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LIQUID DESICCANT DEHUMIDIFICATION MEMBRANE CEILING/DISPLACEMENT VENTILATION SYSTEM EFFECTIVENESS WITH OPTIMIZED OPERATIONAL STRATEGY

by
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AN ABSTRACT OF THE THESIS OF

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Title: Liquid Desiccant Dehumidification Membrane Ceiling/Displacement Ventilation System Effectiveness with Optimized Operational Strategy

The performance of the chilled ceiling (CC) displacement ventilation (DV) systems is constrained by latent load removal capacity and associated cost of supply air dehumidification to prevent condensation on the chilled ceiling.

In this study, a liquid desiccant dehumidification membrane cycle (LDMC) is introduced and mathematically modeled to replace the chilled ceiling to remove directly both latent and sensible load directly from indoor space through the liquid desiccant ceiling membrane. The desiccant system is coupled with displacement ventilation system to enhance the indoor air quality at 100% fresh supply air. Solar energy is used as thermal energy source required for the regeneration of the desiccant.

The operation of the system is simulated using integrated models of the various components of the hybrid system and an optimized operational strategy is adopted based on genetic algorithm to improve performance while maintaining thermal comfort and good air quality. The optimized parameters for system operation are the DV supply flow rate and temperature and the desiccant temperature at the inlet of the ceiling membrane.

The model of the hybrid LDMC-C/DV system and its optimization were implemented for a case study consisting of an office during the month of August in Beirut hot and humid climate. When using optimal set points, a decrease of 49% in energy consumption is observed when compared to the conventional CC/DV system while comfort and IAQ were satisfied. In addition, a ceiling temperature (as low as 16°C) was attained by the membrane ceiling which was lower than air dew point temperature when compared to conventional CC system.

CONTENTS

NOMENCLATURE

A	area (m^2)
c	concentration of water per desiccant (kg_{H2O}/kg_{CaCl2})
CC	chilled ceiling
C_p	specific heat (J/kg·K)
d	discount rate
D	diffusion constant of vapor in the pipe wall material (m^2/s)
DV	displacement ventilation
h_c	heat convection coefficient ($W/m^2 \cdot s$)
h_m	mass convection coefficient (m/s)
h_{fg}	latent heat of vaporization of the water (J/kg)
h_d	enthalpy of the liquid desiccant solution (J/kg)
h_s	enthalpy of supply air(J/kg)
K	thermal conductivity (W/m·K)
L	length of the pipe (m)
LDMC-C/DV	liquid desiccant membrane cycle at the ceiling combined with displacement ventilation (DV) system
$\dot{m}$	mass rate (kg/s)
n	number of years

N	number of dehumidification/regeneration pipes
p	present worth value
$P_{chiller}$	chiller capacity (W)
PLR	part load ratio
Q_s	Cooling energy needed for supply air (W)
Q_d	Cooling energy needed for liquid desiccant (W)
Q_{DV}	sensible load removed by displacement ventilation system (W)
r_o	external radius of the pipe (m)
r_i	internal radius of the pipe (m)
R	chilled ceiling cooling load to room total load = Q_c/Q_s
T	temperature (℃)
T_a	indoor room (℃)
T_∞	ambient temperature (℃)
T_w	wall temperature (℃)
T_c	ceiling temperature (℃)
T_s	supply air temperature (℃)
U	resistance coefficient of permeable pipe per unit length
w	humidity ratio ($kg_{H2O}/kg_{dry\ air}$)
w_a	indoor room humidity ratio ($kg_{H2O}/kg_{dry\ air}$)
w_{sol}	the solution equilibrium humidity ratio ($kg_{H2O}/kg_{dry\ air}$)

Greek Letters

ρ	density (kg/m^3)
$\propto$	cost function weighing factor

σ	Stefan Boltzman constant
ϵ	Emissivity

Subscripts

a	room air
c	heat convection
d	$CaCl_2$ and H_2O desiccant solution
I	indoor conditions
m	mass convection
s	supply air conditions
o	outdoor conditions
w	wall

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CHAPTER I

COMBINED LIQUID DESICCANT CYCLE WITH DISPLACEMENT VENTILATION SYSTEM

1.1. Introduction

People these days spend most of their time in indoor spaces. The indoor air should be treated and fresh air should be supplied to the space, to provide comfortable and healthy indoor space for people. With increasing the amount of supply fresh air, an increase in energy consumption arises in order to condition this required fresh air.

Many studies have focused on new heating ventilation and air-conditioning (HVAC) systems that improve thermal comfort and reduce energy consumption. Among these systems is the displacement ventilation (DV) system. In displacement ventilation, the supply fresh air enters the room at the floor level and displaces the warmer air that rises due to buoyancy effect to reach the ceiling level where they are exhausted (Jiang et al. 1992; Yuan et al. 2001). To prevent thermal drafts at the occupied level, and ensure thermal comfort, DV systems supply air at temperatures not less than 18°C and velocities not more than 0.2 m/s (ASHRAE 2009). For this reason, the maximum load that can be removed from the occupied zone by a displacement ventilation system is approximately 40 W/m^2 of floor area (Yuan et al. 2001). Behne (1999) estimates that to increase cooling capacity in hot climates, chilled ceiling (CC) is combined to the displacement ventilation system and the cooling load limit would increase to 100 W/m^2 of floor area. Due to their energy transfer mostly by radiation and by decoupling ventilation from heat transfer, CC systems provide better thermal comfort, and reduce energy consumption and noise (Joege et al. 2002). Increasing the portion of the load carried by the chilled ceiling will enhance the indoor thermal comfort (Keblawi et al. 2009). However this is not possible with the conventional CC\DV systems, since CC systems are limited with problem of condensation of water vapor. If the CC panel temperature drops below the dew point temperature of the space, condensation takes place on the panel which constrains the ceiling temperature to always be above the room air dew point. To prevent condensation, humidity is removed from the air at the supply section. Humidity removal is usually done by two methods: conventional and non-conventional techniques. The conventional way of removing moisture is by cooling the supply air to a temperature below its dew point temperature to condense the excess moisture and then reheating it to the adequate supply air temperature (Xiao et al. 2011). But this is not energy efficient method due to the additional sub-cooling and reheating energy needed (Wang et al. 2013; Xiao et al. 2011). Researches had considered a sustainable dehumidification technique by using of desiccant technology as a passive method for supply air dehumidification (Wang et al. 2013; Fauchoux et al. 2010).In Conventional desiccant dehumidification techniques; desiccant is utilized in solid or liquid phase. If liquid desiccant is used, tower beds are employed for dehumidification and regeneration (Wang et al. 2013; Mohammad et al.2013), whereas in the solid desiccant case, a desiccant wheel is used (Wang et al. 2013). However, a direct contact takes place between the strong desiccant solution and the supply air during dehumidification, which will lead to carryover of hazardous salts to the ventilation system causing negative impacts related to health issues and corrosion problems (Studak et al. 1988).

To overcome the previous problems, researchers have implemented liquid desiccant membranes to cool and dehumidify air without direct contact with the desiccant (Fauchoux et al. 2009; Eldeeb et al. 2013). The membranes are impermeable to liquid desiccant but permeable to water vapor. A membrane constructed of circular pipes was modeled and experimentally validated by Keniar et al. (2015) for direct indoor dehumidification. This new dehumidification methodology eliminates the dangerous of transferring of hazardous desiccant to the indoor air since there is no direct contact between the dehumidifying air and the liquid desiccant.

The conventional *CC/DV* air conditioning system involves the simultaneous operation of two subsystems with multiple variables (flow rate, temperatures…) to be controlled under dynamic loading unlike conventional systems. Energy consumption is reported to be strongly dependent on the settings of these variables and the selected control strategy of the system operation (Mossolly et al. 2009; Keblawi et al. 2011; Niu et al. 2002). The purpose of a control strategy of the *CC/DV* system operation is to provide acceptable thermal comfort and best indoor air quality (IAQ). Niu et al. (2002) used, while assessing a *CC/DV* system performance, a control strategy by which the supply air flow rate and temperature are kept constant at design values of ventilation requirements and temperature of 15 °C, and further cooling requirement is met by activating the ceiling panel. Mossolly et al. (2009) showed that using a multi-variable control strategy, that varied optimally the *CC* temperature, supply flow rate and supply air temperature in response to given load profile, saved more than 15% *CC/*DV system energy consumption in comparison with single variable control strategy performance based on *CC* temperature.

Keblawi et al. (2011) developed real-time optimized supervisory controller for a combined *CC/DV* system to improve energy efficiency. The supply humidity control of the *CC/DV* optimized operation systems studied by Niu et al. (2002), Mossolly et al. (2009) and Keblawi et al. (2011) was significant to prevent the water vapor condensation on the ceiling. This is an energy intensive process (Xiao et al. 2011, Bahman et al. 2012). Using of liquid desiccant membrane for heat and mass transfer at the indoor space will result in change of optimal set points for the CC/DV system operation since humidity removal is done inside the room. The operational variables ranges and energy consumption associated with the dehumidification system and DV system will change leading to a different set of optimized parameters under transient loading. Such optimization for the proposed liquid desiccant membrane system combined with DV system has not been studied in literature.

In this study, a liquid desiccant dehumidification membrane system is proposed to be used as chilled ceiling for cooling and dehumidifying an internal office space in Beirut City climate. The desiccant system is coupled with displacement ventilation system to enhance the indoor air quality and thermal comfort of an indoor space. Solar collectors are used to provide thermal energy needed for regeneration of the liquid desiccant. The proposed liquid desiccant membrane cycle at the ceiling combined with displacement ventilation (LDMC-C/DV) system overcomes the ceiling temperature limitation in conventional CC/DV systems, where lower temperature could be reached in the proposed system without onset of condensation on the ceiling thus improved thermal comfort. Since humidity removal takes place inside the space, it will prevent carryover of liquid desiccant and result in lower regeneration temperature as compared to solid desiccant system. The membrane liquid desiccant system combined with DV system is integrated with solar collectors for liquid desiccant regenerating. Since the LDMC-C/DV system involves the simultaneous operation of two subsystems with multiple variables (flow rate, temperatures…), an optimized control strategy is proposed using genetic algorithm as an optimization method. The corresponding optimized operational variables will lead to obtaining the best operating conditions to minimize energy cost while considering the thermal comfort and IAQ. The economic feasibility and hourly performance of the LDMC-C/DV system will be compared to a conventional CC/DV air conditioning system to assess the expected increase in DV load and reduction in CC load to obtain the same space operating conditions.

