Hydronic Radiant Cooling
Overview and Preliminary Performance Assessment

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Abstract

A significant amount of electrical energy used to cool non-residential buildings is drawn by the fans used to transport the cool air through the thermal distribution system. Hydronic systems reduce the amount of air transported through the building by separating ventilation and thermal conditioning. Due to the physical properties of water, hydronic distribution systems can transport a given amount of thermal energy using less than 5% of the otherwise necessary fan energy. This savings alone significantly reduces the energy consumption and especially the peak power requirement. This survey clearly shows advantages for radiant cooling in combination with hydronic thermal distribution systems in comparison with the *All-Air Systems* commonly used in California.

The report describes a literature survey on the system's development, thermal comfort issues, and cooling performance. The cooling power potential and the cooling power requirement are investigated for several California climates. Peak-power requirement is compared for hydronic radiant cooling and conventional *All-Air-Systems*.

Introduction

Cooling non-residential buildings in California contributes significantly to the electrical power consumption and the peak power demand. Part of the electrical energy used to cool buildings is drawn by fans transporting cool air through the ducts. This electricity is heating the conditioned air, and therefore, is part of the internal thermal cooling peak load. Usibelli et al. [1] found that the typical thermal cooling peak load for office buildings in California can be divided as follows: 31% for lighting, 13% for people, 14% for air transport, and 6% for equipment. External loads account for only 36% of the thermal cooling peak load.

DOE-2 simulations for different California climates using the California Energy Commission (CEC) base case office building show that, at peak load, only about 10% to 20% of the supply air is outside air [2]. Only this fraction of the supply air is necessary to properly ventilate the buildings to maintain a high level of indoor air quality. For conventional HVAC systems the difference in volume between supply air and outside air is made up by recirculated air. The recirculated air is necessary to keep the temperature difference between supply air and room air in the comfort range. The additional amount of supply air, however, often causes draft[*]) as well as indoor air quality problems by equally distributing pollutants throughout the building.

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[*]) Draft: Air Movement in an occupied enclosure causing discomfort

% OF TOTAL COOLING LOAD

30 —

20 —

10 —

0 —

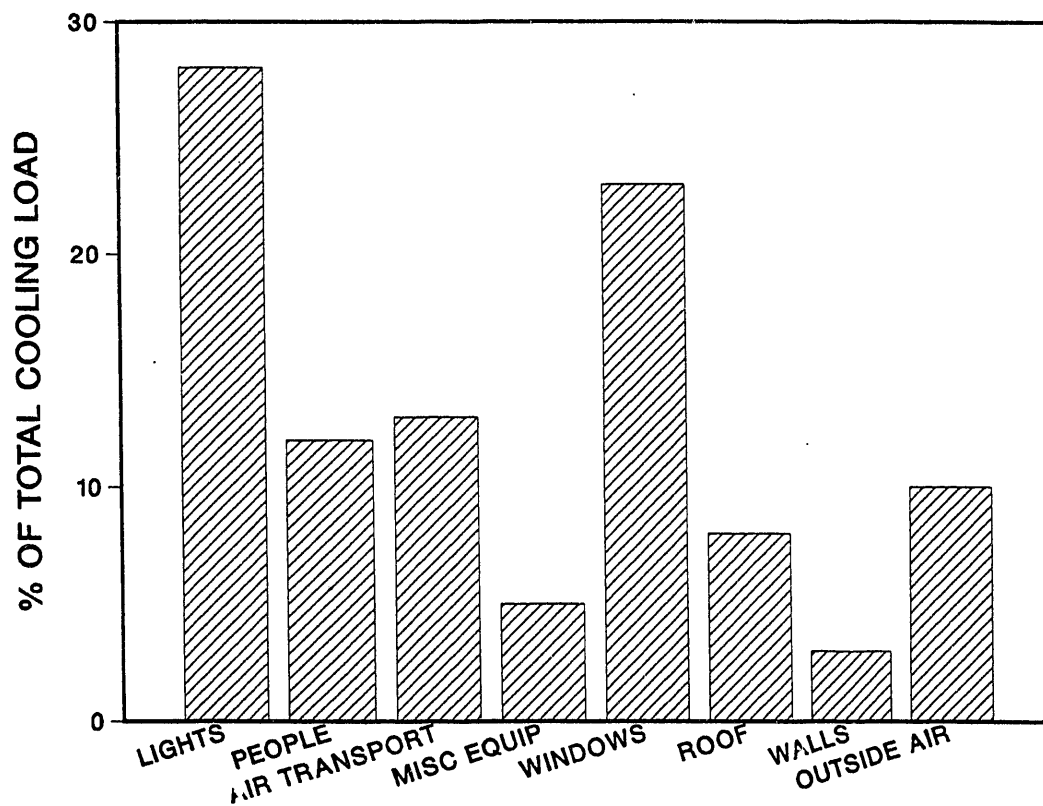LIGHTS   PEOPLE   AIR TRANSPORT   MISC EQUIP   WINDOWS   ROOF   WALLS   OUTSIDE AIR

Figure 1: *Cooling Load Components, Typical Office Building,*
Los Angeles [1]

HVAC systems are designed to maintain indoor air quality and provide thermal space conditioning. Traditionally, HVAC systems are designed as *All-Air Systems*, which means, that air is used to perform both tasks. *All-Air Systems* have been designed as central/decentralized systems, single/dual-path systems, constant/variable volume systems, and low/high velocity systems.

Air-and-Water Systems separate the tasks of ventilation and thermal space conditioning by using the primary air distribution to fulfill the ventilation requirements and the secondary water distribution system to thermally condition the space. These systems reduce the amount of air transported through buildings significantly, as the ventilation is provided by outside air systems without the recirculating air fraction. The cooling is provided mainly by radiation using water as the transport medium. Due to the physical properties of water, hydronic radiant cooling systems can remove a given amount of thermal energy using less than 5% of the otherwise necessary fan energy. The separation of tasks not only improves comfort conditions, but also increases indoor air quality and improves the control and zoning of the system. Hydronic cooling systems combine controlled temperature of room surfaces with central air handling systems [3].

Cooling of buildings can be applied using convection only or a combination of radiation and convection. The latter uses cool surfaces in the conditioned space to cool the air and the space enclosures. Although, only approximately 60% of the heat transfer is due to radiation, these systems are often called "radiative" cooling systems. This term, however, conflicts with another cooling system where the cool sky is used to either cool outer building surfaces or radiative panels. Most literature found refers to radiative cooling or radiant cooling panels when describing hydronic radiant cooling systems. In order to avoid confusion, in the following survey we refer to hydronic cooling, reflecting the thermal distribution system used for radiant cooling systems.

Due to the large surfaces available for heat exchange in hydronic radiant cooling systems (usually almost the whole ceiling), the coolant temperature is slightly lower than the room temperature. This small temperature difference allows the use of either heat pumps with very high COP-values or indirect evaporative cooling to further reduce the electric power requirements. At the same time, hydronic radiant cooling systems reduce problems caused by duct leakage as the ventilation air is significantly reduced and only conditioned to meet room temperature rather than cooling supply air temperature conditions. Furthermore, space needs for ventilation systems and their duct work are reduced to about 20% of their original space requirements. Besides the reduction of space requirements for the shafts that house the vertical air distribution system, floor-to-floor height can be reduced, which offsets the initial cost of the additional system. An integration between water distribution lines for hydronic radiant cooling systems and the sprinkler system might further reduce the initial cost.

The thermal storage capacity of the coolant further helps to shift the peak to later hours. Because of the hydronic energy transport, this cooling system has a potential to interact together with thermal energy storage systems (TES) and looped heat pump systems.

This report intends to give an overview over hydronic radiant cooling. It describes a literature survey on the system's development, thermal comfort issues, and cooling performance. The cooling power potential and the cooling power requirement are investigated for several California climates. Peak-power requirement is compared for hydronic radiant cooling and conventional *All-Air-Systems*.

History of Hydronic Cooling

Hydronic cooling was in use long before *All-Air Systems* were invented. Hauptmann from the University of Heidelberg recently discovered the ruins of a village in Turkey which dates back to more than 7000 B.C. [4]. Houses in this village with the kurdish name Nevala Cori utilized hydronic radiant cooling by re-routing water supplied by the Kantara Creek through channels imbedded into the buildings' slab to cool the stone floors.

The use of hydronic radiant cooling in more modern buildings had been investigated more than five decades ago. The earliest technical papers found in our literature survey date back to 1938 [5,6]. At this time the idea of hydronic radiant systems was not developed further. In 1951, Bilden [7] reports about the advantages of hydronic radiant

heating and cooling and describes demonstration projects. At this time, the cooling effect still was more of a by-product of hydronic panel heating systems rather than the target itself. However, Bilden refers to hydronic radiant cooling systems installed in an office building in Paris, a department store in Zuerich, the Museum for Modern Art in Paris, the Hotel Excelsior in Rome, the Banque de Rome and the Palais des Journaux 9both in Milan). The combination of hydronic radiant cooling and ventilation is mentioned in the paper.

In 1957, Ronge and Lofstedt [8] investigated the physiological effects of hydronic radiant cooling, especially the draft sensation of cold ceilings and their compensation with higher air temperatures. This lead to a comfort chart which shows the interrelation between ceiling surface temperature and the air temperature. Baker [9] describes the advantages of hydronic heating and cooling, based on the heat exchange of the human body with the environment and remarked, that the heat transfer due to radiation has not been given adequate consideration. Baker lists a comfortable, healthful and more invigorating environment, more uniform air temperatures, cleaner surfaces, neater appearance, and improved efficiency, as advantages of panel systems. Condensation might be a problem if an auxiliary dehumidifying coil is not used to control the rooms dew point temperature. Although, cost of operating a cooling panel will be less than for a conventional convection system, the degree of comfort seems to be more important to Baker.

Boyar [10] in his contribution to an ASHRAE symposium in the early sixties shows that radiant panel systems have dynamic behaviour similar to *All-Air Systems*. At this time, panel systems must have been widely used (probably for heating purposes).

In 1973, Obrecht, Salinger and LaVanture [11] reported that radiant panel ceilings were increasingly being applied as terminal heating and cooling devices in various types of buildings. Radiant cooling panels reduce the quantity of conditioned air supply to the space. Part of the sensible cooling required might still be provided by the ventilation system which is required to supply the air necessary for hygienic reasons.

The authors suggest selecting slightly higher summer design temperatures and somewhat lower design relative humidity than for conventional *All-Air Systems* in order to benefit from the lower mean radiant temperature, which allows higher ambient air temperatures and still provides the same degree of comfort. As heat removal from the space is a function of the temperature difference between the water and the room (air and surface temperatures), the design relative humidity and dew point should be selected as low as economically feasible. The supply water temperature is usually 1°C higher than the design dew point and a temperature rise of 4°C throughout the system is used. This usually leads to a temperature difference of approximately 10°C between the design dry bulb temperature and the mean water temperature.

Although, many authors have reported significant advantages of hydronic radiant cooling over *All-Air Systems*, even the two energy crises did not seem to have an effect on the development and the use of hydronic radiant cooling systems. Increasing energy prices did not change the market share of hydronic radiant cooling systems.

