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A NEW DEVELOPMENT IN AIR CONDITIONING

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ABSTRACT

Ventilation by heat sources and a cold ceiling—these two terms and the technology behind them have been the main topics discussed in the European ventilation and air-conditioning trade for about two years. The characteristics of this new air-conditioning system, which many experts call the "system of the future," have been presented at many recent international specialist meetings and congresses. Most of the papers focused on the advantages, such as "increased comfort," "savings in energy and space," "avoidance of draft due to low air velocities," and "decoupling the removal of the thermal load from the necessary volume of supply air." In Scandinavia and the West European countries in particular, the technology has spread. Even if many planners and architects still are skeptical about the system of ventilation by heat sources and a cold ceiling, the technology's international advancement seems to be only a matter of time.

This paper, however, will not go into the scientific fundamentals of the system. It will present the mode of operation of a cold ceiling combined with a system of heat-source ventilation and then give a survey of the state of the technology. Some examples of application, which not only show the manifold possibilities but also the limits and disadvantages of this system, will be presented.

INTRODUCTION

For years, air-conditioning systems in Europe have mainly been designed as all-air systems, which means the supply air volume flow for one room has to be determined in such a way that all loads in the room, such as heat, humidity, and odorous substances, are removed solely by convection. In most cases, larger air volumes are used more to eliminate heat than to control loads from materials. Consequently, the air volume needed to control the thermal loads is larger than what is necessary to produce indoor air conditions acceptable from a hygienic point of view. This was not the case in times when the high-pressure induction system prevailed. The necessary primary air volume (pure outdoor air) was determined according to hygienic demands; the room load was compensated by the coils of the high-pressure induction systems. In Europe, high-pressune induction systems have no longer been built in the past decade for various reasons. Consequently, the air volume supplied in a room was again designed according to the cooling load of the room. With almost all existing air-conditioning systems, many occupants complained of draft. Annoyance because of drafts has been an important reason for public aversion to air-conditioning systems in recent years.

For some time, European—in particular, Scandinavian—air-conditioning engineers have been trying to develop an air-conditioning system that would exclude some disadvantages of the existing systems.

Important aspects of these designs were

1. decoupling of thermal loads and loads from materials,
2. removal of the thermal load of the room via static cooling surfaces installed in the room,
3. removal of heat by convection and radiation,
4. removal of the thermal load of the room by means of chilled water,
5. an air flow as small as possible and correspondingly low air velocity in the areas in which persons are located,
6. avoidance of draft,
7. constant air volume,
8. lowest possible energy consumption, and
9. lowest possible space requirements by the air-conditioning system.

Another marginal condition that has to be taken into account is the fact that most European office buildings no longer need any heating load during office hours. The manifold, often rather high, internal loads from computers, telefax systems, copiers, illumination, sun radiation, etc. (Figure 1), as well as the prescribed insulation of the outside walls, result in a permanent heat surplus as soon as the office is in operation. This also applies to wintertime and extremely low outdoor temperatures.

The above-mentioned problems have been solved by the system of cold ceiling and heat-source ventilation, which in most cases are combined (Figure 2). The ceiling of an air-conditoned room is designed as a large surface cooled by chilled water. There are several construction possibilities to which we shall refer later.

Designing one or several walls as cooling surfaces has also been tried. Better comfort, however, was clearly achieved by a cold ceiling. A cold floor proved to be completely unsuitable for cooling a room; as a heating surface, however, the floor is better than a warm ceiling.

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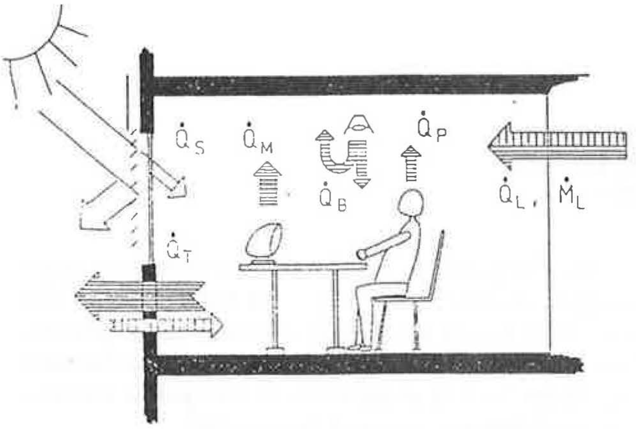


Figure 1 *Thermal load in a modern office room due to radiant heat* (Q_s), *transmission heat* (Q_T), *machine heat* (Q_M), *illumination heat* (Q_B), *heat from persons* (Q_P), *and ventilation heat* (Q_L).

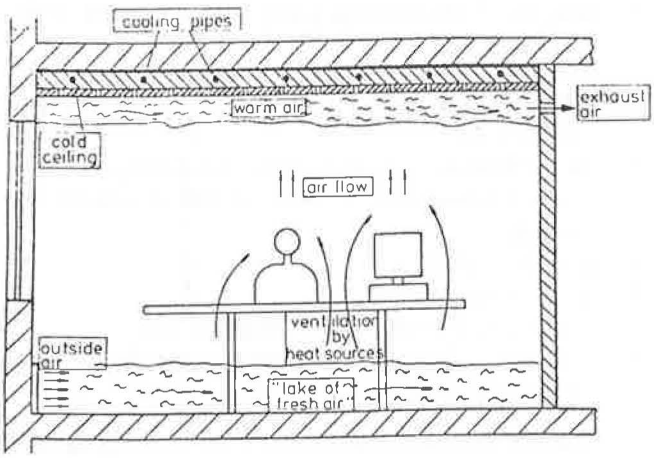


Figure 2 *Cold ceiling and heat-source ventilation system.*

The air volume necessary to meet hygienic requirements is supplied in the room by means of large-surface air outlets near the floor. What we could call a "lake" of fresh air forms on the floor. The return air grilles are situated near the ceiling. The heat sources in the room, that is, occupants and heat-emitting equipment, produce a thermal convection in their immediate surroundings, transporting cool air from the floor to the ceiling. Only the immediate surroundings of the heat sources are supplied with fresh air. The ventilation effectiveness of the load zone is also excellent.