1.2. Objectives

In this study, a liquid desiccant dehumidification membrane system is proposed to be used as chilled ceiling for cooling and dehumidifying an internal office space in Beirut City climate. The desiccant system is coupled with displacement ventilation system to enhance the indoor air quality and thermal comfort of an indoor space. Solar collectors are used to provide thermal energy needed for regeneration of the liquid desiccant. The proposed liquid desiccant membrane cycle at the ceiling combined with displacement ventilation (LDMC-C/DV) system overcomes the ceiling temperature limitation in conventional CC/DV systems, where lower temperature could be reached in the proposed system without onset of condensation on the ceiling thus improved thermal comfort. Since humidity removal takes place inside the space, it will prevent carryover of liquid desiccant and result in lower regeneration temperature as compared to solid desiccant system. The membrane liquid desiccant system combined with DV system is integrated with solar collectors for liquid desiccant regenerating. Since the LDMC-C/DV system involves the simultaneous operation of two subsystems with multiple variables (flow rate, temperatures…), an optimized control strategy is proposed using genetic algorithm as an optimization method. The corresponding optimized operational variables will lead to obtaining the best operating conditions to minimize energy cost while considering the thermal comfort and IAQ. The economic feasibility and hourly performance of the LDMC-C/DV system will be compared to a conventional CC/DV air conditioning system to assess the expected increase in DV load and reduction in CC load to obtain the same space operating conditions.

1.3. System Description

The combine system is composed of 4 main subsystems:

a. Liquid desiccant cycle
b. Displacement ventilation system
c. Parabolic solar collector
d. Space thermal model

Fig.1 illustrates the system components.

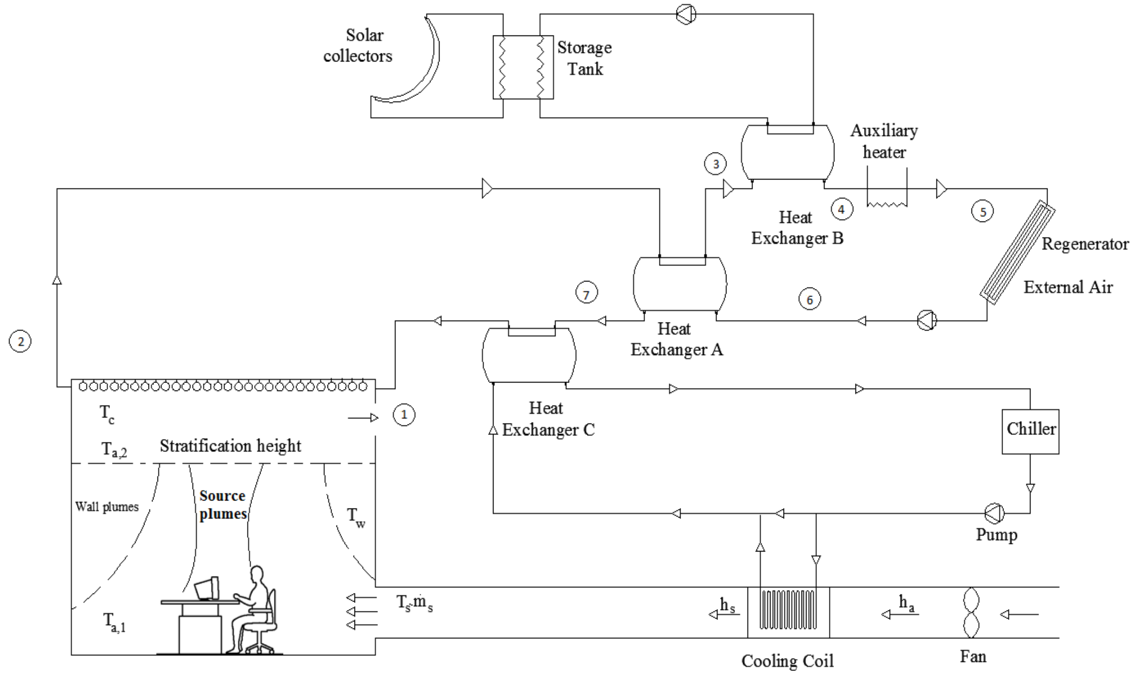

Figure 1: Combined CC/DV system with liquid desiccant cycle.

The liquid desiccant cycle includes the indoor dehumidification permeable pipes, outdoor desiccant regeneration permeable pipes, and heat exchangers. The displacement ventilation system consists of an absorption chiller, fan and cooling coil.

The pipes in the system are solid ones except in the regenerator and dehumidifier parts, where the pipes are made of porous material that is permeable to water vapor only, and not to liquid. The cold desiccant flow enters the membrane pipe at (1) picking the moisture and a part of the sensible load from the indoor space and then leaves the membrane at (2) where it is pumped by a pump through the solid piping system. The desiccant is pre-heated in heat exchanger (A) picking the heat from the liquid leaving the regenerative membrane. After that, the desiccant is heated by the solar energy in exchanger (B) through water stored in a tank. Then the heated desiccant enters the regenerator at (4), to release the absorbed moisture into the ambient air. Before entering the indoor space membrane, the desiccant will be cooled by chilled water supplied by the chiller, through exchanger (C). After that the desiccant enters the dehumidifier again at (1).

In order to heat the liquid desiccant before entering the regenerative membrane, solar energy is used as a heat source. The solar radiation is collected by using the parabolic solar panels that will heat a working fluid stored in a storage tank, which in turn heats the liquid desiccant through a heat exchanger (B) before it enters the regenerator, where the regeneration process takes place with ambient external air. If the solar energy is not sufficient to raise the liquid desiccant to the regeneration temperature, auxiliary heater will be used.The chiller is responsible for the removal of the sensible load of the combined system. It provides chilled water to the cooling coil, where the inlet air is cooled to the supply temperature directly, and also provides chilled water to cool the liquid desiccant entering the indoor membrane. The fan supplies the occupied zone with the sufficient amount of inlet air to meet ventilation and load requirements. For this combined system, several variables are involved in controlling the space thermal comfort and IAQ, which are the supply air temperature, supply air mass flow rate and the liquid desiccant temperature at the inlet of the membrane. These variables need to be related to each other at each prediction period to provide good indoor air quality and thermal comfort while minimizing the energy consumption of the system. Wherefore optimization tool is used.

1.4. Methodology

For this study, each component of the cycle is modeled independently then component models are coupled to form an integrated model to the complete cycle. In addition, optimal set points of the control variables are determined to minimize energy consumption while maintaining good IAQ and thermal comfort. A cost function for the system operation is formulated including the electrical cost function and the thermal comfort constraints. The optimization is performed using the genetic algorithm (Wang et al. 2000; Mitchell 1997) that will handle the following variables:

1) The desiccant temperature at the inlet of membrane (T_1).
2) The supply air temperature (T_s).
3) The supply air mass flow rate ($\dot{m}_s$).

It is possible to fix one of the three variables to study its effect on the system operation, and then the system will be operating with 2 variables. The genetic algorithm will search for an optimal value of the variables that varies between a lower and upper bound in order to minimize energy consumption and maintain thermal comfort (PPD, Temperature gradients) and stratification height in the space. Then the combined LDMC-C/DV system will be applied to a case study for an office in Beirut City where its performance is evaluated and compared to conventional CC/DV air conditioning systems.

CHAPTER II

INTEGRATION OF THE INDOOR MEMBRANE DESICCANT SYSTEM WITH DISPLACEMENT VENTILATION SYSTEM AND PARABOLIC COLLECTORS MODEL

The combined system of desiccant dehumidification with CC and displacement ventilation system for good IAQ, thermal comfort and reducing energy consumption involves complex interactions among its components. In this section, a modeling framework for the liquid desiccant cycle, space model, displacement ventilation system and solar collectors will be described.

2.1. Liquid Desiccant Dehumidification Cycle

The main component of the liquid desiccant cycle is the membrane used in the regenerating and dehumidifying section. The other component is the heat exchanger.

2.1.1. *Regenerator and Dehumidifier Model*

The dehumidifier and regenerator consist of the liquid desiccant flowing in the permeable polypropylene pipes. The permeable pipes couple the mass and heat transfer between the liquid desiccant and the surrounding air. Temperature and concentration of the desiccant solution exiting the permeable pipes must be calculated by the model for given inlet desiccant flow temperature and concentration.

In order to solve for the temperature and concentration changes inside the permeable pipe, conservation laws of mass and energy are applied. The equations were derived under the following assumptions: 1-D moisture and heat variation; no mass or energy storage in the pipe membrane (quasi equilibrium model); the liquid desiccant solution is ideal; and the axial vapor diffusion and heat conduction in the pipe are neglected. With these assumptions, the energy equation becomes (Keniar et al. 2015)

$$h_{fg}\rho_a U_m (w_a - w_d) - \dot{m}_d \frac{d}{dy}[h_d(1+c)] + U_c(T_a - T_d) = 0 \tag{1}$$

Where U_m and Uc represent the overall mass and heat transfer coefficients per unit length, w_a is the humidity ratio of the surrounding air, w_d is the equilibrium humidity of the desiccant solution, T_d is the desiccant temperature, $\dot{m}_d$ is the desiccant flow rate, and c represents the concentration of water per dry basis. U_c and U_m, are represented by

$$U_c = \left\{ \frac{1}{2\pi r_o h_{c,o}} + \frac{\ln(r_i/r_o)}{2\pi k} + \frac{1}{2\pi r_i h_{c,i}} \right\}^{-1} \tag{2}$$

$$U_m = \left\{ \frac{1}{2\pi r_o h_{m,o}} + \frac{\ln(r_o/r_i)}{2\pi k} + \frac{1}{2\pi r_i h_{m,i}} \right\}^{-1} \tag{3}$$

The species conservation equation of the permeable pipes is given by (Keniar et al. 2015)

$$\rho_a U_m (w_a - w_d) - \dot{m}_d \frac{dc}{dy} = 0 \tag{4}$$

The first term represents the vapor transfer from surrounding air to the desiccant solution and the second term represents the net moisture convective flow.

