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Hydronic radiant cooling – preliminary assessment ☆

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Abstract

A significant amount of the electrical energy used to cool non-residential buildings equipped with all-air systems is drawn by the fans that transport the cool air through the thermal distribution system. Hydronic systems reduce the amount of air transported through the building by separating the tasks of ventilation and thermal conditioning. Due to the physical properties of water, hydronic systems can transport a given amount of thermal energy and use less than 5% of the otherwise necessary fan energy. This improvement alone significantly reduces the energy consumption and peak-power requirement of the air conditioning system. Radiant cooling has never penetrated the US markets significantly. The scope of this survey is to show the advantages of radiant cooling in combination with hydronic thermal distribution systems, as compared to the commonly-used all-air systems. The report describes the development, thermal comfort issues, and cooling performance of the hydronic systems. The peak-power requirement is also compared for hydronic systems and conventional all-air systems.

Keywords: Hydronic radiant cooling; Radiant cooling

1. Introduction

Cooling of non-residential buildings equipped with all-air systems significantly contributes to the electrical energy consumption and to the peak-power demand. Part of the energy used to cool buildings is consumed by the fans that transport cool air through the ducts. This energy heats the conditioned air, and therefore adds to the internal thermal cooling peak load. Usibelli et al. [1] found that, in the case of the typical office building in Los Angeles, the external loads account for only 42% of the thermal cooling peak (see Fig. 1). At that time, 28% of the internal gains were produced by lighting, 13% by air transport, 12% by people and 5% by equipment. The implementation of better windows, together with higher plug loads due to increased use of electronic office equipment, have probably caused these contributions to change to some extent since then.

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.

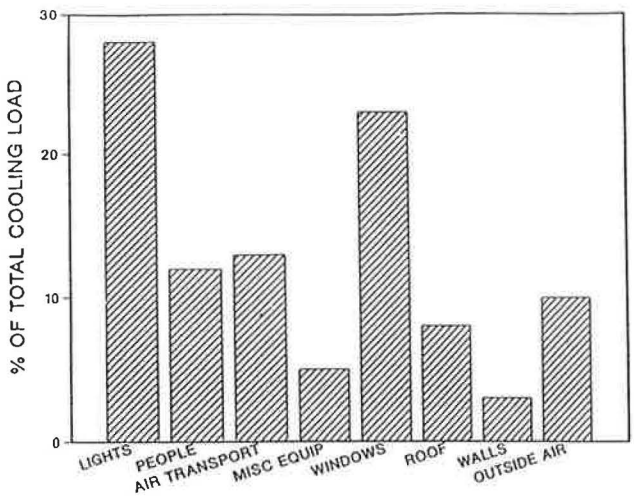

Fig. 1. Peak cooling load components, typical office building, Los Angeles.

DOE-2 simulations for different California climates using the California Energy Commission (CEC) base case office building show that, at peak load, only 10–20% of the supply air is outside air [2]. Only this small fraction of the supply air is in fact necessary to ventilate the buildings in order 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

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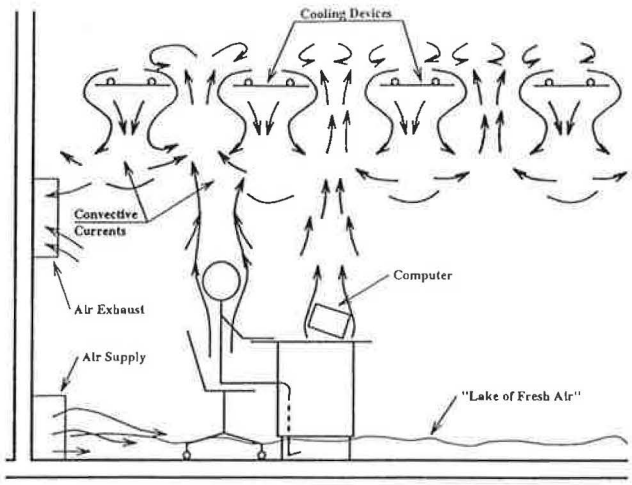


Fig. 2. Air flow patterns in a room with a cooled ceiling.

Upward displacement ventilation shows a characteristic temperature profile caused by the convective currents driven by the heat sources. As supply air enters the room at floor level, the temperature gradient forms a barrier that prevents low energy currents reaching high altitudes in the room. Due to comfort requirements, the temperature gradient between feet and head cannot exceed 3 °C, which limits the cooling capacity of displacement ventilation systems [10]. The fact that displacement ventilation systems use solely outside air further reduces their cooling capacity [12,13].

