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Performance Analysis of a Combined Heat Pump-Dehumidifying System

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Abstract. During the past few decades, all Gulf States have experienced a rapid growth in living standard, which was naturally associated with rapid increase in demand for electricity and potable water. Due to changes of living style, large portion of the electric power was consumed for air conditioning (A/C). Lack of natural water resources and striving needs for water made desalination the main technology to provide potable water. In an effort to search for a novel method for producing fresh water, the present paper investigates a combination of a heat pump (HP) with a dehumidification process to extract water from the atmospheric air. The unit supplies 1.586 m^3/s air to an office building (250 m^2 area) by the sea side in Jeddah, Kingdom of Saudi Arabia (KSA) where the average ambient temperature is 34^oC and relative humidity of 71%. The unit has two series evaporator-coils to maximize the water condensation process by reducing the air exit temperature to 8 ^oC.

The performance of the combined HP-Dehumidifying unit has been investigated for one year employing measured climatic conditions for representative days of the months. The maximum water production was in September (2.23 m^3/day) and the least was in January (0.618 m^3/day). The power consumed by the unit was determined and the mean specific power rate was 0.42 kWh/liter of fresh water. The pilot unit was purchased at a relatively high cost that raises the fixed charges rate, the estimated maximum water cost was 23 \$/m^3 for electricity rate of 0.045 \$/kWh. A more realistic estimate based on sharing the expenses between the A/C and water production processes reduced the water cost to 13.5 \$/m^3 (50.6 Saudi Riyal/m^3). The combined A/C- Dehumidifying unit can be efficient for resort areas of high humidity, low water demand and need for air conditioning.

1. Introduction

Many countries suffer from shortage of natural fresh water. These countries will in the future face a severe problem to provide the required needs. Due to the continuous increase in population and the augmented use of water for agriculture and industry as well as the urbanization of many areas the demand for water will continue to rise. At present, many of the countries rely on desalination technology to provide the necessary water demands. For the near future (20-30 years), sea water desalination seems to be the only proven reliable technology that can be used to secure water supplies. Saudi Arabia is one of these countries; it is the largest producer of desalted water (17.4% of the total world production of 40 million m^3/day with annual growth rate of 12%). The Middle East is the world' foremost area that depends on desalination with 17.8 million m^3/day installed capacity [1]. In Desalting processes the energy consumed for both thermally or membrane desalinating processes is primarily supplied by burning fuel oil, which in turn is the main source of Gulf nations' wealth. It is anticipated that the demand for fresh water will continue to increase forcing high rates of oil consumption; a depleting energy source. The specific fuel consumption rate per m^3 of desalted water depends on the desalination method and the form of energy supplied to the salt removal process. In all cases burning fuel contributes directly to the environmental pollution problems. However, the major desalination units have been in use for the past 40 years, still the cost of water is far from being economic. The water cost from natural resources is in the order of 0.4 to 0.5 \$/m^3 in Canada and USA respectively, while the most recent cost of desalted water in Kuwait is \$ 4.71 /m^3. The present desalting plants capacity and water consumption rate (0.307 m^3/capita-day in KSA) can sustain water needs for few years to come but searching for new novel technology or improving the present methods is underway in many universities and research centers. A comprehensive review on the innovative potable water resources can be found in [2]. Atmospheric water vapor processing (AWVP) is a promising technology, in which the atmospheric water vapor is condensed and collected [3]. However, water yield is small compared to the current technology; it is an option to be investigated for low demand sites. Three different approaches are in consideration; condensation on cold surfaces, absorption of the water vapor by a desiccant and then release the vapor in a regeneration process, and use of a tall tower chimney structure pushing the humid air to cold high altitude zone where condensation takes place [3].

Water desalination using heat pumps has first been considered for the arid areas north of Mexico, Siqueiros and Holland [4] suggesting desalination units assisted by heat pumps (mechanical vapor compression and/or absorption machines) the results showed a feasible economic potential as compared to reverse osmosis plants. Slesarenko [5] suggested incorporating heat pumps with desalination plants. Two schemes were investigated; coupling of R-12 HP and steam-water compression HP for boosting the productivity of multi-stage flash desalter. Al-Juwayhel *et al.* [6] analyzed a combined vapor compression heat pump with a single effect evaporator desalination system.