2.1.2. *Heat Exchanger*

The effectiveness-NTU method is used to model the heat exchangers. Three heat exchangers are used in liquid desiccant cycle. The first heat exchanger, exchanger A in Fig. 1, is used to cold the liquid desiccant entering the room through an absorption chiller. The second heat exchanger, exchanger B in Fig.1, is used as a preheater, it transform the heat from the desiccant living the regenerator to the desiccant entering exchanger C, in order to raise its temperature. The third heat exchanger, exchanger C in Fig.1, is used to supply the thermal energy from the solar collector, this energy is required to increase the desiccant temperature to the regeneration temperature.

The following model is used for the heat exchangers:

$$Q_{\max} = (mc_p)_{\min}(T_{hot,in} - T_{cold,in}) \tag{5}$$

$$T_{hot,out} = T_{hot,in} - \varepsilon \frac{Q_{\max}}{(mc_p)_{hot}} \tag{6}$$

$$T_{cold,out} = T_{cold,in} - \varepsilon \frac{Q_{\max}}{(mc_p)_{cold}} \tag{7}$$

mc_p is the minimum heat capacity of the two fluids entering the heat exchanger, $T_{hot,in}$ is the temperature of the hot fluid entering the heat exchanger and $T_{cold,in}$ is the temperature of the cold fluid entering the heat exchanger. $Q_{\max}$ is the maximum possible heat transfer, and ε is the effectiveness of the heat exchanger.

2.2. Parabolic Solar Collector Model

In order to decrease the operational cost of the system, the solar irradiance will be used as an energy source to supply required heat for regeneration the liquid desiccant. A parabolic solar collector is used because it occupies less space and is very efficient compared to other types of solar heating systems. The performance of the collector-tank system is simulated using the theory of Hottel and Whillier presented by Duffie and Beckman (Duffie et al. 2003). The heat gained in the collectors is found through the following equation:

$$Q_{solar} = SA_r - U_{total} A_r (T_{avr} - T_{ambient}) \tag{8}$$

where S is the solar flux, as a function of time, supplied to the receiver pipe passing through the focal point of the parabolic collectors. A_r is the area of the receiver pipe and U_{total} is the heat loss coefficient associated with average temperature of the fluid inside, T_{avr}, and ambient temperature of the surrounding air, $T_{ambient}$.

This heat supplied is stored in the storage tank, having the following transient equation:

$$\rho_w c_{p,w} \frac{dT}{dt} = Q_{solar} - Q_{load} - U_{\tan k} A_{\tan k} (T_{\tan k} - T_{ambient}) \tag{9}$$

The tank is used to store thermal energy to be used when the dehumidification cycle is running. $\rho_w, c_{p,w}$ are the density and specific heat of water respectively. V is the tank volume and $U_{\tan k}$ is the heat loss coefficient of the tank associated with the temperature of the tank inside, $T_{\tan k}$, and the ambient air temperature, $T_{ambient}$. Q_{load} is the thermal load of the dehumidification cycle required each hour in the cycle.

2.3. Displacement ventilation system and space model

For displacement ventilation system, an absorption chiller provides the cold water required to reduce the supply air temperature passing through the cooling coil. In displacement ventilation, the supply fresh cool air enters the room at the floor level and displaces the warmer air that rises due to buoyancy effect to reach the ceiling level where they are exhausted (Yuan et al. 2001).

The office space considered in this study is modeled in order to assess the comfort level of the occupants in the space. The space is divided into two different zones, a thermally comfortable zone with an acceptable air quality and a stratified zone where neither thermal nor indoor air quality is of a concern. The temperature and humidity are found in each zone as well as the temperature gradient and stratification height. The two different zones are separated by the stratification height formed due to buoyancy effects inside the room induce by air heated from the internal load elements. The stratification height is the level at which air circulating mass becomes negligible, it separates between the two zones in a displacement ventilation system: occupied and contaminated zones (Novoselac et al. 2002). To be able to determine the stratification height, the internal air flow rate due to the plumes is determined. Assuming that the walls are of uniform temperature, the volumetric flow rate of the upward flow (plume) is given by (Mundt 1996) as follows:

$$Q_u = 2.8\times10^{-3}\left|\Delta T\right|^{2/5} z^{6/5} l \qquad (10)$$

where ΔT is the temperature difference between the wall and lumped indoor air, z is the height of the wall, and *l* is the horizontal width of the wall.

The plume generated from a point heat source with a heat transfer rate *q* may be determined by solving equations (8), (9) and (10) for a zone with internal air having a temperature gradient (Mundt 1996) and are expressed as follows:

$$Q_{plume} = 0.00238 \times q^{3/4} \left(\frac{dT_a}{dz} \right)^{-5/8} \times B_1 \tag{11}$$

$$B_1 = 0.004 + 0.039 A_1 + 0.38 A_1^2 - 0.062 A_1^3 \tag{12}$$

$$A_1 = 2.86 \times Z \left(\frac{dT_a}{dz} \right)^{3/8} q^{-1/4} \tag{13}$$

Thus the equation used to determine the stratification height is given by:

$$Q_s = N Q_{plume} + \sum_{k=1}^{n} Q_{u,k} \tag{14}$$

where Q_s is the supply air volumetric flow rate, *N* is the number of point sources inside the room, and *n* is the number of walls.

The sensible energy balance for the lower occupied zone may be written after applying the first law of thermodynamics as follows:

$$m_{a,1} C_{p,a} \frac{dT_{a,1}}{dt} = \sum_{j=1}^{j} h_i A_{w,i} (T_w - T_{a,1}) - \dot{m}_s C_{p,a} (T_{a,1} - T_s) + q_{elec} + q_{people} \tag{15}$$

For the upper contaminated zone, the sensible energy balance may be written as:

$$m_{a,2}C_{p,a}\frac{dT_{a,2}}{dt}=\sum_{j=1}^{j}h_i A_{w,i}(T_w - T_{a,2}) - \dot{m}_s C_{p,a}(T_{a,2}-T_{a,1}) - \sum_{i=1}^{n}\int_0^l U_c(T_{a,2}-T_{d,i})dy + q_{light} \tag{16}$$

In equations (12) and (13), $A_{w,i}$ is the wall area in each zone, m_a is the air mass for the corresponding zone, h_i is the internal convective heat transfer coefficient, is the supply air mass flow rate, T_a is the air temperature for corresponding zone, and $T_{d,i}$ is the desiccant temperature at element i of the pipe.

Unlike the previous study done on CC/DV systems, the humidity is calculated in each zone separately, since the supply humid air enters the room at the lower zone and it rises while carrying more humidity from the occupied zone, then the air is dehumidified at the upper zone before it exist at the ceiling level. The mass balance equation for the lower occupied zone given by:

$$m_{a,1}\frac{dw_{a,1}}{dt}=\dot{m}_s(w_s - w_{a,1}) + \frac{Q_{latent}}{h_{fg}} \tag{17}$$

For the upper, the mass balance may be written as:

$$m_{a,2}\frac{dw_{a,2}}{dt}=\dot{m}_s(w_{a,1} - w_{a,2}) - \sum_{i=1}^{n}\int_0^l U_m \rho_a (w_a - w_{d,i})dy \tag{18}$$

where w_s is the supply air humidity ratio, w_a is the internal room air humidity ratio, and $w_{d,i}$ is the liquid desiccant humidity ratio at element i of the indoor pipe with length l.

CHAPTER III

SYSTEM OPTIMIZATION

The main concern is to find the optimum value for each design parameter for each prediction period for a total simulation time of 12 hours. The simulation is performed on the selected system based on the optimization timeframe with an acceptable accuracy and the optimization process is applied for a prediction period of one. The value of a single design parameter and internal loads are fixed during a prediction period and may vary from one prediction period to another.

3.1. Genetic Algorithm

Modeling the liquid desiccant system with the CC/DV system is complex task with multi-variables involved, several equations are coupled and indirect relations between different parameters are present. Since several non-linear equations are solved, it is advised to use a revolutionary derivative free optimization tool that follows the direct search technique. The simplest optimization tool that could be used for the proposed case is the genetic algorithm optimization tool because it is derivative free, based on numerical analysis, and is somehow efficient if compared with other derivative based optimization schemes. Moreover, it fetches the global minimum of a specific function.

Our choice of using a derivative free algorithm to solve the optimization problem is implemented by the evolutionary genetic algorithm. Genetic algorithms are good methods to solve search and optimization problems, and are based on the genetic process of biological organisms. Genetic algorithms are growing more and more popular and extending from simple design optimization to online process control. The power of the genetic algorithm arises from its robustness, being acceptably good in finding the near optimum solution and being relatively quick. An efficient optimization technique uses two techniques to find the optimal solution, exploration and exploitation, and this is what genetic algorithm does.

3.1.1. The Genetic Algorithm terminology

- The algorithm starts by seeding a set of trial combinations of the variables to be optimized and calculating the numerical value of the objective function for each combination selected. This set is called the "Initial Population".
- The set of numerical values calculated for the objective function from the first trial, is then evaluated according the "Fitness Criteria". The fitness criteria can be defined as the condition for the objective function numerical value to be better convenient than its pears.
- Based on their fitness, some combinations in the previously seeded set are chosen to be "Parents". Parents then undergo either "Crossover" or "Mutation" procedure to produce "Children". Most fitted parents simply jump to the next generated population without any change; such parents are referred as "Elite".
- The current population is replaced by children from the next population.

- Elite children are defined as the individuals of the current generation which have the best fitness values. These individuals will automatically survive to a next generation. Crossover children are formed by combining the pair of parent's vectors. Mutation children are produced by changing the vector or introducing mutations to a single parent.
- The algorithm stops when the "Tolerance" in the objective function values between two generations is less than a certain set error value, or when the maximum number of "Generations" is exceeded, or by any other defined "Stopping Criteria".

