fulfilled by heating of regeneration air. For the heating process (13–14), an electric heater is installed before the entrance of a rotary desiccant dehumidifier, which is used at the end of this air stream, removed moisture from the fresh air stream is and transferred to regeneration air stream (process 14–15) before, finally, being discharged to the environment (process16–17).

II. SECOND LAW ANALYSIS OF THE SYSTEM

The second law of thermodynamics interest with the quality and quantity of energy and this law says that real processes continue in the direction to reduce energy. To achieve the maximum COP of a desiccant air conditioning system, the total system must be fully reversible. The cooling system would be reversible if the heat from the heat source were transferred to a Carnot heat engine, and the work output of this engine is supplied to a Carnot refrigerator to remove heat from the cooled space. In analyzing real performance of these type systems, the actual COP value and Carnot COP(COPc) must be calculated according to the second law of thermodynamics.

In this system, maximum energy consumption takes place in electrical heater unit. Consumed regeneration heat $\dot{Q}_{rh}$ is expresses by:-

$$\dot{Q}_{reg} = \dot{m}_{ra} \cdot c_p \cdot (T_{14} - T_{13}) \quad (1)$$

Cooling capacity of the system $\dot{Q}_{cc}$ is defined by:-

$$\dot{Q}_{CC} = \dot{m}_{fa} \cdot (h_1 - h_6) \quad (2)$$

The regeneration air is taken from outdoor that is fresh air and it is preheated by HEX1 and HEX3. After these processes, measured regeneration temperature is called T13 due to state 13. The last heating operation for regeneration temperature occurs in the electric heater unit. The temperature measured after this unit is called T14 due to state 14. Also; h1 and h6 are enthalpy of state 1 and state 6. These are used for calculating the cooling capacity of the system.

The thermal performance of the system is evaluated based on thermal COP_{th}:-

$$COP_{th} = \frac{\dot{Q}_{cc}}{\dot{E}_{total}} \quad (3)$$

$\dot{E}_{total}$ is the total amount of energy consummated by regeneration heat ($\dot{Q}_{reg}$), fresh, waste and regeneration fans ($\dot{W}_{fan}$), and other electrical components ($\dot{W}_{oth}$):-

$$\dot{E}_{total} = \dot{Q}_{reg} + \dot{W}_{fan} + \dot{W}_{oth} \quad (4)$$

Carnot COP(COPc) that is used for this system depends on ambient air temperature, air conditioned room and regeneration temperature. Also, it is calculated as different from vapor compression refrigeration systems [22]. The Carnot COP of our system is evaluated according to this researcher approach. They found that the COPc can be evaluated as:-

$$COP_C = \left(\frac{T_{reg} - T_{ambient}}{T_{ambient} - T_{room}}\right) \cdot \left(\frac{T_{room}}{T_{reg}}\right) \quad (5)$$

where $T_{ambient}$, T_{room} and T_{reg} are the ambient air temperature, air conditioned room temperature, and the regeneration heat temperature, respectively.

According to our system shown in Fig. 1, T_1 and T_7 are ambient air temperature and cooled room temperature respectively, and the regeneration temperature T_{14} is accepted as the heat source temperature. Based on this approach, Eq. (4) can be derived as:

$$COP_C=\left(\frac{T_{14}-T_1}{T_1-T_7}\right).\left(\frac{T_7}{T_{14}}\right)=\left(\frac{1-\frac{T_1}{T_{14}}}{\frac{T_1}{T_7}-1}\right) \qquad (6)$$

The second law efficiency can be evaluated based on the ratio of the actual COP to the available maximum COPc under the same conditions [25].

$$\text{Efficiency} =\frac{COP}{COP_C} \qquad (7)$$

3. RESULTS AND DISCUSSION

The desiccant based air conditioning designs need new configuration to improve performance of such system. The present work focuses on the second law equations that are applied to the new configuration of a desiccant based evaporative air conditioning system. The system has three air channels that work differently from each other. In these channel, three type air streams that are fresh, waste and regeneration are used to ensure constant volume flow rate with a value of $1000m^3h^{-1}$, $1500m^3h^{-1}$ and $2000m^3h^{-1}$. The ratio of flow rate for all air streams is 1 for all experiments. Also, the effect of regional ambient air conditions that hot and humid ambient air is investigated according to the second law. The ratio of flow rate for all air streams is 1 for all experiments. The regeneration air temperature was regulated to 70°C, 80°C and 90°C for every flow rate in Table 1, Table 2 & Table 3.