Heat pumps have been used for water extraction in a direct approach by condensation of the atmospheric water vapor on the cold evaporators' surface. Gao *et al.* [7] analyzed the performance of a unit in which the air was humidified in an alveolate section (Seldek) and two condensers were then used for dehumidification process. A pilot test unit of 60 kg/day fresh water yield at a compressor power of 500 W was tested. Yuan *et al.* [8] considered an integrative unit for air-conditioning and desalination on basis of direct humidification-dehumidification process. Experimental results showed increase in the fresh water production with the brine temperature and flow rate. The performance of the combined unit was evaluated by introducing a primary energy ratio (PER) measured for different operation modes. For desalting and air-conditioning the PERs were 6 and 3.6 respectively. The maximum recorded water yield was 2.4 kg/h in a two stage-humidifier and 45^{o}C brine feed temperature. There is abundant literature on analysis of the dehumidification process on heat pumps' evaporator coils [9,10,11,12]. These studies were mainly concerned with the finned surface performance and humidity removal patterns rather than water production.

This study is motivated after the pioneer work of Faqih [13] who patented the A/C-Water production coupling concept in (2002). He operated a 3 m^3/day unit in Jeddah (Latitude 22^o N and 49 E longitude) Saudi Arabia. The measured specific productivity was 1.67 lit/kWh. The low cost of electricity in Saudi Arabia (0.05-0.07 SR/kWh) may provide reasonable economic feasibility for combined heat pumps system. The objective of the present study is to estimate the water yield of a system used for both A/C and water extraction from the atmosphere under actual climatic conditions. Conservation equations are used below to calculate the water yield.

2. Analysis

Schematic of the considered combined system is shown in Fig. 1. Fresh moist air from the atmosphere at T_{ai}, and humidity ratio ω_i passes over the evaporator finned coil of a HP and leaves at T_{ao} and ω_o. For air mass flow rates $\dot{m}_a$ the heat transfer rate from the air to the refrigerant at saturation temperature T_r, McQuiston [14] is;

$$\dot{Q}_a = \dot{m}_a\left[\left(h_{ai} - h_{ao}\right) - \left(\omega_i - \omega_o\right)h_{fw}\right] \quad (1)$$

Heat transfers to the refrigerant flowing at $\dot{m}_r$ inside the tubes can be written in the enthalpy potential method as;

$$\dot{Q}_a = \dot{m}_r\left(h_{r3} - h_{r4}\right) = U_o A_s \Delta\bar{h}_m \quad (2)$$

The subscripts 3 and 4 indicate the refrigerant states before and after the evaporator coil, Fig. 1, while $\Delta\bar{h}_m$ is the logarithmic mean enthalpy difference [14].

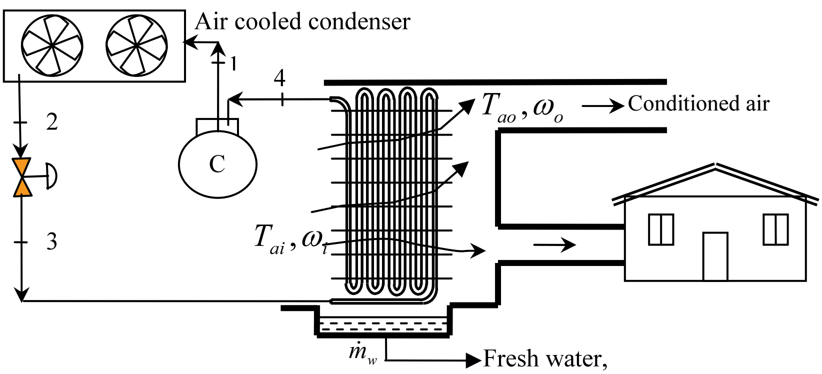

Fig. 1. Heat pump with dehumidifying evaporator for fresh water extraction.

