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PERFORMANCE EVALUATION OF DIFFERENT AIR-SIDE ECONOMIZER CONTROL METHODS FOR ENERGY EFFICIENT BUILDING

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Abstract Energy saving in all-air air-conditioning systems can be significantly improved by using free cooling application. This application, commonly known economizer cycle, is used when outside conditions are suitable, being cool enough to be used as a cooling medium. The most important and characteristic parameter for free cooling is local climatic features. However, detailed analysis of weather data determines the free cooling potential of a region. In this study, energy analysis of the free cooling applications with the outdoor air temperature and enthalpy controls are carried out for a commercial building conditioned by all-air air conditioning system in Antalya. The results obtained from the sample building analysis show that for all working modes the highest energy saving ratio is obtained by enthalpy control strategy.
Keywords: Energy saving; Free cooling; Air-Conditioning; Economizer cycle; Bin number; Antalya.

ENERJİ VERİMLİ BİNALARDA FARKLI EKONOMİZER KONTROL YÖNTEMLERİNİN PERFORMANS DEGERLENDİRİLMESİ

Özet Tam havalı iklimlendirme sistemlerinde serbest soğutma uygulaması ile önemli ölçüde enerji tasarrufu sağlanabilir. Ekonomizer çevrim olarak da bilinen bu uygulamada, uygun koşullardaki dış hava soğutucu olarak kullanılır. Serbest soğutma için en önemli parametre yerel iklim özelliğidir. Ancak, meteorolojik verilerin detaylı analizi ile bir bölgenin serbest soğutma potansiyeli belirlenir. Bu çalışmada, tam havalı bir iklimlendirme sistemi ile iklimlendirilen Antalya'daki bir ticari bina için, dış hava sıcaklık ve entalpi kontrollü serbest soğutma uygulamalarının enerji analizi yapılmıştır. Örnek uygulamada, en yüksek enerji tasarrufu oranı, entalpi kontrol stratejisi ile elde edilmiştir.
Anahtar kelimeler: Enerji tasarrufu; Serbest soğutma; İklimlendirme; Ekonomizer çevrimi; Bin sayısı; Antalya.

NOMENCLATURE

ACEEE	The American Council for an Energy-Efficiency Economy
AHU	Air handling unit
ASHRAE	American society of heating, refrigeration and air-conditioning engineers
CAV	Constant-air-volume
COP	Coefficient of performance of the chiller unit
HVAC	Heating ventilating and air-conditioning system
LCC	Life-cycle cost
M_{full}	The maximum air mass flow rate [m^3/h]
M_{out}	The amount of fresh air [m^3/h]
N_{bin}	Bin number [h/month]
PWR	Present worth cost
Q_{chil_full}	Full cooling capacity of the chiller [kW]
Q_{chil_part}	Hourly cooling demand on chiller (at part-load) [kW]
Q_{fan_full}	Full capacity of the fans [kg/h]
Q_{fan_part}	Hourly load of fans (at part-load) [kg/h]
Q_{room}	The building cooling load,
RTS	Radiant time series cooling load calculation method
SHR	Sensible heat ratio of the building
TES	Thermal energy storage
T_{in}	Indoor air temperature [°C]
T_{out}	Outdoor air temperature [°C]
T_S	Supply air temperature [°C]
TS 825	Turkish standard for thermal insulation in buildings
VAV	Variable-air-volume
W_{chil_bin}	Energy consumption of chiller unit for each bin interval [kWh]
$W_{chiller}$	Monthly total energy consumption of chiller unit [kWh/month]
W_{fan}	Monthly total energy consumption of fans unit [kWh/month]
W_{fan_bin}	Energy consumption of fans for each bin interval [kWh]
W_{full}	Energy consumption of fans at full load [kWh]
W_{in}	Absolute humidity of the indoor air [kg/kg]
W_{out}	Absolute humidity of the indoor air [kg/kg]
W_{total}	Monthly total energy consumption of fans and chiller [kWh/month]
W_{total_bin}	Total energy consumption of fans and chiller for each bin interval [kWh]
η_{fan}	Fan efficiency [%]
φ_{in}	Relative humidity of the indoor air [%]

INTRODUCTION

Economizer cycle

Central air-conditioning systems can be categorized according to the transfer of heating and cooling energy between central plants and conditioned building-spaces. There are four basic system categories: all-air systems, air- and water-systems, all-water systems and packaged unitary equipment systems. All-air systems have been widely used in air-conditioning system applications. Two main air-distribution systems associated with all-air air-conditioning systems are constant-air-volume (CAV) and variable-air-volume (VAV) systems. Energy saving is one of primary reasons that VAV systems are very popular design choices today for some commercial buildings and many industrial applications (Aktacir et al, 2006). With these systems, the volume of the air delivered is reduced whenever operating loads are less than design loads. However, CAV systems are sometimes also preferred due to their low-cost and easy operation advantages.