We use refrigerant R-134a because R-134a is known as Ecofriendly refrigerant because of the absence of chlorine element. The chlorine element which is present in commonly used Refrigerant attach the ozone layer which is situated in stratosphere. These ozone layer will filters out the harmful ultraviolet radiation where was emitted by the sun. The chlorine element depletion the thickness of ozone layer. Therefor we have to use such as refrigerant which has minimum value of ozone layer depletion potential. R134a is 1,1,1,2-tetrafluoroethane (CH_2FCH_3).

As part of the analysis, Carnot COP, cooling capacity, variations of COP, the second law efficiency with respect to the temperature, relative humidity of ambient air and relations with each other's was investigated in this study.

Table-1: NIST RAFRANCE FLUID PROPERTIES, R-134a: V=$1000m^3$

3: R134a: v = 1000.0 m³/kg

	Temperature (°C)	Pressure (MPa)	Volume (m³/kg)	Enthalpy (kJ/kg)	Phase
1	70.0	0.00002796	1000.0	465.63	Gas
2	72.0	0.00002813	1000.0	467.45	Gas
3	74.0	0.00002829	1000.0	469.29	Gas
4	76.0	0.00002845	1000.0	471.13	Gas
5	78.0	0.00002861	1000.0	472.97	Gas
6	80.0	0.00002878	1000.0	474.83	Gas
7	82.0	0.00002894	1000.0	476.69	Gas
8	84.0	0.00002910	1000.0	478.55	Gas
9	86.0	0.00002927	1000.0	480.43	Gas
10	88.0	0.00002943	1000.0	482.31	Gas
11	90.0	0.00002959	1000.0	484.19	Gas

Table-2: NIST RAFRANCE FLUID PROPERTIES, R-134a: V=$1500m^3$

4: R134a: v = 1500.0 m³/kg

	Temperature (°C)	Pressure (MPa)	Volume (m³/kg)	Enthalpy (kJ/kg)	Phase
1	70.0	0.00001864	1500.0	465.63	Gas
2	72.0	0.00001875	1500.0	467.45	Gas
3	74.0	0.00001886	1500.0	469.29	Gas
4	76.0	0.00001897	1500.0	471.13	Gas
5	78.0	0.00001908	1500.0	472.97	Gas
6	80.0	0.00001919	1500.0	474.83	Gas
7	82.0	0.00001929	1500.0	476.69	Gas
8	84.0	0.00001940	1500.0	478.55	Gas
9	86.0	0.00001951	1500.0	480.43	Gas
10	88.0	0.00001962	1500.0	482.31	Gas
11	90.0	0.00001973	1500.0	484.19	Gas

Table-3:NIST RAFRANCE FLUID PROPERTIES, R-134a: V=$2000m^3$

5: R134a: v = 2000.0 m³/kg

	Temperature (°C)	Pressure (MPa)	Volume (m³/kg)	Enthalpy (kJ/kg)	Phase
1	70.0	0.00001398	2000.0	465.63	Gas
2	72.0	0.00001406	2000.0	467.46	Gas
3	74.0	0.00001414	2000.0	469.29	Gas
4	76.0	0.00001423	2000.0	471.13	Gas
5	78.0	0.00001431	2000.0	472.97	Gas
6	80.0	0.00001439	2000.0	474.83	Gas
7	82.0	0.00001447	2000.0	476.69	Gas
8	84.0	0.00001455	2000.0	478.55	Gas
9	86.0	0.00001463	2000.0	480.43	Gas
10	88.0	0.00001471	2000.0	482.31	Gas
11	90.0	0.00001480	2000.0	484.19	Gas

Table-4: Result of Review paper Data

Volume / Temp. °C	$2000m^3$	$3000m^3$	$4000m^3$
90°C – 100°C	11.83	11.57	11.68
100°C – 110°C	11.91	11.97	11.89