The most efficient way to use displacement ventilation is to associate it with a cooling source that does not require air transport inside the room. The logical choice is the coupling of displacement ventilation systems with hydronic radiant cooling, a strategy that allows the separation of the tasks of ventilating and cooling in the building. The theoretical air flow pattern in a room with a cooled ceiling is shown in Fig. 2 [10].

3. Thermal comfort

In order to maintain normal functions, the human body needs to maintain the balance between heat gain and heat loss. Heat can be lost in different ways: radiation to surrounding surfaces, convection to the ambient air, conduction, evaporation, respiration and excretion. The most important loss is due to radiation, followed in order by convection and conduction. Respiration and excretion have less influence on the heat loss of a human body.

To explain the impact of radiation, Baker [14] gives the following example: "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 70s (°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 a person inside an enclosure have a great influence on the thermal comfort of that person, and therefore have to be carefully studied.

If people could not lose heat by radiation, and convection were the only available heat loss mechanism, high air velocities close to the human skin would be required in order to produce a given heat loss. A continuous increase in air velocities would eventually lead to draft, and therefore, to uncomfortable conditions. The possibilities of increasing the heat loss by respiration or excretion are very limited.

Heat loss by radiation is caused by the difference between the body temperature and the mean radiant temperature, which depends on the temperatures of the surrounding surfaces. The mean radiant temperature is easy to define but quite complicated to calculate or measure in practice. Due to the non-uniform distances and angles of persons in relation to the walls, floor and ceiling of a space, each part of the space must be considered separately in the radiation exchange. If a given surface is found to not be isothermal, it has to be divided into smaller isothermal surfaces. Each surface can be assumed to have high emissivity. The radiation emitted and reflected from any surface is distributed as diffuse radiation, which is a good approximation for all normal non-metallic surfaces [15]. The enclosure surfaces often found in a normal room have a rectangular shape and, therefore, the angle factor in the mean radiant temperature calculations is defined between a person and a vertical or horizontal plane. The body posture is also important. The mean radiant temperature in relation to a standing person is not necessarily the same as in relation to a seated one [15]. Likewise, the location and orientation of the person inside the room must also be known, because the mean radiant temperature often varies from point to point.

The first experiments of thermal and comfort sensations to radiation experienced by seated persons were conducted by Schlegel and McNall [16], and McNall and Biddison [17]. Fanger [15] 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."

The combined effects of radiation and convection inside an enclosure can be evaluated by using a parameter called the 'operative temperature'. The operative temperature is calculated by averaging the dry bulb temperature and the mean radiant temperature inside the enclosure. The definition of the operative temperature shows that this parameter does not reflect the presence of radiation asymmetry inside an enclosure.

In the case when this effect is important, the use of operative temperature in evaluating thermal comfort might lead to erroneous results.

Air movement plays a special role among the comfort parameters. According to Esdorn et al. [7], air movement is the largest single cause for complaints (draft). Besides the mean air velocity, the fluctuation of the air velocity has an important influence on the convective heat transfer of the human body. Mayer [18] relates comfort directly to the convective heat transfer coefficient, rather than to the average air velocity. According to Mayer [19], 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 average air velocities for laminar flows of 1.35 m s^{-1}, for transition flows of 0.15 m s^{-1}, and for turbulent flows of 0.10 m s^{-1}. Lower air temperatures significantly reduce the acceptable air velocities.

Extended explanations about the recommended levels of thermal comfort parameters can be found in the ASHRAE Standard 55-92, ISO 7730 and the 1993 ASHRAE Handbook of Fundamentals, Ch. 8.

4. Cooling power

The cooling power of HRC systems is limited due to the fact that in operating these systems, the side-effects associated with the presence of cold surfaces in the space have to be prevented or minimized. A first effect to prevent is condensation. In theory, the surface temperatures of the cooling elements must not be lower than the dew point temperature of the air in the cooled zone. In practice, however, condensation prevention reduces the effective cooling temperature difference (between the cold surface and the air) by a safety margin of approximately 2 °C. In the operation of the system the dew point can also be manipulated, by reducing the humidity content in the ventilation air. A second and more serious concern is the comfort effect of the asymmetrical distribution of the radiant temperature. Kollmar [20] shows that for offices, the lower limit for ceiling temperatures is approximately 15 °C.

To calculate the cooling power of a hydronic radiant cooling system, the heat transfer between the room and the cold ceiling has to be evaluated. There are two components of the heat transfer: radiation and convection. While the radiation term 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 inlet and exhaust.

