NUMERICAL EVALUATION ON FLOOR COOLING CAPACITY IN AN AIRPORT

Bing Yu[1,3], J.P. Ruchti[1] and P.G. Luscuere[1,2]

[1]Division Building Services, Royal Haskoning Group, Postbox 305
6500 AH, Nijmegen, the Netherlands
[2]Delft University of Technology, Postbox 5043, 2600 GA, Delft, the Netherlands
[3]Tianjin University, School of Environment Science and Technology, 92 Weijin Road
300072 Tianjin, China

ABSTRACT

In the modern architectural design, large space open area is widely used, for example, atrium, airport, open office, etc. Radiation heat transfer provides better indoor thermal comfort and makes it possible to apply high temperature cooling and low temperature heating. This will lead to high energy efficiency. Water carrying radiation systems, e.g. cooling ceiling and floor heating system, become an applicable solution for a good indoor climate design.

The floor cooling is less effective than floor heating and can provide usually limited cooling capacity. In order to make a good design of floor cooling system, a theoretic method is described and the numerical evaluation of the cooling capacity is done. The influence parameters for the capacity, for example the distance between pipes, the length of one loop, the water flow rate and the supply water temperature, etc, have been analysed. It can be used for the correct estimation during the design phase.

KEYWORDS

Floor cooling, Capacity, Numerical evaluation, Airport

INTRODUCTION

Large space with open area is widely used in the modern architectural design, to realize such as atrium and halls. Especially, there is a lot of large open space in the airport. In the past, a lower ceiling will be placed to hide all the technical installations, such as pipes, ducts, cables, etc. Nowadays, steel structure is increasing popular. This means a very high roof in the space. Traditional conditioning method with hundred percent of air is not efficient in this situation.

Floor heating works perfect for the large floor area in the airport. Radiation heat transfer provides better indoor thermal comfort. In order to make use of the existed floor heating system, the concept of floor cooling is logically raised. It is supposed to work with supplied chilled water. Some of the airports, such as Suvarnabhumi airport, the new Bangkok International Airport, Thailand and the International Airport of Jeddah, Saudi Arabia applied floor cooling system to condition the indoor climate.

A previous research (Nisson, 1997) shows radiant floor piping can be used to cool a house, but it is only appropriate for dry climates. The floor temperature is held at 20^{o}C by using either a small cooling machine (chiller) connected to the floor piping or the steady 13^{o}C temperature of the ground by means of an earth loop. In most climates, the cool floor can be used to supplement or replace standard ducted air systems. However, in humid climates, problems with over-cooling the floor could lead to wet slippery surfaces and fungus growth.

Argiriou et al (2005) applied TRNSYS program to study the performance of a building with Absorption Heat Pump with floor heating/cooling system. The research was focussed on the application as well and used standard model of floor cooling which is integrated in TRNSYS. A validation on the floor cooling model with test chamber was done and a conclusion was drawn that the TYPE 160 module that was used in order to assess the performance of the total system seems to have systematically underestimated the particular experimental results.

Another researches look at the application of floor cooling system in the residential building as well (Hamada, et al. 2001, Lim, et al. 2003, Lim, et al. 2006).

Normally, floor heating will generate a good temperature gradient vertically because the warmer air will flow upwards. In certain application, floor will be used for both heating and cooling purpose due to the large space, e.g. at airport. The floor cooling will less effective than floor heating and can provide usually limited cooling capacity. This is due to the cold air will stay lower and close to the floor and therefore, the convective heat transfer is weaker than floor heating. This can be proved from the data of heat transfer coefficient for horizontal plate as well (ASHRAE handbook, 2005).

In this study, the authors build up the physical and mathematical model for the hall of Eindhoven airport. Based on the conservation equations, a simulation model is built with Matlab. The influences with the total length, distance between rows, supply water temperature, thermal radiation and water flow rate are analyzed by simulation.

A preliminary design proposal with a floor coling system is chosen as the reference. This has been evaluated and better solution is presented.

MODELING

A typical hydronic radiant floor system is usually placing the water pipe beneath the finished floor or screed floor. The pipe layer is shown in the Figure 1.

