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ANALYSIS OF THE SUMMER AMBIENT TEMPERATURES FOR COOLING PURPOSES

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Abstract—The necessary analysis of the summer ambient temperature data for evaluation of passive and hybrid cooling techniques and components is discussed. A complete analysis using data from Athens, Greece, is presented. Daily, monthly, and annual distributions of the more important parameters, as well as statistical indexes, are estimated and discussed. Correlation techniques are used, and expressions to predict accurately the necessary parameters are proposed. The presented analysis aims to define an appropriate format to report on the summer ambient temperature data used for cooling purposes.

1. INTRODUCTION

Effective energy design of buildings requires detailed and accurate knowledge on the level and distribution of the ambient temperature. The recent important development of passive solar techniques for buildings has offered a detailed analysis of the available information on the ambient temperature data[1-4]. However, compilation of the data is done mainly for heating purposes, and little information is available for the design and evaluation of cooling systems, as well as for the evaluation of summer building performance.

Existing methodologies on the evaluation of building cooling loads are based mainly on the use of the cooling degree days[5]. However, especially in European countries, the cooling degree days are rarely known and values, when existent, are limited to only one temperature base[6-7].

On the other hand, sizing of cooling systems is based primarily on the 1% American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) dry bulb temperature[8]. This technique can lead to oversizing the cooling system, although it is inappropriate for the sizing of passive and hybrid cooling systems and techniques, based on the heat dissipated to natural heat sinks (convective, evaporative, radiative, and ground cooling[9]).

Passive and hybrid cooling systems and techniques require additional information, e.g., the night cooling degree days for the calculation of the night ventilation efficiency[10], the daily variation of temperature, the 24-h distribution, the maximum and minimum values, as well as the frequency distribution of the temperatures and the number of consecutive days over a certain value[11]. Primary climatological data for the development and the calculation of these parameters are made available as they are recorded at standard meteorological stations. Therefore, a classification of the necessary parameters, as well as a reporting format appropriate for cooling purposes, should be defined.

The present paper aims to present a methodology on the treatment of the summer temperature data. An analysis based on multiyear summer temperature data from Athens, Greece, is presented. Tools and formulas for the prediction of important parameters for cooling purposes are also proposed.

2. TEMPERATURE DATA

Athens is characterized by a warm Mediterranean climate with mild and relatively wet winters and warm dry summers. Mean monthly ambient temperatures are between 9-27°C, occurring in January and July to August, respectively. High daytime temperatures also occur during September due to unfavorable wind conditions. Therefore September is included in the cooling season.

Hourly temperature data are taken from the National Observatory of Athens[12], (latitude 37.58°N, longitude 23.43°E, and altitude 107 m). The Observatory, located on a hill at the center of Athens, has been recording measurements of ambient temperature since 1857.

Analysis of multiyear summer temperature data (1857-1990), shown in Fig. 1, indicates that no temperature trends can be defined. Therefore, data for 1977 to 1990 are used without, to introduce important errors. Comparisons of the frequency distribution of the mean monthly temperature data of the last 130 years, given in Fig. 2, with the data of the 14 studied years given in Table 1, show that are exactly in the same zone.

In order to identify possible differences between the used set of data—1977 to 1990—and the data of the past periods, the test of the difference of means of independent samples has been used. The t values between the mean temperature of each of the 25 overlapping decades from 1857 to 1990 (i.e., 1857-1866, 1861-1870, etc.) have been calculated by the t ratio of the test and shown in Fig. 3.

The calculated differences between the mean temperature of the used period and the previous overlapping decades are not statistically significant (c.l. = 0.99). Important differences are observed only for the decades of 1857 to 1866 and 1937 to 1946. However, it should be pointed out that from 1857 to 1866 there are no available systematic screen temperature measurements, whereas 1937 to 1946 is not charac-

