Energy consumption and feasibility study of a hybrid desiccant dehumidification air conditioning system in Beirut

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ABSTRACT: In this work, the transient performance of a hybrid desiccant vapor compression air conditioning system is numerically simulated for the ambient conditions of Beirut. The main feature of this hybrid system is that the regenerative heat needed by the desiccant wheel is partly supplied by the condenser dissipated heat while the rest is supplied by an auxiliary gas heater. The hybrid air conditioning system of the present study replaces a 23 kW vapor compression unit for a typical office in Beirut characterized by a high latent load. The vapor compression subsystem size in the hybrid air conditioning system is reduced to 15 kW at the peak load when the regeneration temperature was fixed at 75 °C. Also the sensible heat ratio of the combined hybrid system increased from 0.47 to 0.73.

Based on hour by hour simulation studies for a wide range of recorded ambient conditions of Beirut city, this paper predicts the annual energy consumption of the hybrid system in comparison with the conventional vapor compression system for the entire cooling season. The annual running costs savings for the hybrid system is 418.39 USD for a gas cost price of 0.141 USD/kg. The pay back period of the hybrid system is less than 5 years when the initial cost of the hybrid air conditioning system priced an additional 1712.00 USD. Hence, for a 20-year life cycle, the life cycle savings of the hybrid air conditioning system are 4295.19 USD.

Keywords: desiccant, dehumidification, hybrid system

1. INTRODUCTION

Emphasis on the design of energy efficient air-conditioning systems for both industrial and comfort applications is becoming a priority in the light of continuing rise in energy demand, costs and the associated environmental problems most notably the climate change. In humid climates such as the Lebanese coast, the humidity issues are a major contributor to energy inefficiency in HVAC devices. The high humidity of the outside air combined with ventilation requirement increases the latent load. Conventional vapor compression air-conditioning systems are not designed to independently control temperature and humidity. The use of desiccant pre-conditioning of supply air can improve the humidity control independent of temperature in air conditioning systems, improve overall energy efficiency, and reduce energy costs. Furthermore, incorporation of desiccant preconditioning into such systems allows higher percentage of fresh air in the supply stream to achieve better air quality at lower energy cost [1] and [2]. Desiccant dehumidification systems use either liquid sorbent equipment or solid-rotary desiccant wheels. Both types of system are used to assist conventional air-conditioning system in residential applications [3]-[7]. Another attractive feature of desiccant dehumidification systems is their suitability for solar or other low-grade thermal energy applications [7]. Solid wheel desiccant system is compact and can be integrated easily into the existing residential air conditioning systems unlike liquid desiccant systems that require tower beds and additional fan power.

In this work, the transient performance of the hybrid desiccant dehumidification wheel and a vapor compression air conditioning system is studied for the ambient conditions of Beirut. The hybrid system of the present study replaces a 23 kW vapor compression unit for a typical office in Beirut characterized by a high latent load. Based on hour by hour simulation studies for a wide range of recorded ambient conditions of Beirut city, this paper predicts the annual energy consumption of the hybrid system in comparison with the conventional vapor compression system for the entire cooling season. The heat dissipated by the condenser supplies part of the wheel regeneration heat and the auxiliary gas heater supplies the rest. An economical analysis will be performed of the integrated hybrid system as compared to the conventional vapor compression system.

2. PROBLEM FORMULATION

A. Conventional system: The objective of the work is to compare the performance and energy consumption of the hybrid desiccant air-conditioning system and the conventional air conditioning system for a high latent load application in the city of Beirut. The selected application is for a 150 m^2 office located in the first floor of a building characterized by typical Lebanese construction materials. The length of the office is 10 m and its width and height are 5 m and 3 m, respectively. The office heat conductance values (U) are: Roof (U = 0.4 $W/m^2.K$), Wall (U = 3 $W/m^2.K$), Partition (U = 2.19 $W/m^2.K$), Floor (U = 1.03 $W/m^2.K$), and Windows (U = 6.17 $W/m^2.K$). The recommended indoor design conditions are taken at 24 °C dry bulb air temperature and 55% relative humidity. The energy analysis of the space is calculated on hourly basis using the DOE software. The sensible and latent loads needed to maintain the office at the specified indoor conditions of temperature and humidity are calculated for the conventional system design. The peak load occurs at 4 p.m. on 25th of July and is equal to 23 kW consisting of a sensible load of 10.9 kW and a latent load of 12.1 kW as shown in Fig. 1. The Sensible Heat Ratio (SHR) for the conventional system is 0.47 for the office and the office fresh air requirement equals to 0.625 m^3/sec for 40 persons and is equal to the supply air of the system. The monthly cooling cost is also evaluated by DOE and is presented in Fig. 2.