VENTILATION BY HEAT SOURCES

For a better description of the construction and effectiveness of ventilation by heat sources, the flow pattern shall first be compared with the known flow pattern of all-air systems.

Probably the most widespread form of air supply and resulting indoor airflow today in the air-conditioning of offices, administration buildings, etc., is the mixing flow ventilation system (Figure 3a). In this system, the supply air enters the room in the ceiling area at high impulse and quite high flow velocity (approximately 0.5 to more than 1 m/s [98 to more than 196 fpm]) and mixes very quickly with the indoor air. As a consequence of this intense mixing and balancing, the temperatures, humidities, and pollution concentrations in the room are largely constant. The local air velocities in the areas around occupants, however, are rather high (approximately 0.15 to 0.3 m/s [30 to 60 fpm]). Because of constantly changing flow directions and high degrees of turbulence, it is very difficult to master the airflow in a room that is extremely unstable. According to the latest findings by Fanger and Mayer (1988), these are precisely the reasons for the annoyances and high rates of complaint caused by drafts. Since not only the average flow velocity, air temperature, and temperature of the surrounding area are common criteria for comfort but also the time fluctuations of the airflow, changing flow directions and velocities exert a substantial influence on the comfort of occupants of an air-conditioned room.

A completely different type of room flow is a displacement flow (Figure 3b). In that case, the supply air enters the room through large-surface openings at low impulse (outlet velocity below 0.2 m/s [40 fpm]) near the floor and is drawn off at the ceiling. If the inlet temperature of the air so supplied is lower than the average indoor air temperature, the supply air will uniformly spread over the entire floor space and will then flow—evenly distributed over the entire room—from that area to the ceiling. The flow velocities then existing in the room are very low (below 0.1 m/s [20 fpm]), unstable, and can only be proved by smoke tests.

A heat source in the flow just described—for example, a person or a heat-emitting office machine—will change the flow pattern. Due to the thermal convection produced by the temperature difference between the warm surface of the heat source and the cooler supply air from the lake of fresh air, the person or machine transports a continuous airflow toward the ceiling. An airflow that is larger than the fresh air volume supplied for this person will result in the flow pattern typical of a heat-source ventilation system (according to Fitzner [1990]), as shown in Figures 3c and 4.

According to Fitzner (1989), the supply air volume that is transported from the lake of fresh air at the floor to the ceiling can be quantified exactly by means of various parameters:

$$V = 5 \times 10^{-3} \times Q^{1/3} \times (z + d)^{5/3} \quad \text{in m}^3/\text{s} \qquad (1)$$

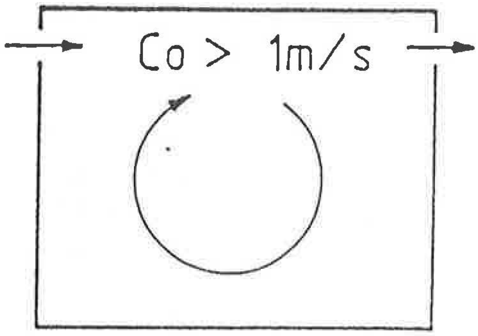


Figure 3a *Mixing flow.*

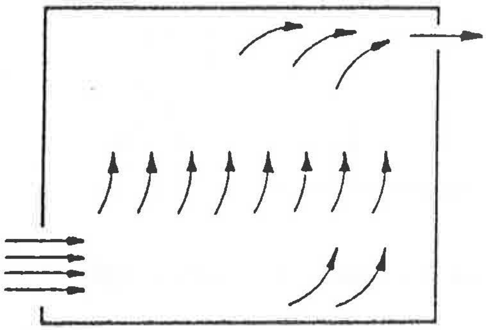


Figure 3b *Displacement flow.*

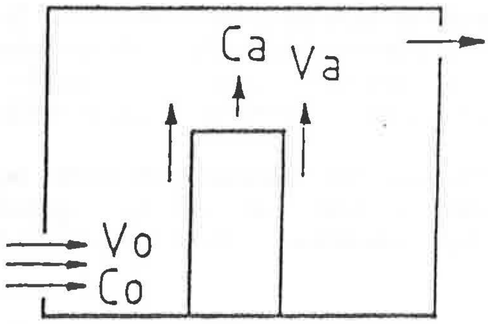


Figure 3c *Ventilation by heat sources.*

where

Q = convective heat dissipation by the heat source, W (approximately 50 % of the entire heat dissipation of the heat source)
z = height of the heat source above floor, m
d = diameter of the heat source, m.

