# Dynamic Simulation of Combination of Evaporative Cooling with Cooled Ceiling System for Office Room Cooling

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***ABSTRACT***

*In view of the emerging radiant cooling technologies in the European market, a dynamic building thermal analysis program ACCURACY is enhanced to be able to do cooling-load calculations and annual energy analysis for rooms with cooled ceiling climate system. The program was validated against the experimental investigation of the dynamic response of a test room to step heating and step cooling. Specifically, the validated program is used to perform a simulation of the combination of evaporative cooling with cooled ceiling technique for office building cooling purpose. The paper will present the construction principles of the program as well as the simulation results.*

## INTRODUCTION

Currently, cooled ceiling technique is getting popular in many European countries and has found its use in many office buildings as an air-conditioning alternative(Mertz 1992; Wilkins 1992). A cooled ceiling can have many variations, but this article will focus on the horizontal plate type. In this type of design, specially made cooling panels are installed as part of a false ceiling, through which cold water flows and extracts heat from the room (Figure 1). Some manufacturers also produce ceiling panels that function as air ducts, through which ventilation air is preheated by internal room heat before entering the room through air diffusers. Various ventilation systems can be combined with the water ceiling system to provide the required outside air and latent cooling. Usually separate heat devices are located conventionally underneath the windows. In some cases the ceiling panels are also used for heating purposes.

The cooled ceiling system has some unique characteristics. Among other things, the horizontal panels will extract heat by both radiation and convection. On the other hand, the existence of the cooled panel surface will lower the radiant temperature in a room. Therefore, special cooling load calculation methods will be required for the design of this system. In the present research, a new methodology is developed, which is to combine the thermal dynamic modelling of building elements and the ceiling panels with each other and to integrate the thermal comfort indices in the calculation procedure. For this purpose, the computer program ACCURACY (Chen 1988; Chen and v.d. Kooi 1988; Chen and v.d Kooi 1990) is enhanced. The program was validated against the experimental investigation of the dynamic response of a test room to step heating and step cooling.

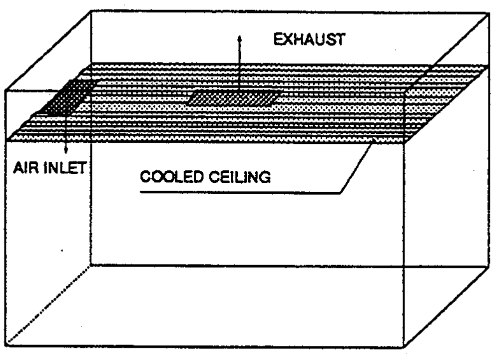


Figure 1. Construction of the horizontal cooled ceiling panels in a room

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The validation shows that the program gives good predictions of the dynamic load and temperature responses for both air-ceiling and water-ceiling systems(Niu and v. d. Kooi 1993).

The large cooling capacity of water ceiling system has some new application potentials. It can use the relatively less cool water from a cooling tower to cool a room. In view of the CFCs' damage on ozone layers, replacing mechanical refrigeration in building cooling will have significant benefit on environment protection. Therefore, using the validated program ACCURACY and the Dutch weather data of 1971, hour-by-hour dynamic simulations are performed for an office room with cooled ceiling systems. The purpose is to explore the possibilities of the combination of evaporative cooling in the Dutch climate. This article will present these preliminary investigation results.

## MATHEMATICAL MODELLING

The program ACCURACY is a combination of transfer function method(for heavy walls) and finite difference method(for ceiling panels) for the modelling of room elements. Specifically, unlike most cooling load programs, the internal air flow patterns and distributed temperatures obtained by CFD (Computational Fluid Dynamics) approach will be used interactively in the simulation procedure for the calculation of surface convection heat transfer. In the present enhancement, the CFD simulation results will be used to calculate the inter-zonal air exchange between the cavity above the ceiling and the room.

