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Numerical Study of Heat Transfer and Aerosol Deposition in a Room Environment with Under-floor or Baseboard Heating Systems

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Abstract

In this study, heat transfer and aerosol deposition in the under-floor and baseboard heating systems have been investigated, numerically. The aim of this study is a comparison between these heating systems. This comparison obtains the optimal heating system with low suspended particles in the air. Computational fluid dynamic with Eulerian-Lagrangian method has been used to simulate fluid and particles flows. The velocity and temperature distribution have been obtained by solving the equations of continuity, momentum and energy. It is resulted that, the radiant heat transfer contains about 63 % and 60 % of overall heat transfer of the under-floor and baseboard heating systems, respectively. Side walls have a same condition for deposing the particles in both of investigated heating systems, approximately. But, in floor heating system, most of the particles are deposited under the roof, while the baseboard heating system has a more percentage of seated particles on the floor.

Keyword: Floor heating system, Baseboard heating system, Heat transfer, Aerosol deposition.

Introduction

Air conditioning systems have been divided to two categories: radiant and based systems on convection, *i.e.* free or forced convectional heat transfer. Air conditioning systems, such as coolers, fan coils and radiators are considered as sub-systems of convection. Air is not heated in the radiant systems, directly. First, transferred heat warms surfaces and then heated surfaces warm the surrounding air by convection mechanism (Wallace, 1996). Baseboard heating system is a low temperature heat radiant that are installed along the bottom edge of walls. In this system, contained thermal energy in the water, inside U-shaped tubes, is transferred to thin metal plate. Then, this energy is transferred to surfaces through radiation mechanism and finally transferred to the room air by convection mechanism (Pope, 2000). On the other side, the performance of under-floor systems is on the convection and radiation mechanisms. Since the temperature

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gradient and buoyancy force are parallel together, the heating process is done easier by convection mechanism (Bears and Banks, 1985).

Olesen (1994) compared under-floor heating and wall panels heating systems. He resulted that, both of these systems provide a thermal comfort environment inside building. Chen and Athienitis (1998) used a 3D finite difference method to model and investigate effects of under-floor heating system on the floor temperature distribution and also its electrical energy consumption. They believed that carpets increase the energy consuming of these systems. Hanibuchi and Hokoi (1998) studied the performance of floor heating systems. They resulted that, first, radiant warms floor of room, then the convention mechanism causes heat exchange between the air and the inner surfaces of the room. Benhidy (1991) examined various types of radiant heating systems in terms of their energy consumption. They used ceiling and floor radiant heating panels and resulted that, with a same thermal comforting, the energy consumption of ceiling radiant heating systems is more than the floor radiant heating systems.

Material and methods

The aim of this study is comparing the heat transfer patterns of under-floor heating and baseboard heating systems in the room and also determining the optimal heating system with low suspended particles in the air. The investigated 2D geometry in this study was a room with 2 *m* width (*W*), 2.6 *m* height (*H*) and plaster walls. Figure 1 shows the defined mesh for this geometry. The floor temperature was constant in the under-floor heating system simulation. 20 cm baseboard was considered along the bottom edge of walls in the baseboard heating system simulation. Used assumptions for these simulations have been listed below:

- The flow was steady state.
- Gaseous phase was compressible.
- Gaseous phase was Newtonian fluid.
- Properties of gas phase were constant.
- Aerosols were solid particles.
- Coagulation of collided particles was ignored.
- Considering very low concentration of aerosols in the air, therefore, effects of these particles on the fluid flow were ignored and the Lagrangian method was used.
- Aerosols were released in the room as a group.
- The weight, lift, drag and Brownian forces affected aerosols.

Computational fluid dynamics (CFD) was used to simulate the heat transfer and also aerosol deposition on surfaces. First, heat transfers of under-floor and baseboard heating system are compared in a room with a combination of radiant and free convection heat transfer. Finally, the dispersion and deposition of aerosols in the room have been investigated.