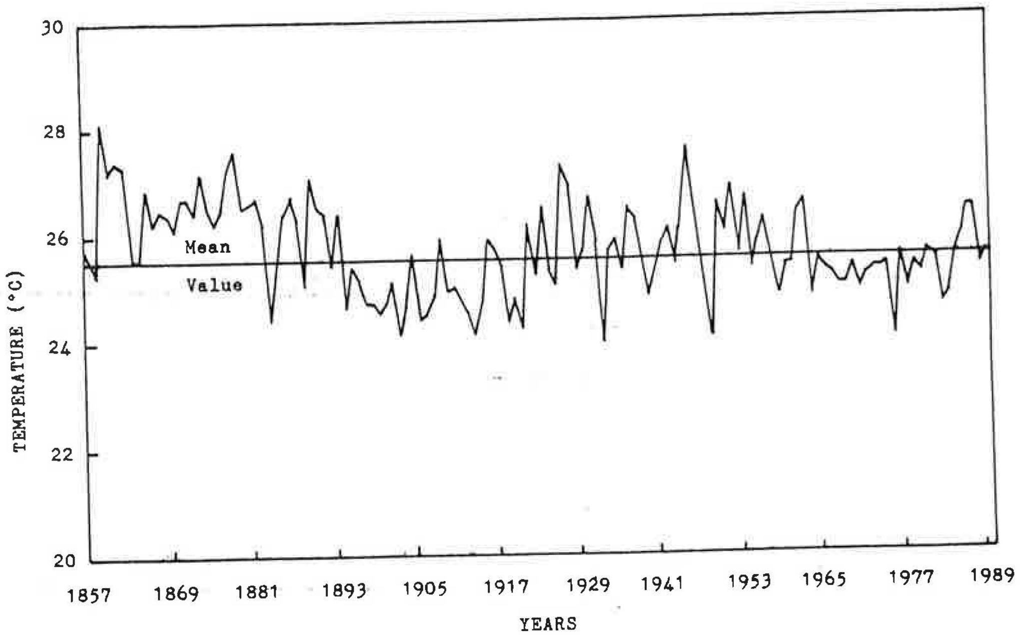


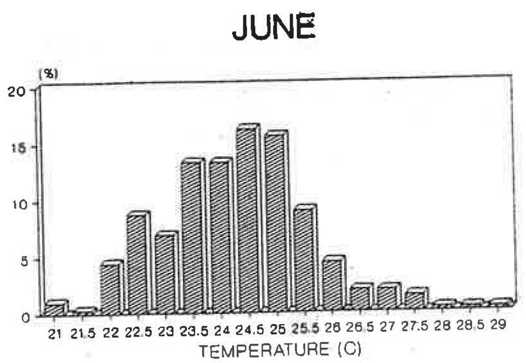
Fig. 1. Mean summer ambient temperature data for the period 1857–1989.

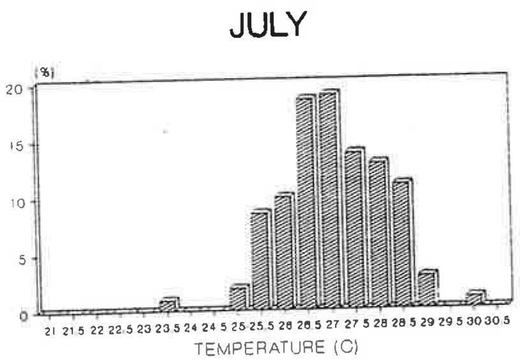
terized by excellent measurements due to war problems. Therefore, data for 1977 to 1990 are used without introducing important errors.

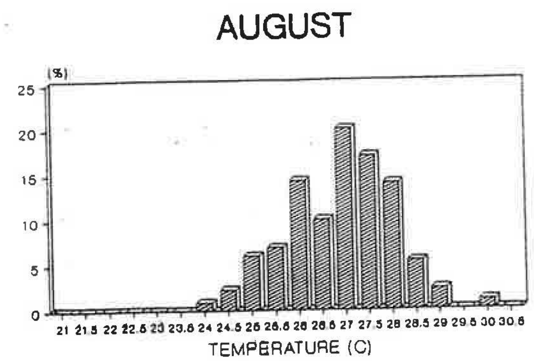
3. DATA ANALYSIS

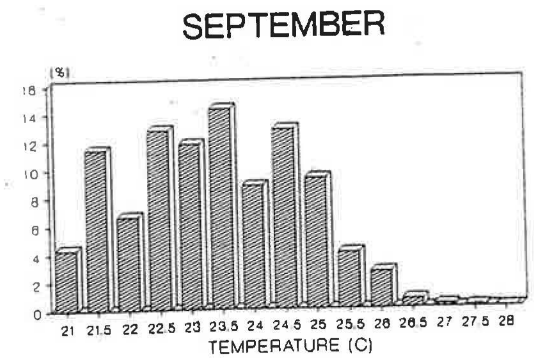
Analysis of data has been completed in various steps. First, the number of hours over a base temperature, HTx, is computed. This type of information is necessary for the evaluation of the performance of buried pipe techniques; it is used to estimate the possible period that the system can operate[13]. Also, it is a very useful index for a first estimation of the cooling needs of the case.