Current Development

During the last decade building inhabitants developed a critical attitude towards air conditioning systems. Terms like *complaint buildings* and *sick buildings* were born. Several publications dealing with occupant satisfaction in air-conditioned and naturally ventilated buildings came to the conclusion that the number of unsatisfied in air-conditioned buildings is significantly higher than in natural ventilated buildings [12-14]. Esdorn et al. [15] state, that there are still air-conditioning systems that do not produce the required standard of comfort. "The existence of air-conditioning systems is actually only noticed when it is not functioning properly."

Draft is a serious problem in many air-conditioned buildings. As air from HVAC systems is normally turbulent in the occupied zone, even small air velocities (less than 0.1 m/s!) might cause an unwanted local cooling of the human body [16]. In order to be able to extract cooling loads from a building, the cooled air either exceeds the outdoor air needed for ventilation (recirculating air system) or temperature differences between the supply air and the room air have to be so large that its supply might cause problems to evenly distribute in the occupied zone (cold air distribution systems).

Whereas the use of recirculation air might cause draft and/or indoor air quality problems due to the distribution of the polluted return air, cold air distribution systems can cause very cold local drafts if imperfect mixing between supply air and room air occurs at the air outlet. Both cases will cause comfort problems. Because of comfort problems and the excessive use of transport energy for *All-Air-Systems* new ventilation strategies were developed [17], such as displacement ventilation.

The idea of displacement ventilation is to overcome the problems of mixing ventilation systems. Contaminants are displaced from the breathing zone and clean air is directly supplied with air flow of low turbulent intensity to the breathing zone [18]. An upward displacing direction is most efficient for cooling purposes. This strategy will always result in increased air temperature with height. The heat sources in the room are the driving forces of the vertical air transport by creating convective air currents (plumes)[†]. This air flow pattern results in greatly improved ventilation efficiency (for definition of ventilation efficiency, see Sutcliff [19]). Ventilation systems with high ventilation efficiencies use solely outdoor air and, therefore, can only extract limited amounts of cooling loads [20,21]. Upward displacement ventilation shows a characteristic temperature profile caused by the convective currents due to the heat sources. As supply air is entering the room at floor level, the temperature gradient forms a barrier for low energy currents to reach high altitudes in the room. Due to comfort requirements, the temperature gradient between feet and head should not exceed 3°C, which further limits the cooling capacity of these ventilation systems [18].

In order to use these energy efficient ventilation systems, another cooling source had to be found. The logical choice was a coupling of efficient ventilation systems with hydronic radiant cooling systems, separating the tasks of ventilating and cooling the

† Displacement ventilation should not be mistaken for "plug flow" or "piston flow"; plume flow ventilation might be a better term
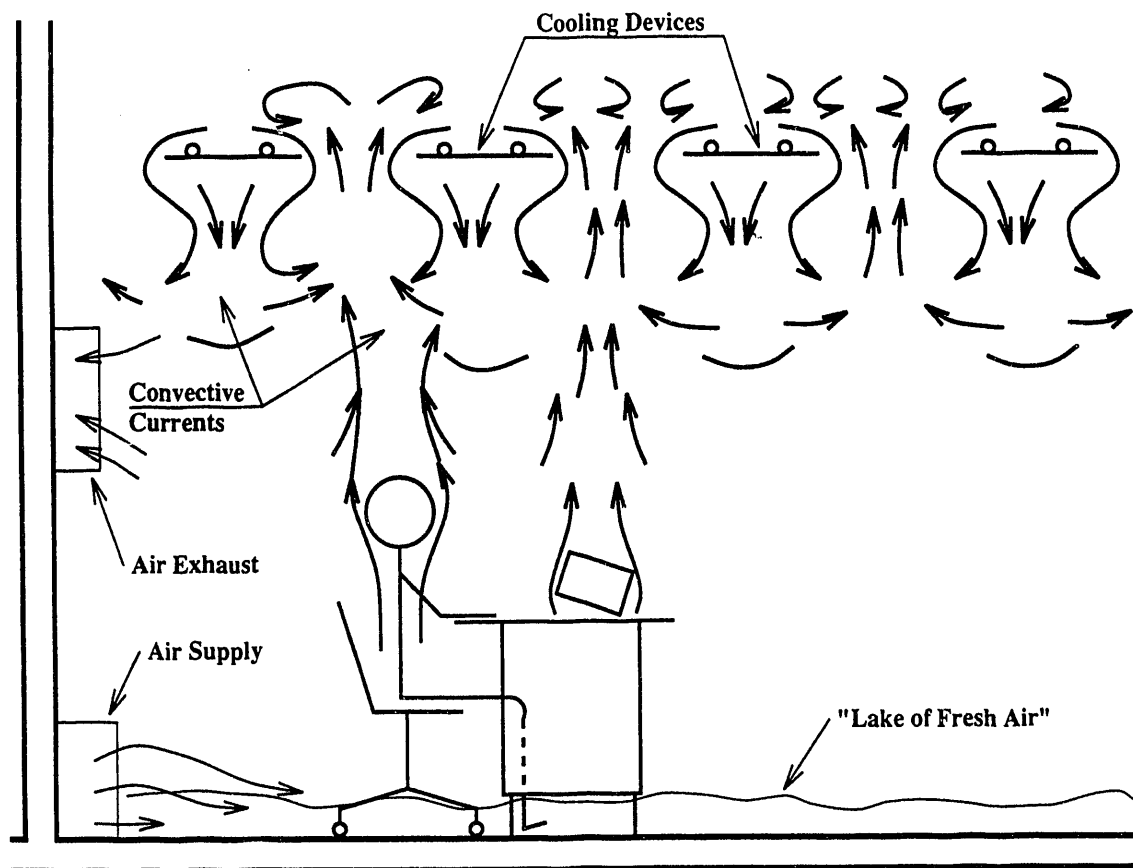
Cooling Devices

Convective
Currents

Air Exhaust

Air Supply

"Lake of Fresh Air"

Figure 2: *Air Flow Patterns in a Room with a Cooled Ceiling [18]*

building. The theoretical air flow pattern in a room with a cooled ceiling is shown in Figure 2.

Thermal Comfort

The human body is a generator of heat. In order to maintain normal functions the balance between heat gain and heat loss must be maintained. Heat can be lost by different ways: radiation to surrounding surfaces, convection to the ambient air, evaporation, respiration and excretion [22-25]. The publications came to the same main conclusion: the heat loss due to radiation has the highest portion, followed by convection and conduction. Respiration and excretion have less influence on the heat loss of a human body.

Baker gives the following example explaining the impact of radiation: "A person sitting out of doors under a clear sky on a summer evening may be chilly although the air temperature is in the high 70's (°F). Were he indoors at this same temperature, he probably would feel uncomfortably warm. The appreciable heat loss by radiation to the clear sky

explains the different sensations of comfort between outdoors and indoors." This example suggests, that the surface temperatures surrounding an occupant within an enclosure has to be considered. Heat loss by radiation and convection may be considered collectively, only if all surface temperatures are close to the ambient air temperature.

Losing the heat by convection only would require a high air velocity close to the human's skin and would finally lead to draft and, therefore, to uncomfortable conditions. The possibilities of increasing the heat loss by respiration or excretion are practically non-existent. This shows that not only the air temperature is important for comfortable conditions, but also the temperatures of the surrounding surfaces which cause heat loss due to radiation.

Because of the irregular surface of the body, the heat loss depends not only on the temperature differences, but also on clothing and activity [26]. ASHRAE Standard 55 [27] specifies the combination of the environmental parameters of air temperature, thermal radiation, humidity, air movement, personal parameters of clothing and activity.

Clothing, through its insulation properties, is an important modifier of body heat loss and comfort. Clothing insulation can be described in terms of its *clo-Value*. For instance, a heavy two-piece business suit with accessories has an insulation value of about 1 clo while a pair of shorts is about 0.05 clo. The insulation of the clothing can be estimated by summing the clo values of the individual items worn and multiplying the sum by a reduction factor [27]. The operative temperature is approximately the average of the air temperature (dry bulb temperature at air speeds of 0.4 m/s or less) and mean radiant temperature (if less than 50°C).

The optimum operative temperature is decreased if the average activity level of the occupants is higher than sedentary. The body will then increase the rate of energy production. The energy production is expressed in met units (metabolic rate). One met is defined as 58.2 W/m^2, which is equal to the energy produced per unit surface area of a seated person at rest. The surface area of an average man is about 1.8 m^2 [27].

The operative temperature for different activities depends on both: activity level and clothing insulation.

According to Table 1, the heat loss due to radiation has the most influence on the human's heat loss and thus on his state of comfort. The mean radiant temperature, which depends on the temperatures of the surrounding surfaces, is responsible for heat loss by radiation. The mean radiant temperature is easy to define but quite complicated to calculate or measure in practice. The first experiments of thermal and comfort sensations of sedentary persons to radiation were conducted by McNall, Biddison [28], and Schlegel, McNall [29]. The most extensive experimental investigations have been made by Fanger [26] in order to show that mean skin temperature and sweat secretion are closely connected with the sensation of thermal comfort. 128 college-age persons (64 males and 64 females) and 128 elderly persons (64 males and 64 females) were used for his experiments. Fanger defined mean radiant temperature as follows: "The mean radiant temperature in relation to a person in a given body posture and clothing placed at a given point in a room, is defined as that uniform temperature of black surroundings which will give the

same radiant heat loss from the person as the actual case under study." [26]

Table 1: Heat Loss of a Human Body [22]	
Radiation	40 - 50 %
Convection	20 - 26 %
Evaporation	18 - 20 %
Respiration	6 - 10 %
Excretion	~ 6 %

Due to the non-uniform distances and angles of persons related to the walls, floor and ceiling, each part of the room must be considered separately. If a given surface is found not to be isothermal, it has to be divided into smaller surfaces, which are isothermal by themselves. Each surface can be assumed to be grey. The radiation emitted and reflected from any surface is diffusely distributed, which is a good approximation for all normal non-metallic surfaces [26].

The enclosure surfaces often found in a normal room are rectangular in form and, therefore, it is the angle factor between a person and a vertical or horizontal plane. The body posture also plays an important part. The mean radiant temperature in relation to a standing person need not be the same as in relation to a seated one [26]. Likewise, the person's location and orientation within the room must also be known, because the mean radiant temperature often varies from point to point.

Air movement plays a special role among the comfort parameters. According to Esdorn et al. [15], air movement is the biggest single cause for complaints (draft). Besides the mean velocity, the fluctuation of the velocity has an important influence on the convective heat transfer of the human body. Mayer [30] relates comfort directly to the convective heat transfer coefficient rather than to the mean air velocity. According to Mayer [31] at an air temperature of 22°C draft is felt if the convective heat transfer coefficient is above 12 W/m^{2}K. This translates to mean air velocities for laminar flows of 1.35 m/s; for transition flow of 0.15 m/s, and for turbulent flow of 0.10 m/s. Lower air temperatures significantly reduce the acceptable air velocities.

Cooling Power

The cooling power of hydronic radiant cooling systems is limited by several parameters. First, the surface temperatures of the cooling elements should not be lower than the dew point of the air in the cooled zone. The dew point can be manipulated by reducing the air's humidity content. A more serious concern is the comfort effect of the asymmetrical distribution of the radiant temperature. Kollmar [32] shows that for offices, ceiling temperatures of approximately 15°C are the lower limit.