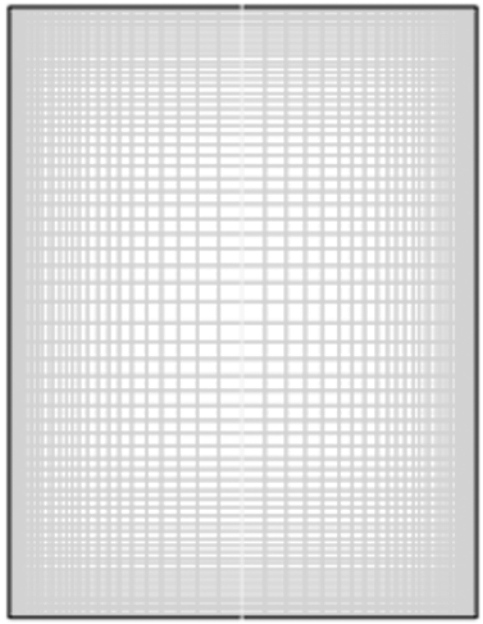

Figure 1. A view of used computational grid

Modelling

The governing equations on the fluid flow are continuity, momentum and energy equations, which have been presented blow, according to the listed assumptions:

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)=0 \tag{1}$$

$$\frac{\partial}{\partial x}\left(\rho u^2+P+\frac{2}{3}\mu\frac{\partial v}{\partial y}-\frac{4}{3}\mu\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\rho uv-\mu\frac{\partial u}{\partial y}-\mu\frac{\partial v}{\partial x}\right)=0 \tag{2}$$

$$\frac{\partial}{\partial x}\left(\rho uv-\mu\frac{\partial u}{\partial y}-\mu\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\rho v^2+P+\frac{2}{3}\mu\frac{\partial u}{\partial x}-\frac{4}{3}\mu\frac{\partial v}{\partial y}\right)=0 \tag{3}$$

$$\begin{aligned}&\frac{\partial}{\partial x}\left\{\left[C_vT+\frac{1}{2}\left(u^2+v^2\right)+P\right]u-k\frac{\partial T}{\partial x}-\frac{4}{3}\mu u\frac{\partial u}{\partial x}+\frac{2}{3}\mu u\frac{\partial v}{\partial y}-\mu v\frac{\partial u}{\partial y}-\mu v\frac{\partial v}{\partial x}\right\}\\&+\frac{\partial}{\partial y}\left\{\left[C_vT+\frac{1}{2}\left(u^2+v^2\right)+P\right]v-k\frac{\partial T}{\partial y}-\frac{4}{3}\mu v\frac{\partial v}{\partial y}+\frac{2}{3}\mu v\frac{\partial u}{\partial x}-\mu u\frac{\partial u}{\partial y}-\mu u\frac{\partial v}{\partial x}\right\}=0\end{aligned} \tag{4}$$

Where x and y are horizontal and vertical directions, respectively. P is pressure and ρ, μ, k and C_v are density, viscosity, thermal conductivity and heat capacity of fluid. Presented velocity elements in above equations (u and v) and also temperature (T) are instantaneous values, which are summations of their averages and tolerances. Existing difference of the surface and fluid temperatures lets us to calculate gas density using the Boussinesq approximation (Boussinesq,

1897). The balance of acting forces on the particles (discrete phase) in the Lagrangian model has been presented as the following equation (Wilcox, 2006):

$$\frac{du_P}{dt} = F_D\left(u - u_p\right) + \frac{\rho_p - \rho}{\rho_p} g + F_B + F_L \tag{5}$$

Where t is time, g is the gravitational acceleration and ρ_p and u_p are particles density and velocity, respectively. The drag (F_D) (Wilcox, 2006), gravity (second term in the right side of above eq.), Brownian (F_B) (Roberts *et al.*, 1992) and lift (F_L) (Olesen, 1994; Wilcox, 2006) have been considered as acting forces on the particles. The discrete ordinates method (DOM) (Chui and Raithby, 1993; Raithby and Chui, 1990) was used to model the radiant heat transfer. The turbulence of compressible fluid flow was simulated using the standard k-ε model (Wilcox, 2006; Smagorinsky, 1963).

Constant surface temperature (300 to 306 K for under-floor heating system and 343 to 353 K for baseboard heating system) and free convection were used as the thermal boundary conditions. No-slip fluid boundary condition and particle trapping condition were considered for all of these surfaces.

Results and Discussions

Natural convection of air in 2D rectangular enclosure with a central heat source at the bottom surface and symmetric cooling systems at the side walls was simulated to valid the modeling of flow pattern. The value of dimensionless heat source length was considered equal to 1/5 and Rayleigh number equal to 10^6. The calculated Nusselt number for this system was 3.48, which had about 3% difference with the reported value by Aydin and Yang (2000). The transport of 1 μm diameter particles in laminar free convection of air in 2D square enclosure with 0.5334 cm width, two adiabatic surfaces and 100 K temperature difference between the other two surfaces and Rayleigh number equal to 10^6 was simulated to valid the modeling of particles motion. Figure 2 compares the calculated the fraction of remaining particles by this study with the obtained values by Akbar *et al.* (2009). The grid independency was checked for this numerical simulation. Figure 3 shows this independency for more than 2500 nodes. These velocity profiles were drawn at x=1 (room symmetric line).