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Modeling the liquid desiccant system with the LDMC-C/DV system is complex task with multi-variables involved and coupled equations with indirect relations existing between different parameters. Since several non-linear equations are solved, a revolutionary derivative free optimization tool was used following the direct search technique (House and Smith, 1995). The simplest optimization tool that could be used for the proposed case is the genetic algorithm optimization tool because it is derivative free, based on numerical analysis, and is somehow efficient if compared with other derivative based optimization schemes (Mossolly et al. 2009; Keblawi et al. 2011; Niu et al. 2002). Moreover, it fetches the global minimum of a specific function.

For the optimized control strategy used for the LDMC-C/DV system and by referring to Fig.1, the variables that are used for cost optimization are the desiccant temperature at the inlet of dehumidification membrane (T_1), the supply air temperature (T_s) and the supply air mass flow rate ($\dot{m}_s$). Each variable in the optimization routine has a lower and an upper bound that defines the interval where the genetic algorithm searches for the optimal cost and are based on physical considerations. The bounds for the different variables are as follows:

- The supply air temperature is considered to vary between 17 °C and 23 °C to reduce draft and ensure thermal comfort (Keblawi et al. 2011).
- The supply air mass flow rate is considered to vary between 0.08 and 0.26 kg/s such that the carbon dioxide content inside the room is within ASHRAE's recommendations for the minimum supply flow rate to meet occupancy schedule for ventilation requirements (Keblawi et al. 2011).
- The temperature of liquid desiccant entering the dehumidifier range between 12 °C and 20 °C.

3.3. Optimization Constraints

There are several non-linear constraints that are applicable to the system. These constraints are related to thermal comfort issues and physical constraints. The constraints may be redefined in the following list (ASHRAE 2007):

- The Percent People Dissatisfied (PPD) inside the occupied zone is less than 10%. This condition is required for the human thermal comfort. The closer the PPD is to zero, it is assumed that the occupants inside the room would be more comfortable noting that the smallest percent people dissatisfaction is 5%.

- The temperature gradient shall not be greater than 2.5 °C/m. This condition is required so that there would not be any large gradients in the human body. Large gradients cause thermal discomfort for living beings.
- The stratification height inside the room is greater than 1 m. This condition is required so that the stratified air does not mix with the breathing zone.
- The relative humidity inside the occupied zone is less than 76%.

3.4. The Fitness Function

To enhance the speed of the genetic algorithm, the electrical cost function and constraints are combined in a single cost function by using penalty functions, thus the fitness cost function may be written as

$$J = \alpha_{elec} \times C_{elec} + \alpha_{tgrad} \times C_{tgrad} + \alpha_{stratH} \times C_{stratH} \qquad (19)$$

The coefficients $\alpha_{elec}, \alpha_{tgrad}$, α_{stratH}, and α_{RH}in the above function are the weight factors for electrical cost (C_{elec}), temperature gradient cost (C_{tgrad}) and stratification height cost(C_{stratH}) respectively and are set according to the system parameter. For the current system, the weight factors are set to unity.

3.4.1. Electrical Cost

The objective function that is to be optimized is the total operational cost of the system; this cost may be divided into:

- The cost of running the chiller.

- The cost of running the pump.
- The cost of running the fan.
- the cost of auxiliary heater

Note that in this work the cost is given in units of KW.

a- Chiller Cost:

The chiller is the main energy consuming component in our system. The cost of the chiller is calculated by using the following equation:

$$E_{chiller} = \frac{Q_{cooling}}{1000 \times COP_{chiller}} = \frac{Q_d + Q_s}{1000 \times COP_{chiller}} \tag{20}$$

Where Q_d is the cooling energy needed for liquid desiccant, Q_s is the cooling energy needed for the supply air and $COP_{chiller}$ is the coefficient of performance of the chiller assumed at value of 3.5.

The cooling energy needed for the supply air in kW is given by:

$$Q_s = \dot{m}_s (h_a - h_s) \tag{21}$$

where $\dot{m}_s$ is the supply air mass flow rate, h_a is the enthalpy of supply air, and h_s is the enthalpy of the outdoor air.

The cooling energy needed for liquid desiccant in kW is given by:

$$Q_d = \dot{m}_d (h_{d,7} - h_{d,1}) \tag{22}$$

where $\dot{m}_d$ is the liquid desiccant mass flow rate, $h_{d,7}$ is the enthalpy of liquid desiccant exiting heat exchanger C, and $h_{d,1}$ is the enthalpy of liquid desiccant entering the dehumidifier.

b- Fan cost

The fan cost is directly related to the air mass flow rate by using the following equation (House and Smith 1995):

$$E_{fan} = \left(\frac{200}{3600}\right)\left(\frac{\dot{m}_s^3}{1.29}\right) \tag{23}$$

where $\dot{m}_s$ is supply air mass flow rate.

c- Pump cost

The pump cost is related to the pump head, liquid desiccant mass flow rate, and the efficiency of the pump. The power of the pump is evaluated by multiplying the pressure difference by the volumetric flow rate and dividing the result by the pump efficiency; mathematically the pump cost equation may be written as:

$$E_{pump} = \frac{\dot{m}_d \times g \times H_p}{1000 \times \eta_{pump}} \tag{24}$$

Note that the pump cost is not included in the cost function, since the desiccant mass flow rate is constant.

d- Auxiliary heater cost

The auxiliary energy needed in case the solar energy is not sufficient is calculated as below:

$$E_{aux} = \dot{m}_d (h_{d,6} - h_{d,5}) \tag{25}$$

where $\dot{m}_d$ is the liquid desiccant mass flow rate, $h_{d,4}$ is the enthalpy of liquid desiccant exiting heat exchanger B, and $h_{d,5}$ is the enthalpy of liquid desiccant entering the regenerator.

Therefore the total energy consumed can be expressed by the following equation:

$$C_{elec} = E_{fan} + E_{chiller} + E_{aux} \tag{26}$$

3.4.2. The Constraints Cost Functions

The cost function for the constraints may be written such that they could be incorporated into the online cost function in a simple manner. These constraints are related to their respective threshold values such that when the constraints are violated, the fitness function would have a very large value.

a- The stratification height cost is bounded to be larger than 1m, thus the stratification height cost is

$$C_{stratH} = \exp\left(\frac{H_{\min}}{H}\right) - 1 \tag{27-a}$$

b- The temperature gradient is to bounded to be less than 2.5 K/m, thus the temperature gradient cost function may be written as

$$C_{tgrad} = \exp\left(\frac{dT/dz}{\left(dT/dz\right)_{\max}}\right) - 1 \tag{27-b}$$

The exponential term helps to restrict the cost function variables between their upper and lower bounds limit. If the value of each variable exceeds the upper bound, the value of the cost function will increase dramatically and the set of variables will be rejected. The use of exponential form to control the constraints' cost and the integration of the constraint terms within the objective function expression were implemented by Keblawi et al. (2011) and Hammoud et al. (2014).

In this study, the optimization will be done for 3 different cases to justify the significance of using LDMC-C in combination with displacement ventilation system. In case A, desiccant temperature will be fixed on 16 °C where supply air temperature and supply mass flow rate will vary to meet the indoor comfort conditions. Case B is similar to the first case with lower desiccant temperature fixed on 13 °C. In case C, the optimizer will select optimal set points of supply air temperature, desiccant temperature and supply mass flow rate to minimize the total operational cost and maintain the indoor comfort and IAQ.

Optimization will be done for two different set of operational variables: In the first set the desiccant temperature will be fixed and the supply air and flow rate will be changed to study the effect of low membrane temperatures on the heating and cooling energy requirements of the integrated system. In the second set the three operational variables will be allowed to change to minimize the total operational cost and maintain the indoor comfort and IAQ.

The component models were coupled with the space model and its requirements. The LDMC-C/DV system model needs the following inputs: the space dimensions, wall properties, ambient conditions, dimensions of the permeable pipes, and the physical properties of the desiccant. The wall model is solved using an explicit finite difference scheme with convergence satisfied for a simulation time step of 60 seconds and a wall increment of 100 divisions, and the average heat gained by the wall is calculated for every hour. The pipes of dehumidifier and regenerator are discretized into 100 elements in order to account for the concentration and temperature variation along the length of the pipe. The solar system is simulated for a period of 12 hours, by taking the needed regeneration energy along with the hourly solar flux as inputs. The model assumed initial walls temperatures in order to compute the energy gained by the multi layers walls of the space model. The dehumidifier model will compute the amount of sensible load removed by the membrane system. After that the space model computes the room temperature and updates the internal wall temperature. The iterative procedure is repeated until convergence is reached. Convergence is considered to be reached if the difference in the norm of the wall temperatures between two iterations is less than 0.01 K. After calculating the internal walls temperatures, the space model computes the stratification height and the space is then divided into 2 zones and the temperature and humidity are calculated at every zone. The regenerator model is used to calculate the regeneration temperature and the temperature at the exit of the regenerator. Knowing the inlet and exit temperatures at the dehumidifier and regenerator, the cooling and heating energies of the desiccant cycle are evaluated. The storage tank model integrated to the solar collector estimates the storage tank temperature at each hour, if this temperature is not sufficient to cover the thermal energy needed, auxiliary energy is used.

All the component models considered in this work; mainly DV system coupled with chilled ceiling and the model of desiccant membrane integrated with solar system; were validated extensively in the literature (Ayoub et al. 2006; Mossolly et al. 2008; Keblawi et al. 2009; Keniar et al. 2015). The CC/DV space model was validated in previous studies of the authors as well as in optimization studies of the operation of the CC/DV system (Ayoub et al. 2006; Mossolly et al. 2008; Keblawi et al. 2009). The membrane ability to transfer heat and mass from an indoor space to a liquid desiccant was validated experimentally by Keniar et al. (2015) and their model is adopted in this current study. The simulation model results will be used in the optimization methodology that assesses performance of the system to determine optimal operational settings for given space load and ambient conditions.