Trogisch [21] 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 overall heat transfer coefficients of 9–12 W m^{-2} K. Given a heat transfer coefficient for radiation of about 5.5 W m^{-2} K (for a temperature difference of 10 °C), the resulting convective heat transfer coefficient would be in the order of 3.5–6.5 W m^{-2} K. These values for the convective heat transfer coefficient are however reached only if forced convection takes place (here forced means that phenomena other 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 directly (convection) and indirectly, by cooling the surfaces of the room envelope. If there is only a small difference between the average surface temperature of the room and the air temperature, the two effects can be estimated jointly [22]. Under this assumption, the specific cooling power (per unit area) of a cooled ceiling can be expressed by the following (empirical) equation (see Appendix):

$$q_{tot} = 8.92(t_{air} - t_{surface})^{1.1} \quad (1)$$

q_{tot} sum of convective and radiant heat transfer (W m^{-2}).

A survey of cooled ceilings [23] shows cooling output ranging from 40 to 125 W m^{-2}. The data are however based on information from manufacturers and do not specify the boundary conditions for the measurements. This problem brings up the necessity of standards for both the measurement conditions and techniques, and several attempts have already been made to set standards for testing radiant panels.

A test facility and a method of testing was developed at the Department of Veterans Affairs [24]. The method describes the testing procedure for thermal performance and pressure drop measurements in the test facility, as well as the accuracy of the instrumentation used.

ASHRAE's technical committee TC 6.5 Radiant Space Heating and Cooling currently sponsors 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 [25].

In Germany, two competing testing procedures have been recently published within five months. The Fachinstitut Gebaeude-Klima (FGK) presented its testing procedure in December 1992 [26]. The FGK industrial standard is based on a measurement in a box (2.4 m × 1.2 m × 1.5 m) with an internal operative temperature of 26 °C and water supply temperatures of 12, 14 and 16 °C. The DIN standard was presented in April 1993 [27]. It measures the performance of radiant panels in the presence of natural convection. The test is based on measurements performed in a closed test chamber

(4 m × 4 m × 3 m) with a conditioned metal envelope. The cooling load is simulated by 12 perforated tubes containing three 60 W bulbs each. The measurements are performed under steady-state conditions, with different coolant mass flows.

While testing procedures and future standards can rate the performance of an HRC system with panels under given boundary conditions, the efficiency of the same system in a specific (but different) application is difficult to determine. The difficulty arises from the fact that the rated performance greatly depends on the testing procedure. For example, a procedure for measuring the efficiency of a cooled ceiling could use the temperatures of the ceiling and of the exhaust air in a test room as a measure for the convective heat transfer between the ceiling and the room air. If, in a hypothetical situation, a shortcut between the supply and the exhaust of the ventilation system in the test room provided high air velocities on the ceiling surface, a large fraction of the exhaust air would in fact be air that has been cooled by the ceiling without having interacted with the room loads at all. The difference between the temperatures of the ceiling surface and the exhaust air would in this case indicate lower convective heat transfer than in reality, the measurements would appear as having been performed under low air flows, and the ceiling would show high efficiency. The functioning of the same ceiling in a normal situation (without the forced convection) is very likely to give different results. These considerations show that a building designer should carefully consider all the details in a testing procedure before deciding to use a specific type of cooling ceiling panel.

5. Cooling performance

Although several papers have been found that describe the cooling power of HRC systems, only a few papers are found that investigate the performance of these systems. Kuelpmann [28] reports on an experimental investigation in a temperature controlled test cell (see Fig. 3). In this experiment, the 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, the room air temperatures measured at different heights did not differ very much (see Fig. 4). The extraction of 100 W m^{-2} internal load by hydronic radiant cooling caused temperature differences of approximately 2 °C between the supply and exhaust grilles. When the temperature difference between the room air and the supply air was increased,