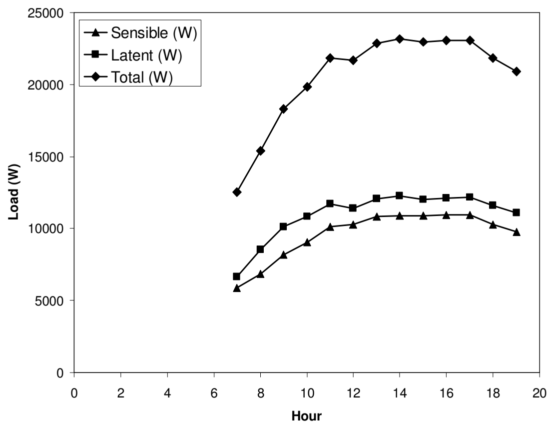

Fig. 1. Cooling load of the office at the peak day

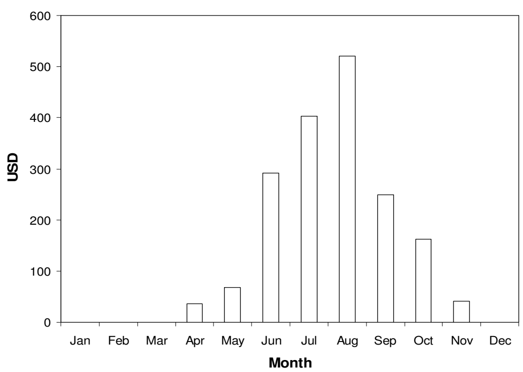

Fig. 2. Monthly energy cost over the entire cooling season

2.1. *Hybrid desiccant air-conditioning system*

B. Hybrid desiccant air conditioning system: Figure 3 shows a schematic of the integrated hybrid desiccant air conditioner system. Ambient humid air at state (1) enters the supply air duct and passes through the dehumidification section of the desiccant wheel where it will be dried up and heated. Exiting the wheel at state (2), the process air enters a heat exchanger where it releases some of its heat energy to the return air stream. Then this hot dry air, state (3), is cooled by a cooling coil and then distributed at state (4) to the room. The desiccant wheel is regenerated by heating the ambient air to a fixed regenerative temperature of 75 °C. The temperature 75 °C is used because the heating auxiliary heater can be replaced by solar or other low-grade thermal energy application. The total heat energy required to regenerate the wheel is supplied partly by the heat dissipated by the condenser of the vapor compression system and the rest is supplied by an auxiliary gas heater. The ambient air is heated from state (1) to state (7) by the heat dissipated by the condenser and then the ambient air is further heated to state (8) by the auxiliary gas heater to increase its regeneration potential. The heated ambient air stream finally regenerates the desiccant wheel and is exhausted at state (9).

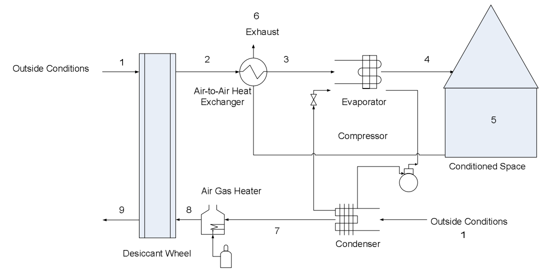

Fig. 3. A schematic of the hybrid desiccant air conditioner system

A brief description of the governing equations of each component in the system is provided in the following sections.

C. Desiccant wheel: A rotary regenerative desiccant wheel is a rotating cylindrical wheel divided into two sections: regeneration and a dehumidification sections. The dehumidification and regeneration air streams are usually in a counter-flow arrangement. The wheel rotates slowly to expose one portion of the desiccant material to the humid process air stream while the other portion simultaneously passes through the hot regeneration air stream. A partition and flexible seals separate the process and regeneration air streams in the dehumidifier. As a result the temperature of the process air is raised and its humidity decreases. In the other hand, the humidity ratio of the regeneration air is increased and its temperature decreases.