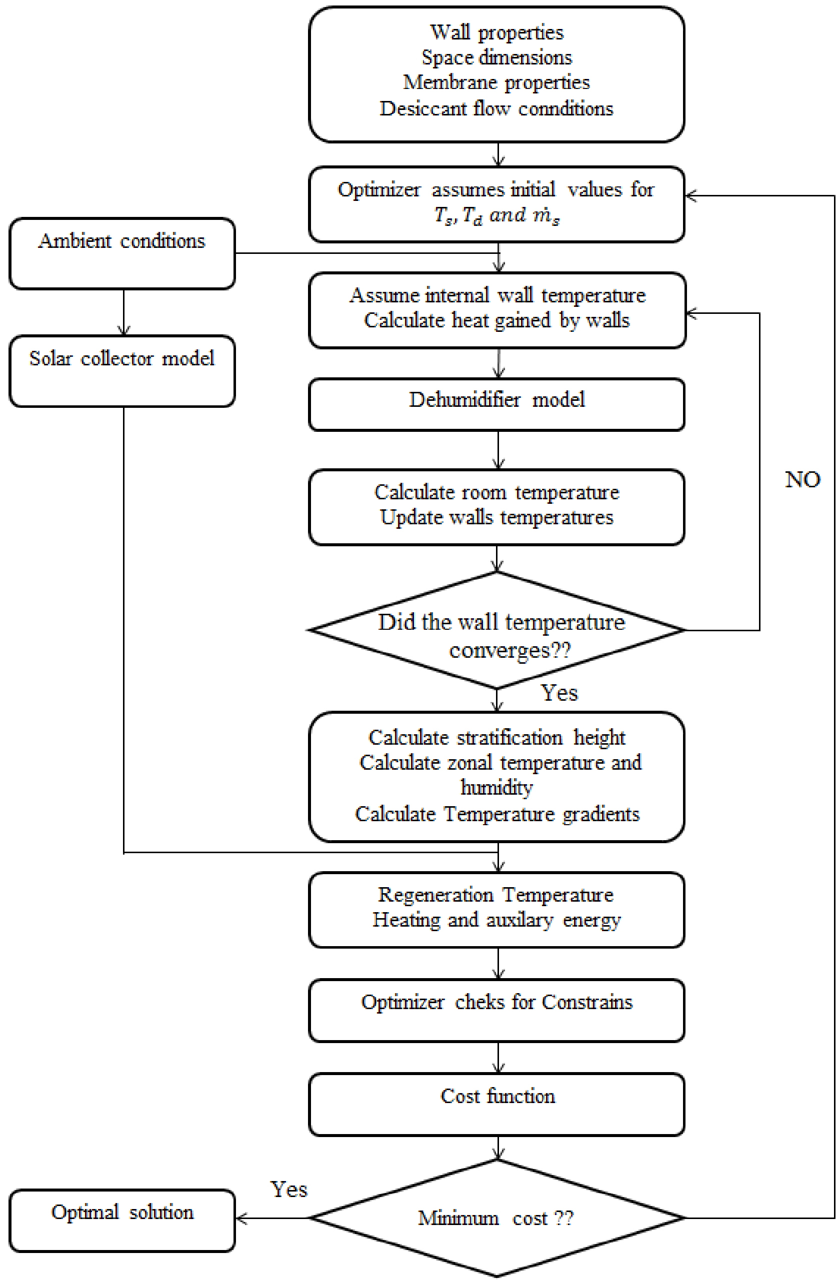

Figure 2: system optimization flow chart.

CHAPTER IV

CASE STUDY

4.1. System Description

In order to investigate the efficiency of the optimized system, it was sized and simulated while implementing optimized operation for an office space in hot humid climate of Beirut City. The office consists of four walls, two external at the south and the west direction, where the east and the north ones are considered partitions. Each wall is composed of three layers of typical Lebanese building construction with properties provided in Table 1, and overall thickness of 219 mm.

Table1: Wall Layer Properties

Layer Number	Layer Material	Thickness [mm]	Density [kg/m^3]	Specific Heat [kJ/kg-K]	Thermal Conductivity	R-Value [m^2-K/W]
1	Gypsum Board	15.88	800.9	1.09	0.161	0.09862
2	LW Concrete Block	101.590	608.7	0.84	0.38	0.26681
3	Face Brick	101.59	2002.3	0.92	1.33	0.07626

The office is considered to be a 5m × 5m × 3m room with 25 m^2 floor area and 3m height. Figure 3 illustrates the sensible and latent load schedules inside the office. The maximum number of space occupants is 6 and the space is occupied from 7 a.m. till 6 p.m. The maximum latent load due to external and internal generation from occupants is 410 W. The total internal and external sensible load that the system should remove reaches a peak of 99 W/m^2, which is higher than the ability of the DV system.

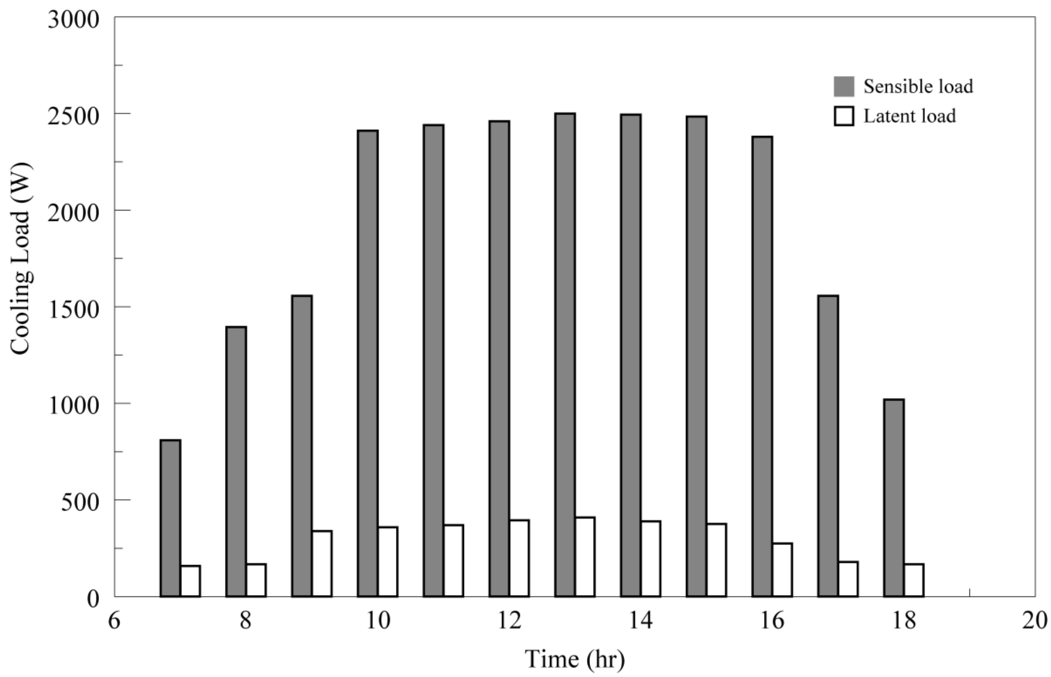

Figure 3: The hourly variation of the internal sensible and latent loads of the office space.

The liquid desiccant pipes are installed horizontally at the ceiling of the room; these pipes are made of a porous material, polypropylene, which is permeable to water vapor only and not liquid. The desiccant system is required to remove the moisture from the upper zone of the space In addition to a part of the sensible load while considering the indoor thermal comfort. The length of each pipe is equal to the ceiling length which is 5 m and the number of pipes is 120 each with diameter of 2.3 cm.

For the liquid desiccant entering the pipes inside the room, the dehumidification increase as desiccant concentration increase, while the maximum allowable value of concentration is 40% to avoid the problem of crystallization of $CaCl_2$ (Audah et al. 2001; Keniar et al. 2015). In this study, the desiccant concentration is set to a fixed value of 38% to decrease the thermal energy needed for regeneration. The desiccant mass flow rate is chosen to be 3.6 kg/hr per pipe (Keniar et al. 2015) in order to cover the whole pipe area and to ensure that enough heat and moisture is transferred to the pipes. The properties of the regeneration tubes (diameter and number) is the same to that of dehumidification pipes; but with length of 1 m to prevent decrease in the desiccant temperature during interaction with the ambient air, because of the large difference in temperature between the regeneration temperature and ambient temperature, and low desiccant flow rate inside the tubes. The regeneration temperature varies to provide the 38% desiccant concentration at the inlet of the dehumidifier. The ambient conditions for the month of August are shown in Table 2 and obtained from the records of the Mechanical Engineering Department at the American University of Beirut.

Table 2: ambient conditios for month of August

Hour	**Ambient Temperature °C**	**Ambient Humidity Ratio kg_w/kg_a**	**Solar Flux W/m^2**
7:00	24.6	0.0177	343.9
8:00	25.8	0.0177	504.7
9:00	27.3	0.0176	646.2
10:00	28.8	0.0177	746.3
11:00	29.9	0.0178	769.3
12:00	30.7	0.0177	790.6
13:00	31.3	0.0178	688.8
14:00	31.6	0.0177	559.6
15:00	31.6	0.0177	403.1
16:00	31	0.0178	241.3
17:00	30.1	0.0177	193.5
18:00	29.1	0.0177	94

After calculating the thermal energy required for regeneration the liquid desiccant, the storage tank and the parabolic solar panels can be sized. A total of 4.8 m ×1.2 m parabolic collectors and a storage tank with 1 m^3 can supply the system with required thermal energy. The specifications of a single parabolic collector are presented in Table 3.

Table 3: Solar collector specifications

Design Parameter	Value
Collector tube diameter	0.06 m
Transparent cover tube diameter	0.09 m
Flow rate of water in collector	0.0535 Kg/s
Width of the collectors	1.2 m
Length of the collectors	2.4 m

The optimization model will be simulated for a typical day in August. The transient simulation is performed at an appropriate time step of one hour over 12 hours for the space model and for the water storage tank. The value of the internal loads and other variables are assumed fixed during every prediction period, and vary from one prediction period to another. The optimization tool searches for the optimum values of the set points for every prediction period. The genetic algorithm maximum population size is chosen to be 60 with a 0.6 crossover value, 10^{-3} tolerance function, and a stall time limit of 150. The weight factors for electrical cost, temperature gradient cost, stratification height cost, PPD cost, and relative humidity cost are set to be unity.