In our simulation approach, a room will be divided into two types of elements - zonal air volumes and enclosure surfaces, which are defined respectively as:

*a zonal air volume* - a volume of air which is geometrically isolated or enclosed by building envelops such as walls, windows, doors as well as internal partitions; according to this definition, the air above the false-ceiling in a room will be mathematically treated as a volume of air that is distinct from the volume of air in a room;

*a surface* - a surface which forms part of the entire enclosure surface facing the air volume, and may be further distinguished into three different sub-types: transparent surfaces(window surface), opaque surface(wall surfaces), and active surfaces (cooled ceiling, or heating radiator). Including the cooled ceiling panel surfaces as one of the basic elements of the simulated building will enable us to analyze the dynamic behaviour in a realistic approach, specifically the radiant effects on thermal comfort and energy consumption will be directly taken into account.

Applying the energy conservation law for each of the air volumes and surfaces will yield the following equations. For an air volume:

$$V_R \rho C_p \frac{dT}{d\tau} = Q_{ic} + \sum_{i=1}^{N} Q_{c,i} + Q_{ext} + Q_{inf} - Q_{exf} \tag{1}$$

where, $Q_{ic}$ is the direct convective heat from internal heat sources, including those from occupants; $Q_{c,i}$ is the convective heat from the individual enclosure surfaces; $Q_{ext}$ is the heat extraction by mechanical ventilation; $Q_{inf}$ and $Q_{exf}$ are the heat carried in and out by infiltration and ex-filtration respectively, they include those caused by the air exchange from adjacent-zones. The values of $Q_{inf}$ and $Q_{exf}$ between the room and cavity above the ceiling will be identified in the experimental validation.

For an enclosure surface:

$$q_s + q_{ir} - (q_c + \sum_{k=1}^{N} q_{r,k}) = q_t \tag{2}$$

where, $q_s$ is the solar radiation through the window(s) that is apportioned to the surface by absorption, $q_{ir}$ is the radiant heat from internal heat sources, $q_c$ is the convective heat from the surface to the air volume, $q_{r,k}$ is the long wave radiative heat exchange with other surfaces, $q_t$ is the heat transmission from the other side of the surface, which will involve different processes for the different surface types mentioned above. For a multi-layer wall, $q_t$ will be calculated by the Z-Transfer-Function method; for a multi-layer window, $q_t$ is calculated by a special procedure which takes into account the multiple reflections and absorptions of the glass panes as well as venetian blinds. These procedures can be found in the literature(Chen 1988; Chen and v.d. Kooi 1990). In this paper only modelling of $q_t$ for the cooled ceiling surfaces will be described in more detail.

It was found by a few researchers(Aiulfi et al. 1985; Morant and Strengnart 1985) in their simulation of a heating radiator in a room that a single element model for the heating radiator is accurate enough for energy simulation purpose while a multi-element model is necessary for dynamic control analysis. Therefore, in the present modelling all cooled ceiling panels will be treated as one element.

One dimensional heat conduction analysis is applied to calculate the heat transmission into the panel surface $q_t$(Fig.2). The energy balance equation for a unit area of the panel shell is

$$q_t = \alpha_w(T_p - T_w) + \alpha_r(T_{p,o} - T_p) - (\delta \rho C)_m \frac{dT_p}{d\tau} \quad (3)$$

where, $\alpha_w$ is the convective heat transfer, $T_w$ is the average water or air temperature inside the panel, $T_p$ is the panel surface temperature, $(\delta\ \rho\ C)_m$ is the thermal capacity of the metal shell, $T_{p,o}$ is the shell temperature of the opposite panel side, and $\alpha_r$ is the radiant heat transfer coefficient between the two sides, and for water panel $\alpha_r = 0$. The conductance resistance has been omitted for the shell since it is rather thin and highly thermal conductive. For the water layer or air layer, the energy balance equation is:

$$\Delta A\, \alpha_w(T_p - T_w) + \Delta A\, \alpha_w(T_{p,o} - T_w) = G C_{p,w}(T_{w,o} - T_{w,i}) + \Delta A(\delta \rho C_p)_w \frac{dT_w}{d\tau} \quad (4)$$

where $\Delta A$ is area of the element, $T_{w,i}$ and $T_{w,o}$ are the inlet and outlet water temperature, and G is the water flow rate, $C_{p,w}$ is the specific heat of the water. $T_w$ is the average water temperature in the whole ceiling and is taken as a mean value of $T_{w,i}$ and $T_{w,o}$.

