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EXPERIMENTAL EVALUATION OF ENERGY SAVINGS IN AIR-CONDITIONING USING METAL CEILING PANELS

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ABSTRACT

Space cooling using metal ceiling panels is analysed experimentally and theoretically. Measurements are performed in a 2.0x2.5x3 m^3 test chamber with a 1.80x2.16 m^2 cooling panel located on the undersurface of the chamber ceiling. Both experimental and theoretical analyses show that the dynamic response of the panel system in conjunction with the thermal comfort conditions is satisfactory for the climate of Greece. Under certain conditions the condensed water vapor may raise dripping problems, for which solutions are proposed. Energy savings exceeding 12.5% may be obtained owing to acceptable increase of indoor temperature, but further savings are possible owing to the higher temperatures of cooling water, which improve the efficiency of solar-driven absorption chillers.

Keywords: - Cooling ceiling panels, energy savings, condensation, thermal comfort

INTRODUCTION

Panel cooling of buildings may be obtained either by piping embedded in the ceiling slab or by a metal sheet incorporating cooling tubes, which is placed onto the undersurface of the ceiling. In the present study the latter case is considered, while the former one has been studied in previous publications [1,2,3]. With such systems, considerable energy savings may be effected, mainly because comfort is obtained with higher indoor temperatures, typically 28^{0}C, owing to direct radiative heat exchange of people's heads with the ceiling. The large cooling surfaces available allow higher temperatures of cooling water and use of high-performance solar-driven absorption refrigerators with improved solar collector efficiency. Also, cool storage in the ceiling and adjacent structural elements reduces peak loads.

Other advantages of panel cooling systems are the higher comfort levels obtained because of the elimination of noise associated with fan coils, the minimization of air motion and the increased uniformity of room surface temperatures. ASHRAE [4] provides a survey on panel cooling literature as well as construction details and design methods. An analytical solution of the related steady-state thermal problem is presented in reference [5], while the transient problem is solved by a combined analytical/numerical method in reference [6].

In the present work the thermal problem of space cooling using metal panels is analysed experimentally and the effects of the parameters involved are studied. Various aspects of practical importance are examined, including the problem of water vapour condensation on panel surface, the dynamic response of the system in conjunction with the thermal comfort conditions, as well as the energy savings obtained.

EXPERIMENTAL SET-UP AND MEASUREMENTS

A test chamber of floor area 2.0x2.5 m^2 and height 3.0 m has been built of prefabricated insulating panels ($U=0.435$ W/m^2 ^oC) composed of two specially formed laminated sheets, in which hard polyurethane foam is injected and cured in a heated press. Any desired value of thermal load within the test chamber is obtained by the use of electric resistances. On the undersurface of the ceiling of the test chamber a cooling panel is placed, which is composed of a 1.80x2.16 m^2 copper sheet incorporating 18 parallel copper pipes. The external diameter of the pipes is 10 mm, their direction is parallel to the 1.80 m dimension of the ceiling and the spacing (distance between the axes of adjacent pipes) is 12 cm. Cold water is supplied to the 18 panel pipes by a distribution pipe while a collection pipe collects the outcoming water, as shown in Figures 1 or 2, so that the pressure drops along the 18 different paths from the inlet to the outlet