The heat transfer between the room and the cold ceiling is based on radiation and convection. Whereas the heat transfer of radiation is relatively easy to calculate, the convective heat transfer is a function of the air velocity at the ceiling level. This velocity is dependent on the room geometry, the location and power of the heat sources, and the location of the air intake and exhaust.

Trogisch [33] compares heat transfer coefficients for cooled ceilings found in the literature with the description of convective heat transfer from a cold flat surface (downwards) as published in textbooks. Investigations dealing with cooled ceilings show heat transfer coefficients of 9 to 12 W/m^2K. With a heat transfer coefficient for radiation of about 5.5 W/m^2K (for = 10K), the convective heat transfer coefficient must be in the magnitude of 3.5 to 6.5 W/m^2K. These values, however, can only be reached if forced convection takes place (here forced means that other phenomena than the cooling at the ceiling are responsible for driving the air flow).

Radiant cooling elements extract heat from a room by cooling the air (convection) and by cooling the surfaces of the room's envelope. The two effects can be described by an empirical equation which is based on the assumption that the mean surface temperature of the room differs only slightly from the air temperature [34]. With this assumption, we can express the specific cooling power (per square meter) of a cooled ceiling by the following equation:

$$q_{tot} = 8.92 \left[t_{air} - t_{surface} \right]^{1.1} \tag{1}$$

q_{tot} sum of convective and radiant heat transfer [W/m^2]

With the information about the heat transfer coefficient ($\alpha_{tot} = 11-12$ W/m^2K) and the lower limit of the ceiling surface temperature ($t_s = 15°C$), one can reach a specific cooling power of approximately 110 to 120 W/m^2 through hydronic radiative cooling. In zones with more than one exterior wall, slightly higher specific cooling powers can be achieved due to higher radiative temperature differences. If forced convection is provided by the ventilation system, the cooling performance of the panel system can be increased.

Although, attempts have been made to numerically determine the cooling power of hydronic systems, there seems to be no standard available for testing cooling panels. ASHRAE's technical committee TC 6.5 *Radiant Space Heating and Cooling* is sponsoring a committee on *Methods of Testing/Rating Hydronic Radiant Ceiling Panels* (SPC 138P). The purpose of SPC 138P is to establish a method of testing for rating the thermal performance of hydronic radiant cooling panels used for heating and/or cooling of indoor space [35]. A test facility and a method of testing was developed at the Department of Veterans Affairs [36]. The paper describes the testing procedure for thermal performance and pressure drop measurements, the test facility as well as the accuracy of the instrumentation to be used.

Whereas testing procedures and future standards will rate the performance of the panel, the efficiency of the system is difficult to determine. A high panel efficiency could be produced by providing high air velocities at the panel surface through a shortcut between the supply and the exhaust of the ventilation system. Although the panel would show high performance, part of the convective heat transfer could be diminished by exhausting cooled air rather than supplying the air to the occupied zone.

In order to avoid condensation, the temperature of the surface of the cooling panel has to be above the dew point. The temperature difference between the room air and the dew point temperature is theoretically the upper limit for cooling power. In practice, however, the effective cooling temperature difference is reduced by a safety-margin of approximately 2K.

To determine the effective temperature differences which can be utilized for radiant cooling some California and non-California climates were analyzed [37] The description of the climates as published by the National Oceanic and Atmospheric Administration [38] is shown in Appendix B.

Table 2 shows some extreme values for air properties for four California and one non-California climate. The humidity ratio and the outdoor air temperatures together with the effective temperature difference available for cooling are listed for the hottest hour and the most humid hour.

Table 2a): *Air Properties for the Hottest Hour for Several Locations*					
Location	x [g/kg$_{dry\ air}$]	t [°C]	τ [°C]	t_{MWT} [°C]	Δt_{eff} [K]
Arcata	8	26	10.5	12.5	13.5
Long Beach	4	37	1.5	3.5	22.5
Red Bluff	3	48	-3	-1	27
San Diego	12	39	16.5	18.5	7.5
Lake Charles	17	35	22.0	24.0	2.0

x = humidity ratio (outside conditions)

t = dry bulb temperature (outside conditions)

τ = dew point temperature (outside conditions)

t_{MWT} = mean water temperature

with: $t_{MWT} = \tau + 2K$

and : $\Delta t_{eff} = t_{Room} - t_{MWT}$; with $t_{Room} = 26\ °C$

We see, that in the case of the most humid hour, the effective temperature difference proves to be relatively small for all chosen climates. Measures have to be taken to reduce the dew point of the room air to obtain reasonable effective temperature differences.

Table 2b): *Air Properties for the most Humid Hour for Several Locations*					
Location	x [g/kg$_{dry\ air}$]	t [°C]	τ [°C]	t$_{MWT}$ [°C]	Δ t$_{eff}$ [K]
Arcata	12	18	16.6	18.6	7.4
Long Beach	16	32	21.2	23.2	2.8
Red Bluff	14	34	19	21	5
San Diego	16	28	21.2	23.2	2.8
Lake Charles	18	34	23	25	1

Cooling Performance

Although, several papers describing the cooling power of hydronic radiant systems have been found in the literature, few articles deal with the performance of these systems. Kuelpmann and Esdorn [39] report on their experimental investigation in a temperature controlled test cell (see Figure 3). The ventilation air was supplied at floor level and exhausted approximately 0.2 m below the ceiling level. Internal loads were simulated by electrically heated mannequins (dummies) standing next to a computer display and by fluorescent lights. External loads were introduced by heating either one of the long side walls or the floor. For displacement ventilation and no cooling with supply air, room air temperatures did not differ very much over height (see Figure 4). The extraction of 100 W/m^2 internal load by hydronic radiant cooling caused temperature differences of approximately 2K between supply and exhaust grille. With increasing temperature difference between the room air and the supply air, the profile became more pronounced. Especially in the lower part of the room, these temperature differences get close to or exceed the comfort limits.

In all examined cases the differences between the room air temperature and the surface temperatures of the "internal walls" were relatively small (±0.4K). Due to the radiation exchange with the cool ceiling, the floor surface temperature is usually below the wall surface temperatures.

Asymmetric or non-uniform thermal radiation may be caused in winter by cold windows, uninsulated walls or heated ceilings. In summer, cool ceiling panels also produce asymmetric thermal radiation. Radiant asymmetry due to a cool ceiling causes less discomfort than a warm ceiling. Based on Fanger's assumption to accept 5% dissatisfied, a radiant temperature asymmetry of 10K is allowable at a cool wall and 14K under a cool ceiling (see Figure 5) [40]. Measurements of radiant temperature asymmetry at 100 W/m^2 cooling power resulted in 5.3K in 1.1m above the floor level in the middle of the room. This corresponds to approximately 1% dissatisfied.

Air flow velocities were measured in 1m distance to the supply air grille in 0.1m height above ground. At an air exchange rate of 3.2 ach and a supply air temperature of 19°C a velocity of 0.12m/s and a turbulence intensity of 20% was measured. The air temperature was 20°C.
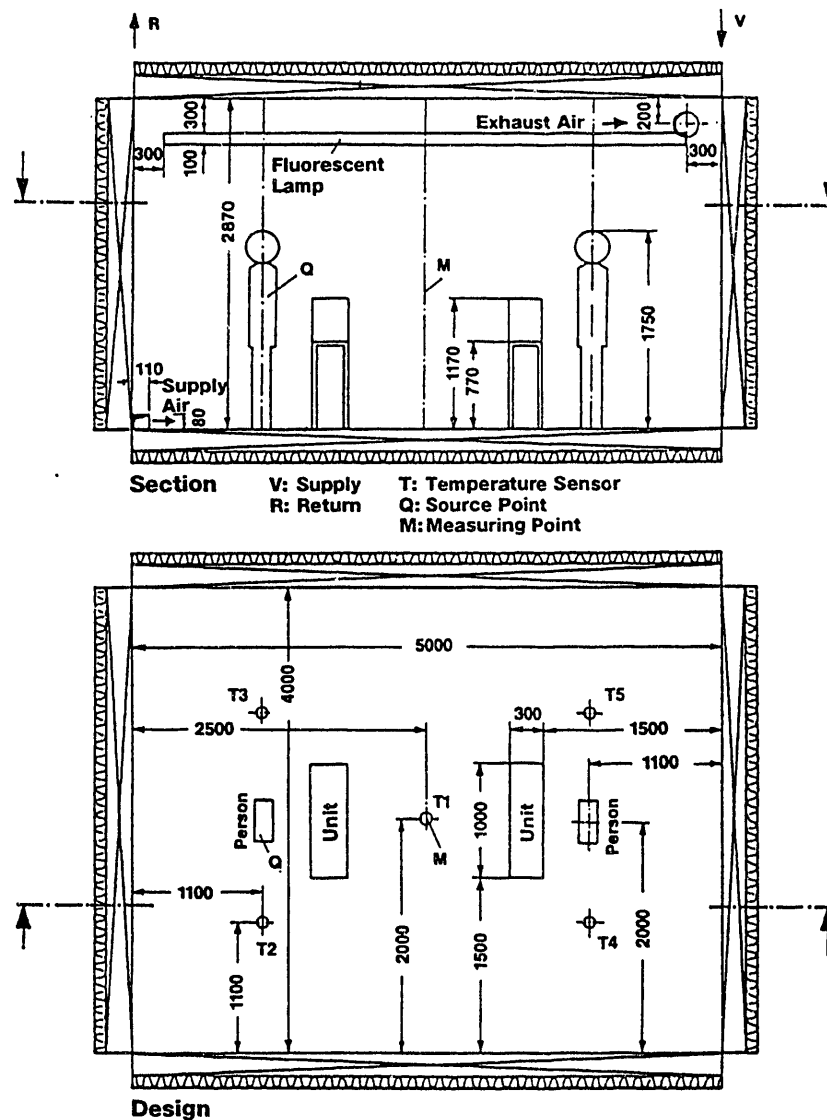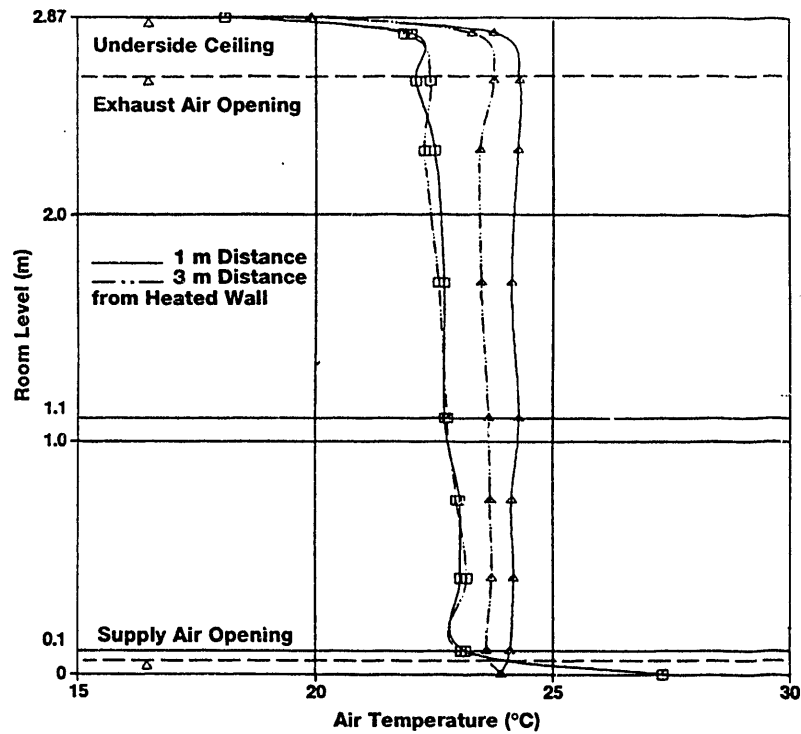
Figure 3: *Test Chamber used to Test Cooling Performance of Hydronic Cooling [39]*

The performance of hydronic radiant cooling was tested in two parliamentarian offices in Bonn, F.R.G. [41]. Dry-bulb temperatures and relative humidity were measured in the ambient air, the supply air and the room air. Furthermore, temperatures were measured in the supply and return pipe of the hydronic system and at three points at the ceiling surface. At ambient temperatures of 30°C air velocities below 0.10 m/s were measured in the occupied zone. Below the ceiling surface velocities between 0.10 and 0.15 m/s were detected.

