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Influence of different outdoor design conditions on design cooling load and design capacities of air conditioning equipments

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Abstract

Outdoor design conditions are important parameters for energy efficiency of buildings. The result of incorrect selection of outdoor design conditions can be dramatic in view of comfort and energy consumption. In this study, the influence of different outdoor design conditions on air conditioning systems is investigated. For this purpose, cooling loads and capacities of air conditioning equipments for a sample building located in Adana, Turkey are calculated using different outdoor design conditions recommended by ASHRAE, the current design data used in Turkey and the daily maximum dry and wet bulb temperatures of July 21st, which is generally accepted as the design day. The cooling coil capacities obtained from the different outdoor design conditions considered in this study are compared with each other. The cost analysis of air conditioning systems is also performed. It is seen that the selection of outdoor design conditions is a very critical step in calculation of the building cooling loads and design capacities of air conditioning equipments.

Keywords: Weather design data; HVAC system; Cooling load; Cost analysis

1. Introduction

Local climatic conditions are important parameters for the energy efficiency of buildings. Because the energy consumption in buildings depends on climatic conditions and the performance of heating ventilating and air conditioning (HVAC) systems changes with them as well, better design in building HVAC applications that take account of the right climatic conditions will result in better comfort and more energy efficient buildings.

Outdoor design conditions are weather data information for design purposes showing the characteristic features of the climate at a particular location. They affect building loads and economical design. The result of incorrect selection of outdoor conditions can be dramatic in view of energy and comfort. If some very conservative, extreme conditions are taken, uneconomic design and over sizing may result. If design loads are underestimated, equipment and system operation will be affected. However, selecting the correct type of weather data is a difficult problem. To overcome the problem, Yoshida and Terai [1] constructed an autoregressive moving average (ARMA) type weather model by applying a system identification technique to the original weather data. Li et al. [2] studied climatic effects on cooling load determination in subtropical regions. They found that the outdoor climatic conditions developed for cooling load estimations are less stringent than the current outdoor design data and approaches adopted by local architectural and engineering practices. Zogou and Stamatelos [3] provided a comparative discussion on the effect of climatic conditions on the design optimization of heat pump systems and showed that climatic conditions significantly affect the performance of heat pump systems, which should lead to markedly different strategies for domestic heating and cooling, if an optimization is sought on sustainability grounds. Lam [4] studied

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climatic influences on the energy performance of air conditioned buildings and found that the predictions of annual cooling loads, peak cooling loads and annual electricity consumption differ by up to about 14%. Bulut et al. [5,6] determined new cooling and heating design data for Turkey. They used the current outdoor design data locally used and the new data presented in their studies [5,6] in order to evaluate the influence of the weather data set on the heating and cooling load. They found up to 25% and 32% differences between the cases considered for cooling and heating loads, respectively.

Outdoor design conditions corresponding to different frequency levels of probability for several locations in the United States and around the world are developed by the American Society of Heating, Refrigeration and Air Conditioning Engineers, Inc. (ASHRAE) [7]. Weather data includes design values of dry bulb temperature with mean coincident wet bulb temperature, design wet bulb temperature with mean coincident dry bulb temperature and design dew point temperature with mean coincident dry bulb temperature and corresponding humidity ratio. These design data are the outdoor conditions that are exceeded during a specified percentage of time. Warm season temperature and humidity conditions correspond to annual percentile values of 0.4, 1.0 and 2.0. Cold-season conditions are based on annual percentiles of 99.6 and 99.0. The 0.4%, 1.0% and 2.0% annual values of occurrence represent the value that occurs or is exceeded for a total of 35 h, 88 h and 175 h, respectively, on average, every year, over the period of record. The selection of frequency as risk level in design conditions depends on the applications. Representing the climatic design data for several frequencies of occurrence will also enable designers to consider various operational peak conditions.

The main goal of this study is to investigate the influence of the outdoor design conditions selected during sizing of on air conditioning system. The analysis consists of three main steps. In the first step, the total cooling loads of a sample building are calculated utilizing different outdoor design conditions such as the data given by ASHRAE [7] and the current design data used by project engineers in Turkey [8]. In the second step, design capacities of the all air central air conditioning equipments selected for the sample building are determined for the various outdoor design conditions considered in the study. Finally, cost analysis of the air conditioning system is performed for the cooling season.