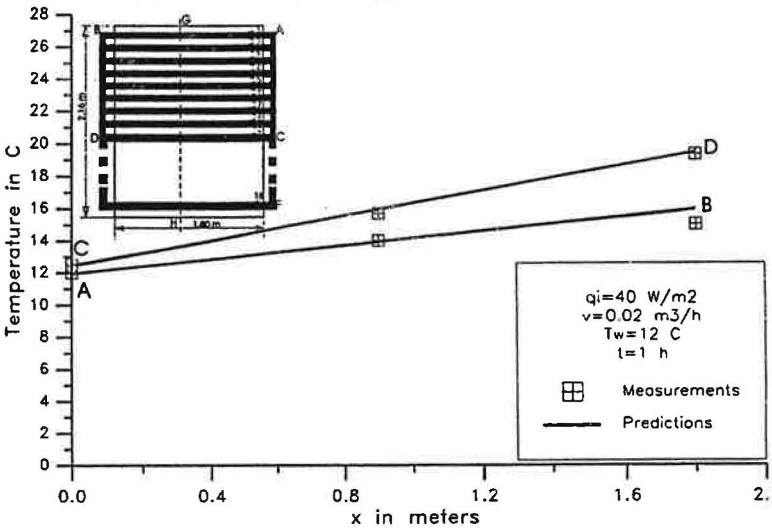

Figure 1. Comparison of measurements and predictions along the 1st and 9th panel pipes

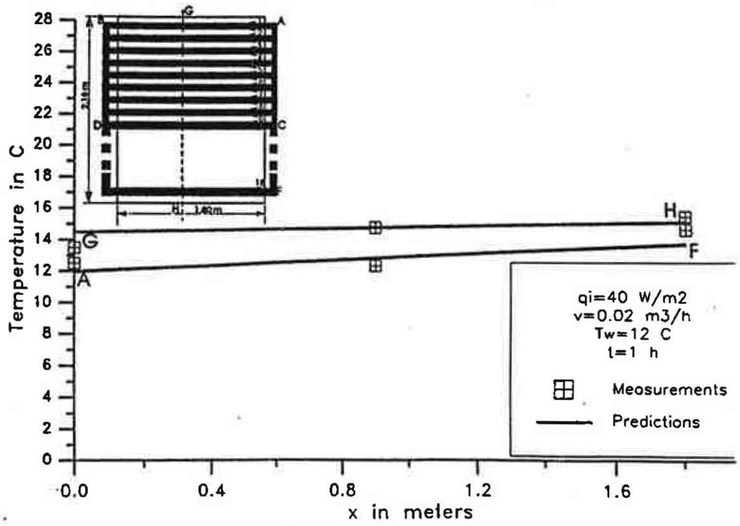

Figure 2. Comparison of measurements and predictions along lines AF and GH

of the panel are equal. Both distribution and collection pipes are made of copper with external diameter 22 mm.

The experimental set-up is completed by a cold-water-producing system, made up of an air-cooled chiller, a cold-water tank, pumps and related control devices. The produced cold water of desired temperature and mass flow rate flows through the pipes of the cooling panel and receives the indoor thermal load.

Iron-constantan thermocouples were installed at various locations on the parallel pipes, the copper sheet between them, the distribution and collection pipes, the inlet and outlet of the cooling water, as well as at various locations inside the test chamber. Temperature measurements, performed with a 1-min time step, were stored and processed in a PC. The parameters varied during the experiments were the inlet temperature and flow rate of the cooling water, T_w=5-20°C and V=0.02-2 m^3/h, respectively, and the indoor heating power density q_i= 0-150 W/m^2.

Examples from a great number of measurements obtained are given in Figures 1 and 2. The former shows the temperature variation along the 1st and 9th panel pipes, while the latter presents the corresponding variations along the distribution pipe AF and the panel central line GH. The curves in both figures correspond to the temperature variations predicted theoretically, as described in [6].

THE CONDENSATION PROBLEM

The usual assumption of linear variation of indoor humidity ratio W in terms of the dry bulb temperature T yields

$$dW/dT = (W-W_s)/(T-T_s) \qquad (1)$$

where subscript s denotes saturation conditions.

Heat quantity q_{L1} (in J/kg dry air), which is absorbed by the panel for a decrease of the humidity ratio dW, may be expressed in terms of the heat of condensation r (in J/kg H_2O) as

$$q_{L1} = r dW \qquad (2)$$

or, because of Equation (1)

$$q_{L1} = r(W-W_s)dT/(T-T_s) \qquad (3)$$

Heat q_{L2} (in J/kg dry air) added to the room during a time interval dt because of the indoor latent load Q_L (in W/m^2) may be expressed as

$$q_{L2} = vQ_L dt/H \qquad (4)$$

where v (in m^3/kg dry air) is the specific volume of the moist air, and H ≅ 3 m is the height of the room expressed in m^3 of moist air per m^2 of floor. The total change of humidity ratio dW during a time interval dt may be analysed with respect to the decrease dW_1, which is due to the water vapor condensation on the panel surface and to the increase dW_2 which is due to the latent heat added to the room, i.e.