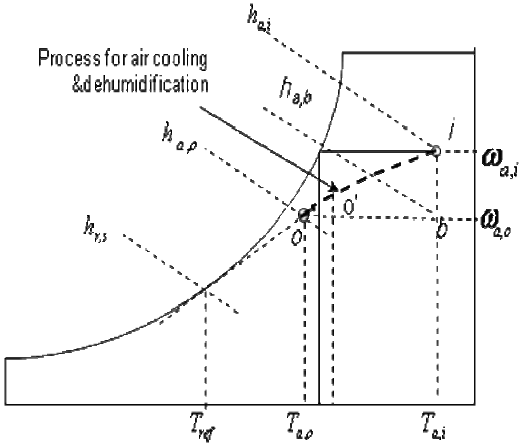

Fig. 2. Psychrometric presentation of the water extraction process from atmospheric air.

In Eq. 3 the term U_o presents the overall heat transfer coefficient for the coil geometry that includes the refrigerant side heat transfer coefficient h_i, the finned tube segments, fin efficiency and the mass transfer process.

$$U_o = \left[\frac{1}{h_i}\left(\frac{b_r' A_o}{A_{pi}}\right) + \frac{b_{wm}\left(1-\eta_{fw}\right)}{h_{ow}\left(\frac{A_{po}}{A_f}+\eta_{fw}\right)} + \frac{b_{wm}}{h_{ow}} \right]^{-1} \tag{3}$$

Equation (3) includes the individual modified thermal resistances of refrigerant ($1/h_i$) (the first term of the denominator), the fin and tube dimensions and the air side thermal resistance$\left(1/h_{ow}\right)$. Details of the different terms in Eq. 3 can be found in [14 &15].

The fresh water collected during the cooling dehumidification process, m_w, depends on the initial and final air states (*i* and *o,* Fig. 2) and the period of operation τ, therefore, the daily water yield may be obtained for a period τ hours/day as;

$$m_w = \int_0^{\tau} \dot{m}_a \left(\omega_i - \omega_o\right) dt \tag{4}$$

However Eq. 4 is a direct relation, it indicates dependency of the system water production on three parameters; the air inlet state that varies with time, ($\omega_i = \omega_i(t)$), the air state at the evaporator exit ω_o and T_o, which depends on the cooling coil capacity and the overall heat transfer coefficient, Eq. 3. The later equation includes a number of parameters that depend on the finned coil geometry, material and surface wetting condition. The third parameter in Eq. 4 is the air mass rate, which can be controlled according to the demand of the conditioned zones. The humidity ratio is time dependent and varies with the relative humidity (RH %), ambient temperature, wind speed and solar insolation. Figure 3 show the hourly dependency of RH and ambient temperature on time for selected days that represent the average of the month in Jeddah, Saudi Arabia (Latitude 22^o N and 49^o E longitude).

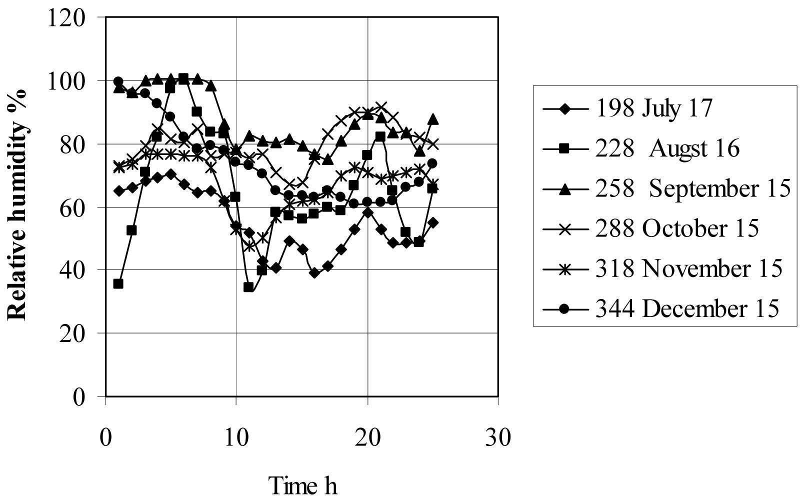

Fig. 3(a). Relative humidity variation for the city of Jeddah, KSA for 6 months.

The variation in Fig. 3(a) indicates clearly that the yield of the system is irregular time dependent. For maximum water yield the air flow rate could be increased but this in turn will increase the evaporator exit temperature T_o and state (o) moves to (o'), Fig. 2. The comfort status inside the air conditioned building will be affected.

