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INFLUENCE OF VENTILATION SYSTEM ON THE PERFORMANCE OF COOLING CEILINGS: APPLICATION TO CHILLED BEAMS

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ABSTRACT

Cooling ceiling systems are controlling only the sensible heat balance of the rooms; they are always combined with a ventilation system foreseen to control indoor humidity and to cover air renewal requirements. Between the types of cooling ceiling in use, the passive chilled beams seem to be the most sensitive to ventilation air influence. In most of the cases, the ventilation outlets are located in the ceiling void, and consequently this generates a penalty on the beam cooling power. The work presented aims at estimating this influence, through results issued from experimental studies. The values obtained show that in usual cases, the beam cooling power can decrease down to 20-25% from its nominal value. Moreover, by using simple modeling assumptions, one should be able to make a preliminary evaluation of this effect. The results of this study are regarded as helpful for design of cooling systems.

KEYWORDS

Free convection, chilled beams, ventilation air, temperature difference, modeling, system performance.

INTRODUCTION

The passive chilled beam system is one of the most common cooling ceiling equipment in use. It is made of a set of horizontal finned water coils installed in the ceiling void of the room and connected in parallel. They are using mainly natural convection heat transfer; the radiation heat transfer is negligible, due to the location of the beam, above a false ceiling (figure 1). In comparison with radiant cooling ceilings, the difference in heat transfer is compensated thanks to the enhancement of external heat transfer area which can be up to 5 times bigger [ASHRAE 1992]. A very easy air circulation must be also maintained between the room and the ceiling void through the false ceiling.

Chilled beams are not reversible for heating regime like the radiant panel ceilings, because of the thermal gradient which nullifies the natural convection effect.

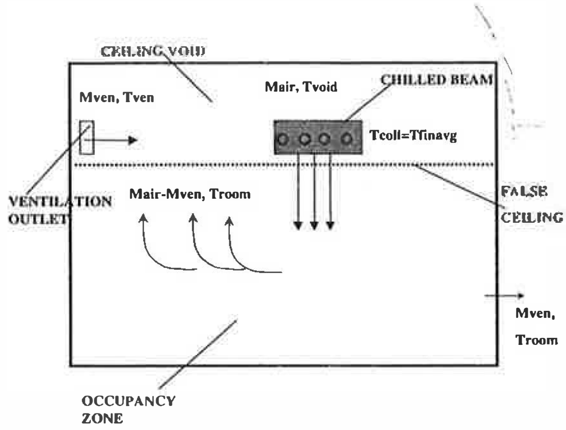

Figure 1. Schematic of a passive chilled beam system.

HEAT TRANSFER AND PERFORMANCE OF CHILLED BEAMS

The chilled beam system can be considered as a heat exchanger (the finned coil) supplied on one side by water and on other side by air aspirated in the ceiling void. In order to define an effective cooling capacity, the air temperature is not taken as reference, but replaced by the space average of globe temperatures measured in the occupancy zone. This temperature is the supply temperature of the beam on air side. A convenient approximation consists in considering a contact factor of 100%, i.e. in assuming that the exhaust air temperature is the same as the average temperature of the finned surface. This temperature can be determined in function of the fin effectiveness. Though, the so-defined cooling capacity is mostly affected by the temperature difference between water and room, by the fin efficiency and by the ventilation mode.
It is proposed here to neglect the water flow rate influence.
Consequently, for a given beam and room configuration (geometry and loads) working in given (nominal) temperature conditions, the performance will still depend on the ventilation mode.
The global heat transfer coefficient of the beam can be defined as :

$$AU = \frac{\dot{Q}_{cool}}{\Delta t_{LOG}} \tag{1}$$

where $\dot{Q}_{cool}$ is the cooling capacity of the system
The logarithmic mean temperature difference is defined as :

$$\Delta t_{LOG} = \frac{(t_{room} - t_{wex}) - (t_{finavg} - t_{wsu})}{ln\left(\frac{(t_{room} - t_{wex})}{(t_{finavg} - t_{wsu})}\right)} \tag{2}$$

Where :
t_{wsu} and t_{wex} - are the water supply and exhaust temperatures.
t_{finavg} - is the average temperature of the fins.
For any combination of the tested conditions the cooling capacity can be calculated using the heat transfer effectiveness :

$$\dot{Q}_{cool} = \varepsilon \dot{M}_{air} c_{pa} (t_{room} - t_{wsu}) \tag{3}$$

where : $$\varepsilon = \frac{\left(t_{room} - t_{finavg}\right)}{\left(t_{room} - t_{wm}\right)} \tag{4}$$

The fin effectiveness can be defined as :

$$\Phi = \frac{t_{room} - t_{finavg}}{t_{room} - t_{fino}} \tag{5}$$

with t_{fino} is the temperature at fins root (pipe contact temperature) and t_{finavg} is the average temperature of fin surface. According to heat transfer theory [Kreith F.1976] it is depending on fin geometry, material (conductivity), and on global heat transfer coefficient between fin and room air .