$$dW = dW_1 + dW_2 = (q_{L1}/r) + (q_{L2}/r) \qquad (5)$$

Substitution of q_{L1} and q_{L2} from Equations (3) and (4), respectively, into Equation (5), gives after rearrangement

$$r dW/dt = vQ_L/H + r(W-W_s)dT/[(T-T_s)dt] \qquad (6)$$

The corresponding differential equation for the room sensible heat Q_S (in W/m²) is

$$C dT/dt = Q_S - h\,(T-T_p) \qquad (7)$$

where C (in J/°C m²) is the room thermal capacitance per m² of floor, and h (in W/m² °C) and T_p are the heat transfer coefficient and the temperature of the panel surface, respectively.

The set of differential Equations (6) and (7) was solved numerically, by using the finite-difference method. An example of the results obtained is shown in Figure 3, which presents the predicted variations of T and W in terms of time, for various values of the inlet temperature T_w of the cooling water in the panel. Temperature T_w is used in the calculation of the panel surface temperature T_p, contained in Equation (7). The remaining parameters are Q_L= 20 W/m², Q_S = 40 W/m², room thermal capacitance corresponding to indoor moist air and to 1 kg of wood per m² of floor area. The initial conditions correspond to the outdoor T and W values at time 12:00 h of 21 June in the Athens area, i.e. it is assumed that when the panel starts operating (t=0) the indoor conditions are identical to the outdoor ones. The latter are known from a statistical processing of 11- years hourly measurements [7]. The calculation of T(t) and W(t) stops at T = 28°C, which is the desired dry bulb temperature for thermal comfort, as explained later. The main purposes of these calculations are (a) to examine the dynamic response of the panel system and estimate the time needed to obtain conditions of thermal comfort, and (b) to quantify the unwanted condensation on the panel surface.