Zhang et al. [8] developed a one-dimensional coupled heat and mass transfer model of rotary dehumidifier to predict the temperature and humidity profiles and analyze and verify performance with experimental data [8]. The one-dimensional model of Zhang et al. [8] of the rotary dehumidifier is adopted in the current work. The mass and energy conservation equations are given as follows:

Moisture balance on the air side

$$\frac{\partial}{\partial t}(\rho_v A_c) + \frac{\partial}{\partial z}(\rho_v V A_c) = \rho_{da} K_m P (Y_w - Y_a) \qquad (1)$$

Moisture balance on the desiccant side

$$\frac{\partial}{\partial t}(m_w)_d = \rho_{da} K_m A_s (Y_a - Y_d) \qquad (2)$$

Energy balance on the air side

$$(\rho_{da} C_{pda} + \rho_v C_{pv} Y_a) A_c \frac{\partial T_a}{\partial t} + (\dot{m}_{da} C_{pda} + \dot{m}_v C_{pv}) \frac{\partial T_a}{\partial z} = K_h P (T_d - T_a) + \rho_{da} K_m P (Y_a - Y_d) C_{pv} (T_d - T_a) \quad (3)$$

Energy balance on the desiccant wall

$$m_d C_{pd} \frac{\partial T_d}{\partial t} + m_w C_{pw} \frac{\partial T_d}{\partial t} + m_m C_{pm} \frac{\partial T_d}{\partial t} = K_h A_s (T_a - T_d) + \rho_{da} K_m A_s (Y_a - Y_d) Q + \rho_{da} K_m A_s (Y_a - Y_d) C_{pv} (T_a - T_d) \quad (4)$$

where Y is the humidity ratio, T is the temperature, t and z are the time and axial direction, K_m and K_h are the mass and heat transfer coefficients, A and P are the area and the perimeter, ρ is the density, V is the velocity, C_p is the isobaric specific heat, and Q is the adsorption heat. The subscripts d, da, l, m, v and w stand for desiccant, dry air, liquid water, matrix material, water vapor and duct wall respectively. Equations (1) through (4) will be solved to determine: the temperature and humidity ratio of the air leaving the desiccant wheel at state (2); and the temperature and humidity ratio of the regenerating air leaving the desiccant wheel at state (9). For a given rotational period, the input parameters to the model are: the state of the process air at state (1) and the regeneration air conditions at state (8). In the current work, the period of the wheel rotation is fixed at 500 s, the wheel thickness is 0.2 m and its diameter is 0.5 m. Also the dehumidification and regeneration areas are considered equal and the air velocity is set at 3 m/s.

D. Heat Exchanger: Since the flow rates of the space return air stream and the process fresh air stream are equal and assuming equal specific heats, the effectiveness of the heat exchanger is assumed to be equal to 0.8 and it can be written as follows:

$$E_{sw} = \frac{T_2 - T_3}{T_2 - T_6} = \frac{T_5 - T_6}{T_5 - T_3} \quad (5)$$

E. Vapor compression size: The cooling load of the vapor compression subsystem is not equal to that of the conventional air conditioning system and its hourly load is calculated as follows:

Vapor compression subsystem Load = Conventional vapor compression system Load - $\dot{m}\,\Delta(h_1 - h_3)$. Where h_1 is the enthalpy of the ambient air and h_3 is the enthalpy of the air leaving the heat exchanger. Hourly cooling load are performed by the visual DOE software over the whole cooling season. In addition, the desiccant wheel model is simulated to obtain the hourly condition of the process leaving the desiccant during the cooling season. The hourly enthalpy, h_3, is easily determined using the heat exchanger effectiveness model equation and thus the hourly load of the vapor compression subsystem can be determined over the whole cooling months.

F. Heat dissipated by the condenser: The amount of heat dissipated by the condenser depends on the cooling load of the vapor compression subsystem in the non-conventional hybrid air-conditioning system and its coefficient of performance (COP). The COP is calculated using a DOE regression formula:

$$COP = \frac{1}{0.011 + 1.18PLR - 0.25PLR^2 + 0.056PLR^3} \quad (6)$$

Where PLR is the part load ratio.

3. SOLUTION METHODOLOGY

The hourly transient cooling load of the space is calculated on hourly basis using the DOE software. The desiccant wheel model is also simulated hourly to determine the condition of the air leaving the heat exchanger using the effectiveness of the heat exchanger. Hence the hourly cooling load of the vapor compression subsystem can be determined to calculate the PLR and COP of the compression subsystem. The next step is to determine the heat dissipated by the condenser and the temperature of the ambient heated air leaving the condenser. The hourly heat supplied by the auxiliary gas heater can then be easily calculated since the regeneration temperature is fixed at 75 °C and the hourly temperature of the heated ambient air leaving the condenser is known.

The simulation result of the desiccant wheel program is validated initially by reproducing the results of the system configuration and model of Zhang et al [8]. The current simulation program has predicted the same performance simulation curves of Zhang et al [8] of humidity ratio variation during the cycle at exit of the desiccant wheel channels.