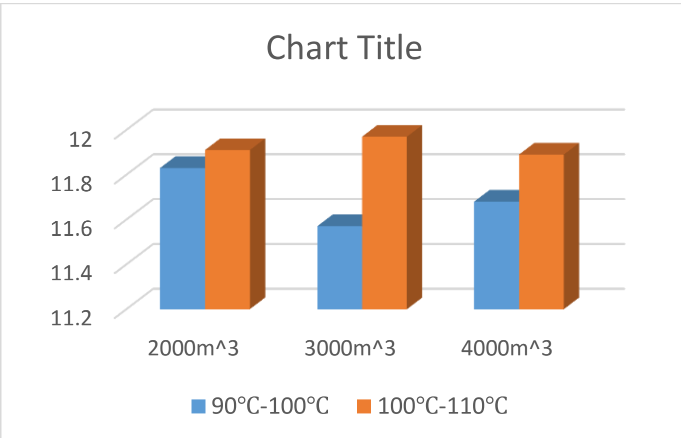

Chart -1: Result of Review paper Data

Table-5: Result of Implemented Data

Volume / Temp. °C	$1000m^3$	$1500m^3$	$2000m^3$
70°C – 80°C	11.23	11.15	11.22
80°C – 90°C	11.56	11.56	11.41

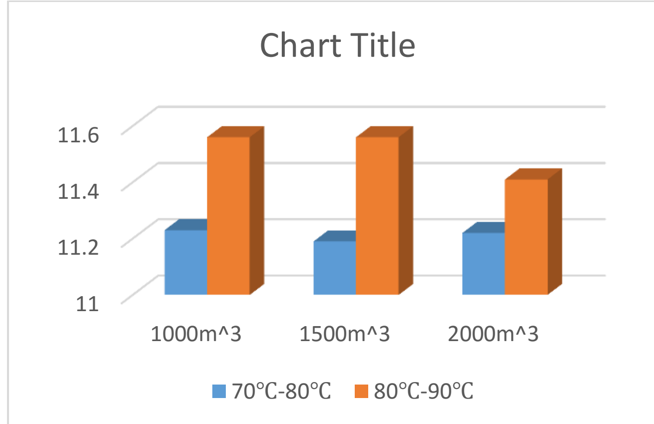

Chart -2: Result of Implemented Data

Table-6: Comparison between Implemented data and Review paper data

Temp. / vol.	$1000m^3$	$1500m^3$	$2000m^3$	$3000m^3$	$4000m^3$
70°C-80°C	11.23	**11.15**	11.22	11.27	11.32
80°C-90°C	11.56	11.56	11.41	11.53	11.44
90°C-100°C	11.75	11.80	11.83	11.57	11.68
100°C-110°C	**11.98**	11.97	11.91	11.97	11.89
AVERAGE	11.630	11.620	11.592	11.585	11.582