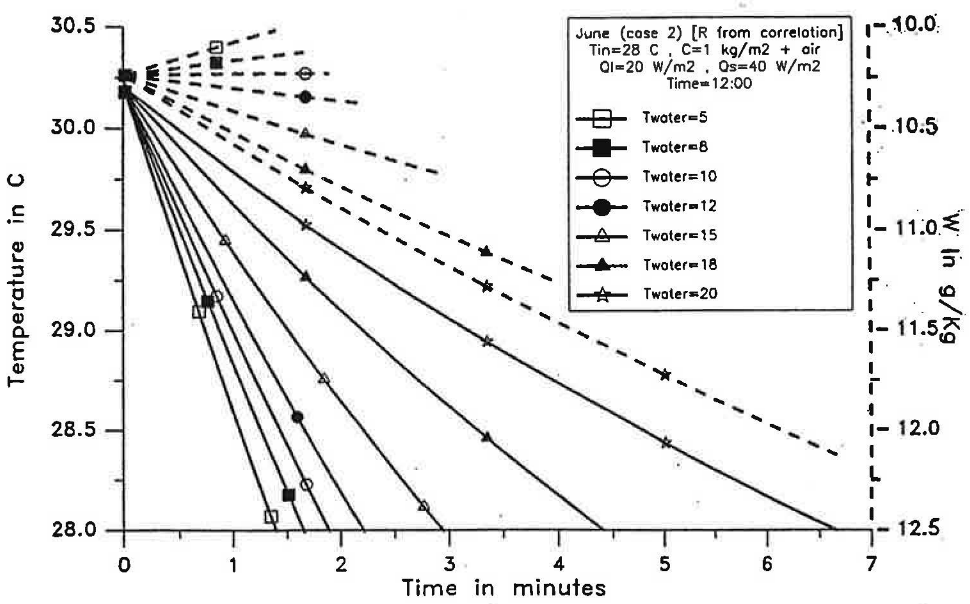

Figure 3. Variations of temperature and humidity ratio in terms of time

Calculations similar to those of Figure 3 were performed with initial conditions the outdoor T and W values of all hours of the typical summer in Athens, for which T is higher than 28°C. An example is shown in compact form in Figure 4, which corresponds to 21 June and gives the mass of condensed water vapor for operation starting at times 10, 11, ..., 18:00 hrs for which it is T>28°C. For example, for T_w = 10 °C, if the system is set on at 12:00 h and off when the indoor temperature drops to 28°C, the mass of condensed water vapour will be 0.28 g H_2O per kg of dry

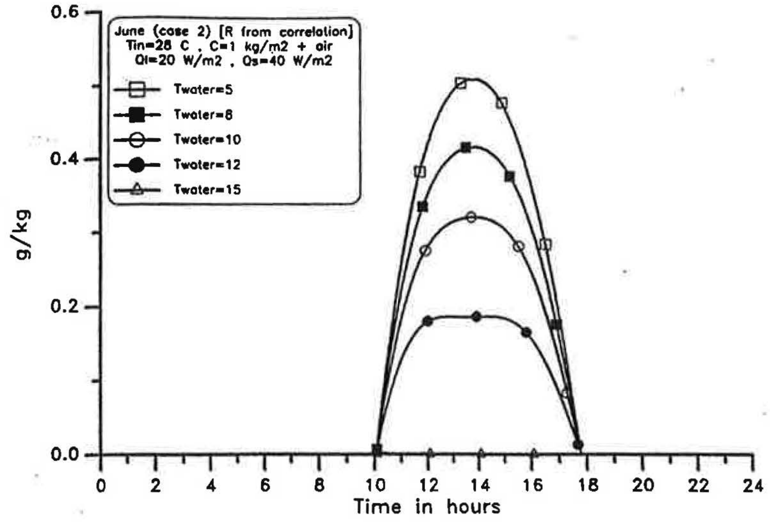

Figure 4. **Mass of condensed water vapor for operations starting at times 10, 11,...,18:00 hrs**