2. Description of the sample building

A high school building was selected in order to conduct the analysis. The sample building is located in Adana, Turkey (36°59′ latitude, 35°18′ longitude and 20 m altitude). Adana, an agricultural and industrial centre and the nation's fifth largest city, is near the Mediterranean Sea. It is hot and humid in the cooling season. The sample building has three almost identical floors. Fig. 1 shows the architectural plan of the first floor. The gross area of the building is 1628 m^2, and the outside surfaces of the walls are light colored. The long sides of the building face north and south. The sample building is used as a high school and is occupied between 08:00 and 17:00 h. The high school has 224 students, 15 teachers, 4 officers and 3 laborers. The building has 14 classrooms, 3 laboratories, 5 offices, 1 library, 1 computer room and 3 corridors. The building complies with the insulation requirement imposed by Turkish Standard-TS 825, "Thermal Insulation in

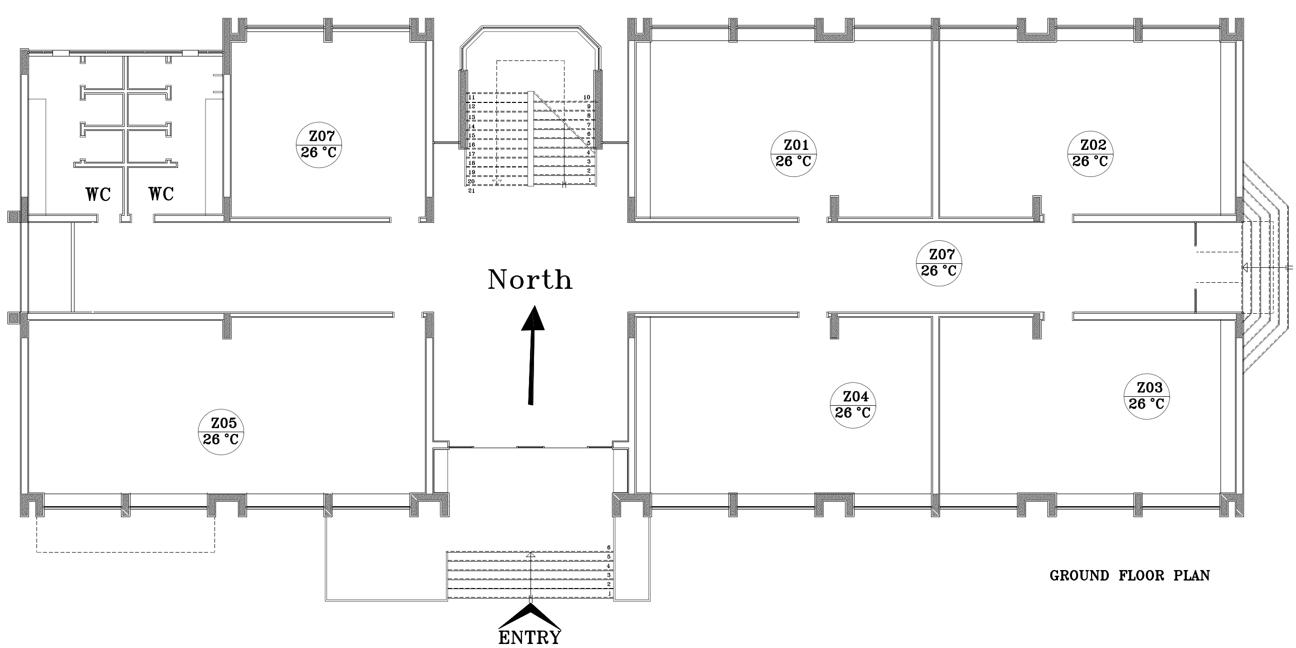

Fig. 1. Architectural plan of the sample building.

Table 1
Overall heat transfer coefficients (U) of the sample building envelope

	Wall	Roof	Floor	Window
U (W/m^2 K)	0.783	0.508	0.757	2.8

Buildings" [9]. Table 1 shows the overall heat transfer coefficients of the high school envelope.