In all-air air conditioning system, air to the room is supplied by mixing the minimum amount of fresh air for ventilation with returning air, so that maximum energy saving can be obtained. Free cooling system is also one of the well-known ways to reduce the energy consumption of an air-conditioning system. Free cooling application, commonly known economizer cycle, is used when outside conditions are suitable; that is, when outside air is cool enough to be used as a cooling medium (ASHRAE, 2008). The free-cooling effect of a building is related to many factors such as the outdoor climate condition, internal heat source intensity, the building's site layout, the applied air flow rate, the efficient coupling of air flow with the thermal mass of the building and ventilation mode, etc (Zhang et al, 2006; Geros et al, 1999; Geros et al, 2005). One of the most strong effect and characteristic parameters is local climatic feature (Budavi, 2001; Pfaferott et al, 2003; Aktacir, 2007). The detailed analysis of local weather data is required to determine free cooling potential of a building. The potential of free-cooling represents a measure of the capability of ventilation to ensure indoor comfort without using mechanical cooling systems (Ghiaus and Allard, 2006). Free cooling is not alternative of mechanical cooling, it must be thought as complementary and supportive application for air conditioning system. Because free cooling alone cannot provide year-round comfort in the building, it would only be effective as part of a hybrid system (Olsen and Qinyan, 2003). Recent research has shown that night ventilation techniques, when applied to massive buildings, can significantly reduce the cooling load of air conditioned buildings. The thermal mass of the structure plays a major role in the efficiency of the night ventilation system by reducing temperature swings. Thermal mass may reduce the temperature swings between night and day, or reduce the indoor daily mean temperature during warmer days by having decreased the thermal mass temperature during previous colder ones (Zimmermann and Anderson, 1998; Yang and Li, 2008). Dinçer and Rosen (2001) pointed out that thermal energy storage (TES) technology can successfully be applied to many existing systems all over the world, based on their work on TES systems for improving cooling capacity. Geros et al. (1999; 2005) have both experimentally and theoretically investigated the cooling potential of night ventilation techniques applied to buildings operating under different conditions and with variable air flow rates. Results of these studies have shown that under A/C conditions, the expected energy conservation vary from 48% to 94% for different air flow rates. Besides, an improvement of the cooling potential of a mechanical ventilation system can be achieved by the integration of short term latent heat thermal energy storage (Zalba et al, 2004; Arkar et al, 2007; Arkar and Medved, 2007). Arkar et al. (2007) presented the results of an investigation into the free cooling efficiency in a heavyweight and lightweight low energy building using a mechanical ventilation system with two latent heat thermal energy storages, one for cooling the fresh supply air and the other for cooling the re-circulated indoor air. They have found that the free cooling technique enable a reduction in the size of the mechanical ventilation system, providing more favorable temperatures, better thermal comfort conditions, and fresher air for the occupants. Corgnati and Kindinis (2007) have analyzed the effectiveness of free cooling ventilation strategies coupled with thermal mass activation to reduce peak cooling loads in Mediterranean climate. Their work has indicated that the night ventilation, even better if coupled with mass activation, can drastically help on reducing summer cooling loads and improving thermal comfort. In supporting side of these studies, the work by Turnpenny et al. (2000) was also valuable to mention especially in terms of describing construction and testing of a novel latent heat storage unit incorporating heat pipes embedded in phase change material for use with night ventilation.

Economizer controls

Air-side economizers use ducts to move the air, dampers to control the flow of the various airstreams, and control systems to control the dampers (ASHRAE, 2008; Honeywell, 2000). The control systems can be integrated with the air conditioning system so that the operation of both the economizer and the air conditioning system can be optimized to reduce energy consumption (ACEE, 2009). Economizer controllers combine all logic functions, control algorithms and switching into a single package. Sensors (temperature or enthalpy), damper actuators, power supply and other miscellaneous wires are connected to the controller. Honeywell (www.honeywell.com) is one of the biggest economizer controller manufacturers. There are two types of control systems in common usage today: dry bulb temperature sensors and enthalpy sensors. Dry bulb economizers only control the outdoor air dampers based on temperature. If it is a cool but rainy day, the outdoor air will be brought in and extra cooling capacity will be required to dehumidify it. Dry bulb systems are appropriate in dry climates, but may cause problems where high humidity and moderate to high outdoor temperatures occur together. For these situations, "enthalpy" controls are better, since they consider the work required to dehumidify the outdoor air (ACEE, 2010).

There are two temperature and enthalpy control strategies available: single (fixed) system and differential dual sensor system. Fixed sensor systems shift from economizer to refrigeration cycle at a specific outdoor air temperature or enthalpy. The dual sensor system, known as differential dry bulb control or differential enthalpy control, allows the economizer system to compare the temperature or enthalpy of the outdoor air to that of the return air to accurately select the air stream with the lower temperature or enthalpy for air conditioning. These differential controls are relatively expensive, but they are very efficient method of controlling outdoor air usage since the return and outdoor air comparison is continuous and automatic year-round. Therefore, they can improve the performance of the economizer system. This can result in lower energy consumption of an air conditioning system. Thereby, free cooling system can improve the building energy performance. Wacker (1989) presented the results of their simulation that enthalpy economizer saved 5%-50% of compressor energy in comparison to dry-bulb economizer with switchover set point of 24 ^{o}C in different locations.

Main objective of this study is to determine the influence of outdoor air temperature and enthalpy controlled economizer cycle on an all-air air conditioning system used in hot and humid Mediterranean climate. For this reason, a commercial building in Antalya, a province in Mediterranean region of Turkey, associated with an all-air air conditioning system is considered. Energy and economic analysis of the free cooling application with the dual sensor controls (differential dry bulb control and differential enthalpy control) are performed for this sample building to examine influence of the parameters under consideration.

MATERIAL AND METHOD

Case study

A financial institution that is conditioned with air conditioning system (24°C dry bulb temperature and 50% relative humidity), located in Antalya (36.53 N latitude-30.42 E longitude), in which hot and humid Mediterranean climate is effective, was chosen as the sample building in this research. Antalya is the largest city on the Mediterranean coast of southwestern Turkey. The area is shielded from the cold northerly winds by the Taurus Mountain range, having a characteristically Mediterranean climate -with hot and humid summers and moderately warm and rainy winters. Around 300 days of the year are sunny, air temperature can climb as high as the 40°C range in July and August, and its location is shown on the Turkey map in Figure 1. The storey plan of the sample building is presented in Figure 2.

Figure 1. Map of Antalya in Mediterranean coast of Turkey.

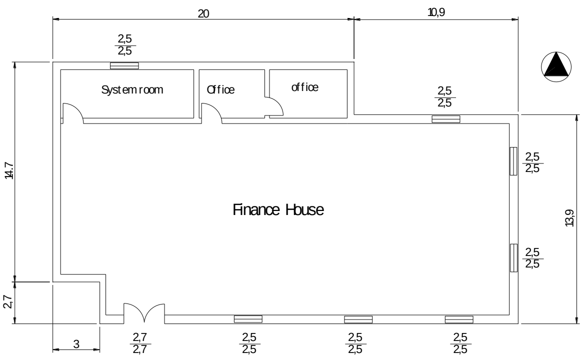

Figure 2. The storey plan of the financial institution conditioned with all-air air conditioning system.

The total heat transfer coefficients and surface areas belonging to the elements of the sample building, determined according to TS 825, are presented in Table 1. TS 825, Turkish standard for thermal insulation in buildings, is in force in Turkey. Windows are double glass, but they are not solar control glass. In hot climates, solar control glass can be used to minimize solar heat gain and help control glare. The operating hours of the financial institution is 9:00-24:00; and the institution has 12 personnel. It is additionally accepted that an average of 10 people requested transactions between the hours of 9:00-12:00 and 13:00-17:00. In Table 2, the internal heat loads of the building, caused by the lighting system, equipments (i.e. computer, printer, photocopy and fax) and people are presented in detail.