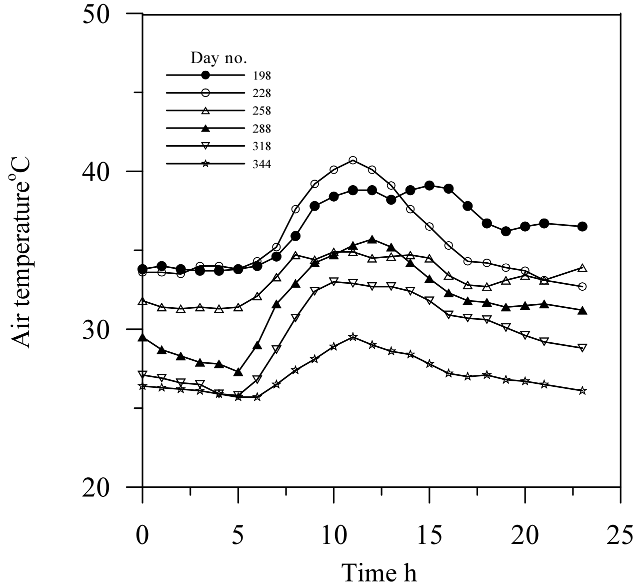

Fig. 3 (b). Dry bulb temperature variation for the city of Jeddah, KSA for 6 months.

Using a HP for both air conditioning and water extraction seems a direct and attractive approach but there are two cases. If the HP is to cover the cooling load the water becomes a by-product and the air state at the evaporator exit is fixed. In this case and in reference to Eq. 4 the extracted water depends only on the air intake state. To avoid the lengthy computation of Eq. 4, it is suggested here to employ the monthly mean Dry bulb temperature (DBT) and RH values of the selected dates for every month. These representative days were based on averaging climatic data for each month [15] (shown in Fig. 3(a) and 3(b)). The lowest RH is 39% and the maximum reaches 100% during morning hours in September. From continuous hourly measurements of the DBT and RH% the annual averages are 34°C and 71% respectively.

In the second case the HP is designed with evaporator surface area and configurations to maximize the water extraction process (low T_r) and the cooling load is of secondary importance. For such case the conditions at the evaporator inlet are known but the exit conditions could be determined by employing either the total enthalpy method of McQuiston *et al.* [14], the NTU- ε method, ASHRAE [16] or the recent fin efficiency weighting method of Habeebullah [17].

3. Results and Discussion

A Heat Pump- Dehumidifying pilot unit has been built by the sea side in Jeddah, KSA with two series DX custom coils, I and II, 44.7 kWh capacity each. The design conditions assumed that the air enters the first evaporator coil at DBT 43.33°C and 27.78 °C WBT; (at this state the relative humidity is 34% and the moisture content is 0.016 kg_w/kg_a) and leaves at 27.49 DBT and 22.48 WBT. The cold air from coil # I pass over the second coil where most of the moisture removal takes place. The temperature drop across the first stage is 15.55°C while that on coil # II is 11.8°C but most of the moisture removal occurs on coil # II.

For the unit features (shown in Table 1) and the mean climatic conditions (Fig. 3(a) and 3(b)) the water production from the unit for air flow rate of 1.586 m^3/s is calculated employing moist air properties from the Engineering Equations Solver, EES [18]. For the selected conservative design conditions (43°C and RH = 39%) the daily yield of coil I is 102.3 *l* and 704.4 *l* from the second coil.

Table 1. Parameters of the HP-Desalination pilot unit.

Stage I			Stage II		
Capacity	42.76	kWh	Capacity	44.7	kWh
Sensible capacity	30.47	kWh	Sensible capacity	22.69	kWh
HP working fluid	R-143		HP working fluid	R-143	
Sat suction temperature	8.54	°C	Sat suction temperature	9.9	°C
Air side data			**Air side data**		
Air flow rate	1585.58	*l*/s	Air flow rate	1585.58	*l*/s
Face velocity	2.13	m/s	Face velocity	2.13	m/s
On-coil DBT	43.33	deg C	On-coil DBT	27.5	deg C
On-coil WBT	27.78	deg C	On-coil WBT	22.5	deg C
Off-coil DBT	27.49	deg C	Off-coil DBT	15.7	deg C
Off-coil WBT	22.48	deg C	Off-coil WBT	15.34	deg C
Sensible heat ratio	0.71		Sensible heat ratio	0.51	
By-Pass factor	0.287		By-Pass factor	0.054	
Δp through coil	42.83	Pa	Δp through coil	99.95	Pa
Coil # I			**Coil # II**		
No of Rows	3		No of Rows	7	
Front size	610 x 1219	mm^2			
Fin spacing	2.5	mm			
Fins/in	10				
Tube outer diameter	9.52	mm			
Material of fins	Copper				
Thermal conductivity of fins and tubes k_f, k	396	W/m K			