Figure 1. Example layout of hydronic radiant floor system (picture from Econosto)

For the application at Eindhoven airport, a preliminary concept is proposed with the section view as Figure 2. This is a system which the pipes are placed above the prefab board and then filling sand concrete above them.

The water pipe is installed inside the concrete layer of 80mm thickness. The centre of the pipe is (47+12) = 59mm below the floor.

Outside long diameter of the pipe is 24mm. The inner diameter is 19mm.

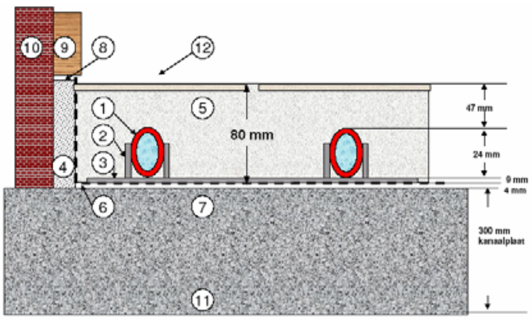


Figure 2. Floor cooling system

The heat transfer process occurs with the floor cooling system will involve several aspects: heat conductivity within the concrete floor; heat convection with the surrounding air; radiation from the floor to the other surfaces; and solar load falling on the floor surface.

The general scheme of the floor piping system is shown as Figure 3. The pipes are placed under the concrete layer with certain distance between different lines.

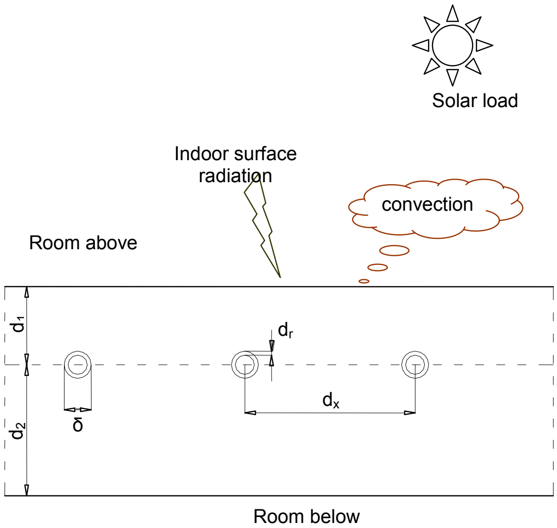
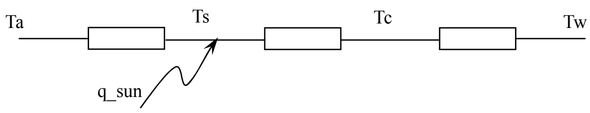


Figure 3. Scheme of the floor piping system with related to the surrounding

Energy conservation:



$$\dot{q}_c = \alpha_t (T_a - T_s) + \dot{q}_{sun} = U_t (T_s - \overline{T}_w) \quad (1)$$

$$\alpha_t = 0.7(T_a - T_s)^{0.33} + 4\sigma\varepsilon C_r (T_s + 273.15)^3 \quad (2)$$

$$U_t = \left(\frac{d_1}{\lambda_b} + R_w + R_r + R_z + R_x\right)^{-1} \quad (3)$$

$$T_s - \overline{T}_w = \frac{T_{wo} - T_{wi}}{\ln\left(\dfrac{T_s - T_{wi}}{T_s - T_{wo}}\right)} \quad (4)$$

$$T_{wo} = T_{wi} + (T_s - T_{wi})(1 - e^{-\frac{U_t A}{\dot{m} c_p}}) \quad (5)$$

Cr is the correction factor to correct the influence of the surrounding objects. In this study, Cr is chosen 0.85. If much furniture or fixed structure with surfaces is placed nearby the floor, the surface temperature will have big deviation from the air temperature and the Cr will be decreased. If large atrium is applied, Cr should be increased.

Total heat transfer coefficient analysis on distributed pipes

It is assumed that the distributed pipes will be simplified as a continuous functioning (concrete core) layer.