Table 1. Surface area and overall heat transfer coefficient of building elements

Building element	Overall heat transfer coefficient (W/m^2K)	Surface area (m^2)
External wall	0.5	239
Windows and Door	2.8	51
Ceiling	0.3	491

Table 2. Building heat gains and other properties

Internal Heat Gain	Load (W)		Split ratio		Diversity ratio
	Sensible	Latent	Radiant	Convective	
People	1650	1210	0.58	0.42	0.85
Lighting	4160	0	0.59	0.41	0.70
Equipment	2380	0	0.30	0.70	0.80

The design cooling loads of the building were calculated by using the values in July 21st; and the changes in those values during the day can be seen in Figure 3. As can be seen from the figure, the maximum design load and sensible heat ratio of the building (SHR) were measured at 16:00 as 27.39 kW and 0.96, respectively. The design values of Antalya in summer are; 39°C dry bulb temperature, 19.6 g/kg absolute humidity, and 11.4°C daily temperature difference (CME, 2002). All-air air conditioning systems consist of cooling unit, fresh air, exhaust and mixed air dampers, aspirator, and ventilating fan. The schematic display of the examined all-air air conditioning system with dual sensor control system is given in Figure 4. As shown from the figure, dual sensor enthalpy/temperature control is equipped with the same outdoor air enthalpy/temperature sensor and an additional second enthalpy/temperature sensor in the return air. Differential enthalpy/temperature controls compare the enthalpy (temperature) of the outdoor air with the enthalpy (temperature) of the return air and change from outdoor air to cooling unit whenever the outdoor air enthalpy/temperature is greater than the return air enthalpy/temperature.

The design values of the air conditioning system were calculated, according to the previously determined design specifications, by means of psychometric analysis. According to these calculations, the design capacity of the cooling unit was found as 34.28 kW; and the total air volume was determined as 10124 m^3/h. An air-cooled cooling unit, with cooling capacity of 37.7 kW, coefficient of performance (COP) of 2.3, and total energy consumption of 16.4 kW at 39 °C nominal working conditions, was chosen from the product catalog of a local firm (Atlantic group, www.atlantikgrup.com). The cooling water regime and the flow rate of the cooling unit were 7°C/12°C and 1.64 l/s, respectively. The flow rates of the aspirator and ventilating fan were calculated to be 11874 m^3/h in the air handling unit; with total nominal power of 4 kW. The amount of total fresh air of the building was determined according to ASHRAE Standard 62 as 616 m^3/h. The supply air temperature, on the other hand, was chosen as 15 °C.

The initial capital cost of the system in the Turkish economic conditions was 34,200 $ without the use of economizer cycle whereas the cost was calculated as 36,690 $ and 37,700 $ when the economizer cycles were used, respectively, with the differential temperature control and with differential enthalpy control.

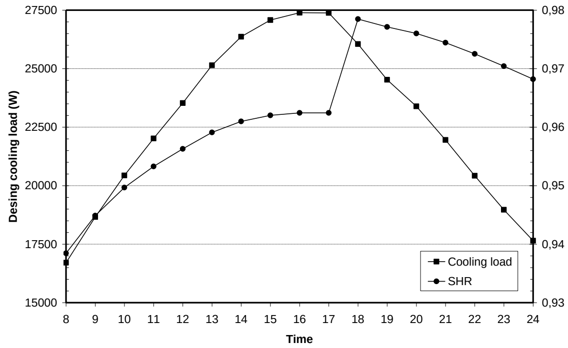

Figure 3. Hourly distribution of the design cooling load and sensible heat ratio (SHR)

Simulation of the system

There are three operating regions on psychometric chart for air side economizer cycle; mechanical cooling, free cooling and partial free cooling (Figure 5). In mechanical mode, when the outdoor air enthalpy/temperature is greater than the indoor air enthalpy/temperature, the outdoor and exhaust air dampers will be at their minimum opening and the mechanical cooling is needed and the cooling unit is operating at full capacity. In free cooling mode, when the outdoor air enthalpy/temperature is less than the supply air enthalpy/temperature, the mechanical cooling is not needed, 100% outdoor air is used and the free cooling capacity of outdoor air is utilized directly, resulting in the cooling unit is shut off. In partial free cooling mode, when the outdoor air enthalpy/temperature is less than the indoor air enthalpy/temperature and greater than supply air enthalpy/temperature, 100% outdoor air is used and the cooling unit is operating at lower capacity due to reducing cooling load.

In this study, the simulation of an all-air air conditioning system with an economizer cycle was performed by using a specific computer code in FORTRAN language. The simulation of the conditioning system was carried out in three different working modes; by using the economizer cycle with outdoor air temperature control, by using the economizer cycle with outdoor air enthalpy control, and without the use of the economizer cycle. The flowchart of the economizer cycle with dual control system is presented in Figure 6. The input data which is required for the simulation as follows: outdoor air temperature (T_{out}) and absolute humidity (W_{out}), indoor temperature (T_{in}) and relative humidity (φ_{in}), supply air temperature (T_S) the amount of fresh air (M_{out}) and maximum air mass flow rate (M_{full}), building cooling load (Q_{room}), and SHR.