Calculations have been carried out for every year, and values have been computed for periods of 10 d,









... for the period 1857–1990

Table 1. Mean monthly temperatures for the period 1977–1990

Year	1977	1978	1979	1980	1981	1982	1983	1984	1985	1986	1987	1988	1989	1990
June	25.0	25.3	25.7	24.0	25.6	25.0	23.0	23.9	25.2	24.7	24.8	25.0	23.6	24.7
July	28.1	27.2	26.5	27.5	26.4	26.0	26.7	26.2	26.5	27.1	28.7	29.1	26.8	28.2
August	27.3	25.7	26.3	26.4	26.4	26.6	25.6	24.9	27.4	27.9	26.6	27.7	27.1	26.4
September	21.9	21.4	23.6	22.7	23.8	24.4	22.9	24.0	23.4	23.9	25.5	23.6	23.6	23.0

starting from the first of June to the end of September. The base temperature values are equal to 25, 26, 27, and 28°C in order to correspond to the design indoor temperature levels. The relative frequency of hours having a temperature over the defined base is given in Fig. 4 for each month. As expected, June is characterized by an increasing number of HTx moving from the first to third 10-d periods. An important increase also is observed during the second 10 d of July and the first 10 d of August. The last 20 d of August as well as of September are characterized by a continuous decrease in the number of hours over the defined bases.

The percentage of hours having a temperature over 25°C observed during the daytime period (0600–2000 h local time) was for June, equal to 63%; for July, 79%; for August, 74%; and for September, 48%. High percentages also have been calculated for the other bases for July and August. Nighttime temperatures over 25°C are observed mainly during July and August. For June and September the percentage of occurrence of night temperatures over the four defined bases is very low (see Fig. 5).

In a second step, the number of successive hours over each base temperature is calculated. It is found that maximums of 298 and 325 successive hours over 25°C are observed during July and August, respec-

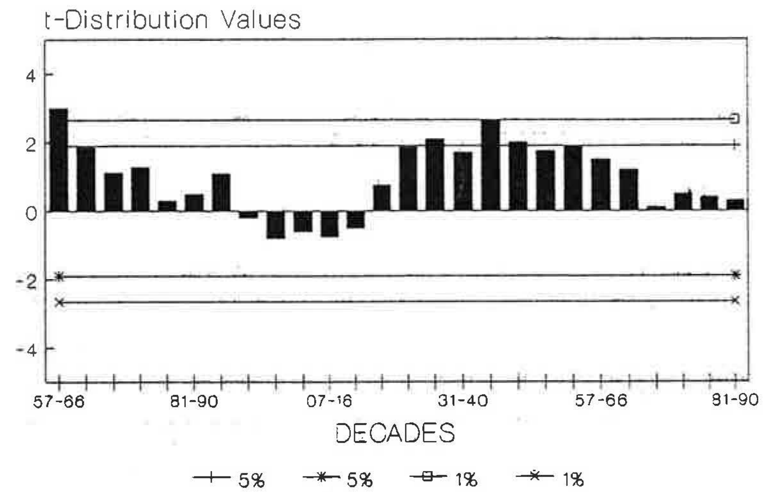


Fig. 3. T-values between the mean temperature of each of the 25 overlapping decades of the period 1857–1990 (lines correspond to 1% and 5% confidence limits).

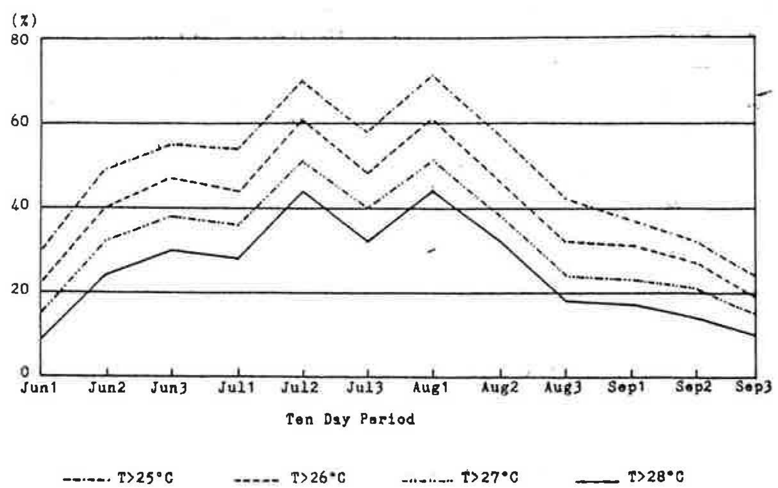


Fig. 4. Distribution of the 10 days percentage of hours having a temperature over the defined bases.