3. Outdoor design conditions

In the analysis, the various outdoor design condition data sets of Adana were used. Details of the data sets are given in Table 2. As shown in the table; there are five data sets. The first data set is the current outdoor design conditions (CURRENT) [8] used by project engineers in Turkey. The second and third data sets are outdoor design conditions for cooling (ASHRAE_04, ASHRAE_1, ASHRAE_2) and evaporation systems recommended by ASHRAE [7] at the 0.4%, 1% and 2% frequency levels (ASHRAE_EVAP_04, ASHRAE_EVAP_1, ASHRAE_EVAP_2), respectively. The fourth data set is the maximum dry bulb and wet bulb temperatures, which are given by ASHRAE [7] at the 0.4%, 1% and 2% frequency levels (ASHRAE_MAX_04, ASHRAE_MAX_1, ASHRAE_MAX_2), respectively. The last data set is the daily maximum dry and wet bulb temperatures of July 21st (DAILY MAX) as design day data, which are calculated from the meteorological data obtained from the Turkish State Meteorological Service (Turkish initials 'DMİ').

4. Air conditioning system

The sample building is conditioned by an all air conditioning system with constant air volume (CAV). The system commonly consists of an air handling unit (AHU), air cooled chiller system, supply and return fans, duct and control units. Fig. 2 is a schematic of an all air central air conditioning system showing typical operating conditions. The returned room air (state R) is mixed with the required outdoor air (state O) at the air handling unit. The mixed air (state M) passes through the cooling coil. The outdoor air is usually warmer and more humid than the return air under typical operation conditions. Therefore, the cooling process

Table 2
Various outdoor design conditions for Adana, Turkey

No.	Description of the data set	Name of the data set	Risk level (%)	DB (°C)	WB (°C)
1	Current design (DBmax-WBmax)	CURRENT	–	38.0	26.0
2	ASHRAE-cooling (DB-CWB)	ASHRAE_04	0.4	36.1	21.6
		ASHRAE_1	1	34.6	21.8
		ASHRAE_2	2	33.2	22.3
3	ASHRAE-evaporation (WB-CDB)	ASHRAE_EVAP_04	0.4	31.7	26.0
		ASHRAE_EVAP_1	1	30.5	25.4
		ASHRAE_EVAP_2	2	29.9	24.9
4	ASHRAE-max (DBmax-WBmax)	ASHRAE_MAX_04	0.4	36.1	26.0
		ASHRAE_MAX_1	1	34.6	25.4
		ASHRAE_MAX_2	2	33.2	24.9
5	Daily-max (DBmax-WBmax)	DAILY MAX	–	35.2	24.1

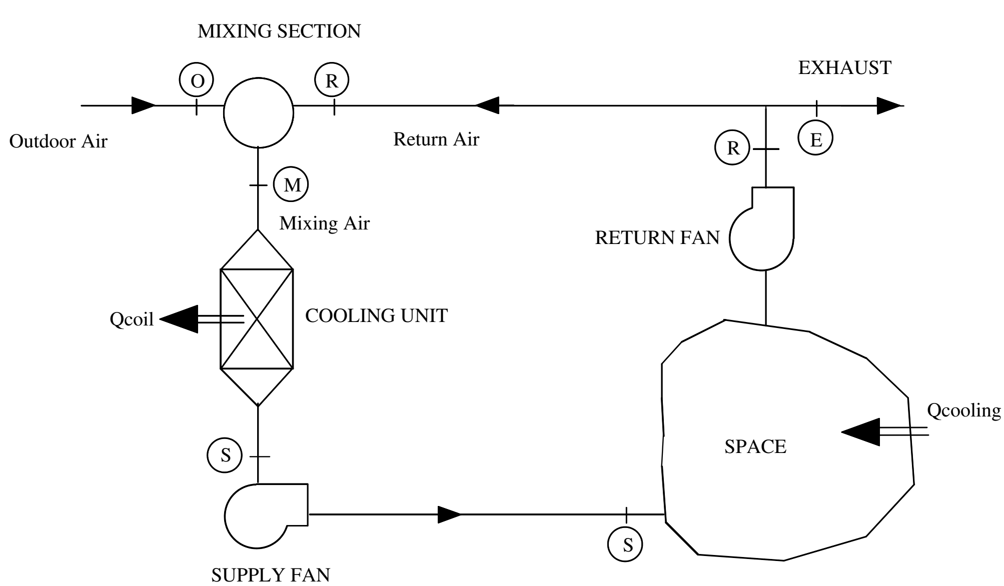

Fig. 2. Schematic of air movement of an all air conditioning system.