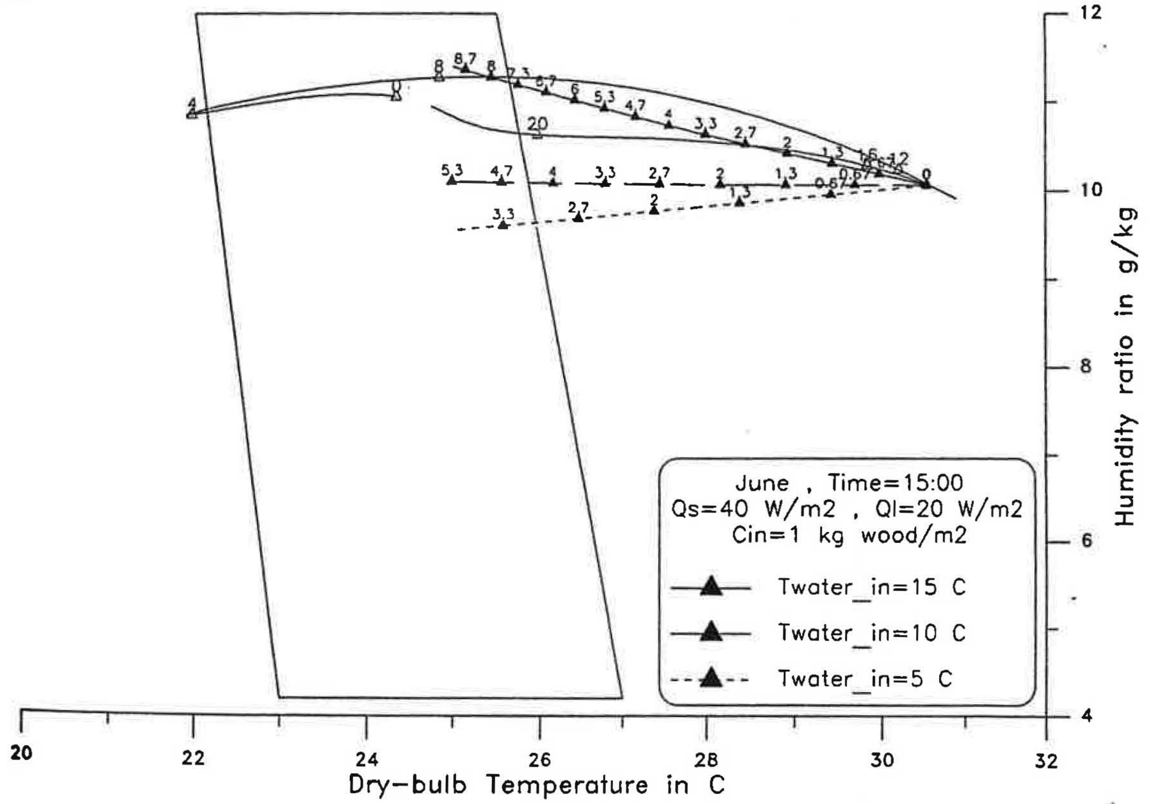

Figure 5. **Psychrometric chart with comfort envelope and indoor point path**

air. Therefore, the total mass of condensed water vapour in our test chamber of volume 2x2.5x3 m^3, is (0.28 g H_2O/kg dry air) x (2x2.5x3 m^3)/(0.867m^3/kg dry air) = 4.84 g H_2O, which is assumed to form, on the 1.80x2.16 m^2 panel surface area, a film of thickness (4.84 g) x (10^{-6} m^3/g)/ (1.80x2.16 m^2) = 1.24x10^{-6}m = 1.24x10^{-3}mm. It is, therefore, obvious that in the present example, the condensed water vapor will not raise dripping problems.

THERMAL COMFORT

The procedure developed in the previous section has been applied to predict the path of the point corresponding to indoor conditions in the psychrometric chart, in terms of time, during panel cooling. The purpose of these calculations is to examine the conditions under which the indoor point can enter the thermal comfort envelope (as defined by ANSI/ASHRAE Standard 55-1981) in a reasonable period of time, starting from the outdoor conditions.

An example of the results obtained is shown in Figure 5, which corresponds to 21 June in Athens for a room with sensible and latent loads 40 W/m² and 20 W/m², respectively. The Figure shows in the W-T plane (a) the summer comfort envelope, (b) the path of the outdoor point during a 24-h period (21 June), with the hours 0,4,8,12,16,20,0 marked by white triangular symbols, and (c) the paths of the indoor point for three values of the inlet temperature (T_w= 5,10 and 15 °C) of the cooling water in the panel. The three paths start from the outdoor point corresponding to 15:00 hrs and are marked by black triangular symbols showing the time from the beginning in minutes. It is seen that for T_w=5, 10 and 15°C, the time needed for the indoor point to enter the comfort envelope are 3.3, 4.7 and 8 min, respectively.