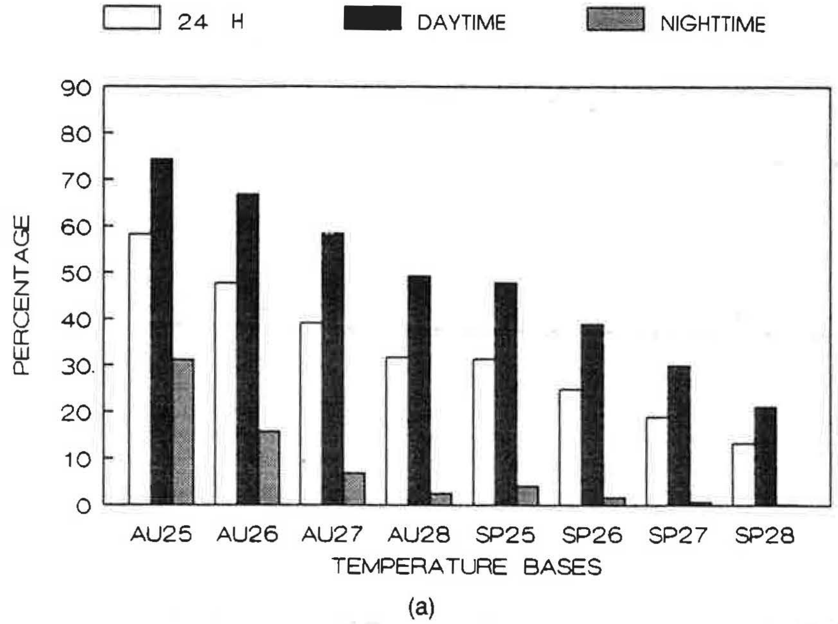


Fig. 5. Mean monthly percentage of the 24 h, daytime and nighttime hours over the temperature bases, for the period 1977–1990. (a) August and September.

tively. The calculated frequency of successive hours over 25°C is given for each month in Fig. 6. For June and September, the maximum frequency is observed between 7 to 11 successive hours, whereas for July and August the maximum observed frequency is close to 12 successive hours.

In order to examine the relation between the monthly hours over 25°C and the corresponding hours over the other base temperatures, 26, 27, and 28°C, correlation techniques have been used. Approximately four monthly values for all years show the same slope. A multiple linear regression is attempted and it is found that the correlation is very satisfactory. The calculated correlation coefficients for every month are given in Table 2. It should be pointed out that the values given by the multiple linear correlation are significant in a significant level of 0.001. Therefore, any increase or decrease, from year to year, of the number of monthly hours over the base temperature considered is always followed by an equivalent linear increase or decrease, of the hours over the other bases.

Regression techniques have been used to correlate

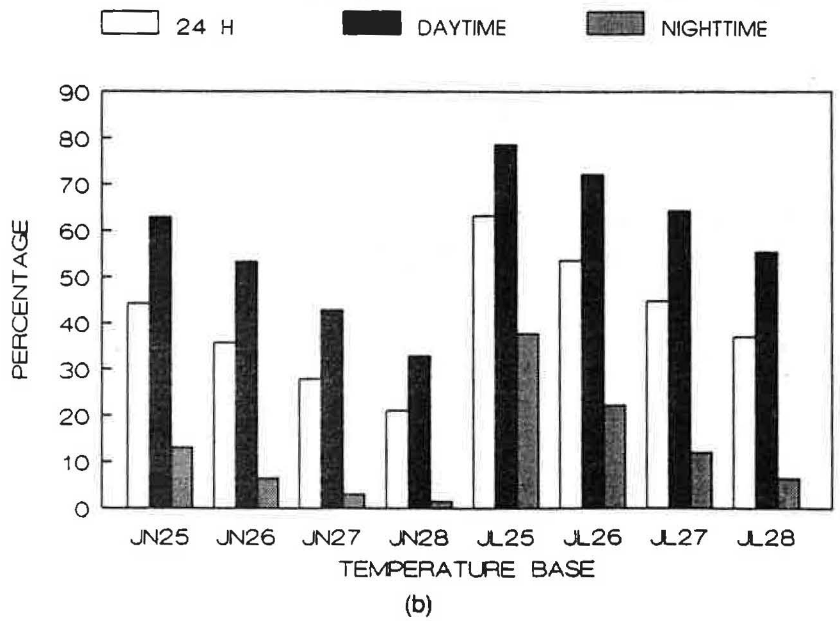


Fig. 5. (*Contd.*) (b) June and July.