Above the head of a sitting person, this convectively transported volume flow amounts to about 100 m^3/h (59 cfm) and, due to the inductive effect of this rising airflow on the stationary ambient indoor air, it will rise to about 200 m^3/h (98 cfm) at a height of two meters. The amount of this air volume flow transported, however, does not depend only on the parameters of Equation 1. There is another important parameter, namely, the vertical temperature gradient existing in the room. As shown by the

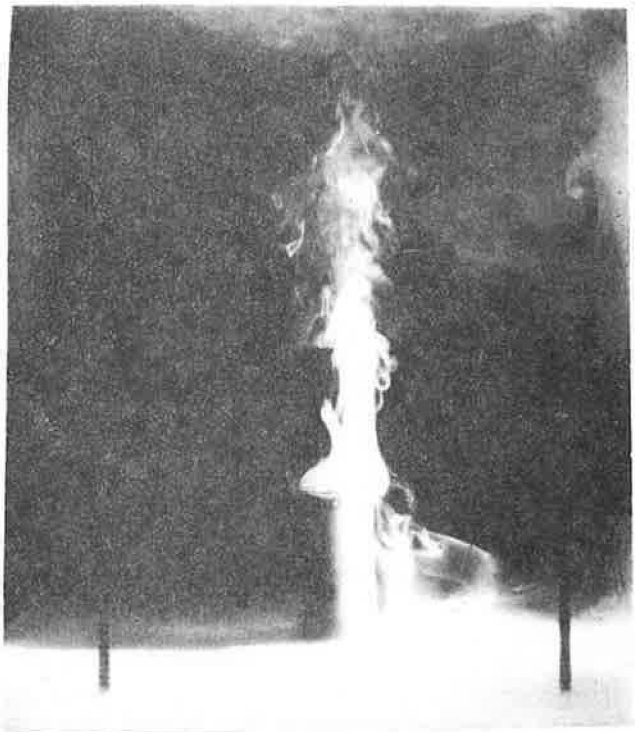


Figure 4 *Airflow in a ventilation system with heat sources during smoke test.*

diagram in Figure 5, this convectively transported air volume flow continuously decreases with rising vertical temperature gradient, and at a gradient of 1 K/m (0.3 °F/ft), it amounts to only roughly half the value of the case described above in which the gradient was 0 K/m. This is due to the reduced thermal effect of the convective flow.

Therefore, the following conditions will have to be fulfilled to enable a heat-source ventilation system to function:

- Large-surface fresh air supply at low impulse near the floor
- Removal of exhaust air in ceiling area
- Effective heat sources in the room transporting the air from the lake of fresh air on the floor to the ceiling.

In the layer of fresh cool air on the floor, a displacement flow prevails. Above this layer, there is a transition to a mixing flow caused by the inductive effect of the convective flow at the heat sources on the stationary indoor air. In the ceiling area, the convective flow is stopped again by the retaining effect of the ceiling, where a displacement flow adjusts itself again. In the entire room, flow velocities are very low and can in most cases only be proved by smoke tests. Except in the areas directly above the heat sources and near the return air grilles, they rarely exceed values above 0.1 m (0.32 ft), the degree of turbulence being lower than 20 %. Accor-

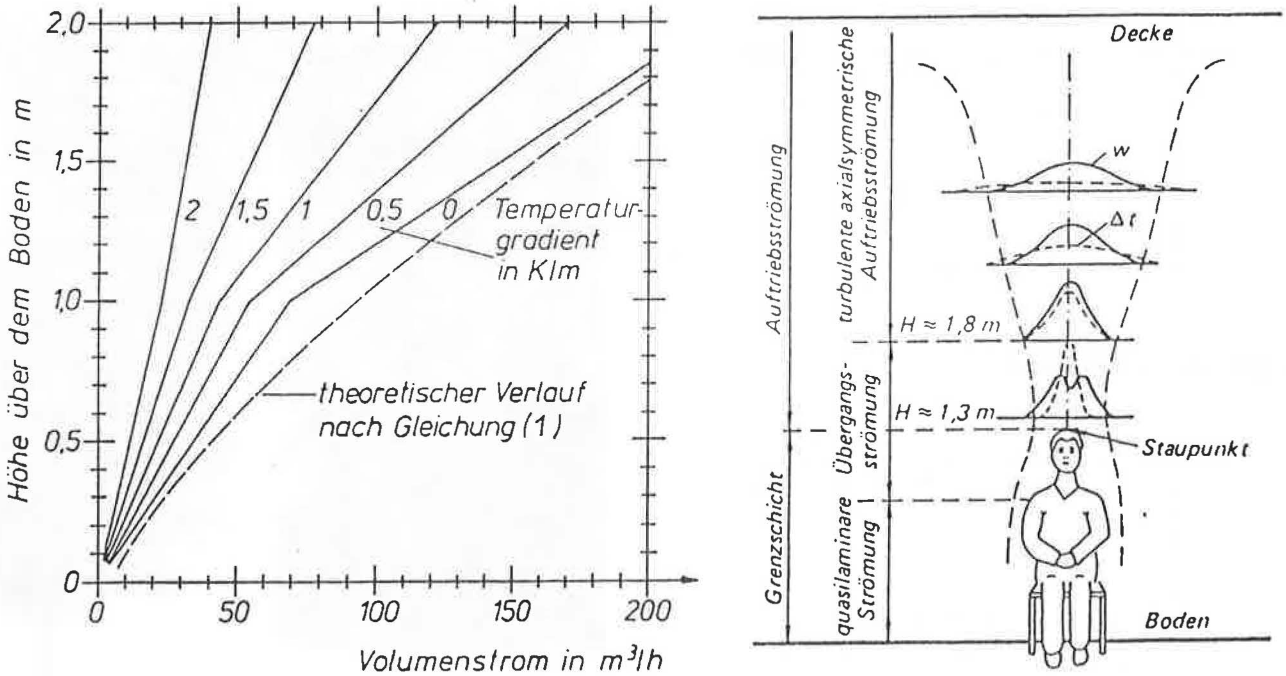


Figure 5 ***Vertical air volume flow induced by a person in a heat-source ventilation system at different temperature gradients in a room (according to Fitzner).***