4.2. Results and Discussion

4.2.1. *Optimized operation for different cases of the case study*

In order to check the effect of lowering ceiling temperature on total cooling and heating energy of the system, optimization is done for two cases with two variables (supply temperature and flow rate) while desiccant temperature is fixed. In the first case, case A, the desiccant temperature is fixed on 16 °C while in case B the desiccant temperature is set to 13 °C. The optimization is then done on case C where the supply temperature, desiccant temperature and flow rate are set to be variable. Table 4 presents all operational set points for Cases A, B, and C. Comparisons will be made with the conventional CC\DV operating with the same operational conditions.

Table 4: Hourly optimized set points for supply air temperature and supply air flow rate, and the ceiling temperature for cases A and B.

Hour	Case (A) Desiccant temperature 16 °C			Case (B) Desiccant temperature 13 °C			Case (C)		
	Supply air temperature (°C)	Ceiling temperature (°C)	Supply air mass flow rate (kg/s)	Supply air temperature (°C)	Ceiling temperature (°C)	Supply air mass flow rate (kg/s)	Supply air temperature (°C)	Desiccant temperature (°C)	Supply air mass flow rate (kg/s)
7	20.19	17.77	0.086	22.28	15.50	0.083	21.24	18.12	0.091
8	19.41	18.01	0.114	21.90	16.00	0.099	21.17	17.23	0.127
9	19.36	18.21	0.137	20.90	16.20	0.119	21.01	16.13	0.139
10	18.44	18.46	0.178	20.41	16.72	0.151	20.83	13.35	0.153
11	18.18	18.50	0.200	20.31	16.79	0.175	20.73	12.72	0.157
12	18.18	18.51	0.204	20.47	16.80	0.186	20.42	12.64	0.176
13	18.03	18.53	0.221	19.27	16.83	0.198	19.83	12.57	0.186
14	18.04	18.49	0.219	19.38	16.79	0.186	19.97	12.61	0.181
15	18.09	18.43	0.214	19.44	16.73	0.166	20.22	13.32	0.175
16	18.23	18.41	0.208	20.10	16.68	0.164	21.24	18.12	0.091
17	19.33	18.29	0.092	21.07	16.38	0.085	21.17	17.23	0.127
18	19.47	17.80	0.087	21.52	15.66	0.082	21.01	16.13	0.139

For case A, as presented in Table 4, the supply temperature of the air was at highest value in the early morning, 20.19°C, since the loads inside the space and the outdoor temperature was low at this period (see Figure 3). Conversely, and for the same reason, the air mass flow rate were low for the beginning of the day with 0.086 kg/s. By increasing the loads in the space and the outdoor air temperature, the supply air temperature decreases to reach minimum values of 18.03°C at the mid of the day, while the supply air mass flow rate increases to maximum values of 0.221 kg/s to remove the additional sensible loads in the space .The supply temperature increases again at the afternoon by decreasing the space loads and ambient conditions as shown in Figure 3, while the mass flow rate decreases to reach 0.087 kg/s at the end of the day. The low loads at the early morning and late afternoon as well as the low desiccant temperature, resulted in low ceiling temperature at these periods, but it showed an increase with the increase of the space loads. For case B, the trend of variation of the supply temperature, ceiling temperature and mass flow rate were similar to that in case A. Since the desiccant temperature in case B is lower than desiccant temperature in case A, the ceiling temperature was lower for case B. as a result, the supply temperature was higher and the supply mass flow rate was lower in case B to maintain the indoor comfort conditions.

The cooling and heating energy needed for cases A and B are calculated and presented in Figure 4. As shown in Figure 4, the energy needed for cooling the supply air and the liquid desiccant as well as the heating energy needed for LDMC-C/DV system increases as loads in the space and ambient temperature increases for both cases A and B. The DV cooling energy was less for case B, because of the higher supply temperatures and lower mass flow rates. The desiccant in case B has lower temperature than the desiccant in case A, so it needs more cooling energy, while the total cooling energy is less in case B with lower desiccant temperature. In case B, decreasing the desiccant temperature resulted in a 10% decrease of the total cooling energy of the LDMC-C/DV system. The heating energy needed for desiccant regeneration in case B was more than heating energy needed in case A since the desiccant is able to carry more sensible and latent load from the space at lower temperatures. For these reasons, it is better to use liquid desiccant membrane as a chilled ceiling in the CC/DV systems to reach lower ceiling temperatures with less cooling energy. The LDMC-C/DV system can overcome the problem of water vapor condensation and a lower ceiling temperature can be used in this system.

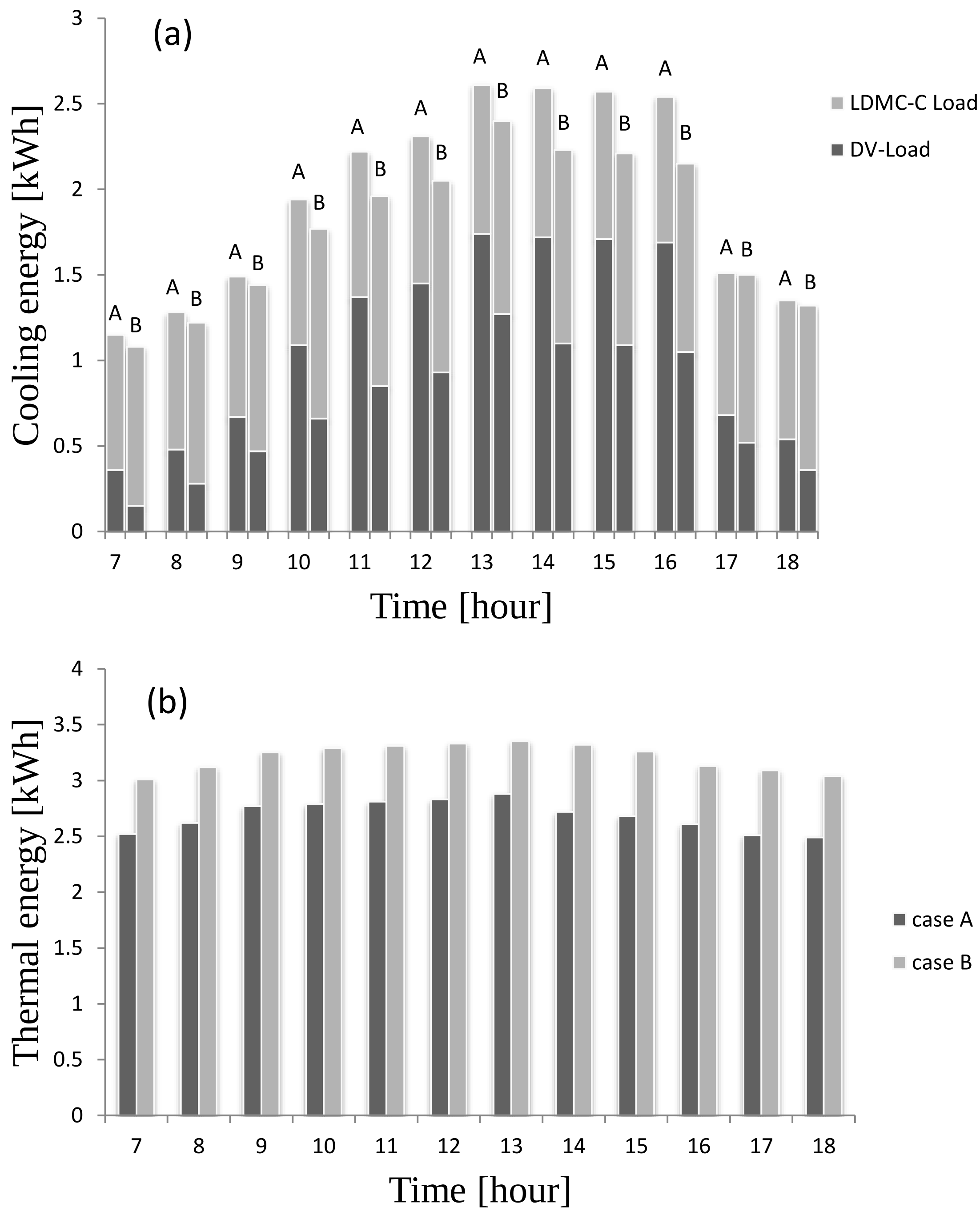

Figure 4: Plots of (a) cooling energy and (b) thermal energy needed for both cases A and B.

For case C of optimized operation in which three variables change (see Table 4), the supply air temperature varied between 21.24°C and 19.83°C, the liquid desiccant temperature between 18.12°C and 12.57°C, and the supply air mass flow rate between 0.091 kg/s and 0.186 kg/s. At low ambient and low sensible and latent loads in the space, the supply air temperature and the desiccant temperature were at high values, and the supply air mass flow rate was low. When both sensible and latent loads reached their peaks at the mid-day at 2,500W and 410W respectively, the optimizer increased the supply flow rate and decreased the supply air temperature and the desiccant temperature, allowing the system to remove the higher load from the space. As a result, the cooling energy for DV system and liquid desiccant and the heating energy needed for desiccant regeneration increased. During the whole operation hours of the system, the available solar energy was sufficient to regenerate the liquid desiccant; therefore no auxiliary heating energy was used. Figure 5 shows the variation with time of (a) the room temperature at the occupied zone, (b) the vertical temperature gradient *dT/dz* and (c) the stratification height for case C. It is clear that the indoor air temperature varied between 24°C and 25.3°C and reached the maximum value at 16:00 where the sensible load and outdoor air temperature are at a high values. The indoor temperature is within the acceptable thermal comfort zone recommended by ASHRAE Standard 55 (Turner, 2011).which recommends a space temperature between 24°C and 28°C during the summer (Turner, 2011). Moreover, the stratification height is higher than 1 m and this condition ensure that the recirculated air in upper zone does not mix with the occupied zone. The temperature gradient *dT/dz* is less than 2.5°C/m which means that thermal comfort is maintained in the office. Therefore, all constraints are met at any time of the operation.