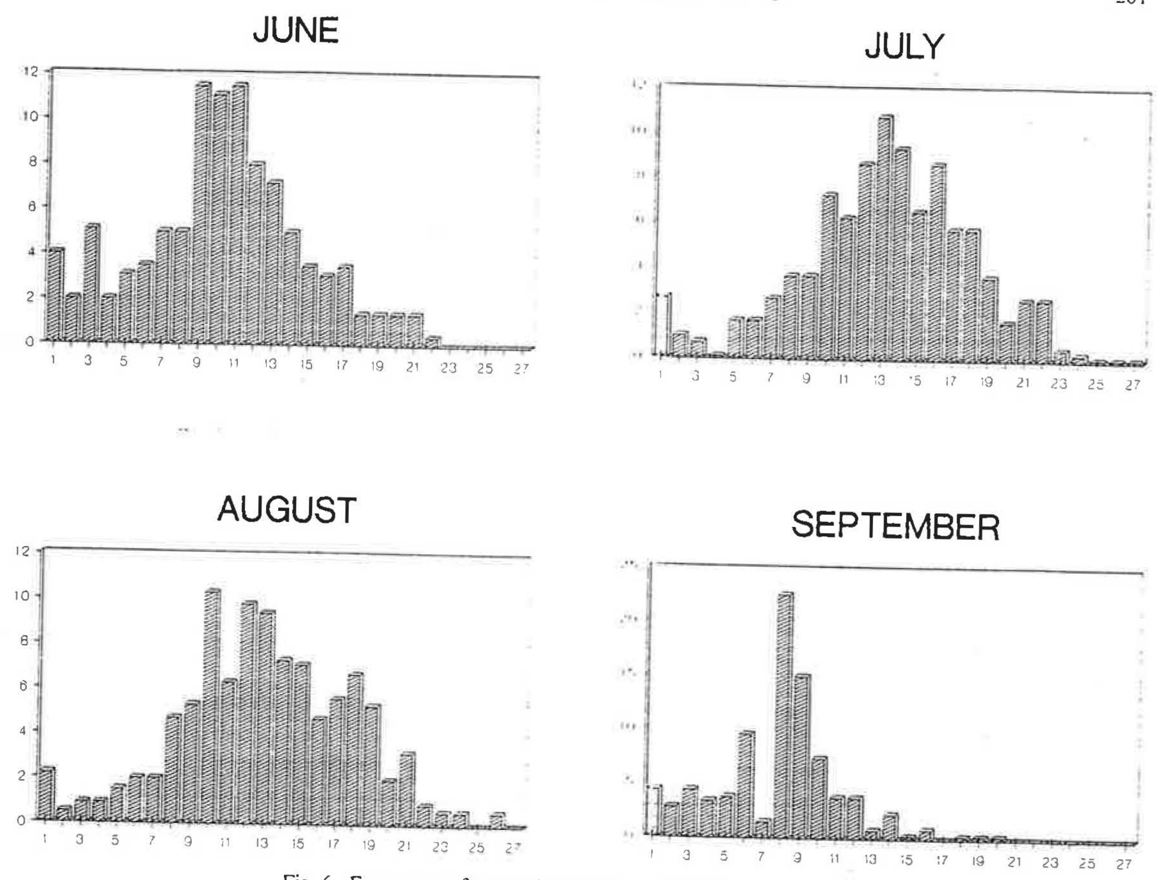


Fig. 6. Frequency of successive hours over 25°C for each month.

the monthly values of HTx for Tx 26, 27, and 28°C with $HT25$. It is found that a simple expression of the form:

$$HTx = a + bHT25 \quad (1)$$

can be used to predict accurately the value of HTx when $HT25$ is known. The corresponding values of a and b and the correlation coefficients r for the three temperature bases are given in Table 3.

Knowledge of the number of cooling degree-hours (DH) is necessary for the estimation of the cooling load of the buildings. Therefore, the number of degree-hours which correspond to the four defined temperature bases was calculated. The monthly values of the cooling degree-hours, calculated per year as well as the percentage of the night degree-hours as a function of the total 24 h degree-hours were computed.

For June, July, and August, the number of degree hours corresponding to $T = 25$°C is greater than 950, with a coefficient of variability between 25 to 41%.

More than 950 degree-hours were also observed during July and August for base temperatures equal to $T = 26$ and $T = 27$°C. It should be pointed out that the increasing coefficient of variability for higher temperature bases is due to the serious annual variability of the monthly degree-hours that correspond to higher temperatures. Also the very high coefficient of variability for September results from the important annual changes of the degree-hours observed for all the base temperatures during this month.