Variation of the yield for constant exit conditions (DBT = 15.7°C and WBT of 15.34°C) for the representative days of the year is shown in Fig. 4.

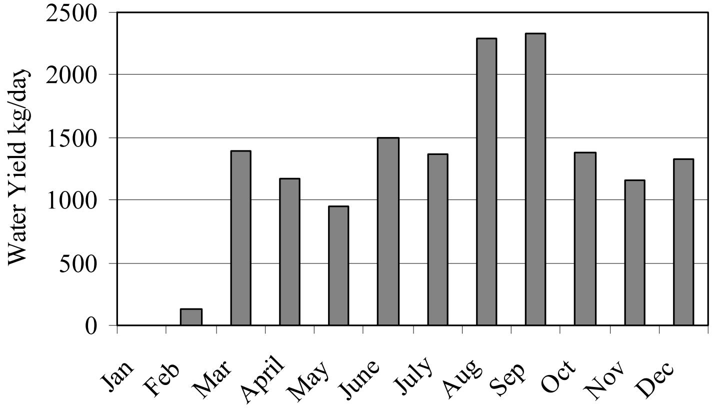

Fig. 4. Daily fresh water yield for representative days of the year in case of fixed off coils conditions (15°C DBT and saturation).

For January it is seen that there is no water production where the DBT is 22.62°C and the humidity ratio is nearly 61%. For this condition only one evaporator coil is used. However the humidity ratio is high but the DBT is low and operation of the two evaporator units does not produce an appreciable dehumidification. By the month of March (DBT = 27 °C and RH = 87%) the unit productivity reaches 1400 *l*/day. The maximum water production is reported in September to be 2330 *l*/day.

To boost the fresh water productivity the HP can be operated at temperatures below the 15 °C outlet air temperature. Table 2 shows the exit air DB and WB temperatures along with the amount of the condensate water. The power consumed by the compressor and evaporator and condenser fans are also given.

Comparing the results in Fig. 4 and that obtained in Table 2 shows that the water yield increases with the reduction of the air exit condition. For the month of January reducing the DBT from 15.7 °C to 8.6 °C increases the water production by nearly 618 *l*/day.

Table 2. Power and water yield based on measured average temperature and relative humidity for the representative days.

Month	Ambient air		Exit condition		Power			Yield
	DBT °C	RH%	DBT °C	WBT °C	Comp kW	Evap kW	Cond kW	*l*/day
Jan	22.62	0.6096	8.6	8.3	10.9	3	2.2	617.9
Feb	25.24	0.5754	10.8	10.3	11.6	3	2.2	645.9
Mar	26.79	0.8694	9.7	9.7	23.4	3	4.4	1923
April	28.13	0.7501	9.3	9.2	23.8	3	4.4	1746
May	29.06	0.6566	8.7	8.6	23.8	3	4.4	1561
June	32.27	0.6601	12.8	12.7	26	3	4.4	1760
July	34.79	0.552	9.4	9.2	25.9	3	4.4	1932
Aug	32.89	0.7902	16.4	16.3	27	3	4.4	2168
Sep	31.73	0.8492	16.1	16.0	26.6	3	4.4	2239
Oct	31.25	0.6723	11.7	11.6	25.3	3	4.4	1741
Nov	27.57	0.7689	9.0	8.9	23.5	3	4.4	1750
Dec	27.01	0.8413	9.6	9.5	23.5	3	4.4	1880