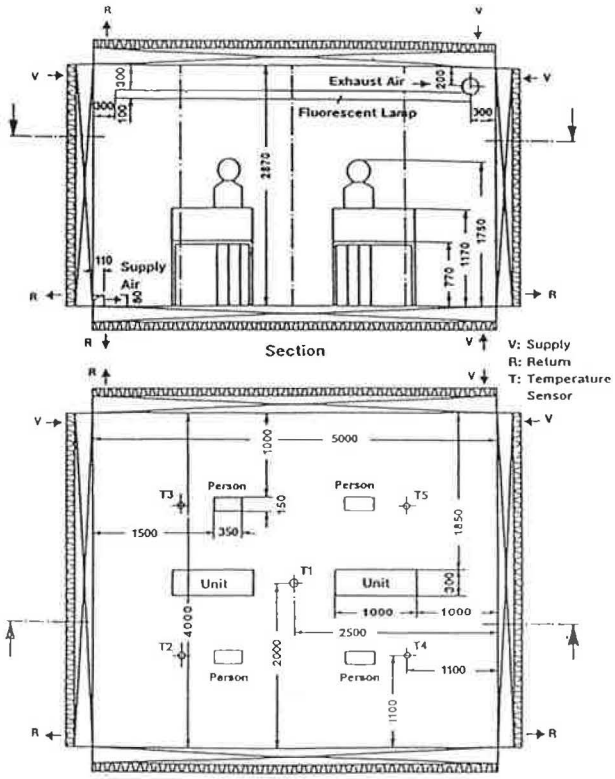
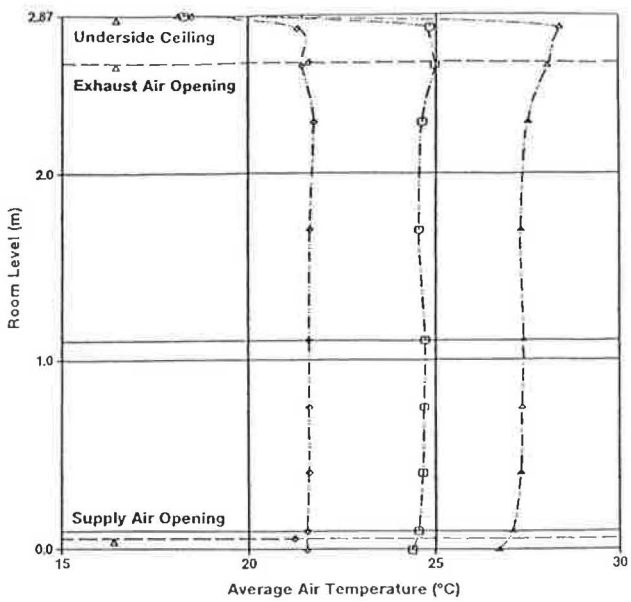


Fig. 3. Test chamber used to test the cooling performance of hydronic cooling.



Symbol	Cooling Load (W/m²)	Room Temperature (°C)	Temp.-Diff. Walls-Air (K)	Performance Part (-) Ceiling ω_{D*}	Ventilation ω_L
◊	35	21.4	0.2	0.96	0.04
□	67	24.4	-0.2	1.00	0.00
Δ	101	26.7	-0.3	1.00	0.00

Fig. 4. Air temperature profile over the room height as a function of external loads; tabulated room temperature is measured at 1.1 m above the floor.

the profile became more pronounced. Especially in the lower part of the room, these temperature differences became close to, or exceeded, the comfort limits.

In all cases examined, the differences between the room air temperature and the surface temperatures of the 'internal walls' were relatively small (±0.4 °C). Due to the radiation exchange with the cooled ceiling, the floor surface temperature was 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, cooled ceiling panels also produce asymmetric thermal radiation. Radiant asymmetry due to a cooled ceiling causes less discomfort than a warm ceiling. Based on Fanger's limit of 5% uncomfortable as a rule for determining the acceptability of a system, a radiant temperature asymmetry of 10 °C is acceptable in the presence of a cool wall, and of 14 °C in the presence of a cooled ceiling (see Fig. 5) [29].

Measurements of radiant temperature asymmetry at 101 W m^{-2} cooling power in the reported investigation resulted in 8 °C difference at 1.1 m above the floor level, in the middle of the room (see Fig. 4). This corresponds to less than 2% of occupants dissatisfied (see Fig. 5).

Air flow velocities were measured at 1 m distance to the supply air grille, at 0.1 m height above ground. At an air exchange rate of 3.2 ACH and a supply air temperature of 19 °C, exceptionally low values were measured for the air velocity (0.12 m s^{-1}) and the turbulence intensity (20%).

The performance of hydronic radiant cooling was tested in two parliamentarian offices in Bonn, Germany [30]. Dry-bulb temperatures and relative humidity were measured for the outside air, the supply air and the room air. Temperature measurements were also made in the supply and return pipes of the hydronic system and at three points on the ceiling surface. For outside air temperatures of 30 °C, the air velocities measured in the occupied zone were below 0.10 m s^{-1}. This value shows that the risk of draft has been eliminated in the tested zones. Below the ceiling, surface velocities between 0.10 and 0.15 m s^{-1} were detected. These low velocities assure that less than 40% of the heat transfer occurs by convection.