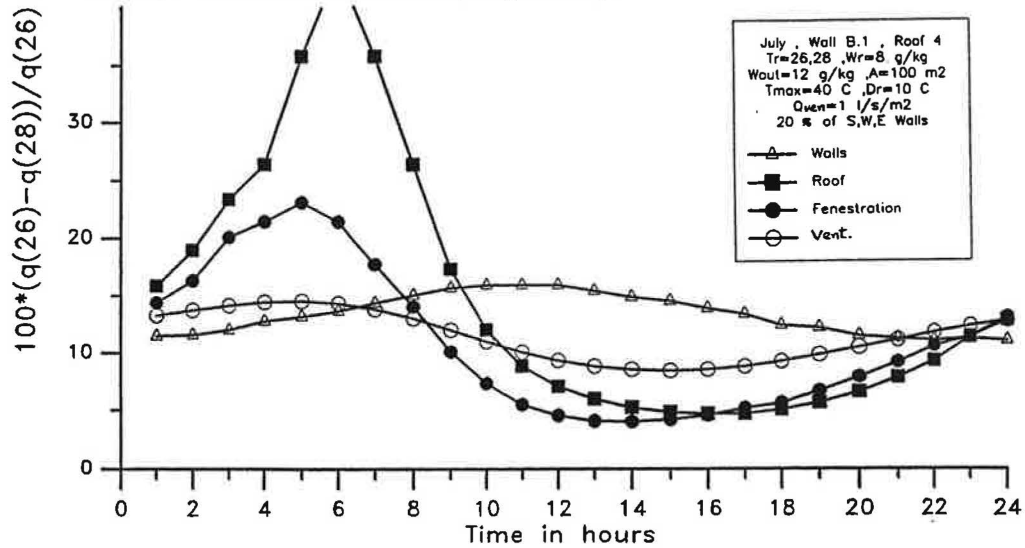

Figure 6. Increase % of cooling loads for 26°C indoor temperature instead of 28°C

ENERGY SAVINGS

The various reasons for which ceiling panel cooling may offer energy savings over convective cooling systems have been mentioned in the introduction. In the present section attention is focused in the increase of the acceptable indoor temperature, which seems to contribute more to the total energy saved. Related experiments in the test chamber, which are still in progress, suggest that in ceiling panel cooling, the summer comfort zone specified in ANSI/ASHRAE Standard 55-1981 [8] might be shifted towards the higher temperatures by up to 2°C. To quantify the resulting energy savings, calculations of cooling load were performed under various conditions, using the ASHRAE [8] method. Figure 6 shows the % increase of each kind of cooling load in terms of time for an 100-m² house, if the indoor temperature drops from 28°C to 26°C, while all other conditions (written on the figure) remain the same. Cooling load increases of up to 40% are observed.

Among the parameters, which have some effect on the percentage of the energy saved are, except from the indoor conditions, the building characteristics (size, envelope composition, orientation, fenestration percentage), outdoor conditions and ventilation rate. Related examples are given in Figures 7 and 8, which compare the total energy consumption for a 24-hour period, under indoor temperatures 26 and 28°C, in terms of building floor area. The percentage of energy saved, which is also given, exceeds 12.5%. The different parameters in Figures 7 and 8 refer to wall and roof compositions (according to ASHRAE codes), to maximum outdoor temperatures and to orientations of fenestration, as stated in the figures.