In order to provide algorithms for the estimation of the cooling degree-hours, regression techniques have been used. It is found that the monthly cooling degree-hours can be predicted accurately as a function of the mean and mean maximum monthly temperature, Tm and Tmm. Tm is calculated as the mean of the 24 hourly temperature values. Therefore, it is proposed to calculate the monthly degree-hours from the following expressions:

$$\text{Base Temperature} = 25°\text{C}: DH(25) = -5941.01 - 264.96Tm + 434.72Tmm; \quad r = 0.977.$$

$$\text{Base Temperature} = 26°\text{C}: DH(26) = -5257.82 + 215.87Tm + 18.04Tmm; \quad r = 0.934.$$

$$\text{Base Temperature} = 27°\text{C}: DH(27) = -4867.18 + 365.54Tm - 118.0Tmm; \quad r = 0.885.$$

$$\text{Base Temperature} = 28°\text{C}: DH(28) = -4463.24 + 215.72Tm - 15.01Tmm; \quad r = 0.931.$$

In case that Tm values are not be estimated due to the lack of hourly data, simpler expressions which involve only the use of the monthly mean maximum temperature, could be used. The proposed expressions are of sufficient accuracy:

$$\text{Base Temperature} = 25°\text{C}: DH(25) = -5548.61 + 207.04 Tmm; \quad r = 0.82.$$

$$\text{Base Temperature} = 26°\text{C}: DH(26) = -5979.22 + 216.11 Tmm; \quad r = 0.922.$$

$$\text{Base Temperature} = 27°\text{C}: DH(27) = -5919.31 + 212.05 Tmm; \quad r = 0.926.$$

$$\text{Base Temperature} = 28°\text{C}: DH(28) = -4964.56 + 175.73 Tmm; \quad r = 0.918.$$

The number of night degree-hours is a parameter required for the estimation of the efficiency of night ventilation techniques[10]. The mean multiyear monthly night degree-hours for all base temperatures, as well as the corresponding standard deviation and the coefficient of variability were computed.

Low values of night degree-hours exist especially for June and September. However, for all months and for 26 and 27°C base temperatures considered, the number of night degree-hours rarely exceeds the 10% of the total daily degree-hours.

Knowledge of the daily maximum ambient temperature is required in order to estimate the mean peak cooling load. The mean monthly maximum ambient temperatures for the period 1977–1990 were computed. The mean maximum temperature varies between 26.4–32.3°C for June and September and between 31.2–35.5°C for July and August.

The absolute maximum, monthly ambient temperatures, Tam, is also a necessary information for the calculation of the maximum peak cooling load. The corresponding absolute maximum ambient temperatures for the period 1977–1990 were computed. Values range between from 30.3–41.7°C for June and September and from 34.8–42.8°C for July and August.

Correlation of the absolute maximum temperatures, with the degree-hours corresponding to the various base temperatures, has shown that the abs-max temperatures can be used as an index for the calculation of the degree-hours. It is found that the degree-hours can be estimated accurately using the following expressions:

$$\text{Tem base} = 25°\text{C}: DH(25) = -4188.35 + 141.60 Tam; \quad r = 0.798.$$

$$\text{Tem base} = 26°\text{C}: DH(26) = -4306.65 + 140.74 Tam: \quad r = 0.832.$$

$$\text{Temperature base} = 27°\text{C}: DH(27) = -4238.86 + 137.02 Tam; \quad r = 0.829.$$

$$\text{Temperature base} = 28°\text{C}: DH(28) = -3535.06 + 112.71 Tam; \quad r = 0.792.$$

The daily amplitude of the ambient temperature variation is a very useful parameter, especially for the calculation of the minimum daily temperatures. Minimum daily temperatures are observed during the night, and their knowledge is necessary for the evaluation of night radiation cooling techniques, radiative cooling. The frequency distribution of the daily amplitude of the ambient temperature for the period 1977–1990 is given for each month in Fig. 7. Daily amplitude values range from 1.9–16.5°C, with the higher frequencies observed from 9–12°C. A correlation of the daily temperature amplitude with the number of hours over 28°C, $HT28$, has shown that high amplitude values are combined with high temperature values, showing a persistance. In this case, high amplitude values are due to an increase of the daily maximum temperature, and they are not due to any drop of the minimum temperature.

Table 2. Monthly correlation coefficients of the multiple linear regression between the hours presenting a temperature over 25°C and these which are over 26, 27, and 28°C

June	July	August	September
0.839	0.939	0.979	0.888

4. APPLICATION TO OTHER LOCATIONS

In order to verify the applicability of the previously proposed techniques in other locations, data from Salonica, Greece for the period 1977–1987 have been analyzed.

It is found that the 28°C base degree-hours for all the summer months are linearly correlated with the corresponding degree-hours for Athens. The coefficient of linear regression is found equal to $r = 0.91$. Therefore, it can be concluded that the temporal variation of the climatic phenomena is similar in both climatic zones.