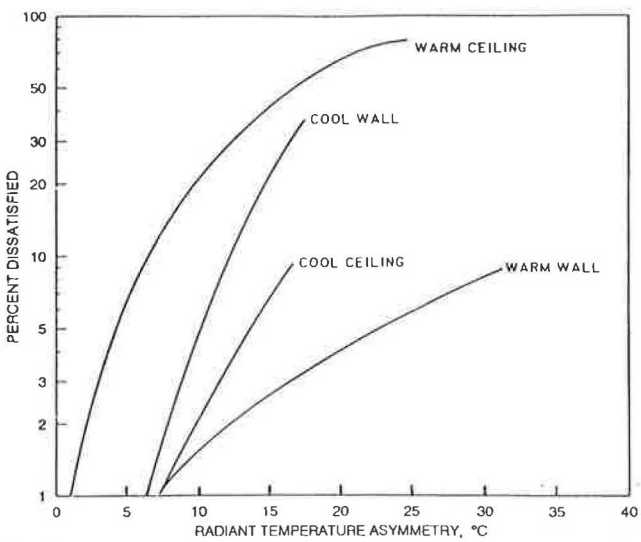


Fig. 5. Measured percentage of people expressing discomfort due to asymmetric radiation.

6. Numerical modelling

The evaluation of the theoretical performance of HRC systems could most conveniently be made by computer models. Energy analysis programs such as DOE-2 do not have the capacity of simulating hydronic cooling systems yet. There have been attempts to adapt DOE-2 so that it roughly performs the task [31], but this approach involves laborious artifices, and is not easily accessible to the average DOE-2 user. The consensus is that a separate module needs to be designed which models the specifics of hydronic radiant cooling. Such a model is presently being developed at the Lawrence Berkeley Laboratory by the Energy Performance of Buildings Group. It is a SPARK-based module (SPARK = Simulation Problem Analysis Research Kernel) that performs most of the tasks of DOE-2, on the particular case of the HRC systems [32].

7. Energy savings

The use of HRC systems is an energy conserving and peak-power reducing alternative to conventional air conditioning, particularly suited to dry climates. A significant amount of the electrical energy used to cool buildings by all-air systems is consumed by the 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 from the fans and the chillers, has a breakdown of approximately 37% for running the fans and 63% for using the chillers.

If the tasks of ventilation and thermal conditioning of buildings are separated, the amount of air transported through buildings can be significantly reduced. In this case the cooling is provided by radiation using water as the transport medium, and the ventilation by outside air systems without the recirculating air fraction. Although the supply air necessary for ventilation purposes is still distributed through ducts, the electrical energy for fans and pumps can be reduced to approximately 25% of the original value.

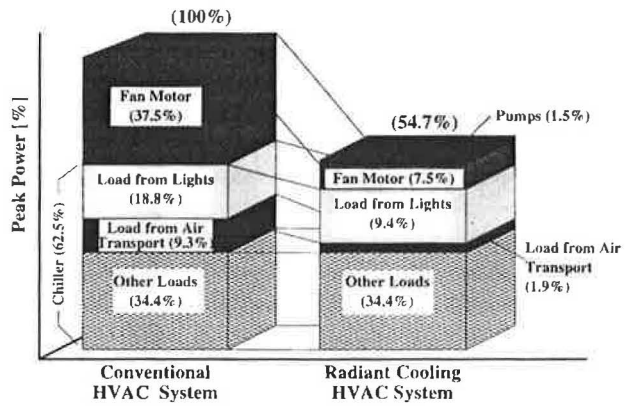


Fig. 6. Comparison of electrical peak power load for an all-air system and an HRC system, based on the peak-load data shown in Fig. 1. Percentages for the HRC system are relative to the overall peak power of the all-air system.

The elimination of recirculation air also increases the efficiency of air-handling luminaries, as the convective heat extracted from the light fixtures is not recirculated as it partly would be in an all-air system, but vented directly to the exterior. 50% of the required thermal cooling energy produced by lighting can be removed in this way. Compared to a constant volume air system, an overall electrical cooling energy savings potential of more than 40% seems reasonable (see Fig. 6).

8. Peak-power requirement

In order to compare the electrical peak-power requirement for conventional systems (all-air systems) and advanced systems (HRC 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, 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 HRC systems. The room temperature is set to 26 °C. Additional assumptions and design considerations used for this example are shown in Table 1.

The all-air system supplies cooling to the room as follows: a cooling coil dehumidifies the outside air according to the required room conditions. ASHRAE Standard 62-1989 requests a minimum air change rate of 36 m^3 h^{-1} per person, which means that for this example the minimum air change rate is 72 m^3 h^{-1}. In order to remove the internal load, a recirculating air volume flow of 678 m^3 h^{-1} is required. The assumed outside air condition of 32 °C leads to a mixing temperature of 25.6 °C.

After having mixed, the air enters a cooler. In order to adjust for the temperature increase due to the fan work, the air has to be cooled further 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, this air handling temperature rise has been assumed to be 1.0 °C.