Figure 4: *Air Temperature Profile over the Room Height as a Function of External Loads [39]*

Symbol	Cooling Type	Load Value (W/m²)	Room Temperature t_R (°C)	Temp.-Diff. Walls-Air (K)	Performance Part Ceiling (-) $\omega_{D\bullet}$	Air (-) ω_L
☐	W	39.7	24.0	0.2	1.00	0.00
△	Fb	40.5	23.0	0.1	1.00	0.00

W: Wall   Fb: Floor

The simulation of the system performance of a cooled ceiling for a continental European climate shows only few hours per year where the set point temperature of the room cannot be met by the system [42]. The office building is made of reinforced concrete with a window area of 33% of the exterior wall. The south-west oriented zone experiences a maximum of 30 W/m² external loads and 57 W/m² internal loads. Occupancy is 16 m²/person. The air exchange rate due to ventilation (0.8 ach) was sufficient to avoid condensation problems.

Cooling Loads

In order to determine the cooling loads one has to expect in California climates, the Base-Case building used by the California Energy Commission (CEC) was simulated using weather data for the four California climates (Arcata, Long Beach, Red Bluff, San Diego) and for Lake Charles. The building used for the simulation runs is the low-rise
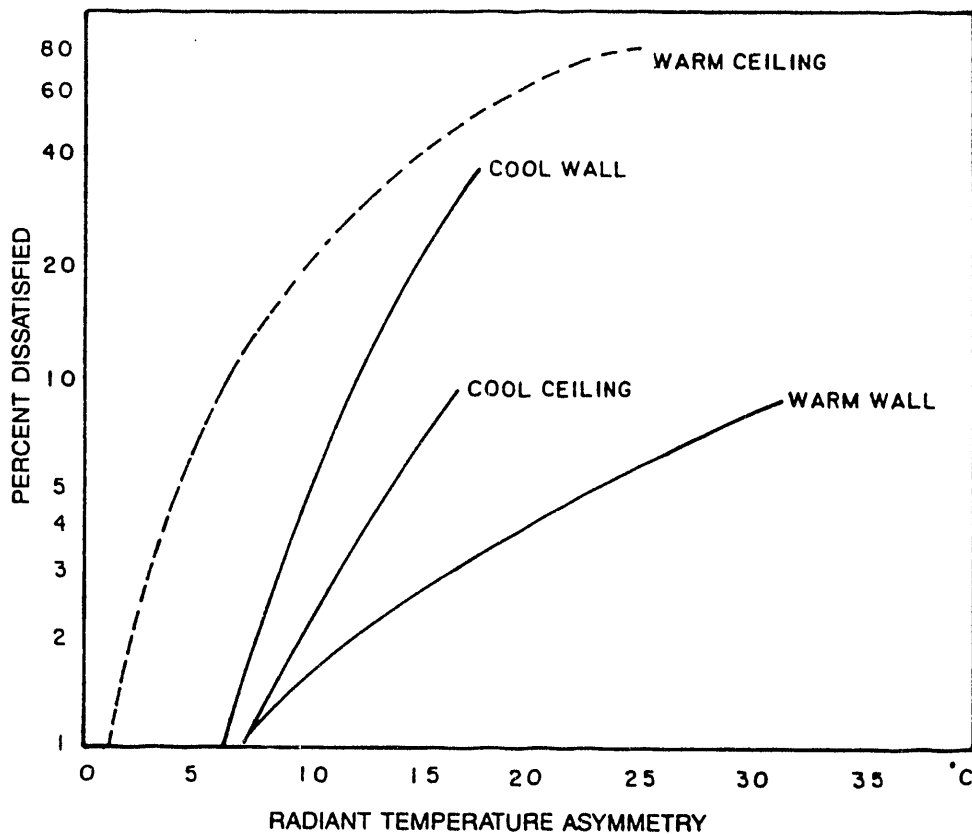
Figure 5: *Percentage of People expressing Discomfort due to Asymmetric Radiation [40]*

office base-case building described in the *Compliance Options Approval Manual*, published by the California Energy Commission [43]. The office building was simulated using the building simulation model DOE - 2.1D [44].

The low-rise office building used for the building simulation is a two story, 1,860 square meters office building, built slab on grade (see Fig. 6). The building is divided into ten different zones. The air to condition each of the zones is supplied by package single zone systems (PSZ). Lighting is provided by recessed fluorescent luminaires, installed to 15 W/m^2. There is an additional load of 5 W/m^2 from receptacles. Wood frame exterior walls are insulated with 1.9 K m^2/W (R-11) batt insulation. Walls are designed with studs (50*100 mm, 400 mm on center) supported by 16 mm exterior Plywood and 16mm Gypsum Board inside. Windows consist of 6mm single pane heat absorbing glass with metal frames. Glass covers 30% of the exterior wall area. The roof is insulated with 1.9 K m^2/W (R-11) batt insulation. The concrete floor includes an 0.8 K m^2/W (R-4.5) insulation.
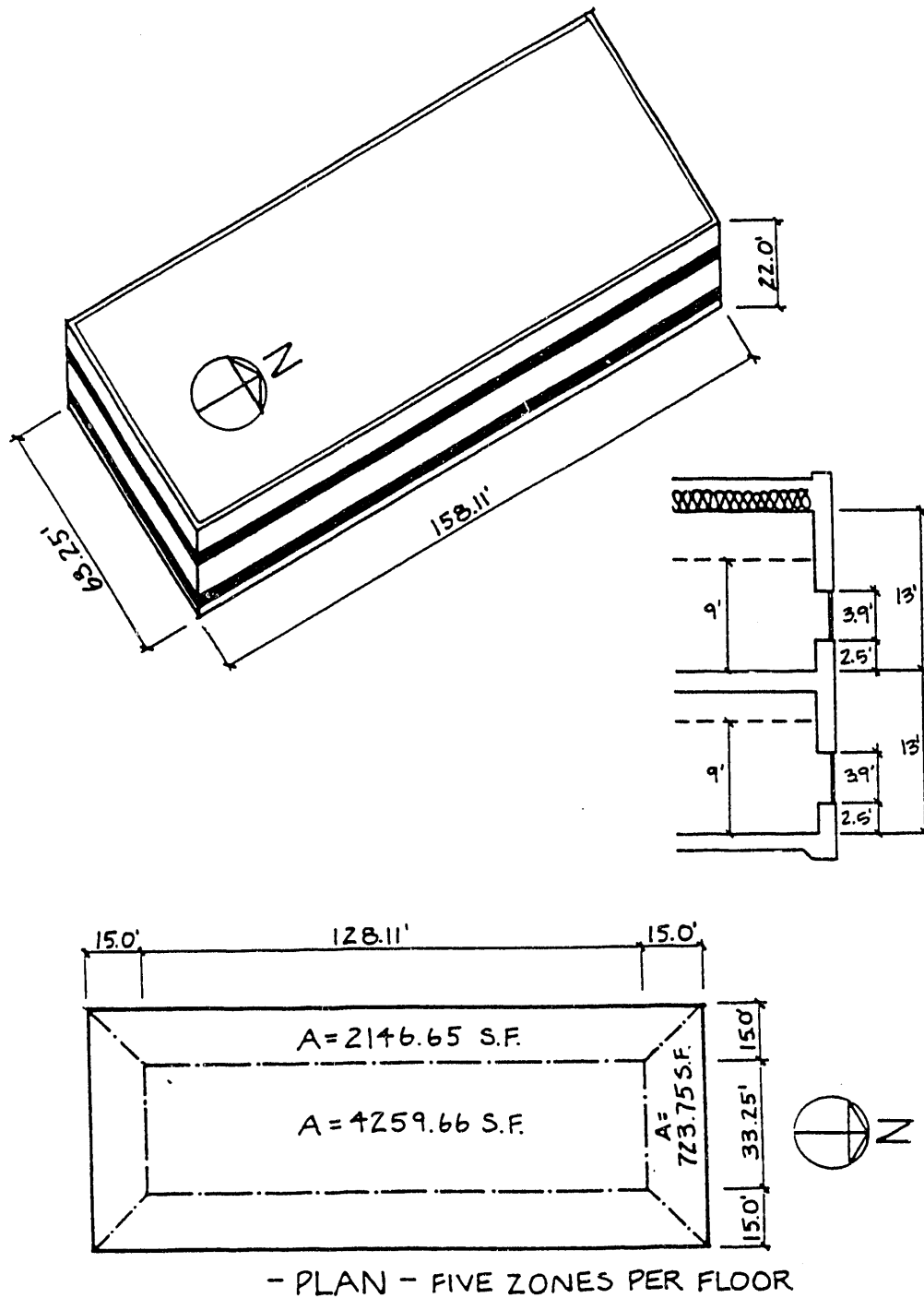
Figure 6: *Low-Rise Office Building [43]*

Occupancy during working hours is 23.3 square meters per person, with 65 Watt sensible and 54 Watt latent heat per person, which translates to a humidity load of 75g/(h pers). The HVAC system is off between 18 p.m. and 7 a.m. from Monday through Friday and during all weekends and holidays. If the fan is off, infiltration occurs at a rate of 0.7 $m^3/(hm^2)$ of exterior wall area.

The cooling peak loads for the different climates show significant differences for the building location, but also for the zones of the buildings. Due to the additional heat transfer, the zones in the second storey experience higher loads than the zones down-stairs. In all cases the South zones and the West zones show the highest specific loads. Table 3 shows the cooling peak loads for the locations simulated.