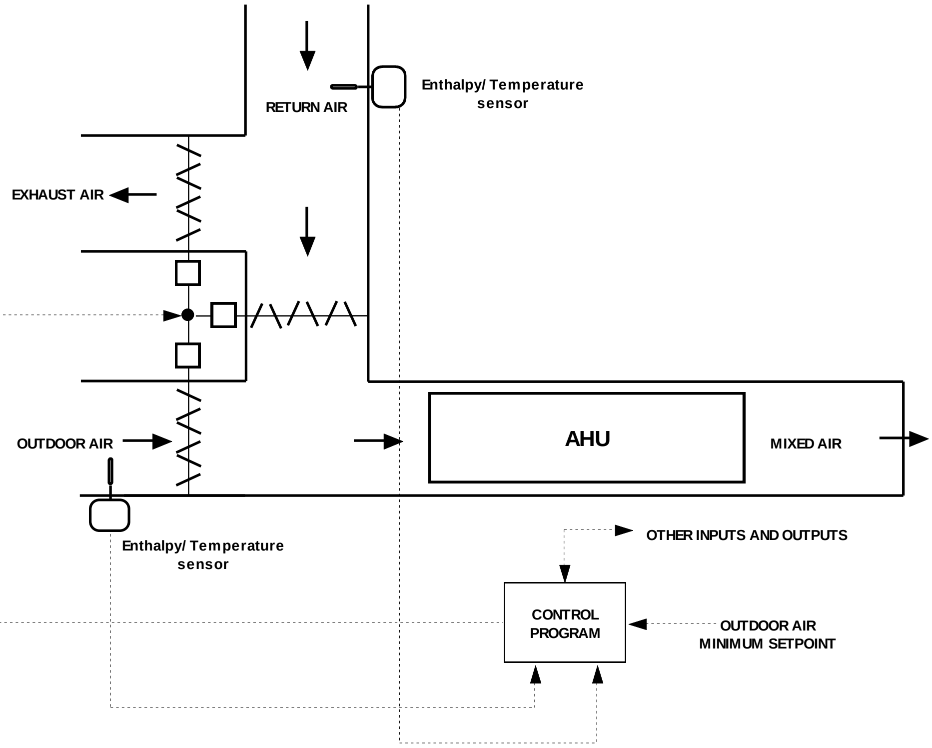

Figure 4. Schematic of an all-air air conditioning system with dual sensor control

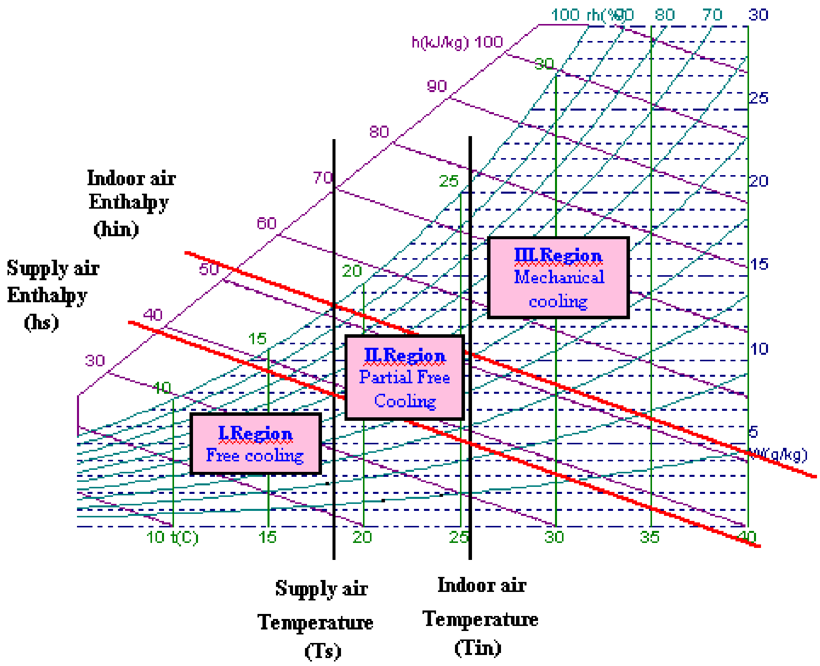

Figure 5. Operating regions on psychometric chart for air side economizer cycle

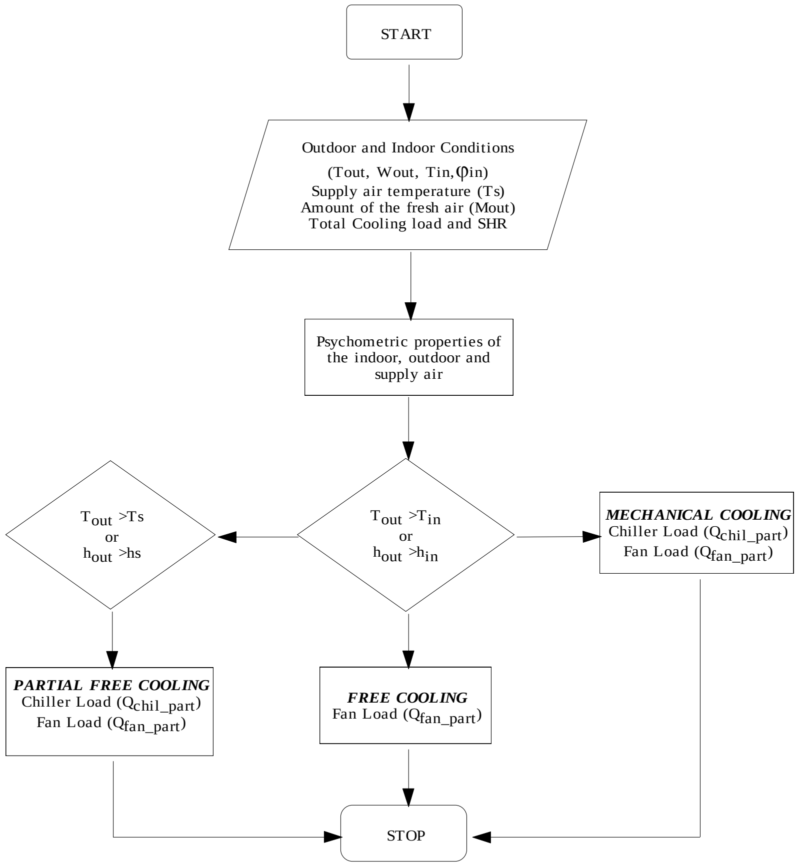

Figure 6. Flowchart of an all-air air conditioning system with dual sensor control

Energy analysis of the system

The long term hourly outdoor temperature and absolute humidity values, spanning over a time period of fifteen years between the years 1981 and 1995, of Antalya were used in this study. In the energy analysis, the outdoor air temperature values with increase of 3 ^{o}C and bin values obtained for 4 hours long in the corresponding day were used. The values in bin display the observance hour of a definite temperature interval in a certain time period. The bin values for Antalya are given in Table 3 (Bulut et al; 2001). As can be seen from the Table 3, 15°C/18 °C temperature interval during the time period of 09:00-12:00 was observed for 35 hours in the month of April whereas 15 °C/18 °C temperature interval was not detected in any time of the time periods in August.