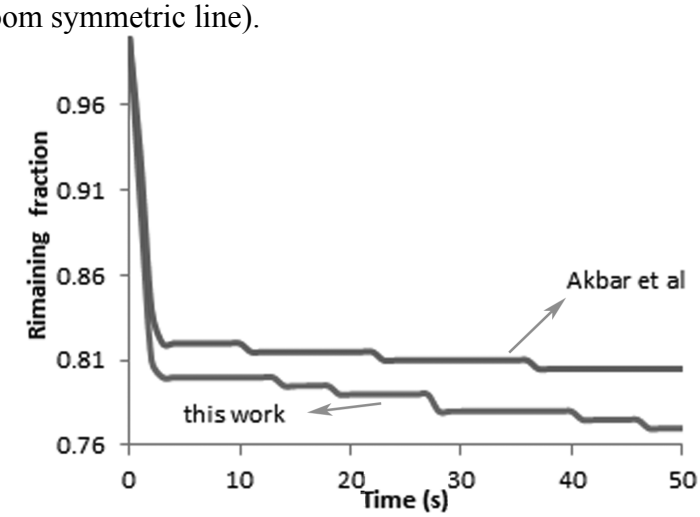

Figure 2. Comparing the calculated fraction of remaining particles by this study with the obtained values by Akbar *et al.* (2009)

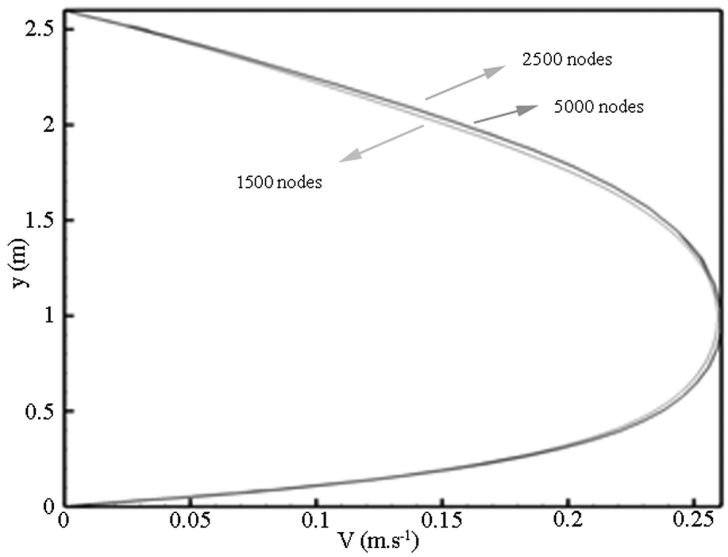

Figure 3. Mesh independence test

Figure 4 compares the velocity vectors of the described geometry with under-floor and baseboard heating systems. As shown in this Figure, geometrical symmetry causes vortex flows in both sides of the centerline of the room. The under-floor heating system creates two large vortex flows (one is clockwise and another is counter clockwise). In this system, the centerline fluid velocity is zero at two points (the floor and roof of the room) and has a maximum value at a certain height. This velocity has been varied with the height, due to the affecting buoyancy force on the fluid. Only one vortex flow is formed in each side of the room. Small heating surface causes that, baseboard heating system creates four vortex flows (two small vortex flows near the baseboards and two large vortex flows at higher positions. It is clear that, upper vortex flows have different directions. In this system, the centerline fluid velocity is zero at three points (the floor, a certain height and roof of the room) and has two extremum values (one minimum and one maximum value) at two certain heights. It shows formation of a vortex flow near the floor and another vortex flow near the room roof.