Table 3: Cooling Peak Loads for the different Zones of the Base-Case Building for the Day of the Building Peak Load [W/m² floor area]					
	Arcata	Long Beach	Red Bluff	San Diego	Lake Charles
First Floor:					
North I	32	58	90	70	62
South I	121	137	149	130	122
West I	93	112	149	115	116
East I	88	102	135	100	107
Interior I	22	22	22	22	22
Second Floor:					
North II	47	72	112	84	83
South II	144	160	177	151	147
West II	109	136	179	139	147
East II	99	120	157	120	131
Interior II	35	37	44	33	40
Whole Building	51	63	84	62	63

The highest loads are associated with the location of Red Bluff. Therefore, the peak load components for this particular case are listed in Tables 4a) and 4b). Results from the simulation runs show relatively high cooling peak loads for all eight perimeter zones. Even the north-zones show peak cooling loads of 90 W/m² floor area for the first floor and 112 W/m² for the second floor. The west zone on the second floor has the highest load with 187 W/m² which has its peak on the late afternoon. Interior zones, which have the same specific interior loads as the other zones, show peak loads of about 22 W/m² for the first floor and 45 W/m² for the second floor. The difference between the two interior zones is based on the additional load the second storey zones get from the roof.

The peak loads calculated for the three perimeter zones cannot be removed by radiant cooling ceilings without special measures to avoid condensation. Furthermore, the large

Table 4a: *Peak Loads for individual Building Components for Base-Case Building, First Floor* [W/m^2 floor area], *Red Bluff*

	North I	East I	South 1	West I	Interior I
	7/14,5pm	7/15,11am	10/6,3pm	7/14,6pm	12/18,5pm
Walls	9	7	8	10	0
Roof	0	0	0	0	0
Glass Conduction	41	26	14	34	0
Glass Solar	21	84	108	91	0
Underground Surfaces	-0.7	-0.7	-0.5	-0.7	0
Occupants to Space					
sensible	2	2	2	1	3
latent	2	2	2	1	2
Light to Space	12	11	11	11	12
Equipment to Space	4	4	4	4	5
Total	90.3	135.3	148.6	151.3	22

Table 4b: *Peak Loads for individual Building Components for Base-Case Building, Second Floor* [W/m^2 floor area], *Red Bluff*

	North II	East II	South II	West II	Interior II
	7/14,5pm	7/15,11am	10/6,3pm	7/15,7pm	7/15,5pm
Walls	9	7	9	12	0
Roof	18	10	11	18	22
Glass Conduction	42	27	15	31	0
Glass Solar	23	94	122	102	0
Underground Surfaces	0	0	0	0	0
Occupants to Space					
sensible	2	2	2	1	3
latent	2	2	2	0	2
Light to Space	12	11	11	10	13
Equipment to Space	4	4	4	4	5
Total	112	157	176	181	45

temperature difference between the cool ceiling and the air might cause a significant down-draft in the room.
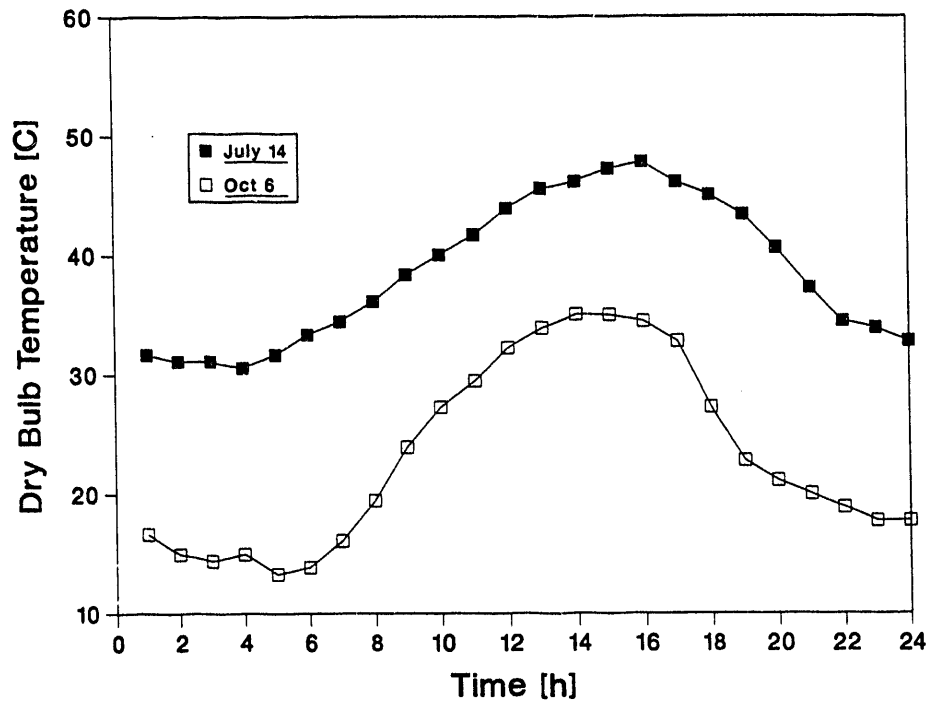
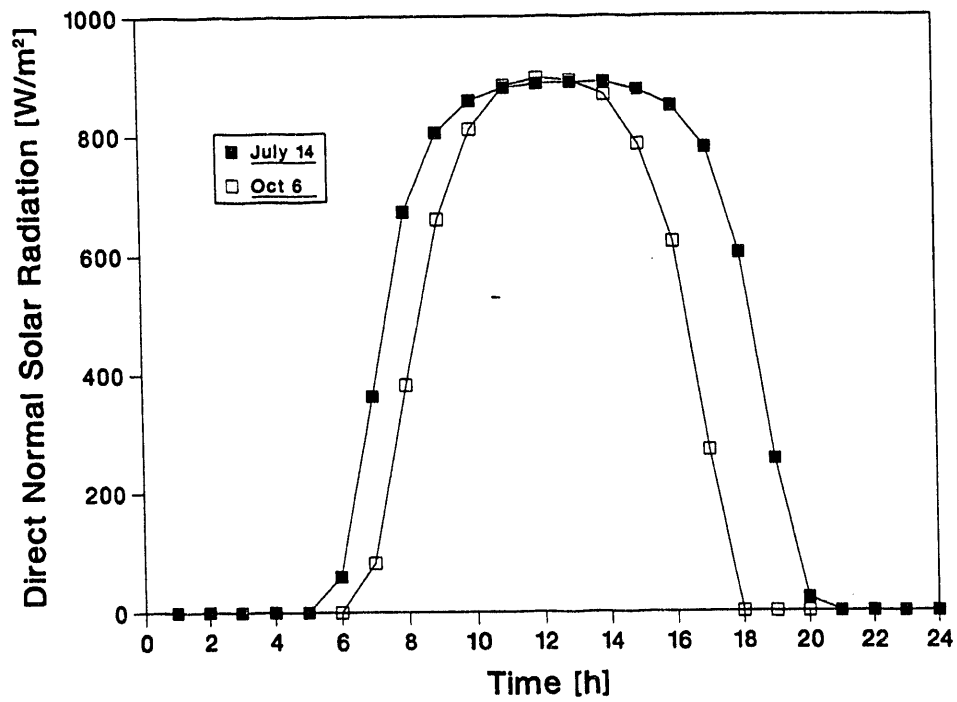**Figure 7a:** *Temperature Profiles for Red Bluff, July 14 and October 6*

Figure 7b: *Solar Radiation for Red Bluff, July 14 and October 6*

Table 5a: Peak Load Components for modified Base-Case Building, First Floor [W/m² floor area], Red Bluff					
	North I	East I	South I	West I	Interior I
Cooling Date	7/14 5pm	7/15 11am	10/6 3pm	7/15 7pm	12/18 5pm
Walls	9	7	9	12	0
Roof	0	0	0	0	0
Glass Conduction	20	12	6	15	0
Glass Solar	19	23	30	25	0
Underground Surf.	-0.7	-0.7	-0.5	-0.7	0
Occupants to Space					
sensible	2	2	2	1	3
latent	2	2	2	0	2
Light to Space	12	11	11	11	12
Equipment to Space	4	4	4	4	5
Infiltration	0	0	0	3	0
Total	67.3	60.3	63.5	70.3	22.0

Table 5b: Peak Load Components for modified Base-Case Building, Second Floor [W/m² floor area], Red Bluff					
	North II	East II	South II	West II	Interior II
Cooling Date	7/14 5pm	7/15 11am	7/14 4pm	7/15 7pm	7/15 5pm
Walls	9	7	10	12	0
Roof	11	6	11	12	13
Glass Conduction	21	13	20	15	0
Glass Solar	21	26	15	28	0
Door	0	0	0	0	0
Underground Surf.	0	0	0	0	0
Occupants to Space					
sensible	3	2	3	1	3
latent	2	2	2	0	2
Light to Space	12	11	11	10	13
Equipment to Space	4	4	4	4	5
Infiltration	0	0	0	3	0
Total	83.0	71.0	76.0	85.0	35.8

As Table 4 shows, the largest amount of heat enters the zones through the windows. Up to 122 W/m² of floor area are transmitted into the south zone. This corresponds to almost 70% of the overall cooling load. It seems to be clear that measures have to be taken to avoid such a high solar gain. Although, solar gain is the primary source of heat load in the south zone, even the north zone accumulates more than 100 W/m².

Whereas the driving source for the cooling load for the south zone is the solar gain, the load for the north zone is dominated by heat transfer caused by high outside temperatures. This is indicated by the time of the peak load shown in Table 5. Not only the hour of the peak is different between the zones, but also the days. Whereas the south zone has it's peak in October, all other perimeter zones experience their peak in July. In order to understand this peak shift phenomenon, ambient temperatures and solar radiation are plotted in Figure 7 for the two days. Solar radiation reaches approximately 900 W/m^2, but outdoor temperatures and the shape of the solar radiation curves are significantly different. Whereas the temperature on the 6th of October stays below 40°C, this value is already reached at 10 o'clock on the 14th of July.

The building components for the base-case building have been varied in a parametric study to find the combination of components which help to reduce the cooling load to reasonable values (i.e., below 100 W/m^2). It should be reported that, with triple reflecting glass, doubling the roof insulation level, vented luminaires and a reduction of the lighting power, cooling loads below 60 W/m^2 are possible. However, even smaller changes help to reduce the cooling load significantly.

The specific loads for individual building components at zone peak power are listed in Tables 5a) and b). To reduce specific peak loads below 100 W/m^2 floor area double glazed windows with reflective coating for East, South and West zones have been used. Insulation levels for the roof have been tripled to reduce the heat flux through the flat roof. These changes reduce solar radiation gain at zone peak power to 30 W/m^2 and keeps the influence of the roof below 15 W/m^2. The highest peaks are experienced in the West and the North (!) zones; but the less than 90 W/m^2 load can be handled by the radiant cooling system (see Section "Cooling Power).

Energy Savings

The use of radiant cooling systems is an energy conserving and peak-power reducing alternative to conventional air-conditioning which is particularly suited to dry climates. A significant amount of electrical energy used to cool buildings by *All-Air Systems* is drawn by fans which are used to transport cool air through the ducts. Part of this electricity used to move the air is heating the conditioned air, and therefore, is part of the internal thermal cooling peak load. The electrical cooling peak load, if defined as the load of the fans and the chillers, can be divided approximately into 37% for running the fans and 63% for using the chillers.

If ventilation and thermal conditioning of buildings are separated, the amount of air transported through buildings could be reduced significantly. In this case the cooling would to be provided by radiation using water as the transport medium. The ventilation has to be provided by outside air systems without the recirculating air fraction. Due to the physical properties of water, radiant cooling systems can remove a given amount of thermal energy using less than 5% of the otherwise necessary fan energy. Although the supply air necessary for ventilation purposes will still be distributed through ducts, the electrical energy for transportation (air and water) can be reduced to approximately 25%

of the original number.