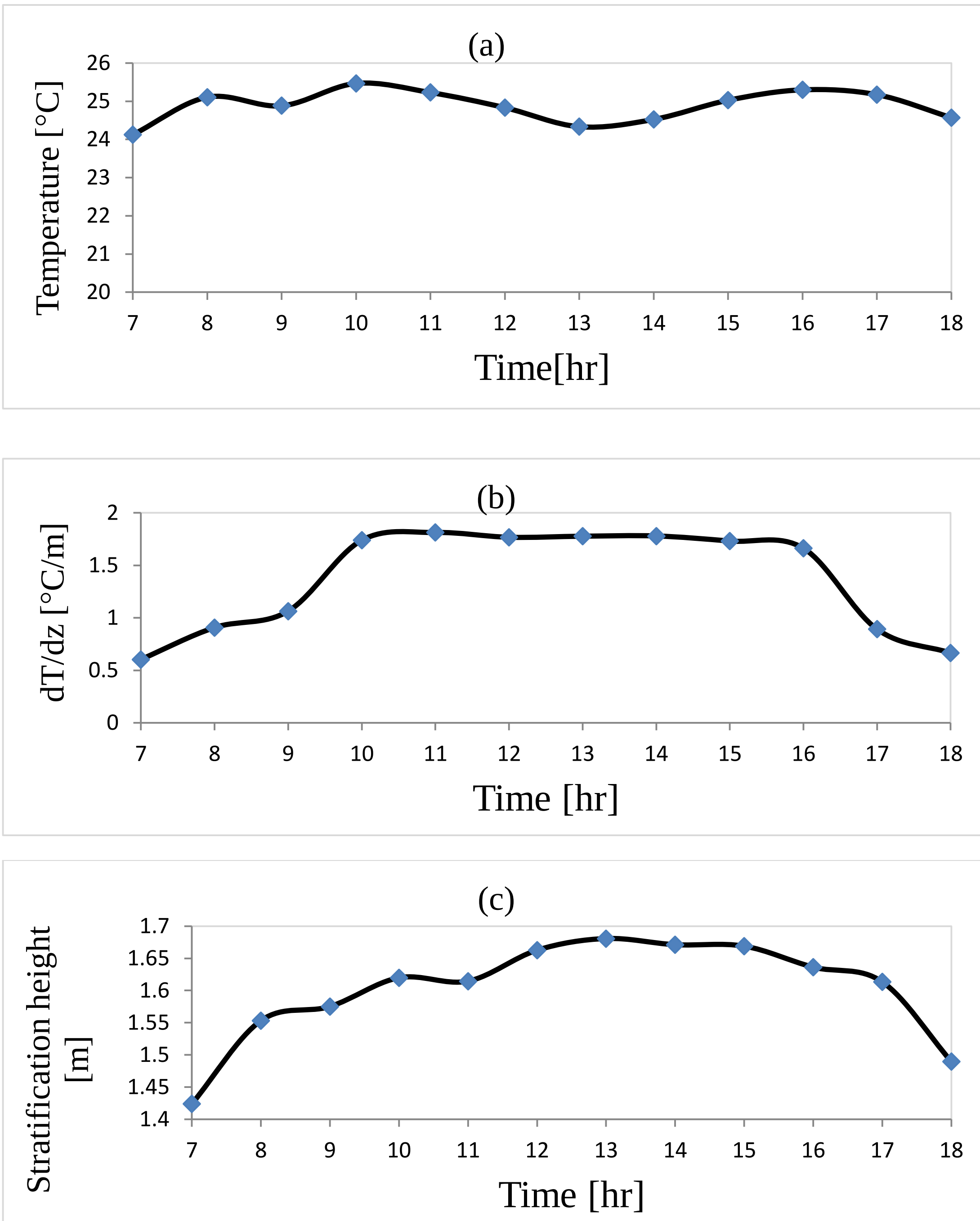

Figure 5: plots of (a) room temperature, (b) vertical temperature gradient *dT/dz*, and (c) stratification height for case C.

By comparing the variation of variables and total energy needed in case C with the previous two cases A and B, it is clear that optimizer selected lower desiccant temperature than the previous cases A and B (see Table 4). Moreover, the supply temperature was higher and the supply flow rate was lower for most of the hours in case C. as a result, the total cooling energy in case C is less than it in cases A and B. As a conclusion, implementing optimization strategy with three variables (supply temperature, desiccant temperature and supply flow rate) will led to energy saving more than the case of optimization with 2 variables (supply temperature and supply flow rate).

4.2.2. Performance comparison between LDMC-C/DV and conventional CC/DV

Another comparison is done between LDMC-C/DV optimized system and a conventional CC/DV system operating with same input conditions. The conventional CC/DV systems have the same supply air temperature, supply air mass flow rate and average ceiling temperature of the LDMC-C/DV system of case C. The DV cooling energy for the LDMC-C/DV system, as shown in Figure 6, is less than that of a CC/DV conventional system operating with same conditions, since the latent load and a part of the sensible is removed in DV section of the conventional system. On the other hand, the CC cooling energy is lower for the conventional system due to removing of humidity by liquid desiccant in the LDMC-C/DV system which requires regeneration process and more cooling energy. The total cooling energy of the LDMC-C/DV system is less than total cooling energy of the CC/DV system; as a result a smaller chiller size is used. The thermal energy needed for desiccant regeneration in the LDMC-C/DV system is higher than the heating energy needed for supply air reheat in the conventional CC/DV systems; however solar system is used as a free renewable energy source to make the LDMC-C/DV more efficient.

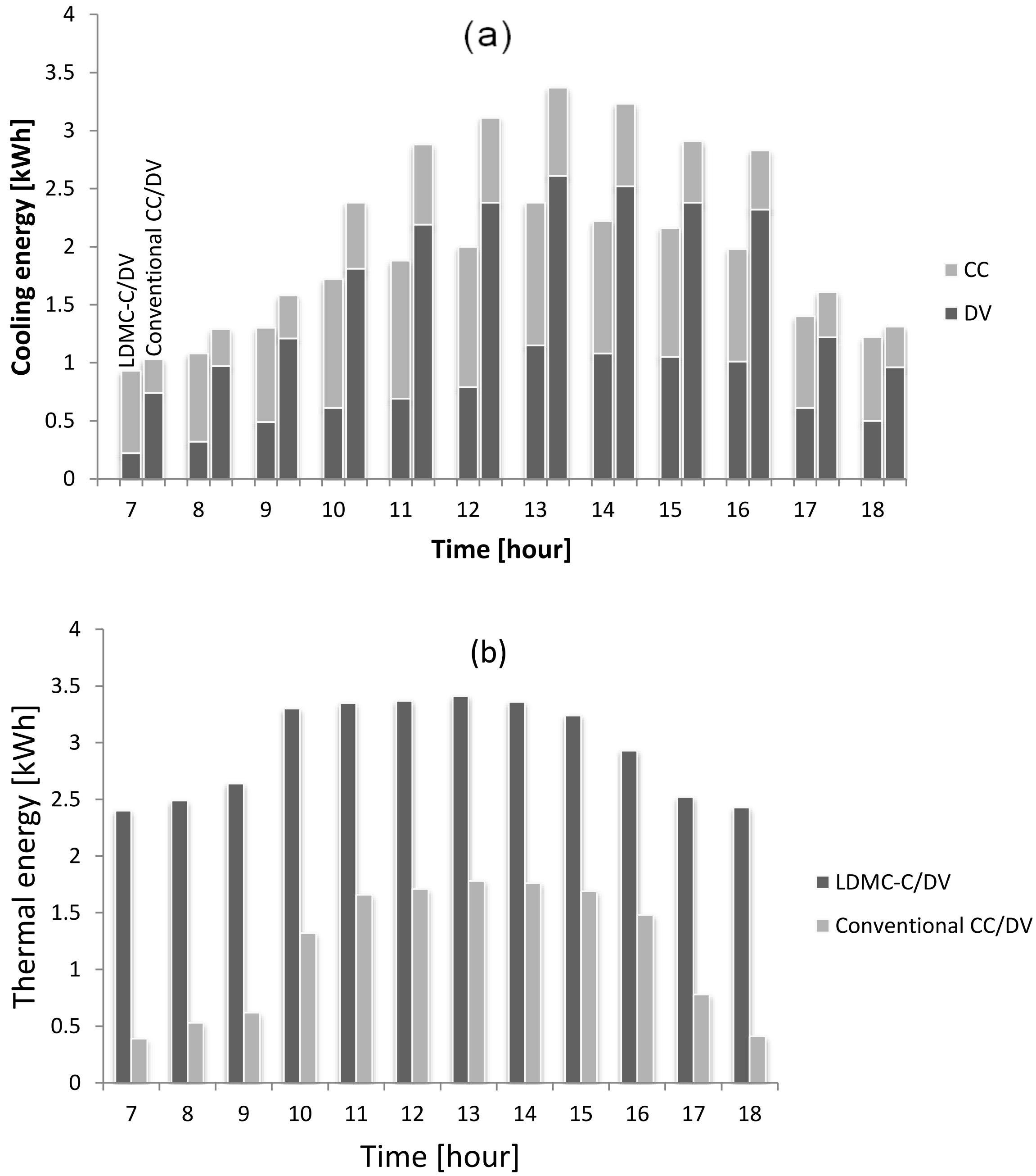

Figure 6: Plots of (a) cooling energy and (b) thermal energy needed for case C of LDMC-C/DV system with comparison with conventional CC/DV system.