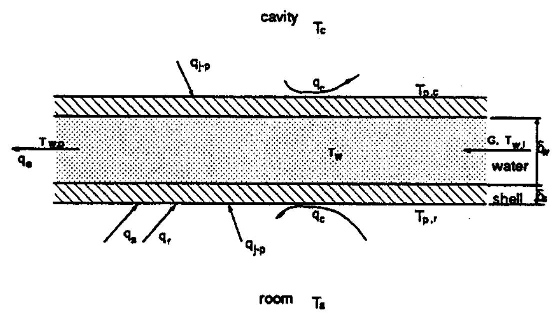


Figure 2. Thermal balances of the ceiling panel element

Different construction modules of the ceiling panels are illustrated in Figure 3. They are a water-pipe clipped with metal sheet(Type-I), a water pipe with closed fins(Type-II), and an air-duct with a large width/height ratio(Type-III). Equations (3) and (4) can apply to all the module types by including the fin effectiveness or the heat resistance between the water-pipe and the clipped-on metal sheet in the $\alpha_w$ value.

## THERMAL COMFORT

The radiant temperature must be taken into account in cooling load calculations. Therefore, the mean radiant temperature will be calculated using the formula

$$t_{mr} = \sum_{k=1}^{N} t_k F_{k-p} \quad (5)$$

where, $F_{k-p}$ is the view factor of the person from the surface k. In combination with the simulated room air temperature $t_a$ and estimated air velocity $V_a$ and partial water vapour pressure $P_a$, the thermal sensation for the occupants with metabolic rate M and clothing level $I_{cl}$ can be predicted in terms of the index PMV(Predicted Mean Vote)(ISO, 1984). Specifically, at known air flow velocity and normal humidity, the operative temperature $t_o$ can be used as a control variable in the simulation procedure for energy estimation. The following formula is used for the calculation of $t_o$:

$$t_o = a_a t_a + (1 - a_a) \sum_{k=1}^{N} t_k F_{k-p} \quad (6)$$

where $a_a$ usually takes the value of 0.55.

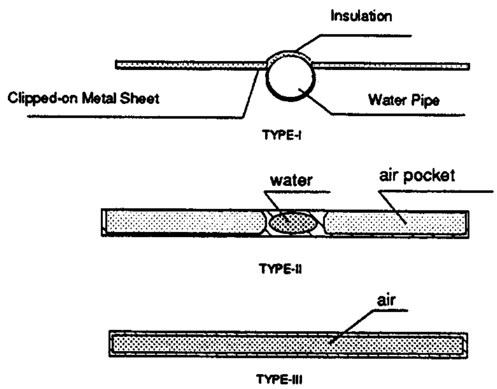


Figure 3. Different types of ceiling panels

## NUMERICAL PROCEDURES

Basically, the equations mentioned above form a set quasi-linear simultaneous equations about the room air temperature, individual surface temperature(including the cooling panel surface temperature that is required to satisfy the thermal comfort). Numerical solutions of the equation set will allow us to further calculate the required supply water or air temperature. Rearranging the equations also allows us to simulate the changes of room air temperature at a given supplied water or air temperature, which simulates the start-up process or the processes occurring in a room when the equipment has reached its maximum capacity.

## EXPERIMENTAL VALIDATION OF THE SIMULATION PROCEDURES

The models introduced involve models of many sub-processes. Parameters describing the convective processes are rather uncertain factors. Particularly for a room with a cooled ceiling system, the air exchange between the cavity above the ceiling and the room has considerable influence on both the heat extraction by the ceiling panels and air flow in the room. Therefore, special measurements are conducted in a test room to identify the amount of this air exchange. By tracer-gas-technique, this rate, when caused by natural convection alone, was found to be about 5 times/hour in terms of the air-change-rate for the room volume. Especially, CFD simulation of this convective process was performed, which gave a reasonable prediction of air exchange (Figure 4). Since this air exchange rate is influenced by many factors in practice, such as the area ratios of gap to panel and panel to ceiling, as well as the internal load and air inlet outlet locations, CFD simulation of this air exchange will be able to account for the variations of these factors. Then the enhanced simulation program ACCURACY was found to give a good prediction of the thermal dynamic behaviour and cooling load of a room with cooled ceiling systems. This validates the feasibility of the fundamental modelling techniques as well as the numerical algorithms(Niu and v. d. Kooi, 1993).