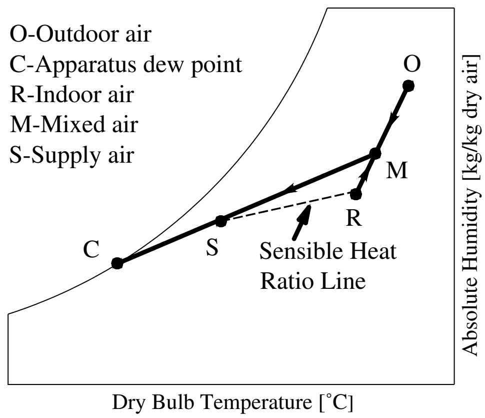

Fig. 3. State points of air during processing for summer operation of air conditioning system on psychometric chart.

generally involves both cooling and dehumidification, with the conditioned air leaving the cooling coil at state (S). The cooled and dehumidified air leaving the coil at state (S) is then supplied to the conditioned space at constant air volume, which is at state (R), to complete the cycle. Fig. 3 shows the state points of air during the process for summer operation of the air conditioning system on a psychometric chart.

5. Calculation of cooling load

To design and select elements of a HVAC system, it is very important to determine the heating and cooling loads of the building. Weather data are very important to compute the loads accurately. However, selection of the most appropriate weather data set can be a difficult problem. Conventional load calculation methods are divided into two classes, i.e. peak load estimation and annual load simulation. Diurnally periodic weather data are used for peak load estimation, but the correlation of weather elements, i.e. temperature, solar radiation, moisture contents, etc. can hardly be taken into account. Reference year weather data are used for annual load simulation, but the results can only give the seasonal summed load, no information being obtained for the detailed load variations owing to the shortness of the data period [1].

The total cooling load of a building consists of the external loads through the building envelope and internal loads from people, lights, appliances and other heat sources. To design and select a properly sized HVAC system, the peak or maximum load for each zone must be computed for a design day based on the required indoor and prevailing outdoor design conditions.

In this study, the indoor design conditions of the building were chosen as 50% relative humidity and 26 °C dry bulb temperature. The radiant time series method (RTS) was used for calculation of the cooling load. The RTS method, introduced by Spitler et al. [10] and the 2001 ASHRAE Handbook, Fundamentals [7], is a new simplified means for performing design cooling load calculations, and it was derived from the “heat balance method”.

The total and the components of the cooling load and sensible heat ratio (SHR) of the sample building calculated using the current outdoor design data (Table 2) are given in Table 3 for different hours of the day. It can be seen from the table that the maximum total cooling load of the building

Table 3
Cooling loads and SHR for CURRENT data set

Time	Parts of the cooling load (W)			Total cooling load (W)			SHR
	External surface	Fenestration	Internal source	Sensible	Latent	Total	
01:00	7603	7356	4193	19,151	0	19,151	1.00
02:00	6488	6379	3885	16,753	0	16,753	1.00
03:00	5496	5415	3577	14,487	0	14,487	1.00
04:00	4600	4643	3299	12,541	0	12,541	1.00
05:00	3805	5148	3020	11,973	0	11,973	1.00
06:00	3156	15,469	3020	21,645	0	21,645	1.00
07:00	3177	21,635	2712	27,524	0	27,524	1.00
08:00	4498	26,535	46,056	62,354	14,735	77,089	0.81
09:00	6888	32,806	50,675	75,634	14,735	90,368	0.84
10:00	9836	39,660	53,446	88,208	14,735	102,942	0.86
11:00	12,926	45,601	55,293	99,086	14,735	113,820	0.87
12:00	15,850	49,601	56,525	107,241	14,735	12,1976	0.88
13:00	18,383	50,906	57,449	112,003	14,735	126,738	0.88
14:00	**20,349**	**49,099**	**58,064**	**112,778**	**14,735**	**127,513**	**0.88**
15:00	21,693	44,505	58,710	110,173	14,735	124,907	0.88
16:00	22,325	39,252	56,028	102,871	14,735	117,605	0.87
17:00	22,132	35,331	56,173	98,902	14,735	113,636	0.87
18:00	21,098	28,935	15,861	65,894	0	65,894	1.00
19:00	19,212	20,204	11,815	51,231	0	51,231	1.00
20:00	16,718	16,400	9192	42,309	0	42,309	1.00
21:00	14,210	13,704	7432	35,347	0	35,347	1.00
22:00	12,095	11,658	6260	30,013	0	30,013	1.00
23:00	10,365	9943	5366	25,674	0	25,674	1.00
24:00	8882	7644	5107	20,285	0	20,285	1.00

(127.51 kW) occurs at 14:00. While 39% of the total cooling load is due to windows, 16% stems from the external surfaces and the remaining portion is from internal heat sources.