According to the theory by Koschenz and Lehmann (2000), the total heat resistance from the water supply temperature to the concrete core layer includes four items:

- Rx is the heat resistance which is supposed the whole plane of the concrete core is the same as the surface temperature of the register
- Rw is the heat resistance from the water to the inner surface of the register
- Rr is the heat resistance due to the conductivity of the wall of register
- Rz is the heat resistance due to the inconsistent temperature of the water in the register

$$R_t = R_x + R_w + R_r + R_z \tag{6}$$

Where

$$R_x = \frac{d_x \ln\left(\frac{d_x}{\pi\delta}\right)}{2\pi\lambda_b} \tag{7}$$

This equation is a simplified version when $d_1/d_x > 0.3, d_2/d_x > 0.3$ and $\delta/d_x < 0.2$. For most situation, these conditions are met.

$$R_w = \frac{d_x}{\alpha_w(\delta - 2d_r)\pi} \tag{8}$$

$$R_r = \frac{d_x \ln\left(\frac{\delta}{\delta - 2d_r}\right)}{2\lambda_r \pi} \tag{9}$$

$$R_z = \frac{1}{2\dot{m}_{sp} c} \tag{10}$$

Heat transfer coefficient

[ASHRAE Handbook, 2005 Fundamentals] Convection from horizontal plates facing downward when heated (or upward when cooled) is a special case. Because the hot air is above the colder air, theoretically no convection should occur. Some convection is caused, however, by secondary influences such as temperature differences on the edges of the plate. As an approximation, a coefficient of somewhat less than half the coefficient for a heated horizontal plate facing upward can be used.

For large space floor cooling,

$$h = 0.7(\Delta t)^{0.33} \tag{11}$$

For force convection,

$$Nu = \frac{hl}{\lambda} = 0.037\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.6} \tag{12}$$

SIMULATION CONDITIONS

The room temperature is 25°C and outside temperature is 30°C. Water supply temperature is 16°C.

The required cooling capacity of the floor pipe system is 25W/m^2.

A preliminary design has been proposed as:

- Length of one pipe: 180m
- The distance between center of the neighbor pipes: 30cm
- Water flow rate: 194 l/h

ANALYSIS AND DISCUSSION

Evaluation of the preliminary design

According to the information above, the following results will be achieved.

For the condition of no movement (no passenger), the natural convection will be applied based on the equation (1). The released cooling capacity is approximate 17W/m^2 and the return water temperature is about 20°C.

For the condition of 0.25m/s movement (some passengers walk relaxedly inside the airport), the released cooling capacity is about 18W/m^2 and the return water temperature is about 20.2°C;

For the condition of 0.5m/s movement (passengers keep moving), the released cooling capacity is about

19W/m^2 and the return water temperature is about 20.5^{o}C.

Possible adaptations

Several adaptations can be chosen to improve the cooling capacity released from the floor piping system.

- Modification of the length of one loop;
- Modification of the water flow rate;
- Modification of the distance between the center of neighbor pipes;
- Change of the supply water temperature.

The following figures show the results with one variable and the rest keep the same as the preliminary design. Every adaptation will show for three type of movement above the floor.

Influence of length

The length of one pipe will influence the cooling capacity from the hydronic radiant floor system. The longer the length of one pipe is, the higher the water return temperature is and therefore, the mean temperature difference between air and water is smaller. It is clearly shown in the Figure 4 and 5 for the change of cooling capacity and water return temperature.