In the first step of the energy analysis, the building cooling load for each bin interval was calculated by using the Radiant Time Series (RTS) calculation method suggested by ASHRAE (Spitler et al, 1997). The psychometric analysis of the system was performed in the following step; and the energy consumption of the chiller unit (W_{chil_bin}) and fans (W_{fan_bin}) for each bin interval (N_{bin}) was determined using equations below (Kreider and Rabl, 1994),

$$W_{chil\_bin} = \frac{Q_{chil\_full}}{COP} \left[0.023 + 1.429 \left(\frac{Q_{chil\_part}}{Q_{chil\_full}} \right) - 0.471 \left(\frac{Q_{chil\_part}}{Q_{chil\_full}} \right)^2 \right] \tag{1}$$

where Q_{chil_full} is full cooling capacity of the chiller unit [kW], Q_{chil_part} is the hourly cooling demand on chiller (at part-load) [kW], COP is coefficient of performance of the chiller unit.

$$W_{fan\_bin} = \frac{W_{fan\_full}}{\eta_{fan}} \left(\frac{Q_{fan\_part}}{Q_{fan\_full}} \right) \tag{2}$$

where Q_{fan_full} is full capacity of the fan [kg/h], Q_{fan_part} is the hourly load of fans (at part-load) [kg/h], W_{fan_full} is the energy consumption of fans at full load [kWh], η_{fan} donates the fan efficiency [%]. Q_{chil_full} and COP values of the chiller unit belonging to each bin interval were obtained from the manufacturer's catalog. For each bin value, partial loads of the chiller and fans (Q_{chil_part} and Q_{fan_part}) were calculated by using simulation program.

As final step, W_{total}; total energy consumption of the fan (W_{fan}) and chiller unit ($W_{chiller}$) was determined using equations below.

$$W_{fan} = \sum_{1}^{n} N_{bin} W_{fan\_bin} \quad (3)$$

$$W_{chiller} = \sum_{1}^{n} N_{bin} W_{chil\_bin} \quad (4)$$

$$W_{total} = W_{chiller} + W_{fan} \quad (5)$$

Economic analysis of the system

Life-cycle cost (LCC) analysis was performed using detailed load-profiles, by considering initial and operating costs to evaluate the economic feasibility of all-air air-conditioning systems based on economizer cycle and without economizer cycle. The present worth cost (PWC) method for LCC was used to evaluate total costs (Elsafty and Al-Danini, 2002; Economic Analysis Handbook, 1993). Results of the LCC analysis are directly affected by the economic measures. According to Turkey's recent economic situation, the following values are used in the simulation: annual interest rate of 18%, annual inflation rate of 10%, and electric energy price of 0.11 $/kWh.

RESULTS AND DISCUSSION

In this study, the simulation of an all-air air conditioning system was carried out for three different working modes: i) using economizer cycle with differential temperature control, ii) using economizer cycle with differential enthalpy control, iii) without using economizer cycle. The working hours of the air conditioning system, in the hours of labor (09:00-24:00), according to the economizer cycle type for the temperature interval and time periods during the month of April are presented in Table 4. As can be comprehended from the Table 4, for the temperature interval of 28.50 ºC, the conditioning system would perform mechanical cooling only for 7 hours for the economizer cycle with differential air temperature control. For the same interval, the system would perform 3 hours partial free cooling and 4 hours of

Table 3. Antalya for some months, the total bin numbers (h/month)

Month	Time Period	Temperature bin (ºC) 0/3	3/6	6/9	9/12	12/15	15/18	18/21	21/24	24/27	27/30	30/33	33/36	36/39	39/42
April	01-04	0	1	14	50	41	11	3	0	0	0	0	0	0	0
	05-08	0	1	13	37	39	21	7	2	0	0	0	0	0	0
	09-12	0	0	0	3	11	35	39	20	9	3	0	0	0	0
	13-16	0	0	0	2	7	29	47	22	9	4	0	0	0	0
	17-20	0	0	1	4	24	52	29	8	2	0	0	0	0	0
	21-24	0	0	4	27	54	29	5	1	0	0	0	0	0	0
May	01-04	0	0	0	11	37	49	22	4	1	0	0	0	0	0
	05-08	0	0	0	6	21	38	31	16	8	3	1	0	0	0
	09-12	0	0	0	0	1	5	23	41	33	13	6	2	0	0
	13-16	0	0	0	0	1	4	21	42	36	13	5	2	0	0
	17-20	0	0	0	0	3	17	43	40	16	4	1	0	0	0
	21-24	0	0	0	3	17	46	43	14	1	0	0	0	0	0
June	01-04	0	0	0	0	2	20	54	33	9	2	0	0	0	0
	05-08	0	0	0	0	1	10	27	32	28	15	6	1	0	0
	09-12	0	0	0	0	0	0	1	7	34	35	26	13	4	0
	13-16	0	0	0	0	0	0	1	8	35	38	24	11	2	1
	17-20	0	0	0	0	0	0	4	28	48	26	11	3	0	0
	21-24	0	0	0	0	1	4	32	58	21	4	0	0	0	0
July	01-04	0	0	0	0	0	1	16	56	35	14	2	0	0	0
	05-08	0	0	0	0	0	1	9	26	39	29	16	4	0	0
	09-12	0	0	0	0	0	0	0	0	7	37	32	29	15	4
	13-16	0	0	0	0	0	0	0	0	3	37	36	27	14	6
	17-20	0	0	0	0	0	0	0	2	27	54	26	11	3	1
	21-24	0	0	0	0	0	0	2	29	66	23	4	0	0	0
August	01-04	0	0	0	0	0	0	14	70	32	6	2	0	0	0
	05-08	0	0	0	0	0	0	12	37	35	25	11	4	0	0
	09-12	0	0	0	0	0	0	0	0	1	39	40	25	16	3
	13-16	0	0	0	0	0	0	0	0	0	34	51	23	12	3
	17-20	0	0	0	0	0	0	0	1	26	64	23	8	2	0
	21-24	0	0	0	0	0	0	2	31	73	15	2	1	0	0
September	01-04	0	0	0	0	1	20	57	35	7	0	0	0	0	0
	05-08	0	0	0	0	2	16	40	31	22	8	1	0	0	0
	09-12	0	0	0	0	0	0	0	3	22	53	29	10	3	0
	13-16	0	0	0	0	0	0	0	2	17	60	30	9	2	0
	17-20	0	0	0	0	0	0	4	27	56	27	5	1	0	0
	21-24	0	0	0	0	0	6	37	58	17	2	0	0	0	0

mechanical cooling the economizer cycle with differential enthalpy control. Energy conservation would be enabled for this condition since the system would work with lesser capacity as a result of the partial free cooling utilization for 3 hours. Though it may appear wasteful to cool outdoor air at a higher dry bulb temperature than return air, the savings are verifiable through psychometric calculations. When the total numbers for the month of April are examined, the system with outdoor air temperature control performs 137 hours of free cooling, 316 hours of partial free cooling, and 27 hours of mechanical cooling, whereas the system with differential enthalpy control performs 321 hours of free cooling, 155 hours of partial free cooling, and 4 hours of mechanical cooling (see Table 4).