$$\Phi = \frac{tanh\left(L_{fin}\sqrt{\frac{2h_{air}}{k_{fin}\,b_{fin}}}\right)}{L_{fin}\sqrt{\frac{2h_{air}}{k_{fin}\,b_{fin}}}} \tag{6}$$

L_{fin} , b_{fin} are fin length and width and h_{air} is the heat transfer coefficient between fin and air. The global heat transfer coefficient of the beam can be defined by associating, three thermal resistances in series:

$$AU = \frac{1}{R_w + R_m + R_a} \tag{6a}$$

These resistances correspond to the convection heat transfers on water and airsides and to metal conduction respectively (the last one includes the effect of fin effectiveness).

The resistance on the airside is the most important term in the definition of the heat transfer, as demonstrated in previous work [M.Bravo et al. 1995]. Consequently, the following empirical equation can be used to describe the relationship between cooling power and air-water mean temperature difference:

$$\dot{Q}_{cool} = K\,\Delta t_{LOG}^{1+n} \tag{7}$$

The two constants K and n of this equation are related to coil geometry and to heat transfer mode respectively.

For horizontal finned tubes the free convection heat transfer coefficient is defined as [Giblin R. 1974]:

$$h = h_{conv} = K_2 \Delta t_{LOG}^{n_2} \tag{8}$$

In pure free convection mode, the exponent "n" might float somewhere between 0.25 and 0.5. Except for metal and water side resistances, he main factor which can "damp" the exponent is the effect of a lower temperature in the ceiling void than in the room. This can be due to the ventilation air blown in the void. The results obtained for cooling capacity confirm these assumptions as "n" value found are in the range 0.15 to 0.3 [A.Ternoveanu et al 1999].

EFFECTIVE INFLUENCE OF THE VENTILATION AIR FLOW

As chilled beam systems are mounted above perforate plate ceilings, the ventilation supply openings are often located in the ceiling void. One must take into account that when using cooling ceilings, the ventilation contribution is limited to cover only air renewal requirements, which corresponds to a small flow rate. However, this small flow rate can modify the temperature in the proximity of the beam and modify its performance, by cooling the ceiling void.
Experimental results on chilled beams exhibit a decrease of the cooling power related to the mentioned cooling effect.
For a chilled beam the cooling power is given by eqn. (7) above. We assume the ventilation flow rate is essentially acting on the effective temperature difference between the two fluids, and not on the shape of the heat transfer law, as the air flow is too small to generate induction effect on the beam.
In that case, the real effective cooling capacity is :

$$\dot{Q}^*_{cool} = K \, \Delta t^{*\,1+n}_{LOG} \tag{9}$$

The mean temperature difference is defined this time by using the ceiling void temperature :

$$\Delta t^*_{LOG} = \frac{(t_{void} - t_{wex}) - (t_{finavg} - t_{wsu})}{ln\left(\frac{(t_{void} - t_{wex})}{(t_{finavg} - t_{wsu})}\right)} \tag{10}$$

In order to determine the average ceiling void temperature one should make the following assumptions :
- Perfect mixing between ventilation air and room air aspirated in the void ;
- Negligible heat losses between ceiling void and adjacent rooms.

Thus the air flow through the chilled beam is given by eqns. (3) and (4) :

$$\dot{M}_{air} = \frac{\dot{Q}^*_{cool}}{c_{pa}(t_{void} - t_{finavg})} \tag{11}$$

According to the above assumptions, the temperature of the void can be determined as (Figure 1) :

$$t_{void} = \frac{\dot{M}_{ven}\, t_{ven} + (\dot{M}_{air} - \dot{M}_{ven})t_{room}}{\dot{M}_{air}} \tag{12}$$

The uncertainty of this evaluation is acceptable, as it appears from figure 2 where are plotted measured cooling capacities on water side against calculated cooling capacities from eqn. (9).
The results above are gathered from tests performed on three different chilled beams systems.
In order to extrapolate the ventilation influence for all types of chilled beams one should define a set of non-dimensional terms:

- The ratio between the cooling capacity of the ventilation air and ceiling system :

$$\omega = \frac{\dot{Q}_{ven}}{\dot{Q}^*_{cool}} \tag{13}$$

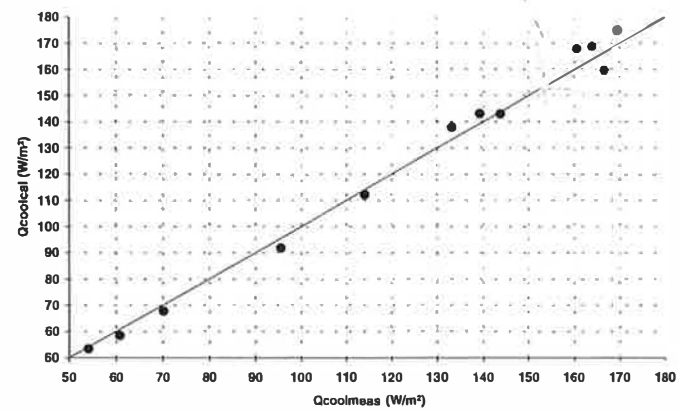

Figure 2. Capacities determined using ceiling void temperature against measured values.