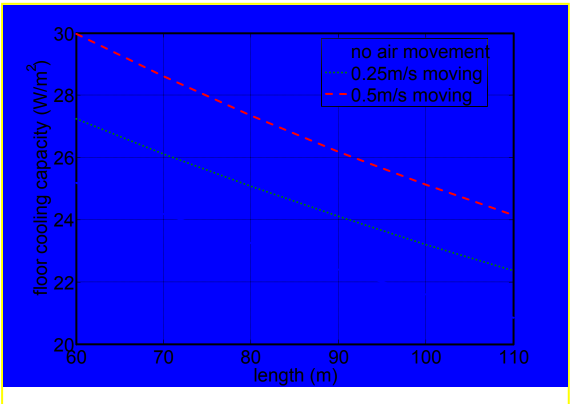


Figure 4. Variation of cooling capacity for different length of one pipe

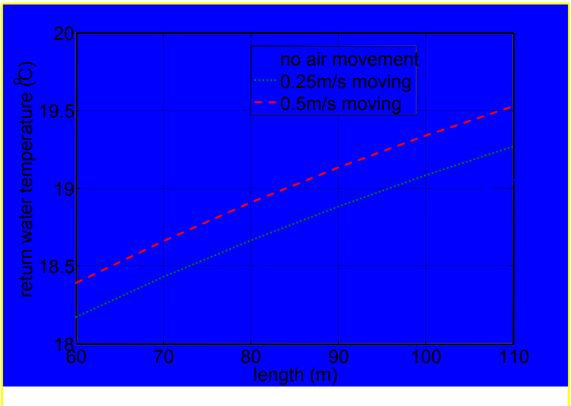


Figure 5. Variation of return water temperature for different length of one pipe

From the above shown results, it is clear that the length of the pipe should be decreased to 60 ~ 100m to get a 25W/m^2 cooling capacity for different movement condition.

Influence of water flow rate

The water flow rate will influence both the thermal performance and the hydraulic system. Higher water flow rate will lead to a higher cooling capacity. This can be found from the result of Figure 6. However, attention should be paid for a higher water flow rate because this might require a too large pump and therefore, it is not cost effective.

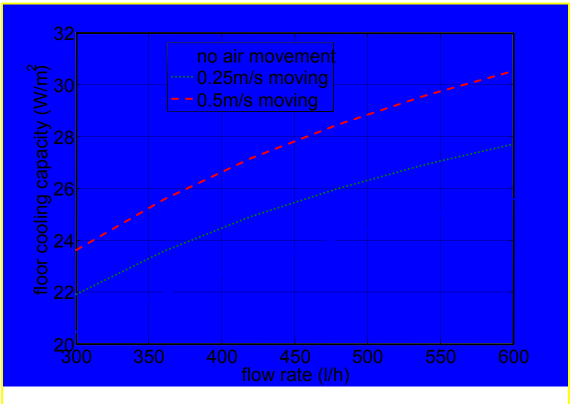


Figure 6. Variation of cooling capacity for different water flow rate

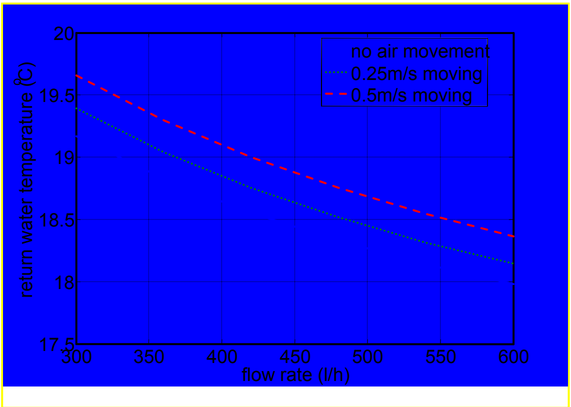


Figure 7. Variation of water return temperature for different water flow rate

From the above shown results, it is clear that the water flow rate should be increased to 540 ~ 340l/h to get a 25W/m^2 cooling capacity for different movement condition. This is about 1.7 ~ 3 times of the preliminary design.

Influence of distance between neighbor pipes

The distributed characteristic of the pipes will generate non-homogeneous temperature distribution horizontally. Figure 8 shows a result from CFD (Computational Fluid Dynamics) on one section along the pipe of floor cooling system. It can be clearly seen that the temperature increases circularly around the pipe and interferes with each other. If the distance between pipes changes, the homogeneous of the temperature at a horizontal level changes as well.

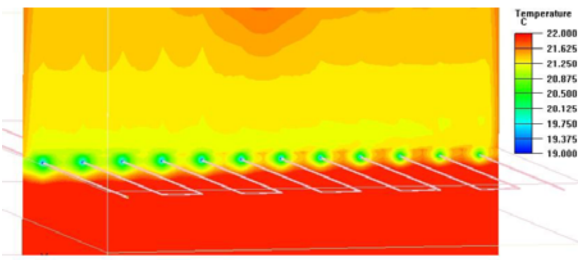


Figure 8. Section view of the temperature distribution along the pipe of floor cooling (an example)

The h.o.h (heart of heart) represents the distance between the centers of the neighbor pipes. It can be seen that the cooling capacity increases when the h.o.h decreases. And this will lead to the decrease of the water return temperature (Figure 10).