## SIMULATION OF EVAPORATIVE COOLING

The simulated system is illustrated in Figure 5. The water is cooled to within 2.5 to 4°C of the wet-bulb temperature $T_{wb}$ of the ambient air. At this stage of the simulation, the dynamic behaviour of a cooling tower is very much simplified: the thermal storage effects of water-tank and the heat exchanging coils for an indirect system are not taken into account. The control schedule is: during the occupied period, the operative temperature of the room is maintained at 22°C; during the night(unoccupied), cooled water is supplied to maintain the room operative temperature in the room at a lower temperature of 18°C, so that more night cooling energy can be stored in the thermal mass of the floor. What will happen is that the operative temperature tends to exceed the required comfort temperature at peak load during the day (Figure 6). Then accordingly the PMV (predicted mean vote) is calculated, according to which the so called PMV weighted number of temperature-exceeding hours (WTH) will be calculated if PMV ≥ 1.5 (Brouwers and Linden 1989). For instance, WTH = 1.5 if PMV = 0.7 for each hour. A room is considered to be thermal-comfortably acceptable if the total WTH in a year are less than 150. Therefore, the total WTH in a year is used as an index to estimate the thermal performances of a cooling system.

The above simulation is performed for an office room, which has the dimension of width × depth × height = 3.6 m × 5.1 m × 2.7 m. 30% of the facade (facing south) is double-glazing window with solar-shading outside. The room is assumed to have a 300 mm thick concrete floor, light gypsum-plate side walls and a heavy outside wall with good insulation layer outside. The room is assumed to be situated in the middle storey of a building with the same rooms below and above. The room is occupied by 2 persons from 8:00am to 18:00pm, and is naturally ventilated with ventilation rate 2

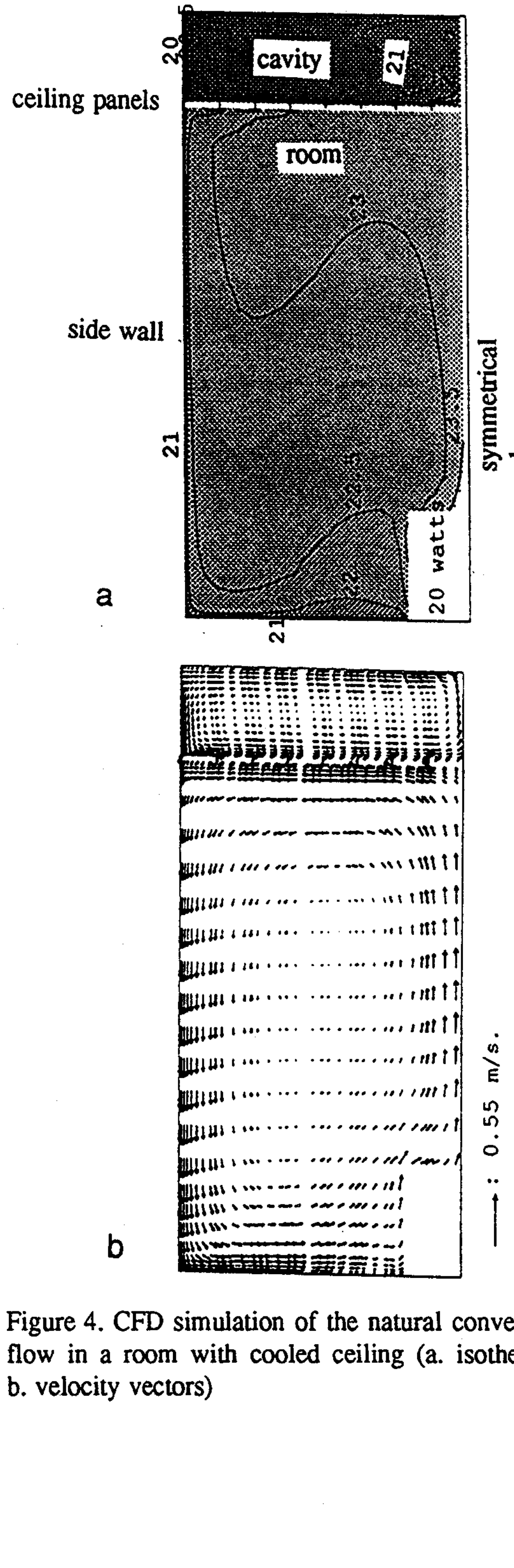


Figure 4. CFD simulation of the natural convective flow in a room with cooled ceiling (a. isotherms; b. velocity vectors)

ach with un-conditioned outside air. The ceiling panel chosen to be simulated has an overall heat transmission value of 16 $W/m^2 \cdot K$ ceiling area, taking into account the heat extraction from the upper side of the panels. The cool water flow rate is determined in such a way that the temperature rise over the ceiling panels is about 1 K at peak load. Up to now, three cases have been simulated.