The used regression analysis, in order to calculate the monthly values of HTx for various temperature bases as a function of the number of hours over 25°C, eqn (1), has shown that the corresponding regression coefficients are also very high for Salonica as shown in the following set of data.

$$Tx = 26°\text{C}; \; a = -25,69; \; b = 0.917; \; r = 0.993.$$

$$Tx = 27°\text{C}; \; a = -49,22; \; b = 0.823; \; r = 0.984.$$

$$Tx = 28°\text{C}; \; a = -61.03; \; b = 0.711; \; r = 0.963.$$

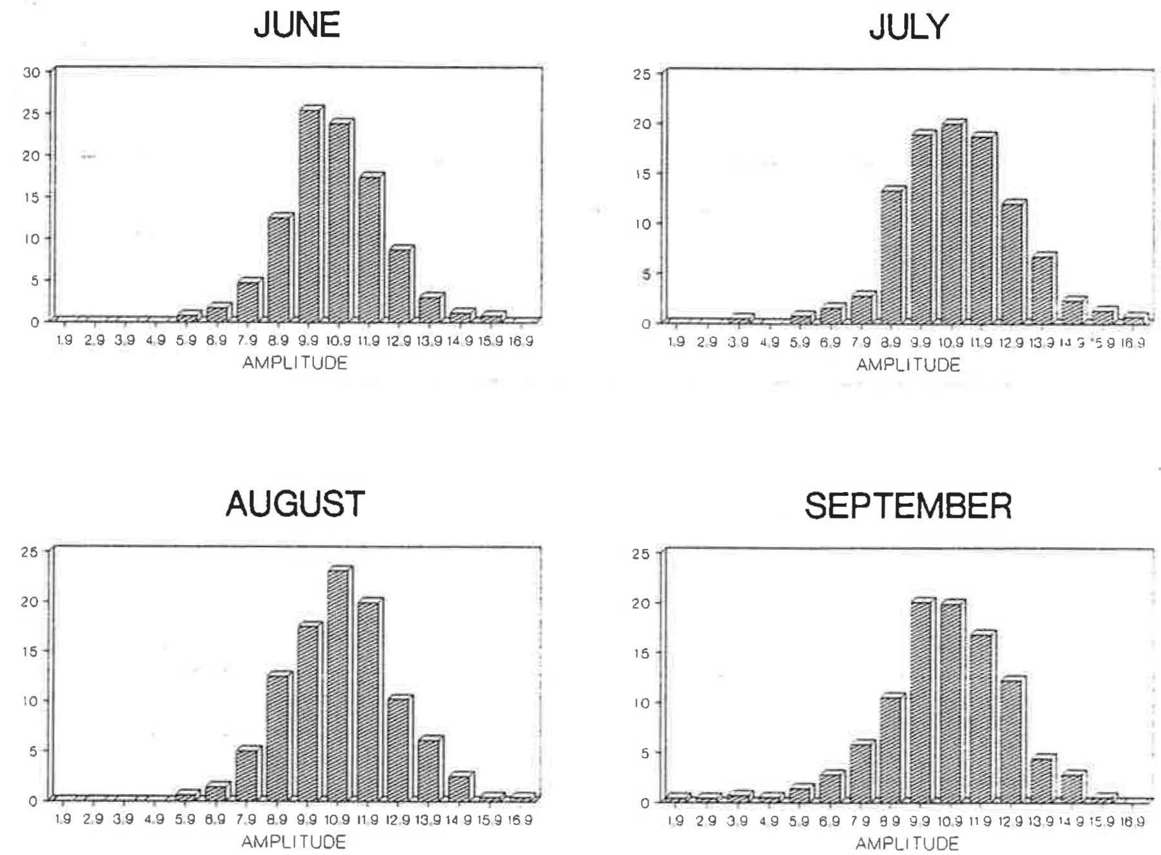


Fig. 7. Frequency distribution of the daily amplitude of the ambient temperature for June, July, August, and September.

The slope of the regression line is very similar for both locations. The difference of coefficient a is due to the different daily temperature swing observed in Athens and Salonica.

As stated previously, the monthly degree-hours (base 28°C) have been calculated as a function of the absolute maximum monthly temperature as well as of the mean monthly temperature. For both cases high correlation coefficients have been calculated. The following equations have been obtained:

$$DH(28) = -2192.03 + 71.74 Tam; \quad r = 0.838.$$

$$DH(28) = -1874.29 + 93.763 Tm; \quad r = 0.821.$$

The statistical important values of the obtained correlation coefficients show that the proposed methods can be applied successfully to other locations. In this case the "local" regression analysis, a and b coefficients, should be calculated.