Table 4. Operating hours in April for temperature interval–time periods; a) differential temperature control, b) differential enthalpy control

	Temp.	**Free cooling (h)**					**Partial cooling (h)**					**Mechanical cooling (h)**				
	℃	**9-12**	**13-16**	**17-20**	**21-24**	**total**	**9-12**	**13-16**	**17-20**	**21-24**	**total**	**9-12**	**13-16**	**17-20**	**21-24**	**total**
	28.50	0	0	0	0	0	0	0	0	0	0	3	4	0	0	7
	25.50	0	0	0	0	0	0	0	0	0	0	9	9	2	0	20
	22.50	0	0	0	0	0	20	22	8	1	51	0	0	0	0	0
a)	19.50	0	0	0	0	0	39	47	29	5	120	0	0	0	0	0
	16.50	0	0	0	0	0	35	29	52	29	145	0	0	0	0	0
	13.50	11	7	24	54	96	0	0	0	0	0	0	0	0	0	0
	10.50	3	2	4	27	36	0	0	0	0	0	0	0	0	0	0
	7.50	0	0	1	4	5	0	0	0	0	0	0	0	0	0	0
	Total	14	9	29	85	137	94	98	89	35	316	12	13	2	0	27
	28.50	0	0	0	0	0	3	0	0	0	3	0	4	0	0	4
	25.50	0	0	0	0	0	9	9	2	0	20	0	0	0	0	0
	22.50	0	0	0	0	0	20	22	8	1	51	0	0	0	0	0
	19.50	39	0	0	0	39	0	47	29	5	81	0	0	0	0	0
b)	16.50	35	29	52	29	145	0	0	0	0	0	0	0	0	0	0
	13.50	11	7	24	54	96	0	0	0	0	0	0	0	0	0	0
	10.50	3	2	4	27	36	0	0	0	0	0	0	0	0	0	0
	7.50	0	0	1	4	5	0	0	0	0	0	0	0	0	0	0
	Total	88	38	81	114	321	32	78	39	6	155	0	4	0	0	4

The working hours of free cooling, partial free cooling, and mechanical cooling during the hours of labor (09:00-24:00), belonging to the air conditioning system of the calculated building, is displayed in Table 5. As can be observed from the Table 5, the chiller unit of the conditioning system, which is at the option of classical working (mechanical cooling), is active for the entire hours of labor (496 hours) during the month of March. When the economizer cycle with outdoor air temperature control is utilized in the air conditioning system, the chiller unit would be disabled since the air conditioning system would perform 100% free cooling for 311 hours while the chiller unit would be activated at partial load for 184 hours in which partial free cooling is performed. The chiller unit would be fully activated for only one hour. When the economizer cycle with outdoor air enthalpy control is utilized in the air conditioning system, however, the chiller unit would be deactivated for the entire hours of labor (496 hours) since the system would perform 100% free cooling.

Differences in the working periods can occur between the enthalpy and temperature controlled economizer cycles, as can be seen from the Table 5, according to the working modes of the system. With the utilization of temperature controlled cycles, 311 hours of free cooling, 184 hours partial free cooling, and one hour of mechanical cooling is observed in March whereas free cooling was observed for the entire hours of labor when the enthalpy controlled system utilized. In the month of June, however, one hour of free cooling, 143 hours of partial free cooling, and 336 hours of mechanical cooling are observed when the differential temperature control system is utilized whereas mechanical cooling is performed for the entire period, except two hours of partial cooling, with the utilization of the differential enthalpy control system. Similar results are obtained for all the months. Such differences are caused by the hot and humid climate of Antalya representing the Mediterranean climatic features. Air transmitted indoor is determined by controlling the outdoor air and return air temperature values in the differential temperature control system. On the other hand, the energy value of air becomes higher for the high values of relative humidity; and air transmitted indoor does fall short for cooling the environment. Thus, the determination of free cooling potential in hot and humid regions should be performed according to the enthalpy control. Otherwise, oversizing may occur.

Table 5. Operating hours of the air-conditioning system with dual sensor control

Month	Mechanical Cooling (h)	Economizer Cycle (h)					
		Temperature Control			Enthalpy Control		
		Free	Partial	Mechanical	Free	Partial	Mechanical
March	496	311	184	1	496	0	0
April	480	137	316	27	321	155	4
May	496	25	339	132	25	398	153
June	480	1	143	336	0	2	478
July	496	0	33	463	0	0	496
August	493	0	34	461	0	0	493
September	480	0	137	343	0	3	477
October	496	40	296	160	48	343	105
November	480	210	260	10	245	229	6

As mentioned above, energy consumption values (of the chiller unit and fans) of the air conditioning system for four hours of working periods, between 9:00-24:00, and 3ºC temperature intervals were calculated according to the equations 1, 2, and 3 during 9 month period between March and November. For the working period of 9:00-12:00 in April, the energy consumption values of the cooling unit (chiller) and fans are presented in Tables 6 and 7, respectively.

The monthly total energy consumption values obtained from the air conditioning system, for the working period of 9:00-24:00, and for the conditions of performing classical cycle (mechanical cooling), utilizing economizer cycle with differential temperature and enthalpy controls are given in Table 8. Minimum energy consumptions of fans and chiller unit for all working mode were seen in March and November according to the table. It is a known fact that free cooling systems use outdoor air to reduce the cooling requirement when outdoor air is cool enough. Maximum energy saving observed in March is thus due to minimum operating hour of the chiller unit.

Monthly total operating cost of the air conditioning system for all working modes, and its saving ratio, described as operating cost of the air conditioning system for the economizer operating mode to operating cost of the air conditioning system for the classical cycle operating mode, were presented in Table 9.