Hourly cooling loads of the sample building were also calculated using different outdoor design data sets (Fig. 4). It can be seen from the figure that the cooling load is affected considerably by the selected weather data set, although the trends are the same. The maximum cooling load is obtained with the data set of CURRENT. It was followed by the data sets recommended by ASHRAE for cooling and evaporation systems, respectively. The DAILY MAX and ASHRAE_1 data sets produce almost the same results.

The maximum design cooling loads and sensible heat ratios (SHR) calculated for all the outdoor design conditions considered are given in Table 4. As can be seen from the table, the maximum design cooling load (127.51 kW) is obtained with the CURRENT data set. Table 4 also shows the ratio of the design cooling load to the design cooling load obtained from the CURRENT data set. The design cooling load with ASHRAE_04, ASHRAE_1, ASHRAE_2, ASHRAE_EVAP_04, ASHRAE_EVAP_1 and ASHRAE_EVAP_2 is 2%, 4%, 7%, 9%, 11% and 12% less than the load with the CURRENT data set, respectively. In the case of the DAILY MAX data set, the design load is found to be 4% less than that for the CURRENT data set. SHR is almost independent of the data sets used, and it is about 0.88 for all the sets (Table 4).

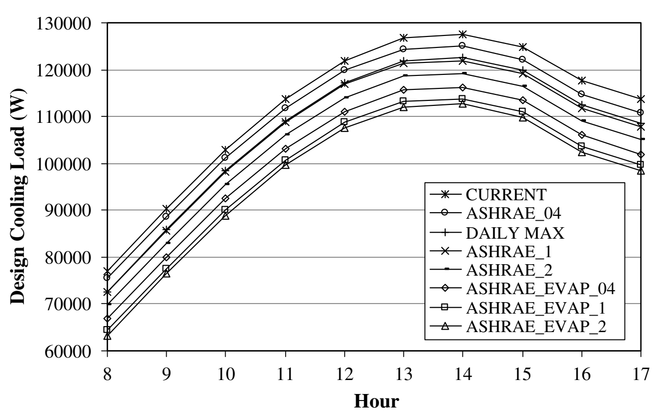

Fig. 4. Total design cooling load for all design data sets.

Table 4
Total design cooling load, SHR and cooling load ratio for different data sets

Data set	Q_{room} (kW)	SHR	Cooling load ratio
CURRENT	127.51	0.88	1.00
ASHRAE_04	124.95	0.88	0.98
ASHRAE_1	121.98	0.88	0.96
ASHRAE_2	119.20	0.88	0.93
ASHRAE_EVAP_04	116.23	0.87	0.91
ASHRAE_EVAP_1	113.85	0.87	0.89
ASHRAE_EVAP_2	112.66	0.87	0.88
ASHRAE_MAX_04	124.95	0.88	0.98
ASHRAE_MAX_1	121.98	0.88	0.96
ASHRAE_MAX_2	119.20	0.88	0.93
DAILY MAX	122.51	0.88	0.96

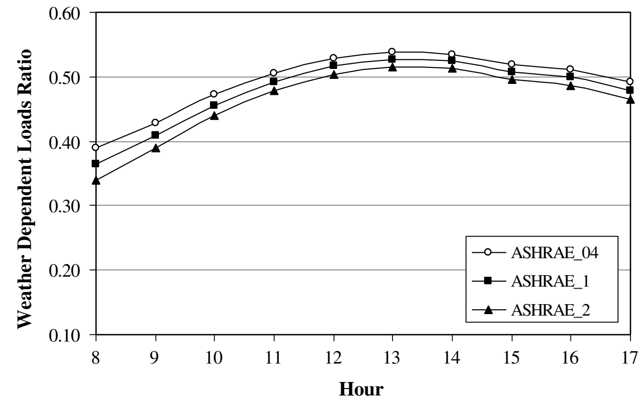

Fig. 5. Ratio of weather dependent loads to the total cooling load for ASHRAE data sets.

The variation of the ratio of the weather dependent component of the total cooling load to the total cooling load during the occupation period is shown in Fig. 5. As can be seen from the figure, the weather dependent components constitute approximately 50% of the total load.