ding to Fanger (1987), this is equivalent to a rate of dissatisfied persons of not more than roughly 5%. The draft annoyance still existing in many systems with turbulent mixing flow does not exist here. As each heat source in the room supplies its own air demands from the lake of fresh air on the floor (conditioned outdoor air), air quality and ventilation effectiveness as well are significantly better than with a system of mixing ventilation. Moreover, this flow pattern largely excludes a horizontal flow; that is, there will not be any overflow of smoke or odorous substances from one area of the room to another.

Despite all this, ventilation by heat sources is not the solution to all problems caused by the airflow pattern. The limits to thermal capacity are determined by

- the fresh air volume supplied (in general 2 to 3 h^{-1}),
- the supply air temperature, and
- the maximum admissible vertical temperature gradient.

To avoid draft in the "ankle area" (the area around the ankles of the occupants), the temperature of the supply air should not fall below 18 to 19°C (64 to 66°F). Another limit is set by the vertical temperature rise in the occupied area, which, according to Esdorn and Külpmann (1989), should not exceed 2 to 2.5 K at heights between 0.1 and 1.1 m (0.32 and 3.6 ft) to maintain comfort. In rooms in which high thermal loads have thus to be eliminated by small air volumes, pure ventilation by heat sources will fail. The maximum load that can be eliminated by such a ventilation system amounts to about 30 to 40 W/m² (10 to 13 Btu/h·ft²) if the above-mentioned marginal conditions are given. If the thermal load of the room exceeds this value and the supply air volume is not raised considerably above the value required for hygienic reasons, other air-conditioning components will have to be added.

One possibility of increasing the thermal capacity of the heat-source ventilation system is the application of a so-called "induction unit." As shown in Figure 6, part of

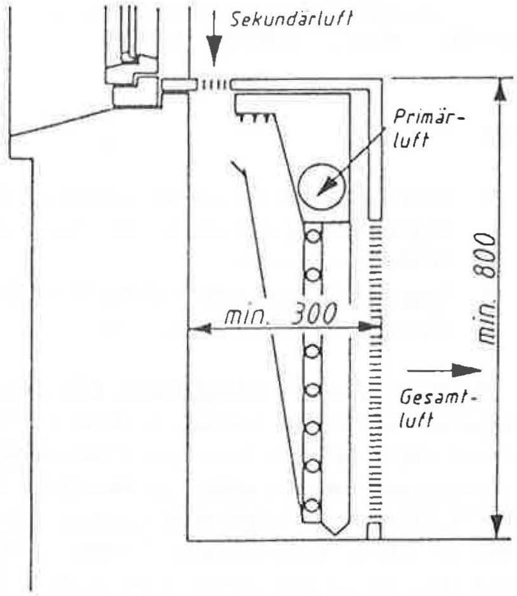


Figure 6 ***Design of an induction unit for a system of heat-source ventilation (according to Laux).***

the primary air V_P (pure outdoor air) in this system transports two to three parts of secondary air V_S (indoor air) through the integrated heat exchanger, through which chilled water is flowing, and thus produces a total of three to four parts of cool supply air V. Subsequently, this supply air volume will leave the room at an outlet velocity of about 2 m/s (390 cfm) via an outlet, especially designed for air by heat sources, which is equipped with a rectifier; finally, the air will return to the room at a low rate of turbulence near the floor. By applying this air-water system, Laux (1989) points out that not only an increase in cooling capacity to about 60 to 70 W/m^2 (19 to 22 Btu/h·ft^2) is achieved but also a further stabilization of the airflow in the room, which, due to the low flow velocities in heat-source ventilation, is naturally susceptible to disturbing factors, such as persons moving around.

An air-conditioning component that, in combination with the heat-source ventilation system, is particularly suitable for eliminating high thermal loads and concurrently fulfilling comfort requirements, is the cold ceiling. The design and operating mode of this component is described next.

COLD CEILING—DECOUPLING OF AIRFLOW AND LOAD REMOVAL

In order to maintain clearly defined conditions in a room with regard to temperature, humidity, or concentration of odorous substances, the respective room loads will have to be continuously eliminated. Most of these loads can be removed by convection—that is, by air—by supplying chilled supply air at low load concentrations and removing the load by the exhaust air. Sensible room loads, in particular heat and cold, may also be eliminated directly via heat exchange surfaces installed in the room. These take effect partially through convection, partially through radiant exchange. The best known example of this type of load removal is the widespread radiator or floor heating system.

It is exactly at this point where the cold ceiling comes in: A great number of laboratory measurements and practical applications proved that cooling loads can also be eliminated by static cooling surfaces in the ceiling area, maintaining very comfortable thermal conditions in the occupied area.

The operating principle of a cold ceiling is very simple: Depending on the room cooling load expected, parts of the ceiling—or alternatively, the entire ceiling—are constructed as a large-surface heat exchanger through which chilled water flows. Various manufacturers offer different systems for this purpose, three of which are described below.