As can be comprehended from the Tables 8 and 9, the amount of savings in energy becomes the highest for the months of March, April and October while no savings in the energy can be established during June, July, August, and September. The highest saving ratio for differential enthalpy control was determined during March (66%) in the air conditioning system utilized for the time period between 09:00-24:00 hours. In the case of the differential temperature control, this ratio was determined as 55%. The highest saving ratio in the sample application for all working modes was observed in enthalpy control strategy (see Table 9). On the other hand, no energy and operating cost savings in noteworthy

Table 6. Energy consumption of the cooling unit for 09:00-12:00 time period in April; a) differential temperature control, b) differential enthalpy control

	Temperature (ºC)	Q_{chil_part} (kW)	Q_{chil_full} (kW)	COP	W_{chil_bin} (kWh)
a)	28.50	15.96	42.20	3.08	6.63
	25.50	13.61	43.50	3.34	5.52
	22.50	3.14	44.90	3.63	1.53
	19.50	0	46.20	3.94	0
	16.50	0	47.60	4.26	0
	13.50	0	49.00	4.61	0
	10.50	0	50.00	4.98	0
b)	28.50	13.89	42.20	3.08	5.91
	25.50	7.94	43.50	3.34	3.49
	22.50	3.14	44.90	3.63	1.53
	19.50	0	46.20	3.94	0
	16.50	0	47.60	4.26	0
	13.50	0	49.00	4.61	0
	10.50	0	50.00	4.98	0

Table 7. Energy consumption of the fans for 09:00-12:00 time period in April; a) differential temperature control, b) differential enthalpy control

	Temperature (°C)	W_{fan_full} (kWh)	η_{fan} %	Q_{fan_part} kg/h	Q_{fan_full} kg/h	W_{fan_bin} (kWh)
a)	28.50	8	90	5978.44	11874	4.48
	25.50	8	90	5283.00	11874	3.95
	22.50	8	90	4600.91	11874	3.44
	19.50	8	90	3889.72	11874	2.91
	16.50	8	90	3241.00	11874	2.43
	13.50	8	90	1628.52	11874	1.22
	10.50	8	90	1059.83	11874	0.79
b)	28.50	8	90	5978.44	11874	4.48
	25.50	8	90	5283.00	11874	3.95
	22.50	8	90	4600.91	11874	3.44
	19.50	8	90	3750.85	11874	2.81
	16.50	8	90	2451.99	11874	1.84
	13.50	8	90	1628.52	11874	1.22
	10.50	8	90	1059.83	11874	0.79

Table 8. Monthly energy consumptions for all working modes of the air conditioning system.

	$W_{chiller}$ (kWh/month)			W_{fan} (kWh/month)			W_{total} (kWh/month)		
		Economizer cycle			Economizer cycle			Economizer cycle	
Month	**Mech. Cooling**	**Temp. Control**	**Enthalpy Control**	**Mech. Cooling**	**Temp. Control**	**Enthalpy Control**	**Mech. Cooling**	**Temp. Control**	**Enthalpy Control**
March	803.63	7.03	3.02	1025.14	809.93	618.91	1828.77	816.95	621.93
April	1274.90	352.75	320.81	1172.31	1120.55	1078.21	2447.21	1473.30	1399.02
May	2083.86	1844.71	1713.86	1494.67	1494.45	1494.45	3578.53	3339.16	3208.31
June	3184.33	3174.82	3172.56	1933.37	1933.37	1933.37	5117.70	5108.19	5105.93
July	4128.67	4128.67	4128.66	2287.77	2287.75	2287.77	6416.43	6416.42	6416.43
August	4177.58	4177.58	4177.58	2311.15	2311.33	2311.15	6488.73	6488.91	6488.73
September	3273.79	3271.67	3271.67	2011.10	2011.10	2011.10	5284.90	5282.77	5282.77
October	2289.77	1686.21	1686.21	1734.19	1732.38	1727.59	4023.96	3418.60	3413.80
November	1272.47	443.18	437.45	1141.24	1124.87	1103.19	2413.71	1568.05	1540.64

amounts was established during June, July, August and September (see both Tables 8 and 9) since the free cooling potential was very low in the region.

Annual variation of PWR values and payback time for all working modes of the air conditioning system were shown in Figure 7. Payback time of the air conditioning system with differential temperature control comparing with no economizer cycle was calculated as 8.30 years; whereas it was 11.49 years for differential enthalpy control. This is due to the fact that capital cost of the enthalpy controlled economizer cycle is higher than the dry bulb temperature controlled economizer cycle. Besides, heat recovery from ventilation system, VAV box and shading elements can be used in order to save energy in free cooling system. Thus, payback time of the air conditioning system is minimized.

CONCLUSIONS

In this study, efficiencies of utilizing differential temperature and enthalpy controlled economizer cycles were determined by evaluating free cooling potential in a selected commercial building located in Antalya/Turkey and utilizing an all-air air conditioning system. The free cooling potential of Antalya, showing typical Mediterranean climatic features, is determined to be changing in significant ratios according to the control systems. It is also found that the most suitable choice for the economizer cycle in the air conditioning system is the enthalpy controlled system since it results in the highest energy savings. The amount of savings in energy becomes the highest for the months of March, April, October and November while no savings in the energy can be established during June, July, August, and September.

Table 9. Monthly total operating costs for all working modes of the air conditioning system and saving ratio.

Month	Mechanical Cooling Operating Cost ($)	Economizer Cycle – Temperature Control – Operating Cost ($)	Economizer Cycle – Temperature Control – Saving Ratio (%)	Economizer Cycle – Enthalpy Control – Operating Cost ($)	Economizer Cycle – Enthalpy Control – Saving Ratio (%)
March	201.16	89.86	55	68.41	66
April	269.19	162.06	40	153.89	43
May	393.64	367.31	7	352.91	10
June	562.95	561.90	0	561.65	0
July	705.81	705.81	0	705.81	0
August	713.76	713.78	0	713.76	0
September	581.34	581.10	0	581.10	0
October	442.64	376.05	15	375.52	15
November	265.51	172.49	35	169.47	36

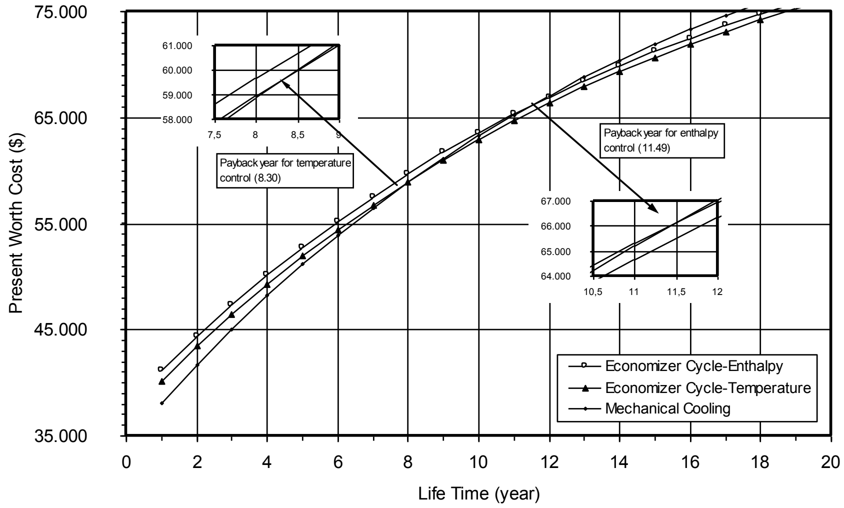

Figure 7. Annual variation of Present Worth Cost (PWR) values and payback time for all working modes of the air conditioning system.