4. Economics of the Combined System

In order to practically evaluate the combination of A/C and water extraction prcesses it is necessary to estimate the cost of the products. The economics of mechanically driven HP is directly related to the capital cost of the unit and the cost of the input energy ($/kWh). For a combined unit which produces $\dot{m}_w$ (kg/s) of water and at the same time remove heat of $\dot{m}_a$ from the ambient condition (*i*) to condition (*o*), Fig. 2. If the fixed capital of the unit is C_f and the estimated life time is n (15 years for the present unit) the fixed charges rate f is

$$f = \frac{i(1+i)^n}{(1+i)^n - 1} \tag{5}$$

The fixed annuity is C_F.

$$C_F = f\,C_f \tag{6}$$

The operation cost of the unit consists mainly of the electric energy expenses consumed by the compressors and fans (see Table 3). If the electricity rate is c_e $/kWh then the operation cost may be expressed as:

$$C_o = \int_0^y c_e L_e dt \tag{7}$$

Where, L_e is the sum of the electric loads and y is period of operation per year.

Equations 6 and 7 give the annual expenses of a combined unit. The unit is of the dual purpose type (where it provides two different products) fresh water and cooling energy in kWh. To calculate the water production cost there are different approaches as follows.

In a biased analysis all expenses are allocated to water production, the maximum water cost becomes;

$$c_{w,\max} = \frac{C_F + C_o}{M_w} \tag{8}$$

M_w is the annual water production of the unit for y operation hours per year.

A unit has been built in Jeddah, KSA, at a cost of $ 37,000. This is a high cost for a pilot unit and a commercial unit would be much cheaper. The electricity rate depends on the consumption rate and varies between 0.02 and 0.07 $/kWh. Assume 15 years lifetime and working hours y = 8000 h/year. For an interest rate of 10% the fixed charges rate f is 0.13. The annual water production of the unit from Table 3 is 590 m^3 (excluding the yield of January). The cumulative monthly water production and the corresponding electricity consumption kWh (compressors and fans) are given in Fig. 5. The estimated annual yield of the unit from the data in Fig. 5 is 610 m^3. This value drops to 590 m^3 when considering only 8000 operation hours per year. The corresponding energy consumption is 248 MWh during the 8000 hours operation period y. From these numbers the specific water productivity is 2.38 *l*/kWh. This result clearly shows that the specific electricity consumption for the operation conditions of Table 2 is 0.42 kWh/ *l* of fresh water.

Now consider both fixed and operation costs, Eq. 8, take a mean value for the electricity rate as 0.045 $/kWh; the cost per unit production becomes;

$$c_{w,\max} = \frac{0.13x37000}{610x10^3} + 0.42x0.045 = 0.027 \ \$/l \tag{9}$$

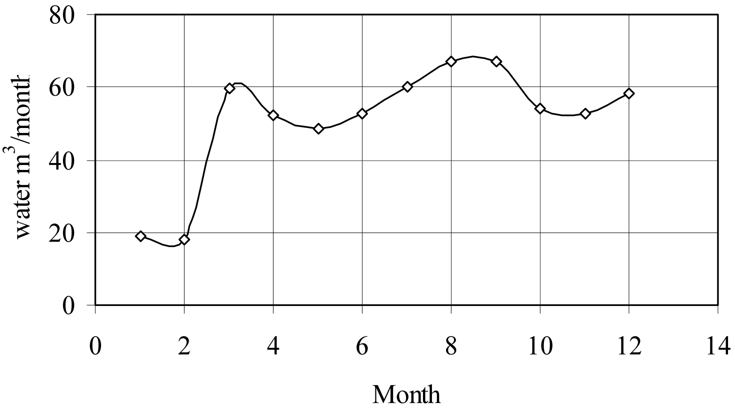

Fig . 5(a). Cumulative monthly water production.