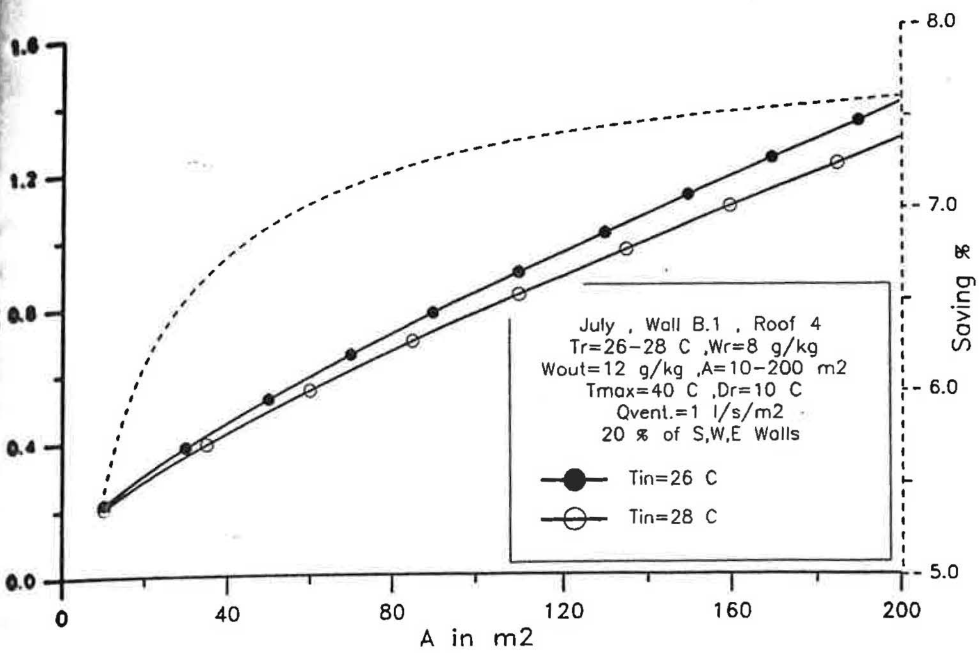

Figure 7. Total energy consumption and percentage of energy saved

CONCLUSION

Some aspects of air-conditioning using metal ceiling panels have been examined experimentally and theoretically. The agreement between measurements and predictions is, generally, satisfactory if it is taken into account that it is not always possible to obtain in the experiment the exact conditions, used in the theoretical solution (i.e. effect of outdoor conditions, effect of oscillations in the inlet temperature of cooling water, distribution of indoor heating load, etc.).

Both experiments and theoretical analysis showed that the dynamic response of the panel system is satisfactory for the climatic conditions of Greece. Starting from the outdoor conditions, the time needed by the system to reach the thermal comfort conditions is, generally, reasonable and can be set to a desired value by regulating the inflow temperature of cooling water. Indicative values for the climatic conditions of Greece may be summarized as follows: For low indoor loads, the time needed is less than 3 min for $T_w<10^oC$ and less than 10 min for $T_w<15^oC$, while for medium to high indoor loads, the corresponding times may be doubled.

It should be emphasized that the ASHRAE comfort envelope shown in Figure 5 corresponds to conventional air-conditioning systems. For ceiling panel cooling, higher indoor temperatures (typically 28°C, as explained earlier) are acceptable and therefore the envelope should have been displaced by about 2°C towards the higher temperatures. In this case, in the example of Figure 5, the times needed for the indoor point to enter the comfort envelope would be 2, 2.7 and 4.7min for T_w=5,10 and 15°C, respectively.

The condensation problem on the surface of the panel has been examined experimentally and theoretically. Both analyses showed that under certain conditions the condensed water vapor may raise dripping problems, which may be solved by locating the panels on the ceiling under a slight inclination or in the upper part of the walls from the height of peoples' heads up to the ceiling.

According to the present analysis, cooling systems using metal ceiling panels obtain energy savings, which may exceed 12.5%. This percentage, which is due to the acceptable increase of indoor temperture, may be considerably increased because of the available large cooling surfaces,

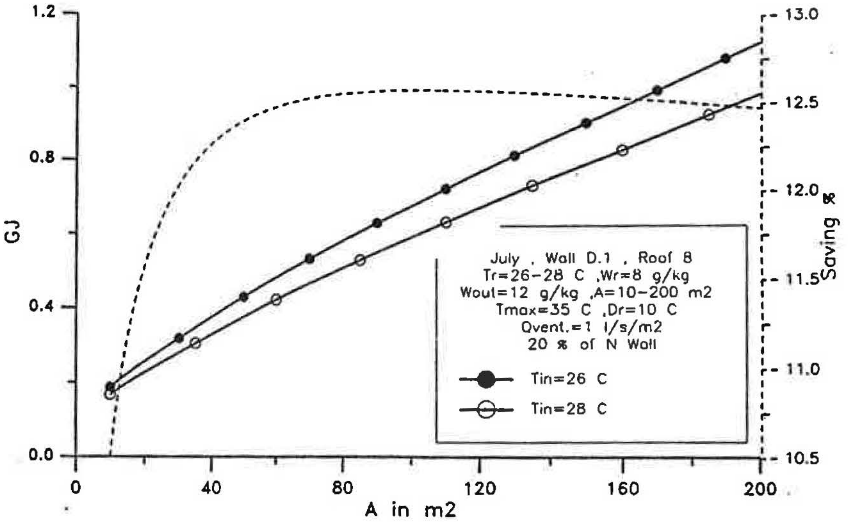

Figure 8. As in Figure 7 under different conditions

which allow higher temperatures of cooling water and, therefore, use of higher performance solar- driven absorption chillers with improved solar collector efficiency.

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