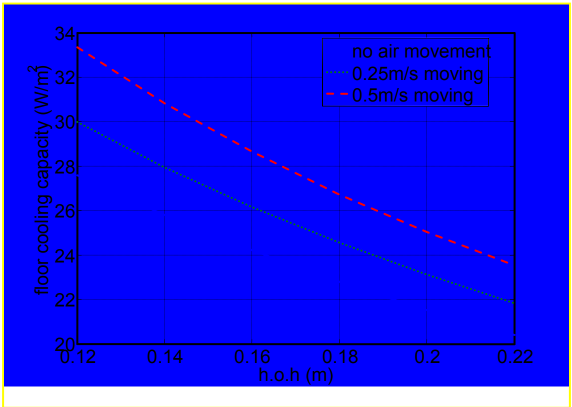


Figure 9. Variation of cooling capacity for different distance between neighbor pipes

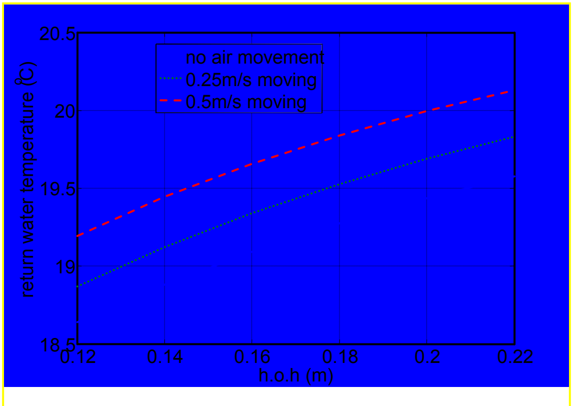


Figure 10. Variation of water return temperature for different distance between neighbor pipes

From the above shown results, it is clear that the distance between the pipes should be decreased to 0.15 ~ 0.2m to get a 25W/m^2 cooling capacity for different movement condition.

Influence of water supply temperature

Water supply temperature will influence the cooling capacity as well. When the supply temperature decreases, the temperature difference between the room air and the water will become bigger and therefore, more cooling capacity can be released to the room. This is almost a linear process (Figure 11). And the water return temperature will decrease following the trend as the supply temperature (Figure 12).

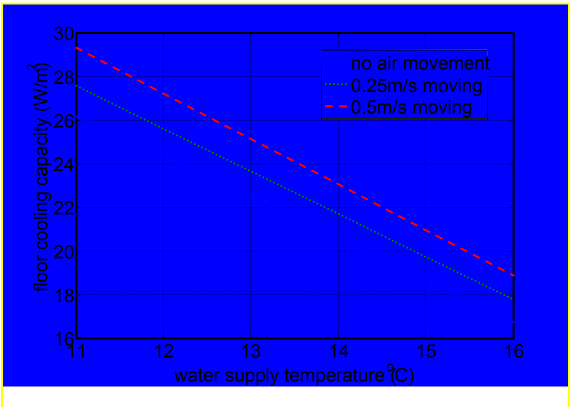


Figure 11. Variation of cooling capacity for different water supply temperature

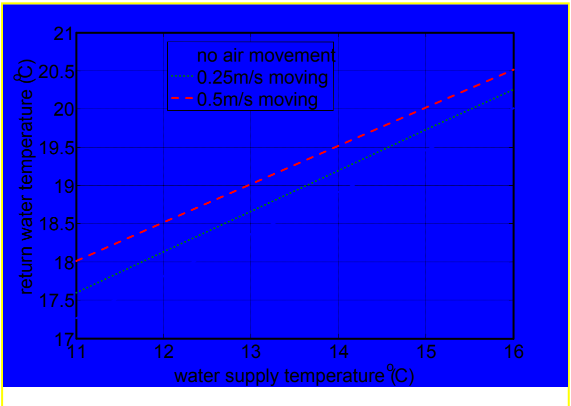


Figure 12. Variation of water return temperature for different water supply temperature

From the above shown results, it is clear that the water supply temperature should be decreased to 11.6 ~ 13.1°C to get a 25W/m^2 cooling capacity for different movement condition.