The electrical power for an all-air system amounts in this example to

$$\sum Q^{el}_{\text{all-air system}} = 1270 \text{ W}$$

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

Whereas the all-air system removes the cooling load by means of supplying cold air, the HRC system removes the load mainly by means of water circulation. The tasks of the ventilation side of the system are thus to supply the room with the necessary air exchange rate for comfort reasons, and to control the dew-point in the room, to avoid humidity buildup. In order to provide a stable displacement ventilation, the supply air volume flow should be about 3 °C below the room air temperature. The required temperature of the supply air is thus 23 °C, which reduces the hydronic cooling load by around 3 W m^{-2}.

In order to control humidity, the cooling of the outside air below the supply air temperature might be necessary. A reheater can be installed which warms the air using waste heat from the compressor. The warming of the air could be done even more efficiently if the air were channelled through building components before arriving to the room inlet. This would save the power to reheat and provide some conditioning at the same time. The electrical power of the HRC system amounts to

$$\sum Q^{el}_{\text{air-and-water}} = 909 \text{ W}$$

Table 2 shows the electrical power calculated for an all-air system and an HRC system. The values in the table show that the power required by the HRC system is only about 72% of the total electrical power required by the all-air system. The savings could be increased by venting the light fixtures in the building.

9. Economics

Although manufacturers of HRC systems claim many installations of their systems, it is difficult to obtain information about the economics of the systems installed. There are only a few papers that deal with the economics of HRC systems.

Feil [33] compared different ventilation/cooling systems for an office. In a first-cost comparison with a VAV system, the break-even point for HRC systems is approximately at a specific cooling load of 55 W m^{-2}.

Table 1
Assumptions used for the comparison of peak-power requirements for an all-air system and an HRC system

	Both systems	
Room conditions		
Cooling load (W m^{-2})	80	
Room air temperature (K)	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	
	All-air system	HRC system
Design considerations		
Outside air flow ($m^3\ h^{-1}$)	72	72
Supply air flow ($m^3\ h^{-1}$)	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

Table 2
Electrical power requirement to remove internal loads from a two-person office with a floor area of 25 m

	All-air system	HRC 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		20 W
Water cooler		641 W
Total	1270 W	909 W
	100%	71.5%

Table 3
Yearly energy consumption (kWh m^{-1}) for an office building in Europe

	VAV system	HRC system
Heating	43	43
DHW	4	4
Lights	34	34
Miscellaneous	10	10
Ventilation	12	8
Fans/pumps	31	24
Cooling	7	8
Sum	141	131

Hoenmann and Nuessle [34] estimated yearly energy consumption for an office building in Europe (see Table 3). The building has 5000 m^2 of floor area distributed over four floors. The specific cooling load is 50 W m^{-2}. The relatively low savings potential for the overall energy consumption of the building (less than 8%), is due to the large energy consumption by heating and lighting. Unfortunately, the authors do not provide consumption data for cooling only. Furthermore, the outside-air-only VAV system utilized an economizer mode, while the same savings potential has not been matched in the HRC system by installing a water-economizer.

The space requirements for the two systems are shown in Table 4 [34]. The largest savings, 36%, appear in the equipment rooms, followed by 28% for the distribution shafts. For systems with false ceilings, the reduction in height per floor is in the order of 0.15–0.20 m. HRC systems which are integrated into the ceiling produce even higher savings.

Table 4
Space requirements for systems in office buildings

	VAV system	HRC system
Shafts	25 m^2	18 m^2
Equipment rooms	165 m^2	107 m^2
Plenum height	0.4 m	0.2 m

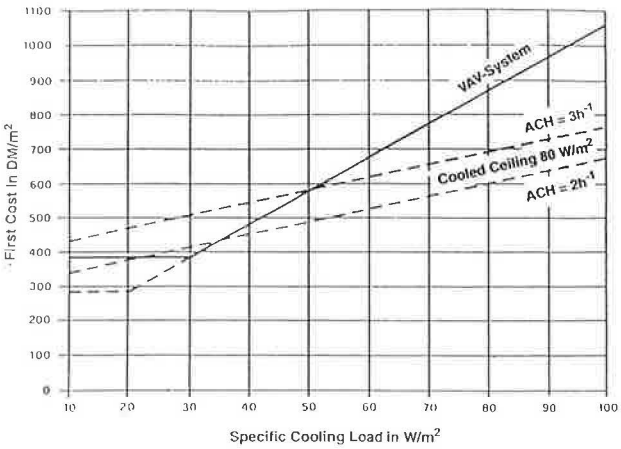


Fig. 7. Comparison between first cost for VAV systems and HRC systems, as a function of the specific cooling load.