System 1

System 1 uses cooling mats consisting of numerous thin synthetic tubes (diameter: approximately 2 mm [0.0066 ft]) that are arranged in parallel (Figure 7). The distance between the individual small tubes through which chilled water flows is so small that a homogenous temperature is produced on the bottom side of the ceiling. As shown by Figures 8a and 8b, the cooling mats in this system can be incorporated in the ceiling in two different ways.

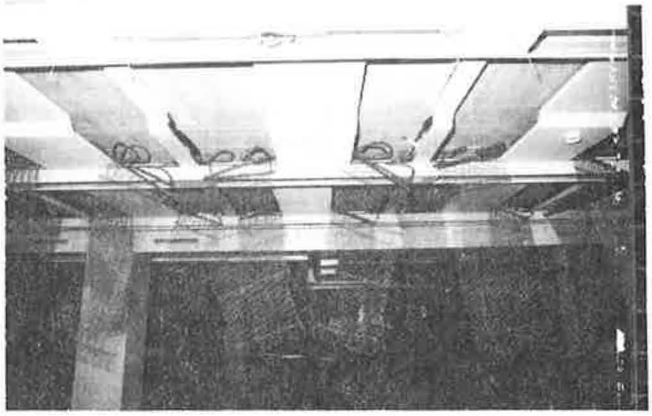


Figure 7 *Cold ceiling according to the KaRo principle (according to Herbst).*

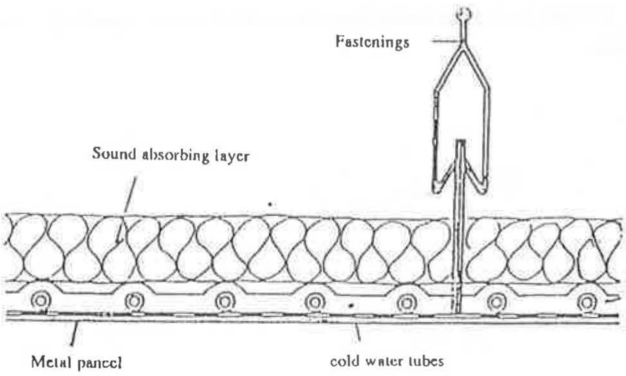


Figure 8a *Metal ceiling panels with cooling mat and insulation (Nickel).*

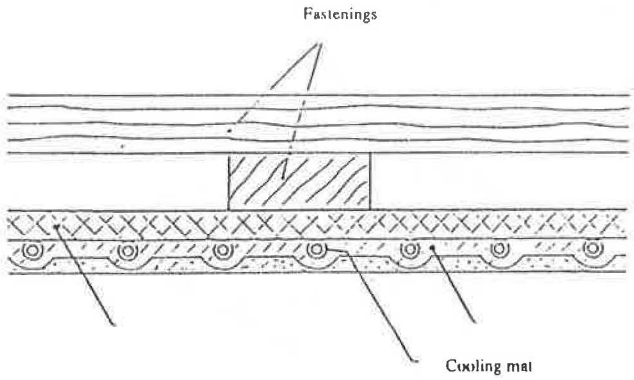


Figure 8b *Concealed cooling mat (Nickel).*

Figure 8a shows the cooling mats placed on suspended metal panel ceilings. A mineral wool layer installed above supplies both the heat insulation toward the hollow space of the ceiling and the sound insulation toward the room. Figure 8b shows the cooling mats placed directly below the ceiling and covered by a thin layer of plaster, the result being a smooth ceiling without unpleasant seams between the single ceiling elements.

System 2

System 2 (Figure 9) consists of a metal ceiling in which the cooling coils, through which chilled water flows, are in direct contact with a smooth-surfaced metal support construction. Due to the heat conduction between the cooling coils and the metal surface, uniform temperatures are produced over the entire surface of the suspended ceiling.

System 3

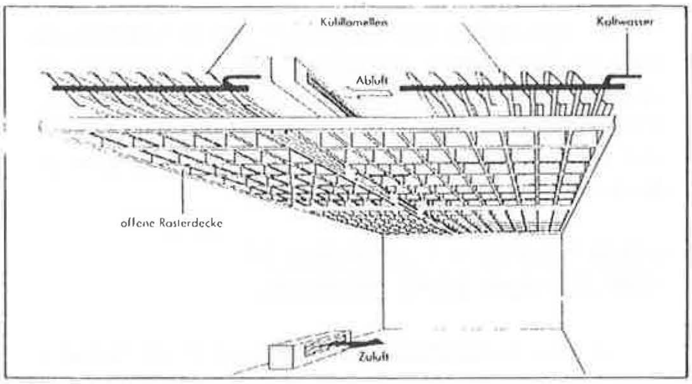
System 3 (Figure 10) is composed of several cooling elements. To enlarge the heat exchange surface, the copper tubes through which chilled water flows are fitted with heat-conducting aluminum fins at an angle of 45°. According to Sodec and Reichel (1990), this achieves an optimum relation between convective and radiant exchanges.