The variations of optimized variables of case C (supply air temperature, supply air mass flow rate and ceiling temperature) are compared to previous studies on conventional CC/DV (Mossolly et al. 2008) at the same load range and same space of our case study. The study of Mossolly et al. (2008) investigated optimized operation of conventional CC/DV using genetic algorithm optimization tool to obtain optimal set points of supply air temperature, supply air mass flow rate, and ceiling temperature while maintain the indoor air quality and thermal comfort . Table 5 shows the values of optimized variables reported by Mossolly et al. (2008), the optimized set points obtained from this study and the total energy consumption of both systems. The energy consumption include the energy consumed by chiller, reheat and fan for the study of (Mossolly et al. 2008), while it include energy of chiller and fan in the current study were no reheat energy is used. The regeneration energy is not included in the energy consumption calculations since it is based on free source solar energy.

Table 5: Hourly optimized set points for supply air temperature, supply air flow rate and ceiling temperature as obtained from study of (Mossolly et al. 2008) and from the current study.

Hour	Load (W/m^2)	Study of (Mossolly et al. 2008)				Current study			
		Supply air temperature (°C)	Ceiling temperature (°C)	Supply air mass flow rate (kg/s)	Total energy consumption (kWh)	Supply air temperature (°C)	Ceiling temperature (°C)	Supply air mass flow rate (kg/s)	Total energy consumption (kWh)
7	**32.4**	20	21.5	0.19	5.3	21.24	19.65	0.091	1.15
8	**55.8**	19.7	21.1	0.195	5.35	21.17	19.31	0.127	1.58
9	**62.3**	19.5	20.8	0.192	5	21.01	18.78	0.139	2.01
10	**96.5**	19.2	20	0.196	5.2	20.83	17.21	0.153	2.95
11	**97.6**	19.1	19.9	0.205	5.75	20.73	16.70	0.157	3.46
12	**98.7**	19	19.8	0.205	6	20.42	16.35	0.176	3.95
13	**99.9**	18.85	19.65	0.206	6.1	19.83	15.98	0.186	4.40
14	**99.8**	18.9	19.5	0.21	6.2	19.97	16.09	0.181	4.14
15	**99.6**	18.5	19.2	0.22	5.8	20.22	16.84	0.175	3.71
16	**95.5**	18.4	19.45	0.021	6.3	20.56	17.63	0.169	3.57
17	**62.3**	18.7	19.2	0.212	6.2	20.82	19.09	0.147	2.07
18	**41.2**	19.2	19.5	0.213	5.6	21.13	19.81	0.109	1.51

The trend of variation of the supply air temperature and ceiling temperature of the LDMC-C/DV system has the same trend of (Mossolly et al. 2008) study. It has a decreasing trend as sensible load in the space increased, and an increasing one as the sensible load decreased. In addition, relatively low supply air temperature and ceiling temperature were attained at peak loads, 19.8°C and 15.9 °C respectively. Moreover, the variation of the supply air flow rate had an increasing trend reaching peak values at peak load hours and a decreasing one as the sensible load in the space decreased. Although the trend of variation of optimized variables is the same compared with (Mossolly et al. 2008)a wider range of variation of the chilled ceiling temperature is recognized when compared with the conventional CC/DV optimized operational values, where it reaches 15.9 °C at the peak load hours since ceiling temperature is not constrained by condensation of water vapor. Moreover, the supply air temperature is higher and supply flow rate is lower most of the times because of the higher load removed by the ceiling in the current study. In addition to that, reheat energy is used to remove humidity from the supply air to prevent condensation on ceiling. For these two reasons, the energy consumption is less in current study. The control strategy implemented by Mossolly et al. (2008) was based mainly on variation of supply and ceiling temperatures while considering the inlet air humidity limit to prevent the problem of condensation on the chilled ceiling, while in our current LDMC-C/DV system, the optimal values of supply air temperature, supply air flow rate and ceiling temperature varied to maintain the indoor air quality and thermal comfort without the limitation on the inlet air humidity.

4.3. Economic Analysis

The economic feasibility of the proposed LDMC-C/DV system is compared to a conventional CC/DV system using the same input and operating conditions of case C (Figure 7). The supply air temperature, supply air mass flow rate and the average ceiling temperature of the conventional CC/DV system are chosen to be the same of the optimized values obtained for the LDMC-C/DV system at each hour. The comparison of the energy consumption for the two systems is done for a typical days for May, June, July, August and September. The total monthly energy consumption can be calculated by multiplying the daily total energy by the numbers of days of the month. Higher values of electric power consumption are considered in the conventional CC/DV system, due to additional cooling and reheat energy. In the conventional CC/DV system, the humidity is removed by reducing the supply air temperature below the ceiling dew point, to prevent water vapor condensation on the ceiling. After reaching the required humidity ratio, the supply air will be reheated to the specified supply temperature.

Figure 7 shows the total electrical energy consumption of the LDMC-C/DV system and the conventional CC/DV system during the five months of the cooling season. The maximum electric energy consumed as well as maximum energy saving was observed during the month of August due to the high levels of ambient temperature and humidity ratios. By comparing the total energy consumption of the tow systems; it’s clear that the LDMC-C/DV system consumes 49% less than the conventional CC/DV system. Therefore, by considering electrical cost of 0.15 $/kWh in Beirut, the amount of electricity saving over the operational period of 5 month per year is calculated to be about $390.

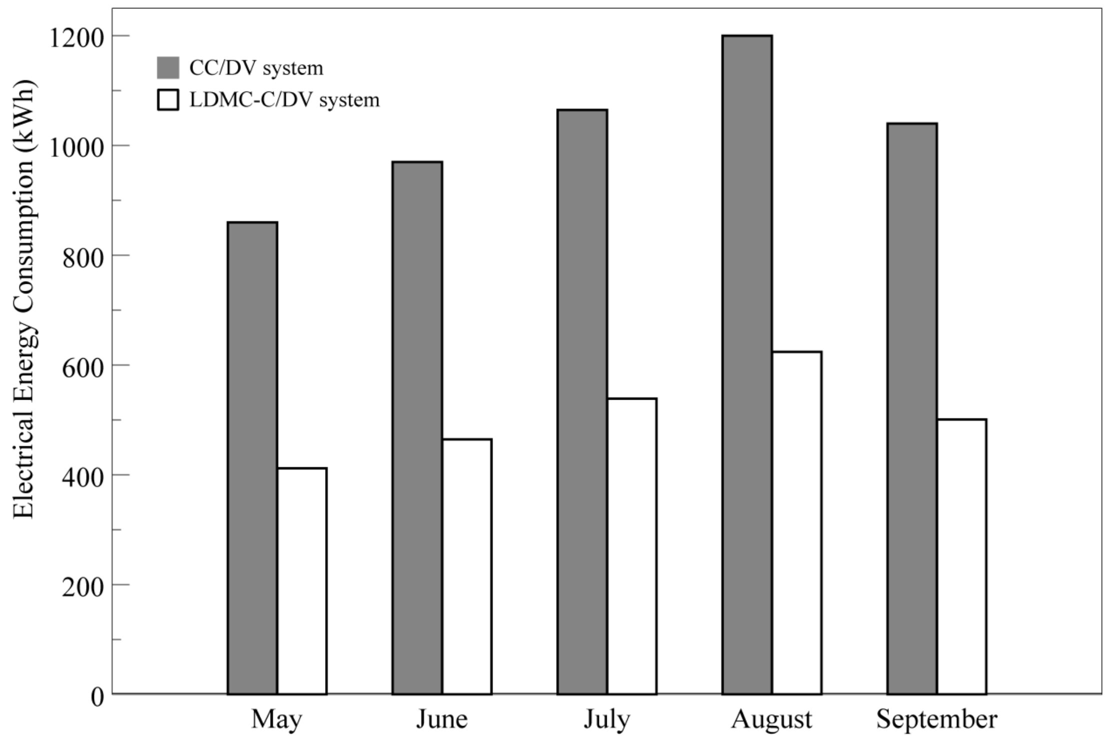

Figure 7: Comparison of energy consumption over cooling season.

The cost of installing the proposed system will be compared to the cost of installing the conventional CC/DV system in addition to the cost electricity consumed every year, in order to determine the payback period. The cost of the proposed system includes the initial costs of the dehumidification/regeneration pipes and the parabolic solar panels. The length of parabolic collectors required to supply the thermal energy for desiccant regeneration is 4.8 m with a 1.2 m width of. The estimated cost of the solar collectors is \$2000 (Audah et al. 2011). As for the membrane, the material used costs 5 $\$/m^2$ (Larson et al. 2006), which implies a total material cost of \$220. The cost of installation and fabrication of the membrane pipes is estimated as \$500. The cost of the three heat exchangers is about 900\$.

Therefore the total cost of proposed system is \$ 3,620. At the other hand, the cost of installing chilled ceiling panels is taken to be 36 \$/m^2, so the initial cost of the panels is \$ 900. The additional cooling required by the CC/DV system was estimated to be 5 kW, resulting with \$500 as incremental initial chiller cost. The initial cost of using the proposed system is thus \$2,220 more than the CC/DV system. In order to determine the present worth value of both systems equations (19a) and (19b) are used.

$$P = A_0 \frac{1-\left(\frac{1+i}{1+d}\right)^n}{d-i} \qquad i \neq d \tag{28a}$$

$$P = A_0 \frac{n}{1+d} \qquad i = d \tag{28b}$$

In equations (19a) and (19b), *P* represents the present worth, n is the number of years, d is the discount rate and i is the annual rate at which electricity costs are increasing. Using a discount rate of 2% and an annual rate of electricity price increase as 3%, the payback period is calculated to be 5 years and 11 months when implementing the proposed LDMC-C/DV system in this study.

4.4. Conclusion

In this study, a dehumidification membrane system has been used as a ceiling panel for heat and moisture transfer inside a typical office in Beirut city to control thermal comfort and IAQ of an indoor space and to prevent water vapor condensation as in conventional CC systems. The desiccant system has been coupled with displacement ventilation system to enhance the indoor air quality. Optimization strategy based on genetic algorithm has been used to predict the optimal set points of the combined system for improving operating performance while considering the thermal comfort. The results showed that the combined system can reach its goals by enhancing indoor air quality and thermal comfort, with low ceiling temperatures and less cooling energy. The economic analysis of the combined system showed that 49 % energy saving has been achieved compared to the conventional CC/DV system.

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