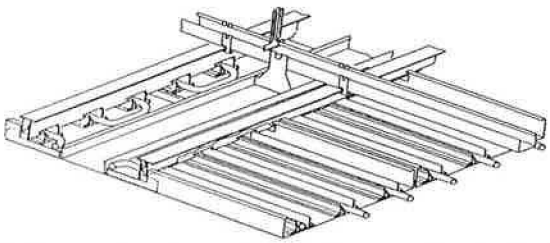


Fig. 8. Construction of a cooling panel system. Drawing by Fläkt Lufttechnik GmbH.

For first cost calculations, Hoenmann and Nuessle [34] show a break-even point for their aluminum panel system at 50 W m^{-2} at an air exchange rate of 3 ACH (see Fig. 7 [34]).

10. Systems

Most of the HRC systems belong to one of four different system designs. The most often used system is the panel system. This system is built from aluminum panels with metal tubes connected to the rear of the panel (see Fig. 8 [34]).

The connection between the panel and the tube is critical. 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 built 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.

In the case of panels suspended below a concrete slab, approximately 93% of the cooling power is available to cool the room. The remaining 7% cools the floor of the room above (see Fig. 9).

Fig. 10 shows a typical installation of suspended ceiling panels. A closed panel arrangement with in-

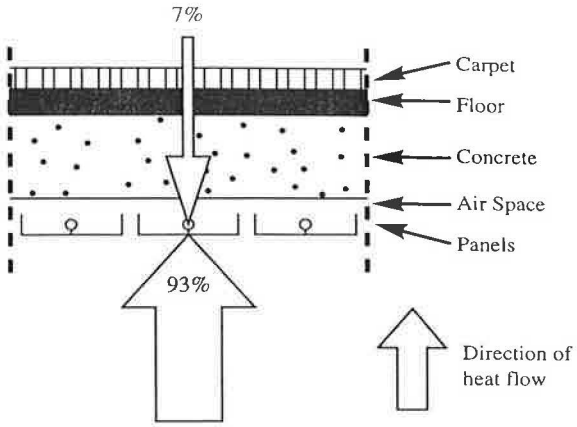


Fig. 9. Heat transfer for panel system (cooling mode).

Fig. 10. Cooling ceiling in an office environment (courtesy of REDEC AG).

sulated panel backsides uncouples the thermal storage of the slab in this arrangement. This arrangement improves the response time for startup conditions, but loses the ability of smoothing cooling load peaks.

The temperature profiles for the different ceiling panel systems have been published by Graeff [35].

Cooling grids made of small plastic tubes placed close to each other can be imbedded in plaster, gypsum board or mounted on ceiling panels (e.g. acoustic ceiling elements) (see Fig. 11). This second system provides an even surface temperature distribution. Due to the flexibility of the plastic tubes this system might be the best choice for retrofit applications. It was developed in Germany and has been on the market for several years.

When the tubes are imbedded in plaster, the heat transfer from above is higher than in the case of cooling panels (Fig. 12). The heat transfer to the concrete

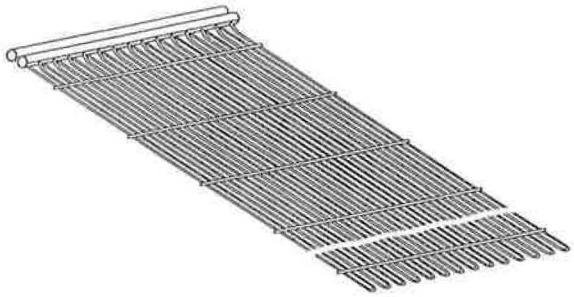


Fig. 11. Construction of a cooling grid (courtesy of KaRo-Information Service).

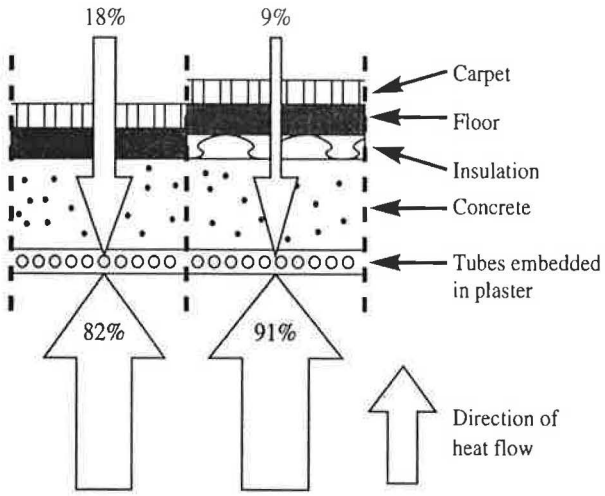


Fig. 12. Heat transfer for ceiling with cooling grid.