Influence of the solar radiation

The solar radiation falls on the floor will increase the cooling demand to the water pipe (therefore the central cooling machine) but release less cooling capacity to the air. This is because the solar radiation will be absorbed by the surface of floor and transferred quickly to the hydronic system. The water return temperature will be higher than the situation of no solar radiation. The mean temperature difference will be smaller and leads to less effective cooling capacity to the surrounding air.

Influence of radiation correction factor Cr

Cr is a correction factor to involve the effect of the temperature distribution of the surrounding surfaces. To explain it in a simple way, when most surfaces are far away from the cooling floor, Cr is closer to 1. In case, many surfaces are closer to the floor, e.g. much furniture stand above the floor, Cr is smaller than 1.

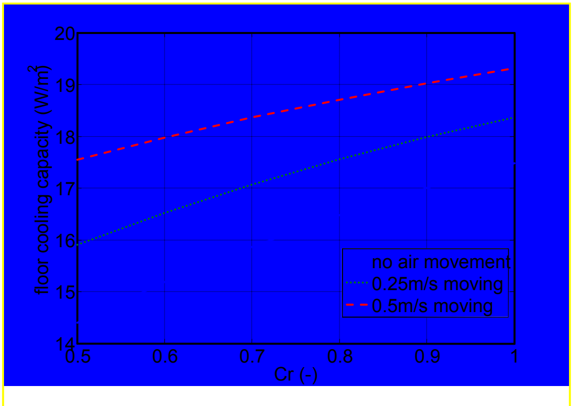


Figure 13. Variation of cooling capacity for different correction factor Cr

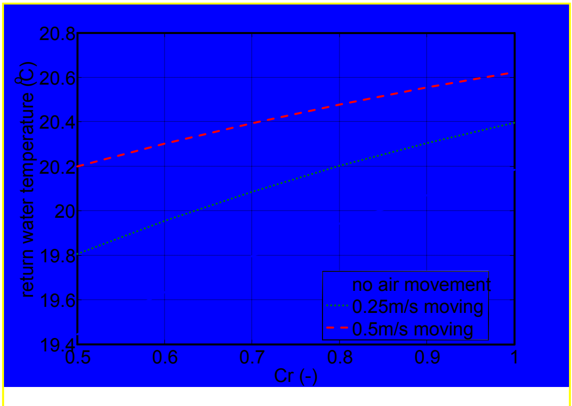


Figure 14. Variation of water return temperature for different correction factor Cr

From the above figures, it can be seen that Cr will correct a little bit on the cooling capacity but not dramatically. The change from 0.5 to 1 of Cr will lead to about 9% variation on the cooling capacity.

Influence of the movement

From Figure 4 to 14, all the results indicate the influence of the movement above the floor. It is clear that the moving of the passengers will give a positive effect to release more cooling capacity. In practice, when a lot of people are moving, more cooling is required. It happens barely that no movement occurs when the cooling is required.

CONCLUSIONS

The analytical model for the floor cooling system of the Eindhoven airport is built based on the conservation equations.

The preliminary design has been simulated and could not be able to deliver 25W/m^2 cooling capacity as expected from floor piping system.

Several parameters can be adapted to improve the cooling capacity:

- Modification of the length of one loop;
- Modification of the water flow rate;
- Modification of the distance between pipes (h.o.h);
- Change of the supply water temperature.

The influence from the above parameter as well as the other factors such as solar radiation and the movement of the passengers has been discussed. The introduced correction factor Cr has been analyzed and the result shows that Cr will lead to a bit change on cooling capacity but not dramatically.

It is possible to optimize all the parameters with the simulation model if an object such as total cost is defined.

NOMENCLATURES

q	Heat flux
α_t	Total heat transfer coefficient
T	Temperature
U_t	Total heat transfer coefficient of distributed pipes
σ	Stephen-Boltzmann constant
ε	Black body coefficient
δ	Diameter of pipe
R	Thermal resistance
d	Thickness; distance
λ	Thermal conductivity
A	Area
m	Mass flow rate
h	Heat transfer coefficient
Cp	Thermal capacity
Cr	Correction factor
Δt	Temperature different
Nu	Nusselt number
Re	Renolds number
Pr	Prandtl number

Subscriptions:

c	cooling
a	air
s	surface
w	water
i	inlet
o	outlet

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