There is a radiant exchange between the cold ceiling of the room and the warmer enclosing surfaces of the room as well as the heat sources present in the room. According to the equation

$$Q_s = A \times e_{1.2} \times C_s ((T_w/100)^4 - (T_D/100)^4), \quad (2)$$

where

Q_s = radiant heat (W) removed from the room
A = surface (m^2)
$e_{1.2}$ = radiation coefficient resulting from the surfaces exchanging radiation
C_s = radiation constant (= 5.67 W/(m^2 × K^4))



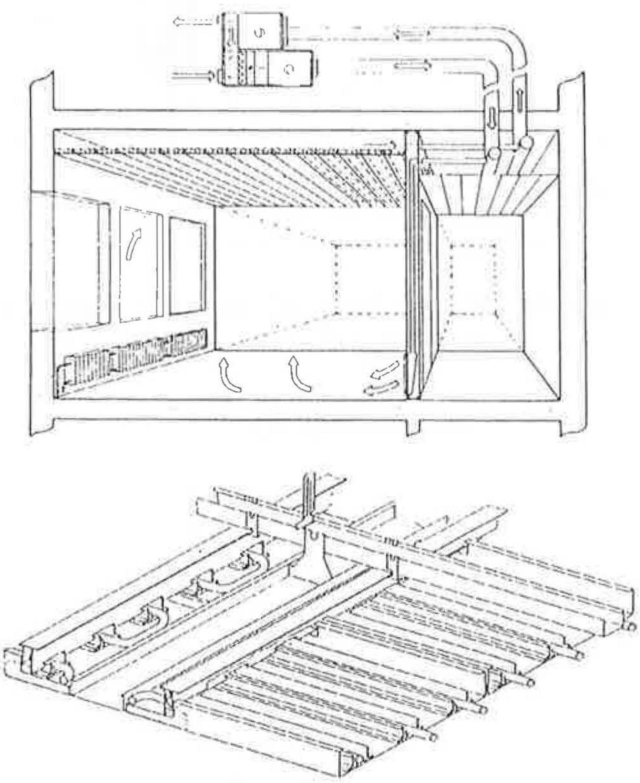


Figure 9 *Design of a cold ceiling system (according to Fläkt).*

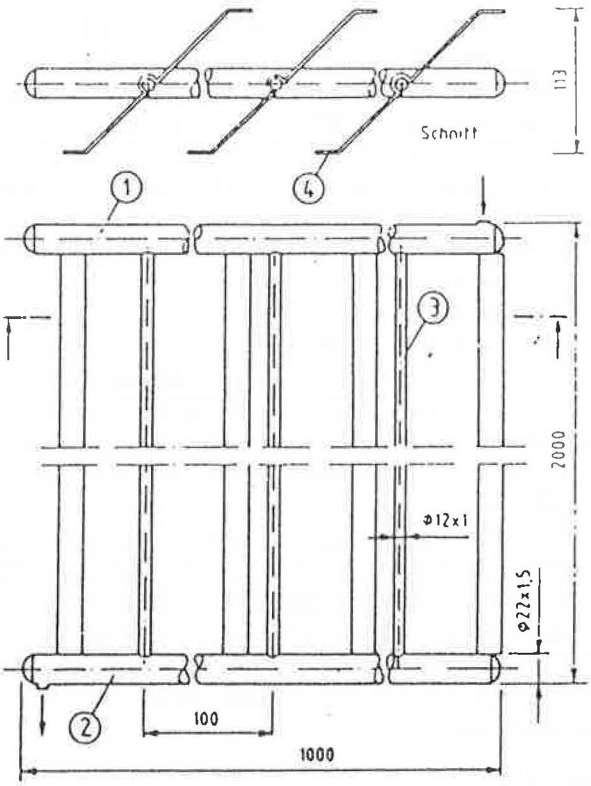


Figure 10 *Design of a cold ceiling system (according to Krantz).*

T_w = average surface temperatures of the heat sources (K)

T_D = average surface temperatures of the cold ceiling (K),

the ceiling eliminates a heat flux QS. According to Equation 2, this heat flux is continuously rising with an increasing temperature difference between the ceiling T_D and the T_w of heat sources and walls if the surfaces and radiation coefficient remain constant. According to Esdorn and Külpmann (1989), Equation 2 can be simplified for a rough estimation of the operating conditions of the cold ceiling and its surface-related capacity.

$$Q_{TOT} = (\alpha_{TOT} \times (T_{1.1m} - T_D) \text{ in W/m}^2. \qquad (3)$$

Depending on the construction of the ceiling, one can expect $\alpha_{tot} = \alpha_{conv} + \alpha_{str}$ to be roughly 9 to 12 W/(m²·K). The value t1.1m is the average room temperature at a height of 1.1 m, and T_D is the average temperature of the cold ceiling surface. Hence, the cooling capacity of the cold ceiling depends directly on the temperature difference between the ceiling surface and the occupied area. But, similar to the heat-source ventilation system, there are thermodynamic marginal conditions and comfort aspects that set limits to this cold ceiling system:

1. To avoid condensate forming in the area of the cold ceiling, the surface temperature of the cold ceiling must never come near the dew point. For this purpose, the entering temperature of the chilled-water flow in most cases is controlled to a value that is at least 2 K above the actual dew point of the indoor and outdoor air. In particular, on warm summer days with comparatively high outdoor air humidity, the windows in rooms with such an air-conditioning system must be kept shut and/or contacts must be provided that stop the chilled-water flow to the ceiling when a window is opened. Special window switches are used for this purpose.
2. Also for reasons of comfort, the surface temperature of the cold ceiling must not be too cold. As already demonstrated by Fanger (1988), the asymmetry of a person's radiant heat emission to surfaces at varying temperatures is an important factor in comfort. Various investigations, however, proved that the normal surface temperatures of cold ceilings of about 18 to 20°C (64 to 68°F) result in an increase in comfort rather than in a decrease. At an asymmetric radiant temperature of 5 K, it was almost impossible to prove a rate of dissatisfaction. On the contrary, the old rule "Cool head and warm feet are comfortable" proved to be true. Moreover, investigations carried out separately by Gräff, Fitzner, and Esdorn (Stahl 1990) demonstated that the cooling capacity of a cold ceiling is affected about 40% to 45% by convection and 55% to 60% by radiation. This relation corresponds to the heat emitted by the usual heat sources of a room. A human being who feels at ease thus emits warmth in about the same relation.