*Case 1*: The internal heat load in the occupied period is 60 $W/m^2$ floor area, 70% of the ceiling are horizontal water-panels, the supplied water temperature is ($T_{wb}$ + 2.5) °C. It is found that during the whole summer, the overall WTH in the room is 77 (which is equivalent to the statement that the hours at which operative temperatures exceed 25°C are less than 50), which meets the thermal comfort requirement that WTH must be less than 150.

*Case 2*: Suppose the cooling tower has a smaller capacity and can only cool the water to about 4 K above the wet bulb temperature, then it was found that the number of hours at which temperatures exceed the comfort range is too large. The WTH is about 220, higher than the criterion 150.

*Case 3*: The room is supposed to have smaller internal heat load: 50 $W/m^2$. The supplied water temperature is *($T_{wb}$ + 4) °C*. The resultant WTH is 130, less than 150.

It can be seen that the internal load is rather high in the cases investigated. It can be expected that, for a room with lower internal load, e.g., 30 or 40 $W/m^2$ floor area, a low percentage of cooled ceiling panels will be required. The assumption that water can be cooled to within 2.5 to 4 °C of the wet-bulb temperature falls well within the capacity of many types of cooling towers available in the market. Therefore, it may be concluded that evaporative cooling in combination with cooled ceiling system is able to provide good thermal comfort for office buildings in the Dutch climate. Detailed dynamic modelling of the cooling tower and the water pump as well as the air fan for the tower will allow us to perform further cost-effective analysis of the system.

## CONCLUSIONS

1. The modelling approach presented in this paper, which is to use CFD results to calculate air temperature distribution as well as inter-zonal air exchanges, was validated in a climate room and it was found that the program ACCURACY thus constructed can give a rather good prediction of the thermal dynamic behaviours;
2. The application of the validated program for the numerical simulation of evaporative cooling of office buildings in combination with cooled ceiling system shows a promising potential of this passive cooling scheme in the Dutch climate.

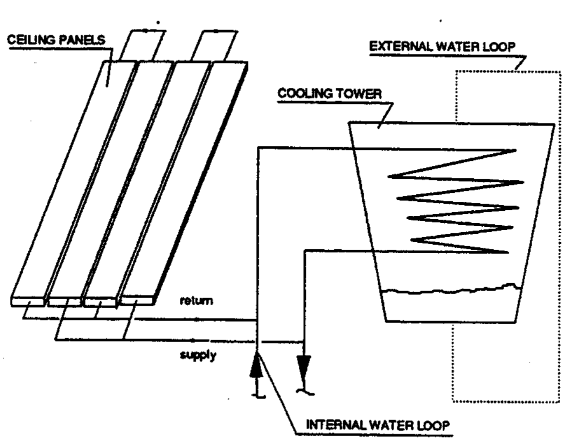


Figure 5. Schematic for the combination of cooled ceiling with a cooling tower (indirect system)

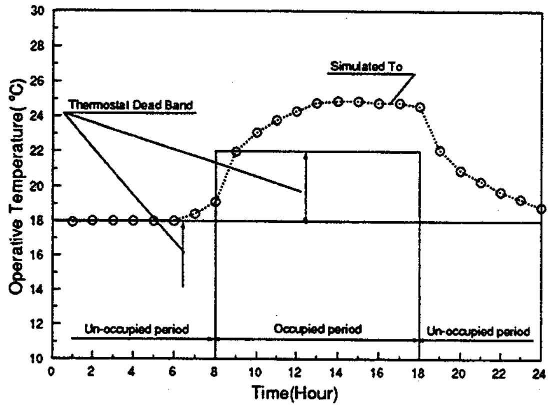


Figure 6. Assumed control schedule and simulated operative temperature variations on the day July 11, 1971

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