6. Calculation of design capacities of air conditioning equipments using different outdoor design conditions

The maximum (design) cooling coil capacity ($Q_{coil,max}$) and the maximum (design) supply air flow rate (M_{tot}) for the sample building were calculated using the maximum building cooling load (Q_{room}) and sensible heat ratio (SHR) given above, the minimum fresh air ventilation requirement (M_{out}), the indoor and outdoor design conditions and the fixed supply air temperature as input parameters. Because of an iterative approach requirement in the calculation procedure, a computer program was written [11] for the calculations.

In the calculations, the temperature of the air supplied to the air conditioned space was selected to be 16 °C. According to the ASHRAE Standard 62 [12] ventilation rate procedure, the minimum fresh air ventilation requirement for the sample building (M_{out}) was determined to be 7000 m^3/h.

Table 5 gives the design cooling coil capacity ($Q_{coil,max}$), total and fresh air mass flow rates (M_{tot} and M_{out}) and mixing ratio ($\Phi = M_{out}/M_{tot}$) for all the outdoor design conditions considered in this study. It is seen from the table that the design cooling coil capacity (184.05 kW) and the total mass flow rate (39525 kg/h) required are maximum with CURRENT data set. Table 5 also presents the ratio of the design coil capacity to the maximum design coil capacity, which is obtained with the CURRENT data set (coil capacity ratio), and the ratio of the total mass flow rate to the maximum total mass flow rate, which is again obtained with the CURRENT data set (fan capacity ratio), for all the outdoor design conditions considered. It can be

Table 5
Design cooling coil capacity and other properties for different data sets for supply air temperature of 16 °C

Name of the data set	$Q_{coil,max}$ (kW)	Coil capacity ratio	M_{tot} (kg/h)	Fan capacity ratio	M_{out} (kg/h)	Φ (%)
CURRENT	184.05	1.00	39,525	1.00	7739	20.0
ASHRAE_04	145.9	0.79	38,720	0.98	7863	20.3
ASHRAE_1	145.2	0.79	37,798	0.96	7888	20.9
ASHRAE_2	146.1	0.79	36,938	0.93	7909	21.4
ASHRAE_EVAP_04	176.5	0.96	35,615	0.90	7866	22.1
ASHRAE_EVAP_1	168.9	0.92	34,886	0.88	7903	22.7
ASHRAE_EVAP_2	163.1	0.89	34,521	0.87	7926	23.0
ASHRAE_MAX_04	183.9	0.99	38,720	0.98	7777	20.1
ASHRAE_MAX_1	175.7	0.95	37,798	0.96	7820	20.7
ASHRAE_MAX_2	168.9	0.92	36,938	0.93	7858	21.3
DAILY MAX	164.7	0.89	37,964	0.96	7834	20.6

seen from Table 5 that the ASHRAE data sets (ASHRAE_04, ASHRAE_1 and ASHRAE_2) used for cooling applications produce the minimum design cooling coil capacities. For these data sets, the design cooling coil capacities are approximately 21% less than that of the CURRENT data set. In the case of the DAILY MAX data set, which is obtained from the daily maximum dry and wet bulb temperatures of July 21, the design coil capacity is 11% lower than that obtained from the CURRENT data set. It is noteworthy that one of the results given in Table 5 is that the design coil capacity obtained with the maximum dry bulb and wet bulb temperature selected from ASHRAE design conditions for the 0.4 risk level (ASHRAE_MAX_04) is approximately equal to the design coil capacity obtained for the CURRENT data set. This shows that the current design conditions used in Turkey (CURRENT) were possibly derived from the maximum dry bulb and wet bulb temperatures. In addition, when the data sets are compared, considering maximum (design) supply air flow rate (M_{tot}), the highest mass flow rate is again obtained with the CURRENT data set. Therefore, it can be concluded that the current outdoor design conditions used in Turkey for design and selection of air conditioning systems are generally stringent. The HVAC equipment designs are oversized and consequently uneconomic. Both the initial and operating costs of the air conditioning system increase because of over sizing the system.

7. Cost analysis of the air conditioning system

In this part of the study, the cost analysis of the all air central air conditioning system is conducted for the CURRENT, ASHRAE_04, ASHRAE_1 and ASHRAE_2 data sets. The results obtained for the CURRENT data set are only compared with the ASHRAE_04, ASHRAE_1 and ASHRAE_2 data sets due to their lower design cooling coil capacity compared with those of the other data sets.