4. RESULTS AND DISCUSSION

It is important to perform an economic feasibility analysis of the hybrid-desiccant system as compared to the traditional vapor compression system. The hybrid-desiccant system is cost effective when its additional cost is less than the present value of the energy savings. The cost of the vapor compression system for the office having a size of 23 kW chiller, is taken equal to $7330. When used as a hybrid system, the desiccant dehumidifier reduces the size of the vapor compression system of the restaurant to 15 kW having a cost of about $5720. The cost of the desiccant wheel is $3000. The total additional initial cost of the hybrid system is priced $ 1712 including installation cost. The operating energy cost of the integrated system, over the whole cooling season, is given by:

$$COST = \frac{E_{elec} C_e}{3.6x10^6} + \frac{Q_{aux} C_g}{H_g} \quad (7)$$

Where E_{elec} is the monthly electric energy consumption and H_g is the heating value of gas. Figure 4 shows the monthly cost of cooling for the hybrid desiccant vapor compression system for a gas price of 0.141 US $/kg. The annual running costs savings for the hybrid system is 418.39 USD for a gas cost price of 0.141 USD/kg. In this study, an initial extra expenditure “C”, $1712, is invested at a market discount rate d = 0.08, in order to make a saving “S” in the yearly energy bill that is inflating at a rate e equal to 0.05 over a life cycle of N years. The system is assumed to have no resale value. The life cycle savings is then given by the following equation:

$$LCS = -C + PWF(N,e,d)S \tag{8a}$$

Where,

$$PWF(N,e,d) = \frac{1}{d-e}\left(1-\left(\frac{1+e}{1+d}\right)^{N}\right) \quad \text{if } e \neq d \tag{8b}$$

$$PWF(N,e,d) = \frac{N}{1+e} \quad \text{if } e = d \tag{8c}$$

Figure 5 shows the variation of the life cycle savings with the life cycle period. The pay back period of the hybrid system is less than 5 years when the initial cost of the hybrid air conditioning system priced an additional 1712.00 USD. Hence, for a 20-year life cycle, the life cycle savings of the hybrid air conditioning system are 4295.19 USD.

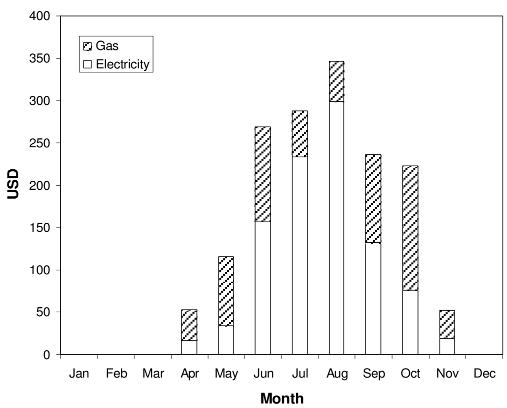

Fig. 4. Monthly energy cost of the hybrid system

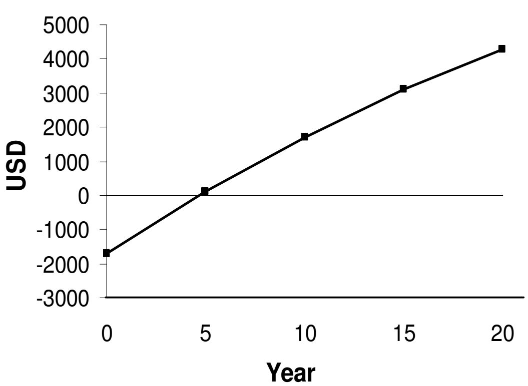

Fig. 5. Variation of the life cycle savings of the hybrid system

5. Conclusion

This work studies the economic feasibility of integrating a rotary solid desiccant wheel with the conventional vapor compression air conditioning system in the city of Beirut. To increase the economic practicality of such a hybrid system, the combined system utilizes the heat dissipated by the condenser and natural gas heat energy in regenerating the desiccant wheel. A heat and mass transfer desiccant model is developed to study the hourly performance of the hybrid system when used in air conditioning a typical office (150 m^2) in Beirut over the entire cooling season. The vapor compression subsystem size in the hybrid air conditioning system is reduced to 15 kW at the peak load when the regeneration temperature was fixed at 75 °C. Also the sensible heat ratio of the combined hybrid system is increased to 0.73. The economic performance of the hybrid system is compared with the conventional air conditioning system. The annual running costs savings for the hybrid system is 418.39 USD for a gas cost price of 0.141 USD/kg. The pay back period of the hybrid system is less than 5 years when the initial cost of the hybrid air conditioning system priced an additional 1712.00 USD.

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