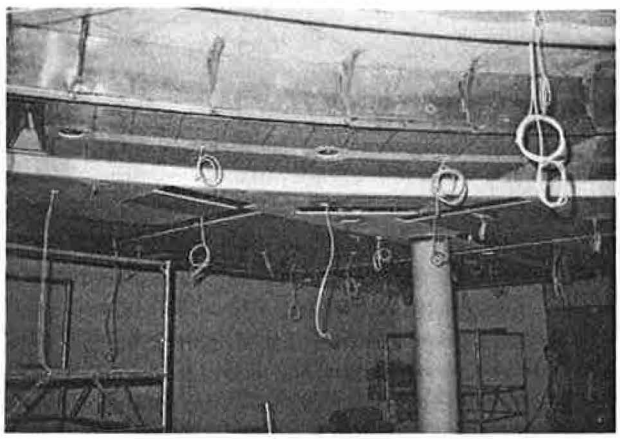


Fig. 13. Cooling grid attached to concrete ceiling before being covered with plaster (courtesy of KaRo-Information Service).

couples the cooling grid to the structural thermal storage of the slab. Plastic tubes mounted on suspended cooling panels show thermal performance comparable to the panel systems described above. Tubes imbedded in a gypsum board can be directly attached to a wooden ceiling structure without a concrete slab. Insulation must be applied to reduce cooling of the floor above. Fig. 13 shows cooling grids attached to a concrete ceiling before being covered with plaster.

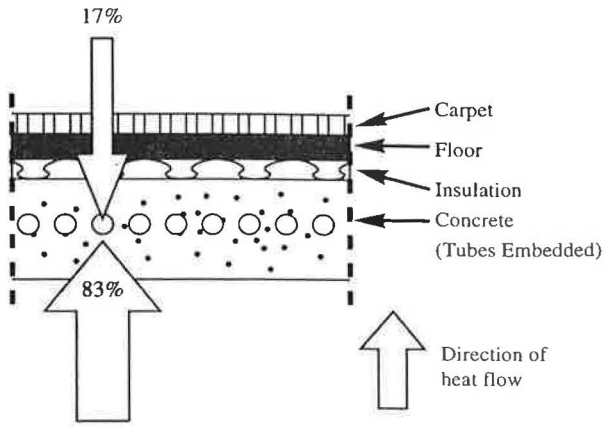


Fig. 14. Heat transfer for slab cooling.

A third system is based on the idea of a floor heating system. The 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 in association with alternative cooling sources. Due to the thermal storage involved, the control of this system 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 [36]. This system is particularly suited for alternative cooling sources, especially the heat exchange with cold night air. The faster 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.

Due to the location of the cooling tubes in this system, a higher portion of the cooling is applied to the floor of the space above the slab. Approximately 83% of the heat removed by the circulated water are from the room below the slab, while 17% are from the room above (Fig. 14).

A fourth system has been developed in Germany, which is also 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 [37].

11. Control issues

As mentioned before, the cooling power of radiative heat exchange is limited by the dew-point of the room air. In order to avoid condensation the cooling surface is kept above the dew-point for all operation conditions. If the dew-point is reduced by dehumidifying the supply air, higher thermal loads can be removed by means of

Table 5
Summary of HRC 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
Use large cooling surfaces	cool at high temperature level
Eliminate recirculation air	improve indoor air quality
Reduce convection	improve comfort
Reduce size of thermal distribution system	improve space usage reduce building cost
Can utilize alternative cooling sources	reduce energy consumption reduce peak-power requirement
Limited cooling output	need accurate sizing
Increased risk of condensation	need good humidity control

candidates to be associated with cooling sources other than compressors. Alternative cooling sources should therefore be investigated, and their energy savings potential, peak-power reduction and interaction with the HRC systems should be determined.

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Appendix: Cooling power

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

$$q_c = \alpha_c(t_{air} - t_{surface})$$

for convection, and

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

for radiation, where

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 K^4)
T_r mean radiant temperature of an unconditioned surface (K)
T_p mean radiant temperature of a cooled surface (K)
F_a configuration factor (–)
F_e emissivity factor (–)

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 means of heat transfer coefficients. The combined heat transfer coefficient can be calculated using an empirical equation. Glueck (1990) developed the following equation based on measurements of cooled ceilings:

$$\alpha_{tot} = 8.92(t_{air} - t_{surface})^{0.1}$$

where 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:

$$q_{tot} = 8.92(t_{air} - t_{surface})^{1.1}$$

q_{tot} = sum of convective and radiant heat transfer (W/m^2).

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