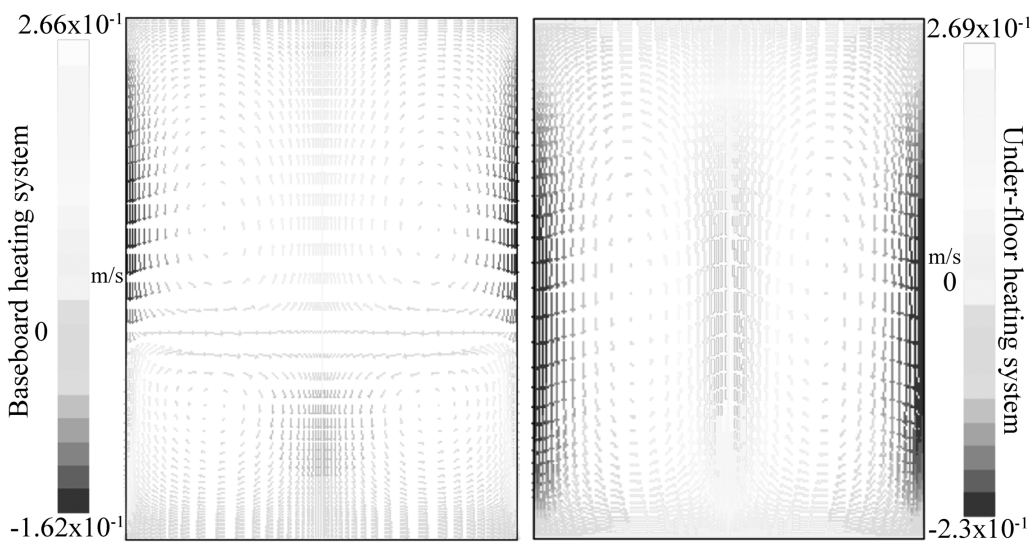

Figure 4. Velocity vectors

Figure 5 compares the temperature contours of the described geometry with under-floor and baseboard heating systems. As shown in this Figure, the under-floor heating system has a lower temperatures gradient than the baseboard heating system. Therefore, the under-floor heating system has a more uniform temperature.

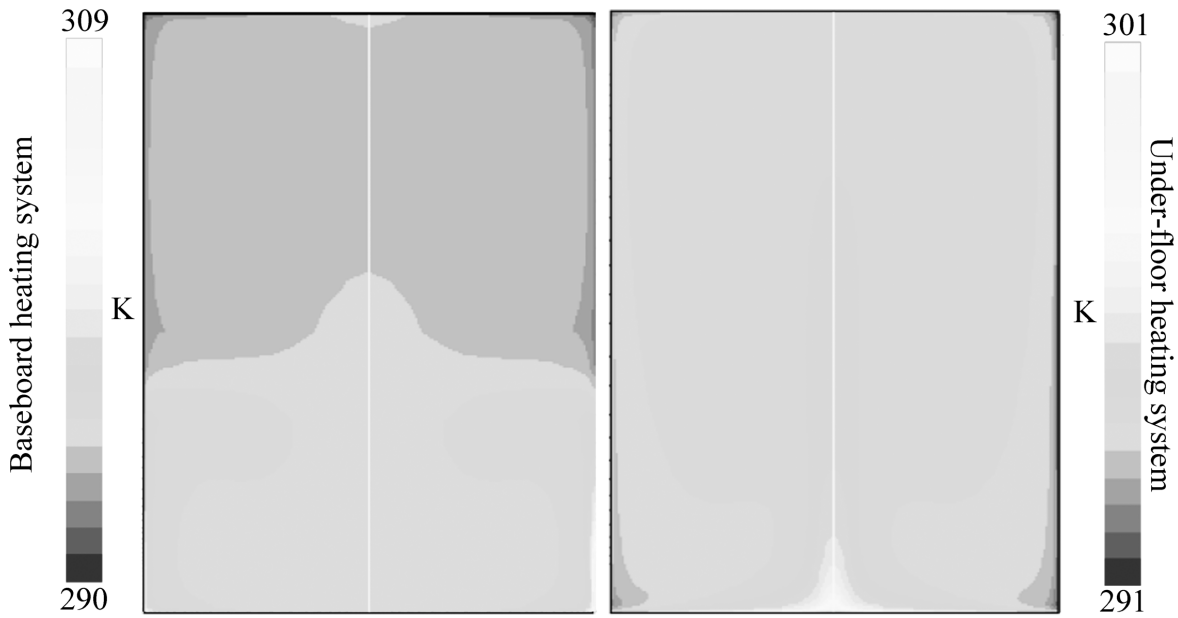

Figure 5. Temperature contours

Figures 6 to 8 compare the depositions of various sized aerosols on the surfaces (floor, roof and side walls of the room) in these two heating system. As seen in Figure 6, both of them have a similar aerosols deposition on the side walls. Other two Figures shows that, in the under-floor heating system, the aerosols have a more deposition on the room roof and a less deposition on the floor. Therefore, this heating system is more suitable for human healthy.