The elimination of recirculation air also increases the efficiency of air-handling luminaires, as the convective heat extracted from the light fixtures is not recirculated as it would be in an *All-Air-System* but vented directly to the outside. 50% of the thermal cooling energy due to lighting can be removed this way. Compared to a constant volume air system an overall electrical cooling energy savings potential of more than 40% seems reasonable (see Figure 8).

Peak-Power Requirement

In order to compare the electrical peak-power requirement for conventional systems (*All-Air-Systems*) and advanced systems (*Air-and-Water Systems*), the power requirement for a simple example has been calculated.

The example is based on an office with a floor area of 25 m^2, a two person occupancy and a total heat gain of 2000 W. The specific cooling load amounts to 80 W/m^2, which is in the range of radiant cooling systems. The room temperature is set to 26°C. Additional assumptions and design considerations used for this example are shown in Table 6.

The operation of an *All-Air-System* and an *Air-and-Water-System* can be seen in Figures 8 and 9. ·

The *All-Air-System* supplies the cooling to the room as follows: the outside air is treated by a cooler which dehumidifies the air according to the required room condition. ASHRAE Standard 62 (ASHRAE 1989) requests a minimum air change rate of 36 m^3/h Person. This means that for this example the minimum air change rate is 72 m^3/h. In order to remove the internal load, a recirculating air volume flow of 678 m^3/h is required. The assumed outside air condition of 32°C leads to a mixing temperature of 25.6°C

After mixing, the air is treated by a cooling system. In order to adjust for the temperature increase due to the fan work, the air has to be cooled further down than the 18°C specified as supply air temperature. The temperature adjustment depends on the pressure drop, fan efficiency and volume flow. In our example the temperature rise has been assumed to be 1.0K.

The electrical power for an *All-Air-System* amounts in this example to:

$$\Sigma Q_{el,All-Air-System} = 1270 \text{ W}$$

In order to be able to compare the two systems the boundary conditions have to be equal. This includes the efficiencies of fans and motors, pressure losses for supply and exhaust ducts and chillers' COPs.

Table 6: Assumptions used for the Comparison of Peak Power Requirements for All-Air -Systems and Air-and-Water-Systems		
	Both Systems	
Room Conditions:		
Cooling Load [W/m^2]	80	
Room Air Temperature [°C]	26	
Relative Humidity [%]	50	
Humidity Ratio [g$_{water}$/kg$_{dry air}$]	10.6	
Number of People [-]	2	
Outside Air Conditions:		
Air Temperature[°C]	32	
Relative Humidity [%]	40	
Humidity Ratio [g$_{water}$/kg$_{dry air}$]	12.1	
Enthalpy [kJ/kg]	63.0	
	All-Air-System	Air-and-Water-System
Design Consideration:		
Outside Air Flow [m^3/h]	72	72
Supply Air Flow [m^3/h]	750	72
Temperature Differences:		
Room Air - Supply Air [K]	8	3
Room Air - Ceiling [K]	0	8
Supply Water - Return Water [K]	--	2
Efficiencies:		
Fan: Hydraulic/Mechanical/Electrical [%]	60/80/98	60/80/98
Water Pump [%]	--	60
Pressure Drop:		
Supply Duct/Return Duct/Water Pipe [Pa]	500/250/--	500/250/40000
COP [-]	3	3

Whereas the *All-Air-System* removes the cooling load by means of supplying cold air, an *Air-and-Water-System* removes the load mainly by means of water. The air system's tasks thus are to supply the room with the necessary air exchange rate for hygienic reasons and to avoid humidity built up (control of the dew-point). In order to provide a stable displacement ventilation the supply air volume flow should be 3K below room air temperature. Therefore the supply air temperature is reduced to 23° C, which allows a reduction of the hydronic cooling load by 2.88W/m^2.
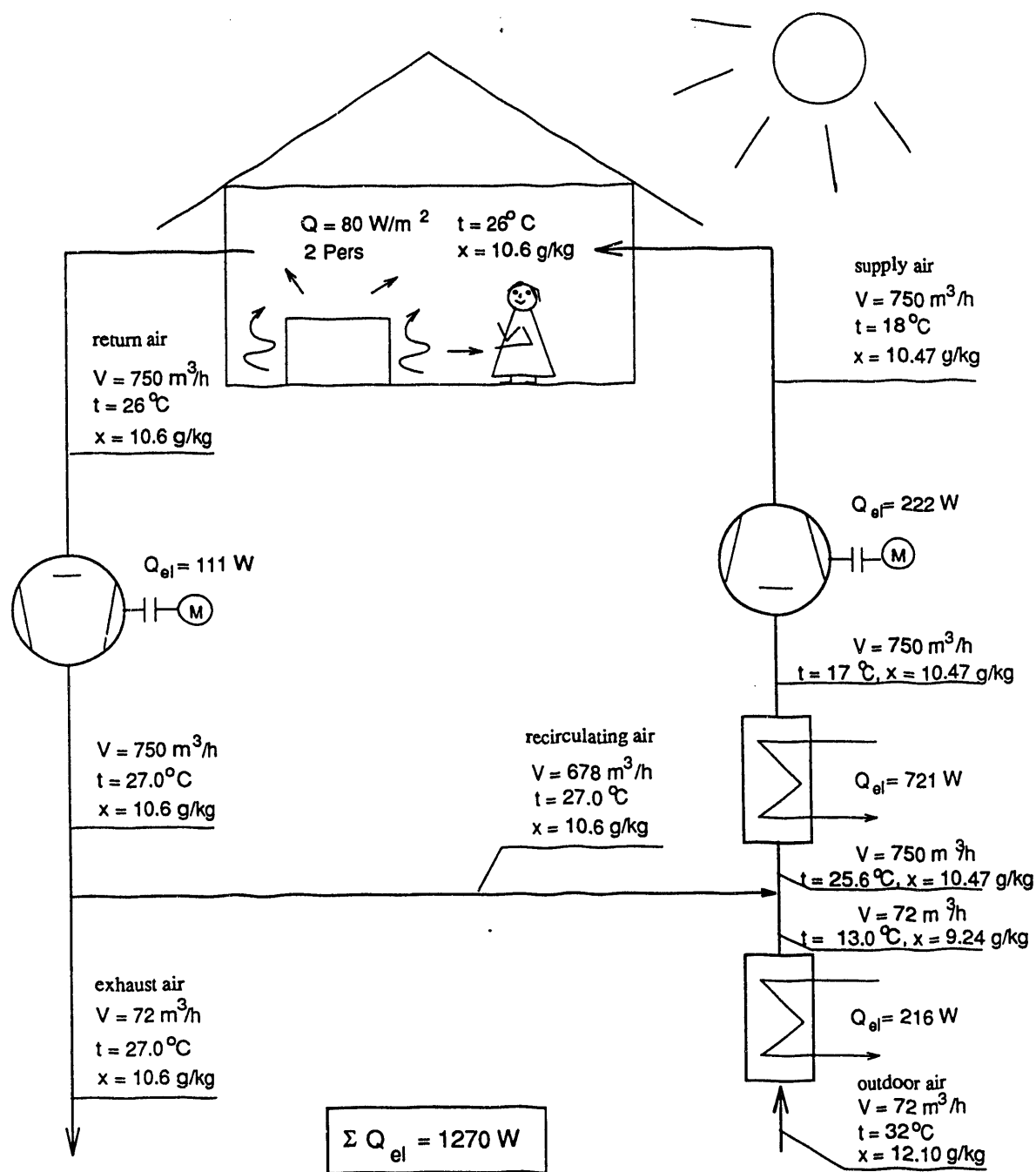
Q = 80 W/m^2 t = 26° C
2 Pers   x = 10.6 g/kg

return air
V = 750 m^3/h
t = 26 °C
x = 10.6 g/kg

supply air
V = 750 m^3/h
t = 18 °C
x = 10.47 g/kg

Q_{el} = 111 W

M

Q_{el} = 222 W

M

V = 750 m^3/h
t = 17 °C, x = 10.47 g/kg

V = 750 m^3/h
t = 27.0°C
x = 10.6 g/kg

recirculating air
V = 678 m^3/h
t = 27.0 °C
x = 10.6 g/kg

Q_{el} = 721 W

V = 750 m^3/h
t = 25.6 °C, x = 10.47 g/kg

V = 72 m^3/h
t = 13.0 °C, x = 9.24 g/kg

exhaust air
V = 72 m^3/h
t = 27.0 °C
x = 10.6 g/kg

Q_{el} = 216 W

outdoor air
V = 72 m^3/h
t = 32°C
x = 12.10 g/kg

Σ Q_{el} = 1270 W

Figure 8: *Peak Power Requirement for All-Air-Systems*

Figure 9: *Peak Power Requirement for Air-and-Water-System*

In order to control humidity, the outside air might have to be cooled below the supply air temperature. A reheater is installed which increases the air temperature using waste heat from the compressor. A more efficient way to use the supplied cooling energy would be to channel the air through building components and provide some of the conditioning with the air. The electrical power of the *Air-and-Water-System* amounts to:

$$\Sigma Q_{ei,air+water} = 891 \text{ W}$$

Table 7 shows the electrical power calculated for an *All-Air-System* and an *Air-and-Water-System*.

Table 7: Electrical Power Requirement to remove Internal Loads		
	All-Air-System	Air-and-Water-System
Supply Fan	222 W	21 W
Air Cooler	721 W	--
Pre-Cooler/Dehumidifier	216 W	216 W
Exhaust Fan	111 W	11 W
Water Pump	--	2 W
Water Cooler	--	641 W
Total	1270 W	891 W
	100 %	70.2 %

The *Air-and-Water-System* has a power requirement of only about 70% of the total electrical power required by the *All-Air-System*.

A comparison of electrical peak power requirement for *All-Air-Systems* and *Air-and-Water-Systems* was performed based on the data shown in Table 7 [46]. Due to the higher specific cooling loads typically found in office buildings, the peak power requirement for the *Air-and-Water-System* is reduced to approximately 55% of the requirement for *All-Air-Systems* (see Figure 10).