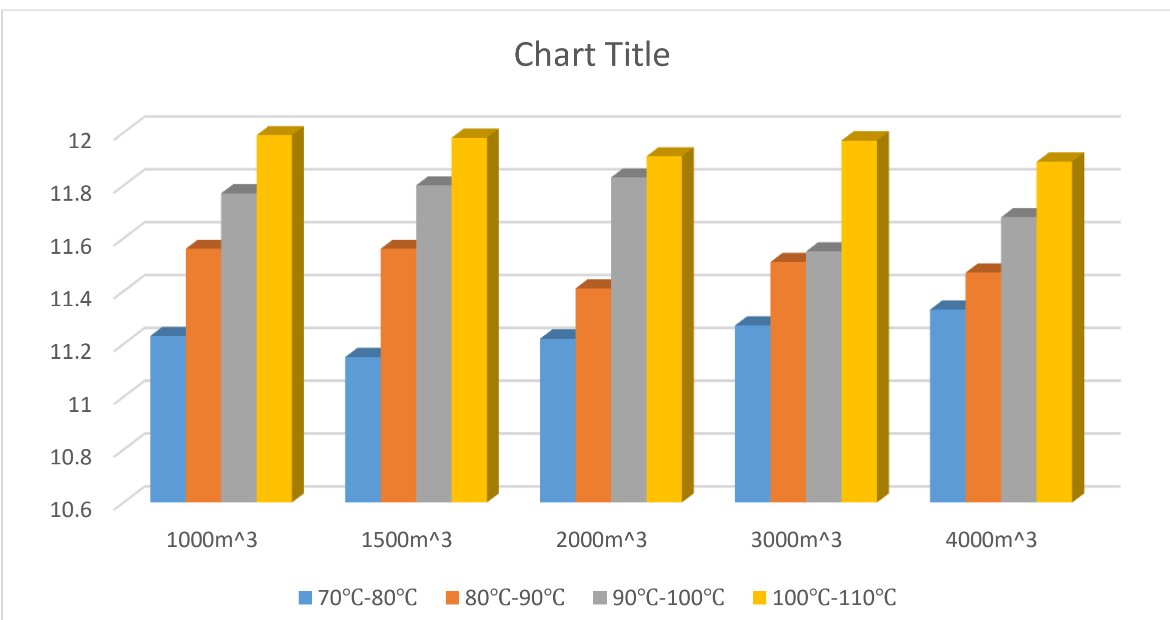

Chart -3: Comparison between Implemented data and Review paper data

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$$\text{Percent (\%) change} = \frac{COP_{max} - COP_{min}}{COP_{min}}$$

$$= \frac{11.98 - 11.15}{11.15} * 100$$

$$= 7.44 \text{ (rise)}$$

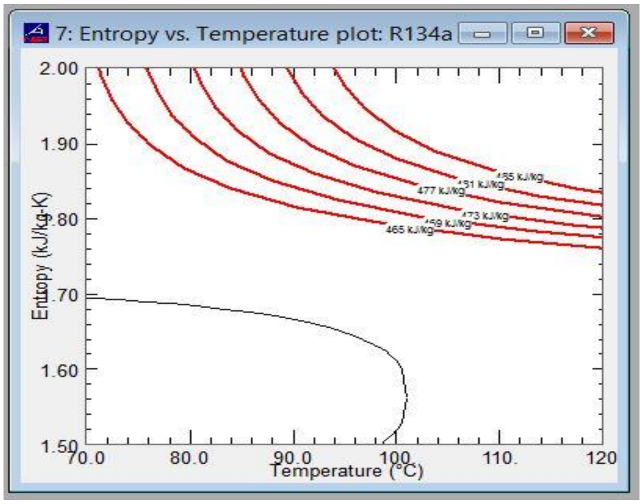

Fig -2: R-134a refrigerant T-s graph

Area under the curve on T-s diagram represent heat interaction for a reversible process. All reversible clockwise cycle on T-s diagram are work producing cycle and vice versa. As a gas expands in a system, entropy increases because it is unavailable energy. For reversible (ideal) processes, the area under the T-s curve represent net heat interaction which is also equal work interaction. Area under T-s diagram has no meaning for an irreversible process. T-s diagram is used to check the changes to temperature and specific entropy throughout a thermodynamic process or cycle.

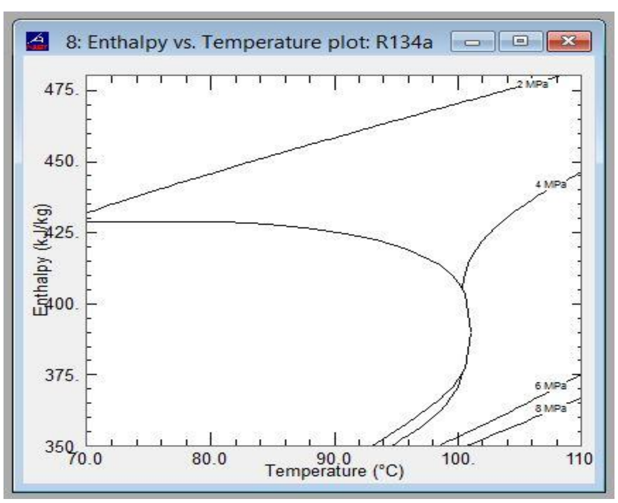

Fig -3: R-134a refrigerant T-h graph

T-h diagram can be constructed for any pure substance. The region between the saturated liquid line & the saturated vapor line represents the area of two phases existing as the same time. The vertical distance between the saturated liquid to saturated vapour, represents the latent heat of vaporization. If an amount of the heat was added to the saturated liquid line, it is equal to the latent heat of vaporization. Then the water would change phase from a saturated liquid to a saturated vapour, while maintaining a constant temperature. As shown figure 3 operation outside the saturation lines results in a sub-cooled liquid or superheated steam.