The air handling unit (AHU) and chiller system were selected from the products of a local HVAC equipment supplier. The AHU contains fans, cooling coil, filter, mixing and exhaust air elements. The total mass flow rate of the AHU for all the data sets is 40,000 kg/h. The supply and return fans in the AHU provide air and have 15 kW and 12.5 kW power requirements, respectively. For the air conditioning system, the mass flow rate is constant throughout the operation of the system, therefore, even for part load conditions, the fans require maximum power.

For the CURRENT design data set, the net cooling capacity of the chiller system under nominal operating conditions (38 °C condenser air inlet temperature, 10 °C evaporator inlet and 6 °C outlet temperature) is 185 kW and the power requirement of the unit is 80 kW. The compressor in the chiller unit is controlled by a five stepped proportional control system for part load operations.

It is clearly known that the compressor in the chiller unit usually operates at part load under real operating conditions because of the varying cooling load. Whenever the operating load is less than the design load, the capacity of the compressor in the chiller unit should be reduced by the five stepped proportional controller for saving energy.

In cases of ASHRAE_04, ASHRAE_1 and ASHRAE_2, the net cooling capacity of the chiller system under nominal operating conditions is 146 kW, and the power required is 66 kW. The compressor in the chiller unit has a four stepped proportional control for part load operation.

The operating cost of the air conditioning systems consists of the electricity consumptions in the fans and the chiller unit. The cooling period for Adana covers 184 days between April 15 and September 15. The daily operating time of the central air conditioning system is 9 h (from 8:00 to 17:00). The electric price is 0.10 $/kW h.

In this study, the procedure given by Aktacir et al. [13] is used to calculate the seasonal operating cost of the air conditioning systems. Firstly, the hourly cooling coil capacity (Q_{coil}) for the 21st day of each month during the cooling season was computed using the hourly cooling loads. Calculation of the hourly cooling loads requires hourly outside air data. Hourly values of weather data were calculated using a weather data model given by Bulut et al. [14].

The coil capacity for days other than the 21st day of each month was not calculated. Therefore, the results obtained for the 21st day of each month were integrated on an hourly basis by the Simpson integral method and seasonal average values of the hourly coil capacity ($Q_{coil,av}$) were obtained.

Secondly, the variation of the coefficient of performance (COP) of the chiller unit with outside air temperature and the variation of COP with part load ratio were considered. The part load ratio (PLR) was defined as

$$\mathrm{PLR} = Q_{chil}/Q_{chil,full} \quad (1)$$

where Q_{chil} is the hourly cooling demand on the chiller, which is approximately equal to the hourly coil load (Q_{coil}), and $Q_{chil,full}$ is the full cooling capacity of the chiller. The automatic control system of the chiller unit will select a suitable operating step for the compressor depending on the value of PLR.

Using the data provided by the manufacturer and the hourly outside air temperature, hourly values of cooling capacity (Q_{chil}) and power required (P_{chil}) of the chiller at full load and at part load (for each of the five steps of the chiller) were obtained for the 21st day of each month during the cooling season. Seasonal average hourly values of Q_{chil} and P_{chil} ($Q_{chil,av}$ and $P_{chil,av}$) were then calculated utilizing the Simpson integral method. Using the seasonal average hourly values of coil load ($Q_{chil,av}$) obtained previously and the chiller capacity at full load ($Q_{chil,full,av}$) and the seasonal average hourly part load ratio (PLR_{av}), the seasonal average hourly operating steps of the compressor (ST_{av}) were determined.

Finally, from the corresponding part load $Q_{chil,av}$, $P_{chil,av}$ and operating times, it was possible to calculate the seasonal energy consumption and, then, the operating cost of the chiller unit.

Tables 6 and 7 give the seasonal average operating cost of the chiller and the fans selected for the data sets CURRENT and ASHRAE_04, ASHRAE_1 and ASHRAE_2, respectively. The seasonal average hourly part load ratio (PLR_{av}), seasonal average hourly operating step (ST_{av}), seasonal average hourly power requirement ($P_{chil,av}$) and seasonal average hourly electric energy consumption are also given in the tables. The compressor in the chiller unit usually operates at part load under real operating conditions because of the varying cooling load. It can be seen from Table 6 that the chiller system selected according to the CURRENT data set operates 1 h at step 2 (25–50%), 4 h at step 3 (50–75%) and 4 h at step 4 (75–100%). In the cases of ASHRAE_04, ASHRAE_1 and ASHRAE_2 data sets, (Table 7), the chiller system operates 1 h at step 2 (20–40%), 4 h at step 3 (40–60%) and 5 h at step 4 (60–80%).