Table 3. The coefficients a, b, and r of the linear regression analysis (eqn [1])

$T_x = 26°\text{C}$	$a = -22.26$	$b = 0.885$	$r = 0.994$
$T_x = 27°\text{C}$	$a = -48.35$	$b = 0.803$	$r = 0.989$
$T_x = 28°\text{C}$	$a = -76.22$	$b = 0.741$	$r = 0.983$

5. CONCLUSION

Summer ambient temperature data have been analyzed to fulfill design requirements of passive and hybrid cooling components and techniques. Data from Athens, Greece have been used. The necessary parameters for the evaluation of the above systems have been defined and calculated. Tools and expressions to estimate the required information have been proposed. The present analysis can be seen as a guideline whenever summer ambient temperatures data is used for cooling purposes.

NOMENCLATURE

DH	monthly degree-hours
HTx	number of monthly hours over the temperature base (Tx)
Tm	mean monthly temperature (°C)
Tam	absolute maximum monthly temperature (°C)
Tmm	mean maximum monthly temperatures

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PERFORMANCE OF AIR-HEATING COLLECTORS WITH PACKED AIRFLOW PASSAGE

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Abstract—This paper deals with a comparative theoretical parametric analysis of solar air-heating collectors with and without packing in the flow passage above the back plate. The air heater without packed airflow passage is the conventional type, comprising a glass cover in conjunction with an absorber and a back plate with the air flow beneath the absorber. The packed channel air heaters are of two types: the conventional type, with packing in the flow passage between the absorber and the back plate, and the other type consisting of two glass covers and a back plate with the black-painted packings (which function as absorbers) in the flow passage between the inner cover glass and the back plate. For comparing the performance, packings of different materials, shapes, and sizes with different void fractions are considered. To estimate the effective energy gain, the energy expended in pumping the air through the air heaters with different duct depths and lengths and with different airflow rates is computed.

1. INTRODUCTION

In the search for a suitable design of solar air-heating collectors for high-temperature applications, several improved designs, which have been the subject of many theoretical and experimental investigations, include a finned-channel air heater[1–2], a corrugated-absorber-plate air heater[3], a jet plate air heater[4], a packed-bed or porous absorber air heater[5–8], etc. The transfer of heat to/from fluid flowing through packed beds has been of industrial importance for some time, and interest has therefore been focused by numerous investigators on such systems[9–11]. The main reason for this interest is because the large surface area of the packing material and the turbulence producing airflow path through the bed provide for a rapid increase of heat exchange. However, a higher heat transfer coefficient to the flowing air in any solar air heater is invariably associated with a higher pressure drop in the system. On the one hand, it is necessary to ascertain that the pressure drop across the packed bed is large enough to ensure good flow distribution across the bed: on the other hand, it is necessary to keep the pressure drop very low so that the energy spent in pumping the air through the bed is low enough to make the system cost effective.

The solar heating literature reveals that although extensive research has been performed on exploring the use of packed beds as heat storage devices[12–14], little has been published on its use in solar air-heating collectors[7]. In addition, the implications of necessary trade-offs between the pumping power and heat transfer coefficient, and hence between the pumping power and the efficiency of packed-bed air heaters, are still not fully resolved. The objective of the present work, therefore, is to carry out a detailed theoretical parametric analysis of two different types of packed-flow-passage solar air heaters, one comprising a cover plate, an absorber, and a back plate with packings in the airflow passage between the absorber and the back plate, and the other consisting of two glass covers and a back plate with black-painted (absorbing) packings in the flow passage between the inner glass cover and the back plate. A parallel study has also been carried out on the conventional air-heater configuration with a single glass cover and air flow beneath the absorber to compare its efficiency with those of the packed channel air heaters. Computations of the electrical power expended in pumping the air through the collectors have been done to estimate the net or the effective gain of the systems for air channels of different lengths and depths, having different airflow rates with and without packings of different materials, shapes, and sizes.

2. THEORETICAL ANALYSIS

The three air-heater configurations considered for comparative investigations in the present paper are shown schematically in Figs. 1(a), (b), and (c) as type I, II, and III, respectively. The type I air heater is of the conventional type, consisting of a single cover plate of glass, a black-coated absorber plate, and a back (rear) plate with the air flow beneath the absorber. The type II air heater is of the conventional configuration of type I with the flow passage filled with packings. The type III air heater consists of two sheets of cover glass and a back plate; the passage between the inner glass cover and the back plate is provided with black-painted packings, which act as absorbers of solar radiation and transfer the heat to the air flowing through the packed channel.

The steady-state energy balance equations for the different components of the three air-heater types are presented below:

Type 1, cover plate,

$$I\alpha_1 + (h_{r21} + h_{c21})(T_2 - T_1) = U_t(T_1 - T_a); \quad (1)$$

absorber plate,

$$I\tau_1\alpha_2 = (h_{r21} + h_{c21})(T_2 - T_1) + h_{r23}(T_2 - T_3) + h_{c2f}(T_2 - T_f); \quad (2)$$