Taking into account all these parameters, a cold ceiling is capable of eliminating a heat load of up to about 70 to 80 W/m² (22 to 25 Btu/h·ft²). However, the capacity data presently indicated by manufacturers for the different types of cold ceilings vary a great deal. Depending on the measuring system adopted, on the sensor and load locations in the room to be measured, and on marginal conditions, values indicated in the brochures vary between 50 and 150 W/m² (16 and 48 Btu/h·ft²). This was the reason why, about half a year ago, the German sellers of cold ceilings set up a study group to establish standardized measuring systems and common marginal conditions for the determination of cold ceiling capacity and incorporating these in guidelines.

SAVINGS IN FAN ENERGY WILL PAY

Various studies and recent publications have dealt with the economy of this new air-conditioning system compared to conventional systems. Both installation and operating costs have been considered.

Under the management of Professor Esdorn (Esdorn and Külpmann 1989), a German study demonstrated that at a room cooling load of between 20 and 60 W/m² (6 and 19 Btu/h·ft²) a system of "cold ceiling, static heating, and ventilation by heat sources" is capable of saving energy, namely, 20% to 50% compared to a variable-volume system) and 40% to 60% compared to a constant-volume system. These savings are mainly due to the significantly lower energy consumption of the air transporting fan.

Hönmann and Nüßle (1990) compared investment costs of cold ceilings combined with heat-source ventilation systems and those of a variable-volume system. As shown in Figure 11, the breakeven point is said to be at a room cooling load of about 35 to 50 W/m² (11 to 16 Btu/h·ft²). A similar result was achieved by the indepen-

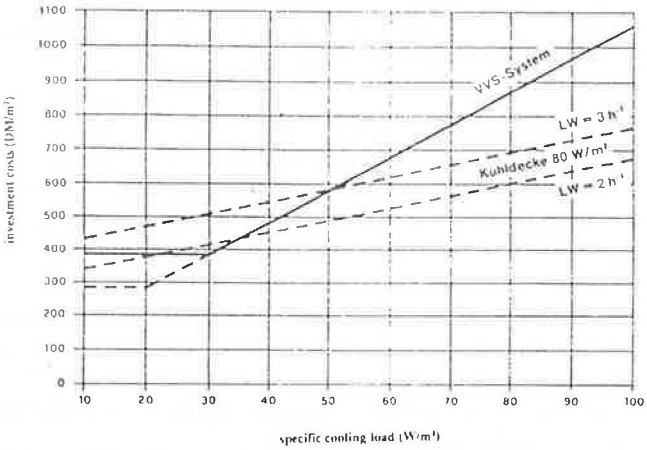


Figure 11 *Comparison of the investment costs of conventional all-air systems with those of a new cold ceiling system (according to Hönmann).*

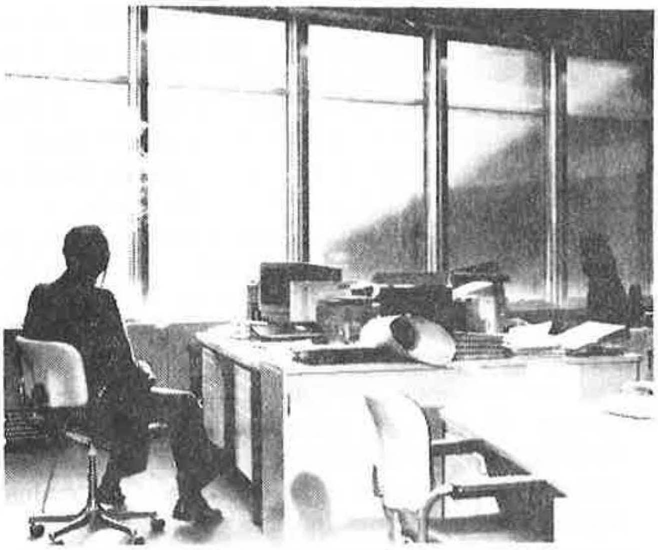


Figure 12 *Supply air inlet of a heat-source ventilation system in an office building.*

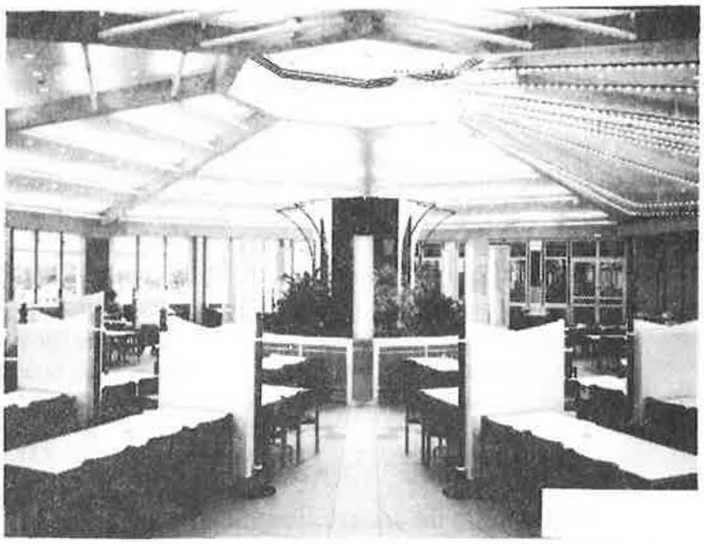


Figure 13 *Air outlets in heat-source ventilation systems at the parapet and the center of a hall with a diameter of 28 m (Fitzner).*