Table 6
Seasonal average operating cost of the selected chiller and fans for CURRENT data set

Device	Operating time	Part load ratio, PLR_{av}	Operating step, ST_{av}	Power required, $P_{chil,av}$ (kW)	Electric consumption (kW h/year)	Operating cost ($/year)
Chiller	08:00–09:00	0.37	2	24.58	4523	452
	09:00–10:00	0.45	3	37.00	6808	681
	10:00–11:00	0.52	3	38.12	7014	701
	11:00–12:00	0.59	3	39.13	7200	720
	12:00–13:00	0.64	4	54.49	10,026	1.003
	13:00–14:00	0.67	4	55.34	10,183	1.018
	14:00–15:00	0.67	4	55.81	10,269	1.027
	15:00–16:00	0.65	4	55.65	10,240	1.024
	16:00–17:00	0.61	4	54.83	10,089	1.009
	Total operating cost of the chiller ($/year)					**7.635**
Fans	08:00–17:00	1	–	27.5	45540	4.554
	Total operating cost of the fans ($/year)					**4.554**

Table 7
Seasonal average operating cost of the selected chiller and fans for the data sets ASHRAE_04, ASHRAE_1 and ASHRAE_2

Device	Operating Time	Part load ratio, PLR_{av}	Operating step, ST_{av}	Power required, $P_{chil,av}$ (kW)	Electric consumption (kW h/year)	Operating cost ($/year)
Chiller	08:00–09:00	0.45	2	23.81	4381	438
	09:00–10:00	0.55	3	38.61	7104	710
	10:00–11:00	0.65	3	39.79	7322	732
	11:00–12:00	0.72	3	40.97	7538	754
	12:00–13:00	0.78	4	57.24	10,532	1.053
	13:00–14:00	0.82	4	57.97	10,667	1.067
	14:00–15:00	0.83	4	58.35	10,736	1.074
	15:00–16:00	0.80	4	58.28	10,723	1.072
	16:00–17:00	0.75	3	42.15	7755	775
	Total operating cost of the chiller ($/year)					**7.676**
Fans	08:00–17:00	1	–	27.5	45,540	4.554
	Total operating cost of the fans ($/year)					**4.554**

Table 8
Initial and yearly operating cost of the selected air conditioning system

Cost	Current ($)	ASHRAE for all risk level ($)
Initial	94.635	87.850
Yearly total operating for proportional control	12.189	12.230
Yearly total operating for on–off control	13.630	13.009

Consequently, the chiller system never operates at full load during the whole cooling season.

The initial and total operating costs of the air conditioning system are presented in Table 8. When the system is designed according to the ASHRAE data sets, the initial cost is about 8% less than that when designed according to the CURRENT data set. However, there is almost no difference between the operating costs of the air conditioning system with proportional control for all the data sets considered. This is due to the fact that the compressor in the chiller unit of the air conditioning system is adjusted by a stepped proportional control for part loads. However, if a chiller unit with on–off type load control is selected instead of the proportionally controlled one, the operating cost for the ASHRAE data sets is approximately 5% less than that for the CURRENT data set.

When the control systems are compared for the same design data, it is seen that the operating costs of the on–off control type are approximately 12% greater for the CURRENT data set and 6% greater for the ASHRAE data sets than that of the proportional control type. In the calculation of operating cost for the on–off control, it is assumed that the maximum starting current is five times higher than the nominal current.

8. Conclusion

In this study, the influence of different outdoor design conditions on cooling loads and air conditioning system is investigated. It is found that a significant part of the cooling load depends on outdoor weather conditions. For the sample building located in Adana, Turkey, approximately half of the cooling load originates from the building envelope, which is weather dependent.

The findings indicate that the current outdoor design conditions used in Turkey for design and selection of air conditioning systems are generally stringent. The HVAC equipments designed are oversized and consequently uneconomic. Both the initial and operating costs of the air conditioning system increase because of over sizing the system.

It seen that the control system of the chiller unit, which is the main component of a HVAC system, is of great importance for energy saving. Under real operating conditions, the HVAC system operates at part load. Therefore, equipments that have a high efficiency at part loads should be selected. Engineers and building designers should select and assess the appropriate outdoor design conditions in order to achieve optimum air conditioning equipment sizing according to their applications and acceptable risk levels. Designers and engineers should also consider additional operational peak conditions in the design and selection steps of the HVAC system.

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