- The ratios between air-water mean temperature difference, capacity, AU and effectiveness, with and without ventilation:

$$\delta = \frac{\Delta t^*_{LOG}}{\Delta t_{LOG}} \; ; \quad \zeta = \frac{\dot{Q}^*_{cool}}{\dot{Q}_{cool}} \; ; \alpha = \frac{AU^*}{AU} \; ; \beta = \frac{\varepsilon^*}{\varepsilon} \qquad (14\text{-}17)$$

The diagrams of Figure 3 show how the factors of temperature difference (δ) and of cooling capacity (ζ) may vary against the ventilation cooling factor ω. It appears that the penalty on the cooling capacity is almost the same as the ventilation cooling capacity itself. This means that it is useless to blow the air at a temperatures lower than room average temperature. The ventilation flow rate is only depending on the number of occupants.

In the case of a small office room, the penalty may reach 40 % of the nominal capacity.

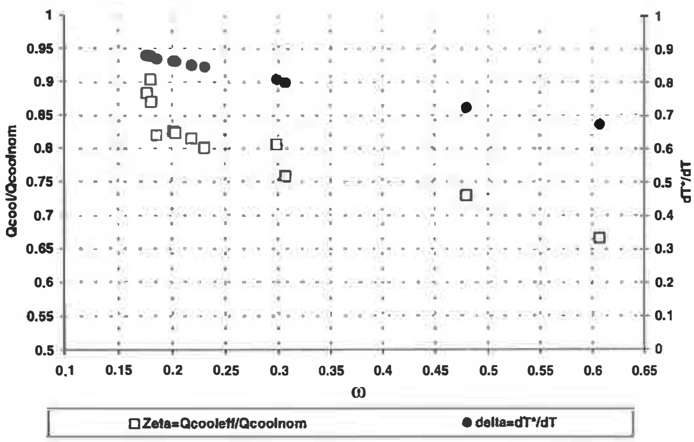

Figure 3. Mean temperature difference and cooling capacity ratios against ventilation energy flow ratio.

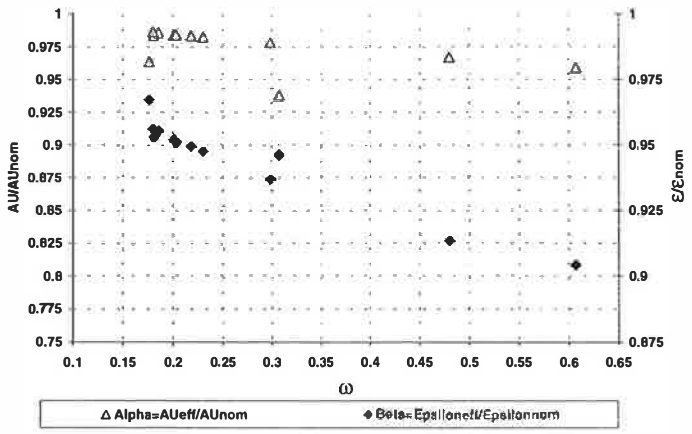

Figure 4. AU and effectiveness ratios against ventilation energy flow ratio.

The figure 4 shows the slope of AU (α) and effectiveness (β) ratios against ω. The decrease is much smaller than for cooling power as the global heat transfer coefficient is less affected by the temperature difference variation. The dispersion of certain values for α is due to the exponent "n" in the equation (7) and (9) which varies slightly for the different types of chilled beam system studied.

CONCLUSIONS

Introducing ventilation air in the ceiling void provides a penalty on the chilled beams performance, as the system is working at lower temperature difference. Consequently, for the same room and water supply average temperatures the chilled beam will provide less cooling power. The penalty is proportional to the ventilation energy flow rate, and may reach up to 40% of the total cooling capacity of the beam.

This influence of ventilation air should be taken into account during the design phase when choosing the optimal compromise between the cooling capacity needed and ventilation parameters as: outlets location, air flow rate and temperature.

Two alternative solutions should be considered:

1) To provide the air directly to the occupancy zone, but with some risk of draught;

2) To use adiabatic air drying and supply the air directly inside the ceiling void without any preliminary cooling (this introduces a supplement of sensible load in the ceiling void, which is easily compensated by the chilled beam.

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