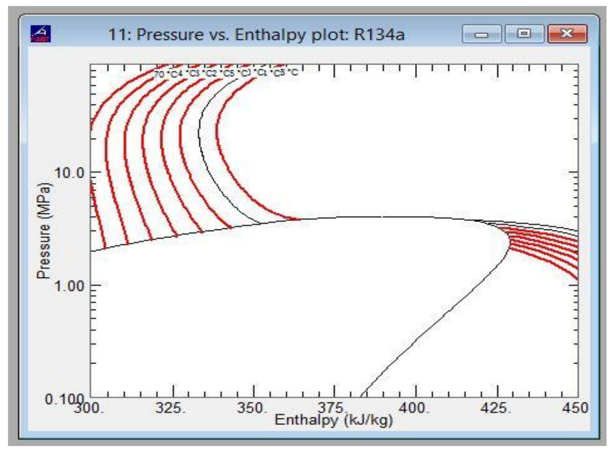

Fig -4: R-134a refrigerant p-h graph

Area under the curve on p-h diagram is a figure 4 with a vertical axis of absolute pressure and a horizontal axis of specific enthalpy. It is an essential diagram used frequently for a performance calculation of a refrigerating machine. Area under curve on p-h diagram is made respectively for R-134a refrigerant. All temperature lines come into the curve at saturated liquid and passes through out at critical point of curve.

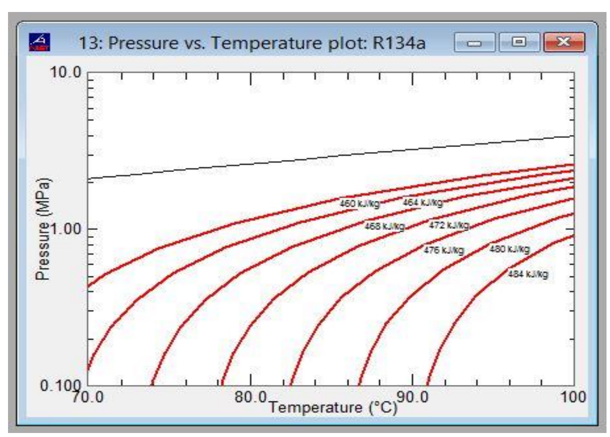

Fig -5: R-134a refrigerant P-T graph

The simplest phase diagrams are P-T diagrams of a single simple substance. The phase diagram shows in P-T space, the lines of equilibrium or phase boundaries between the three phases of solid, liquid, and gas. Area under the curve on P-T diagram is a vertical axis of pressure and a horizontal axis of temperature. The specific enthalpy line is increases under the saturation line curve on P-T diagram. Triple points are points on phase diagrams where lines of equilibrium intersect and three different phases can coexist. For example, the water phase diagram has a triple point corresponding to the single temperature and pressure at which solid, liquid, and gaseous water can coexist in a stable equilibrium The open spaces, where the free energy is analytic, correspond to single phase regions. Single phase regions are separated by lines of non-analytical behavior, where phase transitions occur which are called phase boundaries.

4. CONCLUSION

After using the refrigerant R-134a in an abbreviated version of NIST Refprop software the main results can be listed as follows:

The thermal COP (COPth) increases with increasing ambient air temperature. According to operating parameters, the maximum COPth occurs at the highest ambient air temperature in different flow rate. Daily average COPc increases when ambient air temperature increases. It is observed that flow rate of air has considerable effect on COPc. When the flow rate of air is increased, average COPc decreases at the same time. According to operating parameters in this system, the daily average COPc varies between 11.15 and 11.98 with flow rate of $2000m^3h^{-1}$ and $1000m^3h^{-1}$ respectively.

One of important operation condition is flow rate of air that affects the second law efficiency. It is seen that increasing of flow rate, increases also the second law efficiency. Calculations show that the thermal coefficient of performance COPth, the COPc, the second law efficiency and the cooling capacity strongly depends on the ambient air conditions.

REFERENCES

[1] K.F. Fong, T.T. Chow, C.K. Lee, Z. Lin, L.S. Chan, Advancement of solar desiccant cooling system for building use in subtropical Hong Kong, Energy Build. 42 (2010) 2386–2399.

[2] D.B. Jani, M. Mishra, P.K. Sahoo, Performance studies of hybrid solid desiccantvapor compression air-conditioning system for hot and humid climates,Energy Build. 102 (2015) 284–292.

[3] H. Lee, X. Lin, Y. Hwang, R. Radermacher, Performance investigation on solid desiccant assisted mobile air conditioning system, Appl. Therm. Eng. 103 (2016) 1370–1380.

[4] X. She, Y. Yin, Y. Dong, X. Zhang, Investigation on air flow patterns of evaporative cooling and dehumidification process for a hybrid refrigeration system, Appl. Therm. Eng. 95 (2016) 79–94.