Economics

Although, manufacturers of hydronic radiant cooling systems claim many installations of their system, it is difficult to obtain information about economics of the systems installed. Only few papers deal with the economics of hydronic cooling systems. Feil [45] compares different ventilation/cooling systems for an office. In first-cost comparison with a VAV-system, the break-even point for hydronic systems is approximately at a specific cooling load of 55 W/m². Hoenmann and Nuessle [46] give yearly energy consumption for an office building in Europe (see Table 8). The building has 5000 m² floor area distributed over four floors. The specific cooling load is 50 W/m².
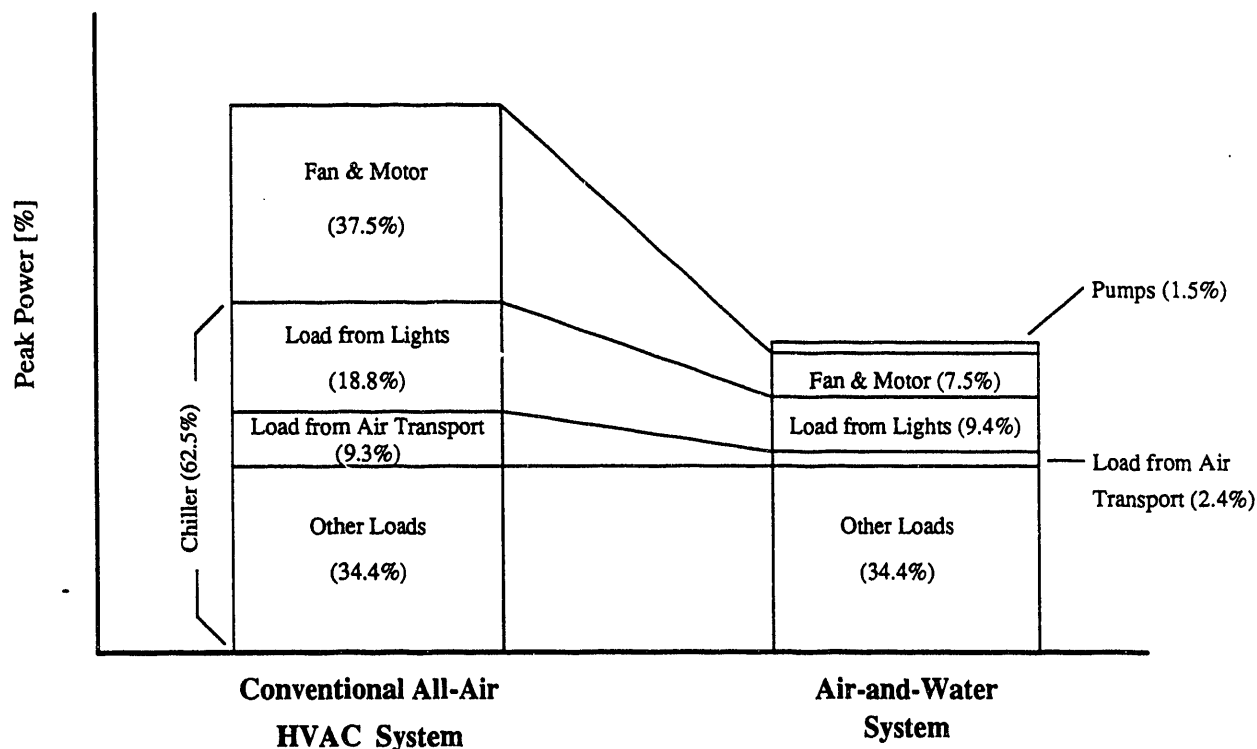
Figure 10: *Comparison of Electrical Peak Power Load for All-Air Systems and Air-and-Water Systems. (For the Air-and-Water System, Percentages are relative to Overall Peak Power for the All-Air System)*

The relatively low savings potential of less than 8% for the building's overall energy consumption is based on the large energy consumption due to heating and lighting. Unfortunately, the authors do not give consumption data for cooling only. Furthermore, the economizer mode savings potential utilized by the outside-air-only VAV-System has not been accomplished for the hydronic radiant system by installation of a water-economizer.

The space requirement for the two systems are shown in Table 9. The biggest savings with 36% appear in the equipment rooms, followed by 28% for the distribution shafts. For systems with false ceilings, the reduction in height per floor calculates to 0.15 to 0.20 m. Systems which are integrated into the ceiling produce higher savings.
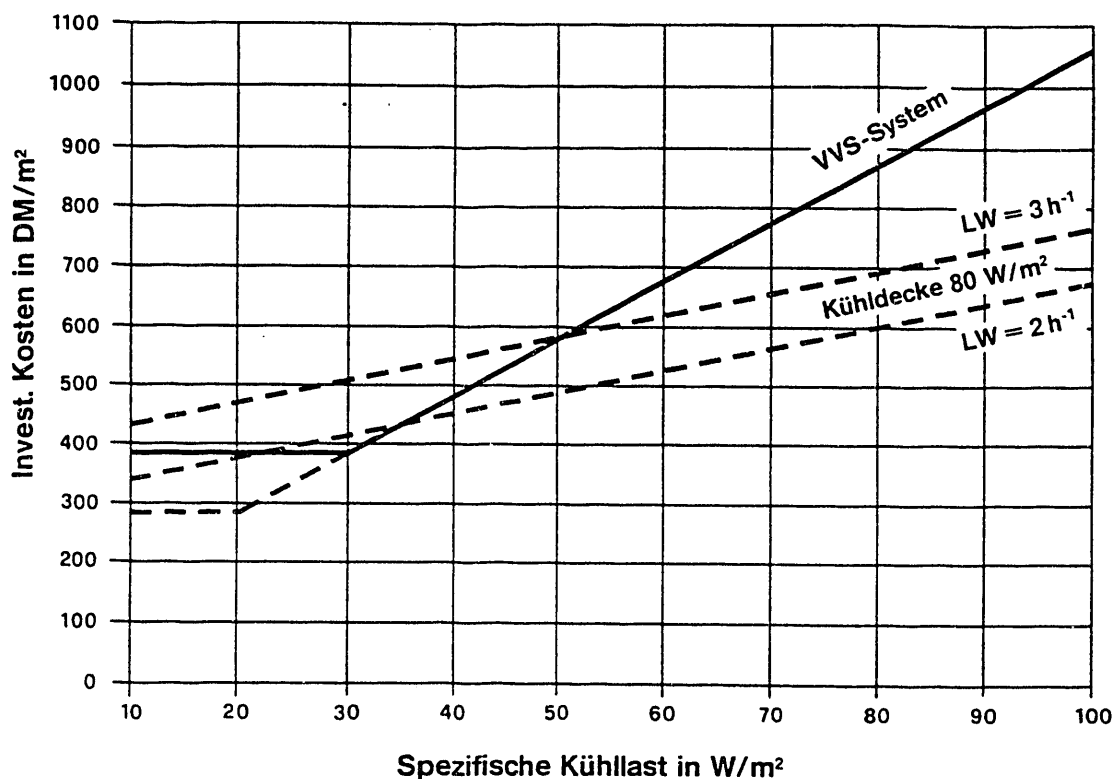
Figure 11: *Comparison between First Cost for VAV-Systems and Hydronic Cooling Systems as a Function of the Specific Cooling Load [46], (VVS-System = VAV-System, Kuehldecke = Cooled Ceiling, LW = Air Change per Hour, Invest. Kosten = First Cost, Spezifische Kuehllast = Specific Cooling Load)*

For first cost calculations, Hoenmann et al. show a break-even point for their aluminium panel system at 50 W/m^2 if an air exchange rate of 3 ach is provided by the ventilation system (see Figure 11).

Table 8: *Yearly Energy Consumption in* kWh/m^2 *for Office Building in Europe [46]*		
	VAV-System	Hydronic Cooling
Heating	43	43
DHW	4	4
Lighting	34	34
Miscellaneous	10	10
Ventilation	12	8
Fans/Pumps	31	24
Cooling	7	8
Sum	141	131

Table 9: *Space Requirements for Systems in Office Buildings [46]*		
	VAV-System	Hydronic Cooling
Shafts	25 m^2	18 m^2
Equipment Rooms	165 m^2	107 m^2
Plenum Height	0.4 m	0.2 m

Systems

Most of the hydronic radiant cooling systems can be categorized by four different system designs. The system used most often is the panel system. This system uses aluminum panels with metal tubes connected to the rear of the panel (see Figure 12). Critical is the connection between the panel and the tube. Poor connections provide only limited heat exchange between the tube and the panel, which results in increased temperature differences between the panel surface and the cooling fluid. Panels build in a sandwich system, include the water flow paths between two aluminum panels (like the evaporator in a refrigerator). This arrangement reduces the heat transfer problem and increases the directly cooled panel surface.

Cooling registers formed out of small plastic tubes which are placed close to each other, can be imbedded in the plaster or mounted on ceiling panels (e.g., acoustic ceiling elements) (see Figure 13). This second system provides an equal surface temperature distribution. Due to the flexibility of the plastic tubes, this system might be the best choice for retrofit applications. This system has been developed in Germany and has been on the market for several years.
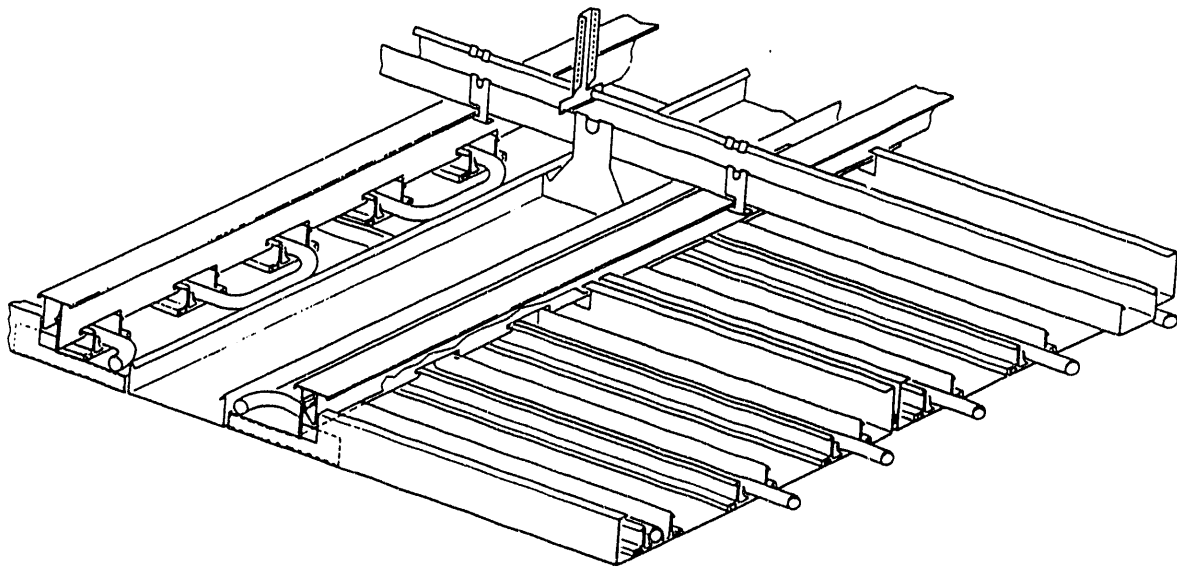
Figure 12: *Construction of a Cooling Panel System [48]*