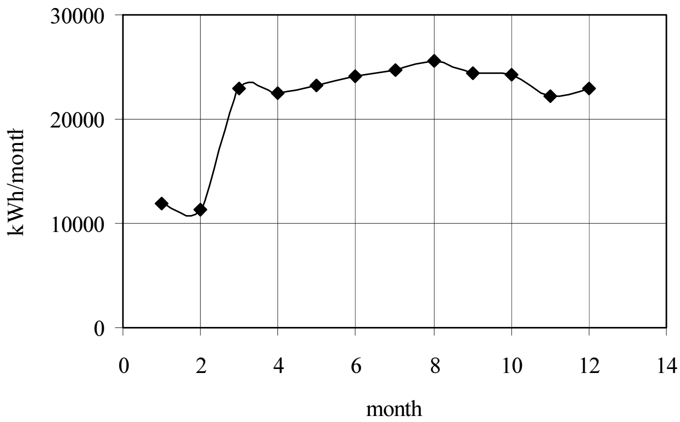

Fig. 5(b). Monthly power consumed by the combined A/C-dehumidification system.

The obtained cost of 2.3 cents per liter (23 $/m^3$) was based on the high capital cost of the pilot plant where the capital charges presents 33% of the total expenses and the rest is energy cost.

In this economic analysis it has been assumed that the fixed capital cost and the energy costs were both allocated for water production. More realistic approach is to share the annual expenses between the cooling load and the water output. It is suggested here to share the expenses according to the ratio of energies utilized for each process, the sensible heat for the A/C effect and the latent heat for the water extraction process.

1- A/C Desal unit
2- Air ducts
3- 250 m^2 floor area building
4- 0.5 m^3 glass tank for water measuring
5- condenser fans

Fig. 6. A combined A/C dehumidification unit installed at Al-Norous resort, Jeddah KSA, size 3 x 2.25 x 1.5 m^3, weight 1140 kg.

Let us define δ as the ratio between the latent heat ratio to, from Fig. 2

$$\delta = \frac{(h_{ai} - h_{ab})}{(h_{ai} - h_{ao})} \tag{10}$$

The annual variation of δ with the average climatic conditions for the representative days is shown in Fig. 7. For all months except January and February, where the DBT was low there is no appreciable change in δ with an annual average of 0.6. This simply means that 60% of the power consumed has been utilized for the latent heat removal or in other words for the water production process.

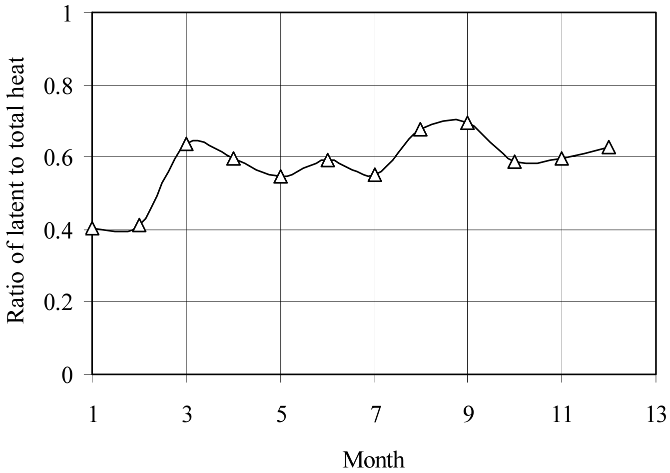

Fig. 7. Annual variation of δ for combined HP water production system.

It seems reasonable now to share the annual expenses by 60% for water and the rest goes for the A/C process this gives 0.0162 \$/*l*. Further decrease in the water cost could be achieved by assuming the lowest electricity rate in Jeddah (0.02 \$/kWh), which results a water cost of 0.0135 \$/*l* (50.6 Saudi Riyal/m^3). The combined system could be utilized for hot humid recreation areas where air conditioning and potable water are needed at limited amounts.

5. Conclusions

In many countries (Gulf States) where the weather is hot and humid on coastal areas, mechanical driven heat pumps have been extensively in use consuming around 60% of the supplied electricity. Further more lack of natural water resources forced these nations to depend on the desalination technology for potable water. A unit that combines a HP with extraction of water from atmospheric air is investigated. The unit covers the cooling load of a building and in the same time provides limited amounts of fresh water. To maximize the water production the HP was made up of two separate series evaporators (copper-copper finned tube coils) with R-134-a refrigerant. The unit provides air at 15 ^oC to an office building (250 m^2) by the sea side in Jeddah, KSA. Under Jeddah climatic conditions the yield of the combined unit changes from 618 *l*/day in January to a maximum of 2.23 m^3/day in September. The specific power was estimated by 0.42 kWh/ litter of fresh water.