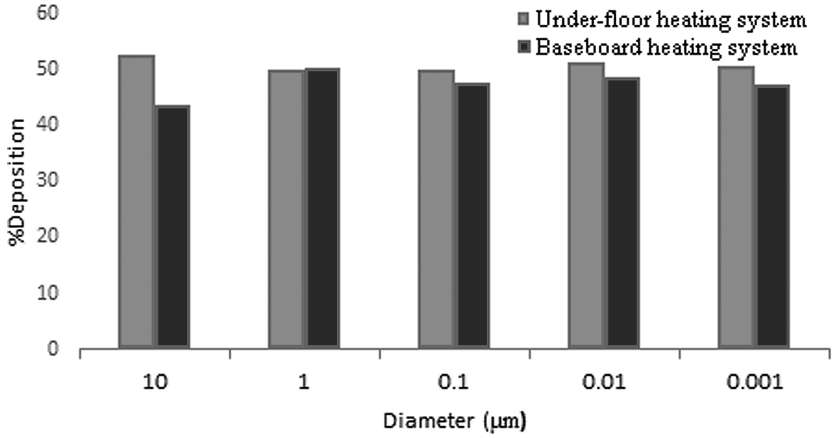

Figure 6. Deposition of particles with various diameters on the side walls

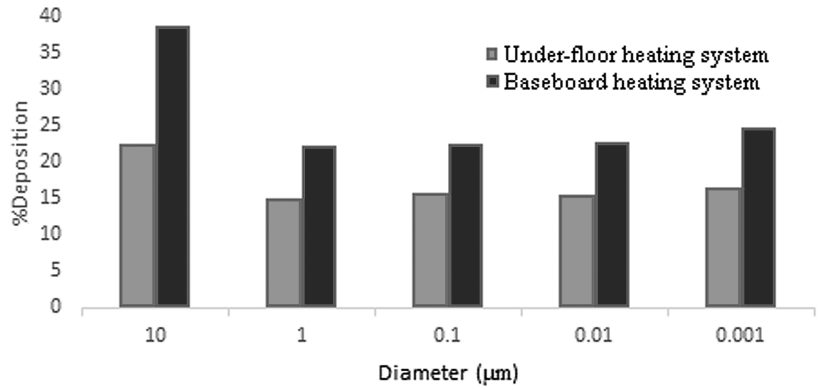

Figure 7. Deposition of particles with various diameters on the floor

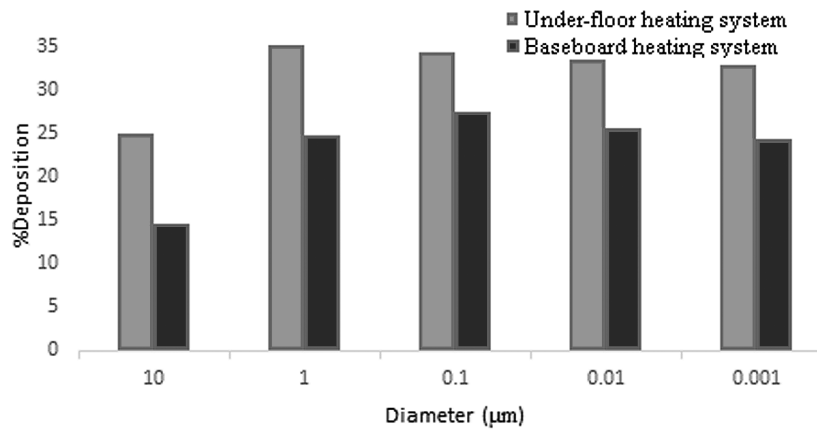

Figure 8. Deposition of particles with various diameters on the roof

Conclusion

A computational fluid dynamics modeling was developed to compare two heating systems in the room, under-floor and baseboard. Computational fluid dynamic with the Eulerian-Lagrangian method was used to simulate fluid and particles flows. Velocity and temperature distribution were obtained by solving the equations of continuity, momentum and energy. This model has been validated using the results of previous published studies. Then, the velocity vectors, temperature profiles, and also the depositions of various sized aerosols on the surfaces (floor, roof and side walls of the room) have been compared for these heating systems. The extracted results were listed below:

- About 60% and 63% of heat is transferred by radiation in the baseboard and under-floor heating systems, respectively. The rest is transferred by free convection.
- Under-floor heating system has a more uniform temperature distribution in the room.
- Saving energy of the under-floor heating system is more, because of low temperature of its warm water.
- Both heating systems have similar distribution of particles on the side walls.
- Particles deposition on the floor is more in the baseboard heating system.
- The under-floor heating system has s more particles deposition on the room roof.

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