The third system is based on the idea of a floor heating system. Tubes are imbedded in the core of a concrete ceiling. The thermal storage capacity of the ceiling allows for peak load shifting which provides the opportunity to use this system for alternative cooling sources. Due to the thermal storage involved control is limited. This leads to the requirement of relatively high surface temperatures to avoid uncomfortable conditions in the case of reduced cooling loads. The cooling power of the system is therefore limited [47]. This system is particularly suited for alternative cooling sources, especially the heat exchange with cold night air. The fast warming of rooms with a particular high thermal load can be avoided by running the circulation pump for short times during the day to achieve a balance with rooms with a lower thermal load. The DOW Chemicals European
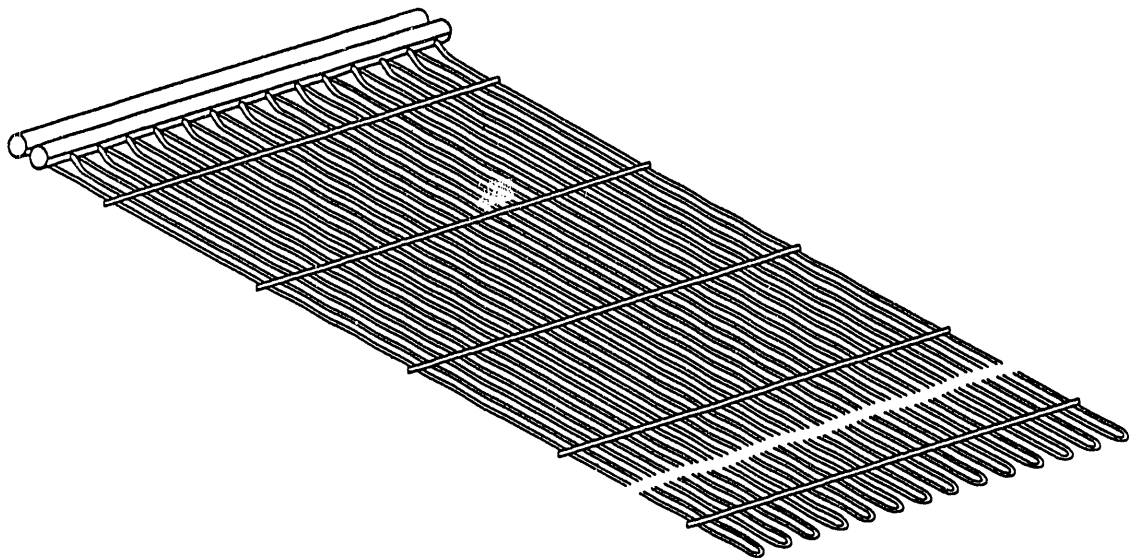
Figure 13: *Construction of a Cooling Register [50]*

headquarter in Switzerland is equipped with such a system.

A fourth system has also been developed in Germany, but is commercially available in California. It provides cooling to a raised floor. The floor provides space for the tubes and the supply plenum. Air is supplied below the windows; reducing the radiative effect of cold window surfaces in winter and hot window surfaces in summer [48].

The temperature profiles for the different ceiling panel systems have been published by Graeff [49].

Control Issues

As mentioned before, cooling power of radiative heat exchange is limited by the dew-point of the room air. For safety reasons the cooling surface is kept above the dew-point for all operation conditions. In order to remove high thermal loads by means of radiation, one can manipulate the dew-point by dehumidifying the supply air. Consequently, the surface temperature of the cooled area can be reduced to increase the operative temperature difference. Precautions should be taken to keep within the specified comfort limits.

Besides the option to reduce the dew-point to avoid damage due to condensation, there is the possibility to switch off the supply of cold water as soon as the relative humidity reaches dangerous levels. Furthermore, buildings have been equipped with window contacts, cutting off the water supply when windows are opened and the ventilation system cannot guarantee trouble-free operation.

Temperature controls for different radiative cooling systems have very different response times. All systems working with thermal mass are relatively slow in response to load changes. If however, operation allows the room temperature to swing and cooling loads can be matched by these systems, they are the most energy efficient systems available. Systems with water supply close to the cooling surface and with little thermal mass (panel systems) have a response time comparable to *All-Air-Systems*.

Although, controlling the dew-point by dehumidifying the outside air requires lower cooler temperatures in the air handling system (than using outside air together with recirculation air for this task), the process is usually more energy efficient because of the smaller amount of air which has to be cooled before the dew-point is reached.

Summary

Whereas hydronic radiant cooling has been applied in the U.S., a significant market penetration was never reached. In Europe too, hydronic cooling was more or less abandoned after some applications in the late thirties and in the fifties. However, user complaints about *All-Air Systems* changed the designers' attitude towards these systems and have led to new system designs with better control. Together with efficient ventilation systems and humidity control, the hydronic radiant cooling system provides several advantages compared to conventional HVAC systems.

The reviewed literature shows that hydronic radiant systems provide draft-free cooling, reduce space requirements, increase indoor air quality, reduce the energy consumption for thermal distribution and for space conditioning, and might even have lower first-cost if specific cooling loads are above 55 W/m^2.

Unfortunately, literature has not been found describing the dynamic thermal behavior of the system and the building. The (dynamic) surface temperature distribution of the enclosure together with the air temperature determine the comfort temperature. Therefore, the dynamics might be an important parameter for further studies. As thermal building simulation programs do not provide the data necessary to judge the performance of radiant

systems, development of dynamic models is needed to better understand comfort issues. Table 10 summarizes the features and their consequences of hydronic radiant cooling.

Table 10: *Summary of Hydronic Radiant Cooling Systems*	
Features	Effect
Separate ventilation and thermal conditioning	Reduce air movement Improve comfort
Transport cooling energy by means of water	Reduce transport energy Reduce peak-power requirement
Eliminate recirculation air	Improve Indoor Air Quality
Use large cooling surfaces	Cool at high temperature level
Can utilize alternative cooling sources	Reduce energy consumption Reduce peak-power requirement
Reduce size of thermal distribution system	Improve space usage Reduce building cost
Reduce convection	Improve comfort
Limit cooling output	Need accurate sizing
Increase risk of condensation	Need good humidity control

Conclusions

In Europe, hydronic radiant cooling became definitely an alternative to conventional *All-Air Systems*. Although, research results have been found supporting the claims made by manufacturers in Europe, there is not sufficient data available for US climates. An analysis based on typical US climates would be necessary to fully assess the applications of radiant cooling for other than the dry climates of California or Arizona.

Due to the availability of limited cooling power emphasis has to be on the building design focusing on the reduction of the building's cooling peak power requirement. A study should be performed, outlining the measures to be taken to satisfy the limited power for different California climates.

Current energy analysis programs, such as DOE-2, cannot simulate hydronic radiant cooling systems. As a result, there is no way to predict the expected performance, which inhibits their use. A computer model should be developed that allows users to calculate heat extraction rates and room surface temperature distributions for radiant cooling systems.

Several hydronic radiant cooling systems are already installed in California. There is some anecdotal evidence that some of these systems do not perform to the occupant's satisfaction. They either seem to consume more energy than predicted, or seem to have problems providing thermal comfort. These buildings should be studied to determine the

status of the systems and to unveil the causes of the performance problems.

Performance test-methods should be developed to compare different products under standard conditions, and field tests should be performed showing the influence of climate, building design and room layout on hydronic radiant cooling performance.

As mentioned earlier, due to their relatively high temperature level, hydronic radiant cooling are obvious candidates to be mated with cooling sources other than compressors. Alternative cooling sources should be investigated to determine their energy savings potential, peak-power reduction and their interaction with the radiant cooling system.

Acknowledgments

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Appendix A: Cooling Power

Radiant cooling elements extract heat from a room by cooling the air (convection) and by cooling the surfaces of the room's envelope. The two effects can be described by:

$$q_c = \alpha_c \left[t_{air} - t_{surface} \right] \qquad (2)$$

for convection, and

$$q_r = \zeta \, F_a F_e \left[\left[\frac{T_r}{100} \right]^4 - \left[\frac{T_p}{100} \right]^4 \right] \qquad (3)$$

q_c	heat transfer by convection [W/m^2]
α_c	convective coefficient [W/m^{2}K]
t_{air}	room air temperature [°C]
$t_{surface}$	surface temperature [°C]
q_r	heat transfer by radiation [W/m^2]
ζ	Stephan-Bolzmann constant [W/m^2]
T_r	mean radiant temperature of unconditioned surface [K]
T_p	mean radiant temperature of cooled surface [K]
F_a	configuration factor [−]
F_e	emissivity factor [−]

for radiation. This shows that the overall heat extraction is a function of the temperature differences between the cooling panel and the air as well as the different surfaces.

Both, convection and radiation, can be expressed by heat transfer coefficients. The combined heat transfer coefficient can be calculated using an empirical equation. Glueck (in [34]) developed the following equation based on measurements of cooled ceilings:

$$\alpha_{tot} = 8.92 \left[t_{air} - t_{surface} \right]^{0.1} \qquad (4)$$

α_{tot}	sum of convective and radiant heat coefficient [W/m^2 K]

This empirical equation is based on the assumption that the mean surface temperature of the room differs only slightly from the air temperature. With this assumption, we can express the specific cooling power (per square meter) of a cooled ceiling by the following equation:

$$q_{tot} = 8.92 \left[t_{air} - t_{surface} \right]^{1.1} \qquad (5)$$

q_{tot}	sum of convective and radiant heat transfer [W/m^2]

Appendix B: Climates used to determine Cooling Loads

The climates used for this study are described by a publication of the National Oceanic and Atmospheric Administration as follows:

Arcata is located at the coast in northern California (Longitude 124.2°, Latitude 40.4°, Altitude 13 m). The climate is completely maritime with high humidity prevailing the entire year. Rainy season begins in October and continues through April. The dry season is marked by considerable fog or low cloudiness that usually clears in the late morning. Temperature as well as enthalpy, are below the conditions of indoor air most of the year.

Temperatures in Long Beach, California (Longitude 118.2°, Latitude 33.8°, Altitude 8 m), rarely peak above 30°C. The enthalpy only exceeds the indoor condition (46.2 kJ/kg), when the temperature is very high. Therefore, this climate can be described as moderately humid. The Pacific Ocean has also a moderating effect on temperatures. The Palos Verdes Hills form a natural border between the weather station and the sea; causing slightly greater ranges between minimum and maximum temperatures than stations on the immediate coast. Precipitation is sparse during the summer months. During the summer, low clouds are quite common in the late night and morning hours.

Red Bluff, California, is located in the northern end of the Sacramento Valley (Longitude 122.2°, Latitude 40.2°, Altitude 104 m). Mountains surround the city on three sides, forming a huge horseshoe. The Coastal Range is located about 45 kilometers west, the Sierra Nevada about 60 kilometers east, and the Cascade Range about 75 kilometers northeast and north. Precipitation is confined mostly to rain during winter and spring months. Temperatures are highest during the months of June through September, when daytime readings frequently surpass the 35°C mark. Nighttime temperatures, however, are usually comfortable.

San Diego (Longitude 117.2°, Latitude 32.7°, Altitude 4 m), is located in the southwest corner of southern California. Weather is tempered by the Pacific Ocean, with the result that summers are cooler and winters are warmer than other places along the same general latitude. Dry easterly winds sometimes blow in the vicinity for several days at a time, bringing temperatures above 30°C. As on the rest of the Pacific Coast, a dominant characteristic of the spring and the summer is the nighttime and early morning cloudiness.

Lake Charles in Louisiana (Longitude 93.2°, Latitude 30.1°, Altitude 4 m) is located on the lake of the same name and the Calcasieu River. Lake Charles' climate is humid subtropical with strong maritime character. The climate is influenced by the water surface provided by the lake, flooded rice fields and the Gulf of Mexico. Rainfall is substantial in all seasons. The absolute humidity rises with the seasonal rise of temperature, thus, the enthalpy increases more then proportionally to the temperature. The winter months are normally mild with cold spells usually of short duration. Summer relative humidity exceeds 80% for about 12 hours a day.

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