For economic evaluation of the combined system it is suggested to share the annual expenses by the ratio of energy utilized for water production and A/C processes. The cost of water depends on the electricity rate and the capital of the pilot unit. For the conditions of Jeddah, 0.02 \$/kWh, and capital cost of 37000 \$ the water rate has reached 13.5 \$/m^3. The combined system could be successful for hot humid recreation areas where air conditioning and potable water are needed at the same time.

Acknowledgements

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Nomenclature

A	Area, m^2
c	specific cost, \$/unit
h_a	specific enthalpy of air, kJ/kg dry air
h_{fg}	specific enthalpy water vapor in the air, kJ/kg
k	thermal conductivity, kW/m K
k_f	fin thermal conductivity, W/m K
L_e	electric load, kW
$\dot{m}$	mass flow rate, kg/s
NTU	number of heat transfer units
$\dot{Q}_a$	heat transfer rate, kW
t	time, s
T	temperature, K
U_o	overall heat transfer coefficient, kW/m2 K
M_{day}	daily water yield, kg/day
y	operation hours per year, h.

Greek Symbols

δ	ratio of latent to total heat
ε	effectiveness
τ	time period, s
ω	RH humidity ratio, kg/kg dry air.

Subscripts

a	dry air
e	electricity
f	fin
i	inlet
m	mean
o	outlet
r	refrigerant
w	water

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تحليل أداء نظام مزدوج لمضخة حرارة وإزالة رطوبة

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المستخلص. شهدت دول الخليج في العقود القليلة الماضية نموًا سريعًا في مستويات المعيشة، وصاحب ذلك بطبيعة الحال زيادة مضطردة على طلب الكهرباء والمياه، ونتيجة للتغيرات في نمط الحياة، فإن الجزء الأكبر من الطاقة الكهربائية يستهلك في تكييف الهواء. إن نقص الموارد الطبيعية للماء مع الحاجة الشديدة له جعل تحلية المياه التقنية الرئيسية لتوفير مياه الشرب وكجهد للبحث عن طرق غير مألوفة لإنتاج مياه عذبة، فإن الورقة الحالية تحقق في عملية مزدوجة لمضخة حرارية مع إزالة الرطوبة لاستخراج الماء من الهواء الجوي. تقوم الوحدة بضخ ١,٥٨٦م^{٣}/ث لمبنى مكاتب (مساحته ٢٥٠م^{٢}) يقع على جانب البحر في مدينة جدة في المملكة العربية السعودية، حيث يبلغ متوسط درجة الحرارة ٣٤ درجة مئوية، ونسبة الرطوبة ٧١٪ وتتكون الوحدة من ملفي تبخير لزيادة عملية تكثيف الماء بخفض درجة الهواء الخارج إلى ٨ درجات مئوية .

وقد تم اختبار كفاءة أداء وحدة التكييف والتكثيف المزدوجة لسنة واحدة بتطبيق أوضاع جوية مقاسة لأيام محددة تمثل كل شهر من السنة، وكان أكثر إنتاج للماء في شهر (سبتمبر) بواقع (٢,٢٣م^{٣} /يوم) والأقل في شهر (يناير) بواقع (٠,٦١٨م^{٣}/يوم). تم حساب كمية الطاقة المستخدمة وكان متوسط معدل الطاقة الخاص ٠,٤٢ كيلوات ساعة/لتر من الماء النقي، وقد تم شراء الوحدة

التجريبية بقيمة عالية نوعاً ما، مما أدى إلى زيادة معدل التكلفة الثابتة.

وقد بلغت القيمة المتوقعة الأعلى للماء المكثف ٢٣ دولار/م^{٣} لمعدل سعر الكهرباء ٠,٠٤٥ دولار / كيلوات ساعة إذا تم توزيع التكلفة بين التكييف وإنتاج الماء، فإن السعر المتوقع الأكثر واقعية هو ١٣,٥ دولار /م^{٣} (٥٠,٦ ريال سعودي)، ومن الممكن أن تكون الوحدة المزدوجة للتكييف والتكثيف أكثر كفاءة لمناطق المنتجعات ذات الرطوبة العالية والطلب الأقل على الماء والحاجة لتكييف الهواء.