[5] G. Angrisani, C. Roselli, M.I. Sasso, Experimental assessment of the energy performance of a hybrid desiccant cooling system and comparison with other air-conditioning technologies, Appl. Energy 138 (2015) 533–545.

[6] A. Khalid, M. Mahmood, M. Asif, T. Muneer, Solar assisted, pre-cooled hybrid desiccant cooling system for Pakistan, Renew. Energy 34 (2009) 151–157.

[7] J. Jeong, S. Yamaguchi, K. Saito, S. Kawai, Performance analysis of four-partition desiccant wheel and hybrid dehumidification air-conditioning system, Int. J. Refrig. 33 (2010) 496–509.

[8] D. La, Y. Li, Y. Dai, T. Ge, R. Wang, Effect of irreversible processes on the thermodynamic performance of open-cycle desiccant cooling cycles, Energy Convers. Manage. 67 (2013) 44–56.

[9] D.Q. Zeng, H. Li, Y.J. Dai, A.X. Xie, Numerical analysis and optimization of a solar hybrid one-rotor two-stage desiccant cooling and heating system, Appl. Therm. Eng. 73 (1) (2014) 474–483.

[10] N. Wang, J. Zhang, X. Xia, Desiccant wheel thermal performance modeling for indoor humidity optimal control, Appl. Energy 112 (2013) 999–1005.

[11] C.R. Ruivo, G. Angrisani, F. Minichiello, Influence of the rotation speed on the effectiveness parameters of a desiccant wheel: an assessment using experimental data and manufacturer software, Renew. Energy 76 (2015) 484–493.

[12] N. Giannetti, A. Rocchetti, K. Saito, S. Yamaguchi, Entropy parameters for desiccant wheel design, Appl. Therm. Eng. 75 (2015) 826–838.

[13] L.G. Harriman, The Dehumidification Handbook, second ed., Munters Cargocaire, Amesbury, 1990, pp. 77–85.

[14] B. Tashtoush, M. Tahat, A.A. Hayajneh, V.A. Mazur, D. Probert, Thermodynamic behavior of an air conditioning

system employing combined evaporativewater and air coolers, Appl. Energy 70 (2001) 305–319.

[15] J. Zhu, W. Chen, Energy and exergy performance analysis of a marine rotary desiccant air-conditioning system based on orthogonal experiment, Energy 77 (2014) 953–962.

[16] E. Elgendy, A. Mostafa, M. Fatouh, Performance enhancement of a desiccant evaporative cooling system using direct/indirect evaporative cooler, Int. J. Refrig. 51 (2015) 77–87.

[17] I. Uckan, T. Yilmaz, E. Hurdogan, O. Buyukalaca, Exergy analysis of a novel configuration of desiccant based evaporative air conditioning system, Energy Convers. Manage. 84 (2014) 524–532.

[18] Z. Lavan, J.B. Monnier, W.M. Worek, Second law analysis of desiccant cooling systems, ASME J. Solar. Energy Eng. 104 (1982) 229–236.

[19] A. Kodama, M. Goto, T. Hirose, T. Kuma, Experimental study of the optimal operation for a honeycomb rotor adsorber operated with thermal swing, J. Chem. Eng. Jpn. 26 (1993) 530–535.

[20] E. Van den Bulck, S.A. Klein, J.W. Mitchell, Second law analysis of solid desiccant rotary dehumidifiers, J. Sol. Energy Eng. 110 (1998) 2–9.

[21] C.M. Shen, W.M. Worek, The second-law analysis of a recirculation cycle desiccant cooling system: cosorption of water vapor and carbon dioxide, Atmos. Environ. 30 (1996) 1429–1435.

[22] M. Kanoglu, M.O. Carpınlıoglu, M. Yildirim, Energy and exergy analyses of an experimental open-cycle desiccant cooling system, Appl. Therm. Eng. 24 (2004) 919–932.

[23] M. Kanoglu, A. Bolatturk, N. Altuntop, Effect of ambient conditions on the first and second law performance of an open desiccant cooling process, Renew. Energy 32 (2007) 931–946.

[24] I. Uckan, T. Yilmaz, E. Hurdogan, O. Buyukalaca, Experimental investigation of a novel configuration of desiccant based evaporative air conditioning system, Energy Convers. Manage. 65 (2013) 606–61.