Figure 14 *Supply air inlet and airflow pattern in a heat-source ventilation system.*

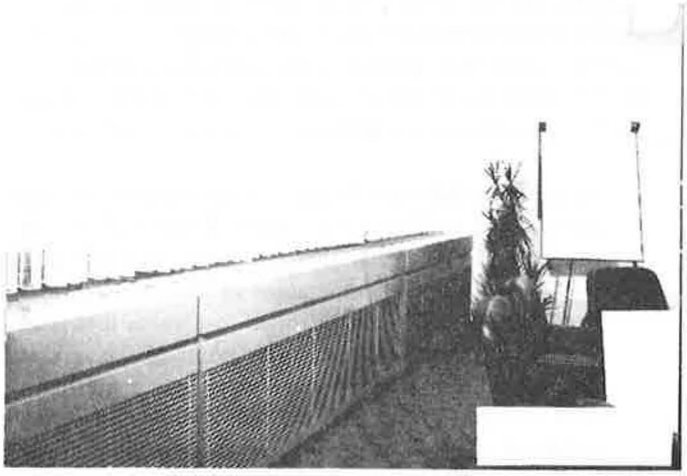


Figure 15 *Induction unit for a heat-source ventilation system in a conference room.*

dent planner Pfeil (Stahl 1990). He compared three different air-conditioning systems with a system of cold ceiling and heat-source ventilation, considered both investment and operating costs, and concluded that the system does pay from loads of about 55 W/m^2 (17 Btu/h·ft^2) onward.

A thorough study of the economy of the system was made by Sodec and Reichel (1990), who concluded that the total investment for a cold ceiling combined with a heat-source ventilation system amounts to 400 to 550 DM per m^2 floor area ($40 to $50/ft^2).

Following are two examples of existing systems that combine a cold ceiling with a heat-source ventilation system.

Example 1

Example 1 (shown in Figures 12 and 13) is the application of a heat-source ventilation system in an office building with a load of 40 W/m^2 (13 Btu/h·ft^2). The cold supply air enters the room through an air outlet at a height of approximately 0.5 m (1.6 ft), which fills the entire periphery of the hall. The center of this hall, which has a diameter of 28 m (92 ft), is also provided with air outlets. Figure 14 shows the smoke absorption of heat-source ventilation. The supply air flows along the floor to the center of the hall and drafts do not even occur in the immediate proximity of the supply air grilles.

Example 2

Example 2 refers to a modernization in which a previously installed high-pressure induction unit was replaced by a new induction unit for a heat-source ventilation system. As already described, the indoor air is drawn off at the upper edge of the parapet; primary and secondary air flows through the heat exchanger and is returned to the room near the floor, from where it is evenly distributed by heat sources (Figure 15, from Fitzner [1990]).

SUMMARY AND OUTLOOK

"The ventilation and air-conditioning industry should be careful not to call the cold ceiling without concurrent ventilation an acceptable air-conditioning system and offer it to customers: It is nothing but an integral component of an aggregate system that, without controlled ventilation to eliminate humidity and odorous substances, is doomed to failure." This is the résumé of Professor Steimle (Fitzner 1990) at the end of a symposium on cold ceiling and room ventilation in which 200 air-conditioning experts participated at Stuttgart in September 1990. The experts agreed that the cold ceiling is not the solution to all air-conditioning problems but that, if correctly applied and operated, it is capable of contributing greatly to an increase in thermal comfort in a room. With that, the supply air volume can be reduced to the flow necessary to eliminate odorous substances or humidity. Compared to conventional air-conditioning systems, where the cooling load must be eliminated by convection, this means a reduction of the supply air volume of approximately 60% to 80%. On one hand, this leads to considerable savings in fan energy and to a reduction of the peak power loads. On the other hand, the low airflow velocities of heat-source ventilation prevent drafts. The thermal capacity of the heat-source ventilation system amounts to about 30 to 40 W/m^2 (10 to 13 $Btu/h \cdot ft^2$) and that of the cold ceiling to about 40 to 80 W/m^2 (13 to 26 $Btu/h \cdot ft^2$). Summing up the figures for the two air-conditioning components, one gets an overall capacity of about 100 W/m^2 (32 $Btu/h \cdot ft^2$). Normally, this capacity suffices to ensure a removal of the load, even in modern office rooms, lounges, and assembly rooms with high internal heat sources. Should the internal thermal loads exceed 100 W/m^2 (32 $Btu/h \cdot ft^2$), the responsible people will have to rethink the overall planning of the room and/or building. In those cases, either the architectural planning is wrong or someone forgot to provide the windows with effective sun protection. Considering that this system also has advantages with regard to installation and operating costs over conventional air-conditioning systems in the case of room loads of approximately 50 W/m^2 (16 $Btu/h \cdot ft^2$) onward, it should only be a matter of time until this system prevails not only in the West European countries but also internationally.

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