As a result of these findings, it is suggested that the free cooling applications of HVAC system in the coastal provinces located in the Mediterranean countries (such as Turkey, Italy, Spain and Greece), which have hot and longer summers and warm winters, should be designed according to enthalpy control strategy. Savings in the amounts of building energy consumption can be established by considering the envelopes of the buildings and free cooling implementations. It is verified here that performing night cooling after the hours of labor in the commercial buildings would have a decreasing effect on the cooling load of the building.

REFERENCES

ACEEE, The American Council for an Energy-Efficiency Economy, Emerging technologies report: Air-side economizer control strategies, 2009. <www.aceee.org/emertech/2009_AirEcon.pdf> (accessed 25.03.2010)

ACEEE, The American Council for an Energy-Efficiency Economy, Online Guide to Energy-Efficient Commercial Equipment. <http://www.aceee.org/ogeece/ch3_air.htm> (accessed 10.02.2010)

Aktacir, M.A., Büyükalaca, O., Yılmaz, T., Life-cycle cost analysis for constant-air-volume and variable-air-volume air-conditioning systems, *Applied Energy*, 83(6), 606-627, 2006.

Aktacir, M.A., Potentials of the free cooling system of the all-air air-conditioning system in different climate regions of Turkey, *Journal of the HVAC& Sanitary Engineering*, 99, 66-74, 2007.

ANSI/ASHRAE Standard 62, *Ventilation for Acceptable Air-quality*, Atlanta (GA): American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc, 1989.

Arkar, C., Medved, S., Free cooling of a building using PCM heat storage integrated into the ventilation system, *Solar Energy*, 81, 1078-1087, 2007.

Arkar, C., Vidrih, B., Medved, S., Efficiency of free cooling using latent heat storage integrated into the ventilation system of a low energy building, *International Journal of Refrigeration,* 30, 134-143, 2007.

ASHRAE, *Fundamental-Handbook*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, 2008.

Budaiwi, I.M., Energy performance of the economizer cycle under three climatic conditions in Saudi Arabia, *International Journal of Ambient Energy*, 22(2), 83-94, 2001.

Bulut, H., Büyükalaca, O., Yılmaz, T., Bin weather data for Turkey, *Applied Energy*, 70, 135-155, 2001.

CME, Chamber of Mechanical Engineers, *Klima tesisatı* (in Turkish), technical publication: 296-2, Ankara, 2002.

Corgnati, S.T., Kindinis, A., Thermal mass activation by hollow core slab coupled with night ventilation to reduce summer cooling loads, *Building and Environment*, 42, 3285-3297, 2007.

Dinçer, I., Rosen, M.A., Energetic, environmental and economic aspects of thermal energy storage systems for cooling capacity, *Applied Thermal Engineering*, 21(11), 1105-1117, 2001.

Economic Analysis Handbook, Naval Publications, (SN 0525-LP-543-5200), Philadelphia 1993.

Elsafty, A., Al-Daini, A.J., Economical comparison between a solar powered vapor absorption air-conditioning system and a vapor compression system in the Middle East, Renewable Energy, 25, 569-83, 2002.

Geros, V., Santamouris, M., Karatasou, S., Tsangrassoulis, A., Papanikolaou, N., On the cooling potential of night ventilation techniques in the urban environment, *Energy and Buildings,* 37, 243–257, 2005.

Geros, V., Santamouris, M., Tsangrasoulis, A., Guarracino, G., Experimental evaluation of night ventilation phenomena, *Energy and Buildings,* 29, 141-154, 1999.

Ghiaus, C., Allard, F., Potential for free-cooling by ventilation, Solar Energy, 80(4), 402-413, 2006.

Honeywell Engineering Manual of Automatic Control for Commercial Buildings, Honeywell Inc, USA, 1997.

Kreider, J.F., Rabl, A., *Heating and Cooling of Buildings,* McGraw-Hill, New York, 1994.

Lazaro, A., Dolado, P., Marin, J.M., Zalba, B., PCM–air heat exchangers for free-cooling applications in buildings: Experimental results of two real-scale prototypes, *Energy Conversion and Management*, 50, 439-443, 2009.

Olsen, E.L., Qinyan, Y.C., Energy consumption and comfort analysis for different low-energy cooling systems in a mild climate, *Energy and Buildings,* 35, 561-571, 2003.

Pfafferott, J., Herkel, S., Jäschke, M., Design of passive cooling by night ventilation: evaluation of a parametric model and building simulation with measurements, *Energy and Buildings,* 35, 1129-1143, 2003.

Spitler, J.D., Fisher, D.E., Pedersen, C.O., The radiant time series cooling load calculation procedure, *ASHRAE Transactions*, 103(2), 503-515, 1997.

Turkish Standard 825, *Thermal insulation in buildings*, Ankara: Official gazette (23725), 14 June 1999.

Turnpenny, J.R., Etheridge, D.W., Reay, D.A., Novel ventilation cooling system for reducing air conditioning in buildings. Part I: testing and theoretical modeling, *Applied Thermal Energy*, 20, 1019-1037, 2000.

Wacker, P.C., Economizer savings study, *ASHRAE transactions,* 95(1), 47-51, 1989.

Yang, L., Li, Y., Cooling load reduction by using thermal mass and night ventilation, *Energy and Buildings*, 40, 2052-2058, 2008.

Zalba, B., Marin, J.M., Cabeza, L.F., Mehling, H., Free-cooling of buildings with phase change materials, *International Journal of Refrigeration*, 27, 839-849, 2004.

Zhang, Y., Lin, K., Zhang, Q., Di, H., Ideal thermo physical properties for free-cooling (or heating) buildings with constant thermal physical property material, *Energy and Buildings*, 38(10), 1164-1170, 2006.

Zimmermann M., Anderson, J., *Annex 28 – Low Energy Cooling August 1998*, International Energy Agency, Energy Conservation in Buildings and Community Systems Programme, 1988.

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