EXPERIMENT AND SIMULATION OF RADIANT/CONVECTIVE SPLIT FROM PASSENGER IN AIRCRAFT CABINS

Weibing HE, Hejiang SUN*

School of Environmental Science and Engineering, Tianjin University, Weijin Road, Nankai District, China

*Corresponding email: sunhe@tju.edu.cn

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SUMMARY

The objective of the present work was to investigate the method of separation between radiant and convective components from total sensible heat loss of a passenger represented by a nude thermal manikin seated in a climate chamber which had a kind of radiant boundary condition similarly to that in the real cabin. The authors want to split the radiation released from occupants by experiment and the rest of convection was calculated by CFD to verify the accurate experimental measurement. A validation of CFD proved that CFD had the capacity of predicting occupant’s heat loss with highly accuracy in a confined space. Results showed that the ratio of convection to radiation was 33:67, lower radiative heat transfer coefficient and natural convection heat transfer coefficient for the whole body was acquired.

INTRODUCTION

Occupant is often a necessary boundary condition to be studied in both experiment and Computational fluid dynamics (CFD). Several studies had found the main factors of human occupants impacting on the micro-environment including the geometric configuration (Topp et al., 2003), posture (Quintela et al., 2004), and the ratio of convective to radiative heat loss from occupants($C:R$ ratio) (Srebric et al., 2008). Among these factors, the $C:R$ ratio was the critical one through its influence on the transportation of contaminants and the distribution of velocities as well as temperatures around the human body. Srebric et al. (2008) compared the different distributions of concentrations and velocities in terms of a representative case (Yuan, 1999) using CFD by setting the convection to radiation ratio ranged from 70:30 to 30:70. They found that a stronger thermal plume and higher airflow velocity would appear in the vicinity of human simulator when the $C:R$ ratio is higher, which introduced fresher and cooler air from head below up to the ceiling, but had the unexpected contaminant possibility. Hence, the radiative/convective split from human’s sensible heat loss is essential to indoor air quality control and design.
Techniques had been applied to study the separation between radiation and convection from occupant’s sensible heat rejection. Experimental methods for a nude body are presented based on the following formulations,

\[ C = h_c(\bar{T}_s - t_a) \quad (W/m^2) \]  
\[ R = h_r(\bar{T}_s - \bar{T}_r) \quad (W/m^2) \]

Where \( C \) is the convective flux; \( R \) is the radiative flux; \( h_c \) is the convective heat transfer coefficient; \( h_r \) is the radiative heat transfer coefficient; \( \bar{T}_s \) is the mean skin surface temperature; \( \bar{T}_r \) is the mean radiant temperature; \( t_a \) is the air temperature surrounding the human body. De deur et al. (1997) adopted an approach that the mean radiant temperature was regulated to equal the ambient temperature \((\bar{T}_r = t_a)\) to obtain total heat transfer coefficient \((h_c + h_r)\), then isolated the convective component by coated a low emissivity foil on the surface of manikin through a series of formula derivations with an assumption that \( h_c \) remains constant under both conditions of the foil coated and uncoated.

A validated CFD program has been widely used to solve problems referring to human occupants in buildings with accepted accuracy. As for the separation of convective and radiative flux, two approaches could be adopted on CFD settings of a computational thermal manikin. First, the convective heat transfer is calculated by giving the skin surface temperature without considering the radiant heat exchange between the occupant and the environment (Srebric et al., 2008). Second, two components of heat flux were calculated simultaneously by giving the total dry heat rejections activating the radiant model (Murakami et al. 1999).

However, all above studies are limited in the aspects of the environmental conditions and the methods of splitting radiant or convective component. Different environmental conditions such as the relative space geometry, ventilation systems and distributions of wall temperature could result in different ratio of radiation to convection. Aircraft cabin is typically characterized with highly passenger density, almost sedentary activity levels and mixing ventilation systems. Under these conditions, the equivalent relationships among \( \bar{T}_s, \bar{T}_r, t_a \) are difficulty to be regulated. Few researches has been done on the percentage of radiation or convection from sensible heat release of passengers in aircraft cabins while more attention were paid to that in residential buildings. It is therefore necessary to explore the percentage of radiation and convection from a passenger in the cabin. The objective of the present work was to investigate the method of separation between radiant and convective components from total sensible heat loss of a passenger represented by a nude thermal manikin seated in a climate chamber which had a kind of radiant boundary condition similarly to that in the real cabin.

**METHODOLOGIES**

The Climate Chamber and Environmental Control System (ECS)
The configuration of experimental platform was a half simplified but highly accurate cabin mockup accounting for keeping the main characteristics such as the air supply mode, air supply parameters. As shown in Figure 1(b), the simplified half-cabin mockup has dimensions of $2m \times 2m \times 2m$ as width, depth and height, respectively. Air supply mode was adopted by a mixed ventilation that supplied fresh air from the higher level of inlet and discharged air from the bottom of a side wall.

(a) Real cabin settings  

(b) Simplified half-cabin mockup

![Figure 1. The simplified process of the cabin mockup](image)

The environmental control system of the chamber was comprised of a wall temperature control system and an air temperature control system. Regarding to the flexibility of regulating thermal boundary condition of walls, three series of water circulation were used. The air temperature control system consisted of a supplied air conditioner with 1200W cooling capacity and a PID regulated heater with 1000W heating capacity.

**Radiant boundary modification of the chamber**

Aircraft cabin, a confined space, was the main studied environment with high occupant density. It is critical to build a similar cabin which could reflect the radiant characteristics when passengers seated quietly. The main factors, illustrated by Figure 2, influencing the passenger’s heat loss in a real cabin include the mutual radiation among the passenger and the cold wall, the radiation between the human body and the ceiling as well as from the human to human. In order to split the approximate radiation successfully from the sensible heat in the climate chamber, the radiant boundary settings were displayed in Figure 3.

![Figure 2. Main factors influencing passengers’ heat loss](image)  

![Figure 3. Boundary settings of the chamber](image)
Boundary conditions for the standard cruise flight situation

According to ASHRAE Standard 161 (2007) and ASHRAE Standard 55 (2004), the temperature design and velocity requirement surrounding the passenger must be met. Under comfort state and with completely sedentary level, an occupant may have a sensible heat loss of 75W regarding to the heat balance of human body. In addition, the subject of our research was a female manikin who had the same heat loss characteristics of a real person and its heating power was regulated to 75 watts.

The practical wall boundary conditions were adopted, most wall temperatures were nearly approach to the local air temperature except that the lower temperature of ceiling, left wall and floor. The supply air condition in the chamber was designed accounting for the comfortable condition like the velocity limit and the vertical temperature difference around the passenger. The inlet temperature was 297.84K, and the mean supplied velocity was 1.54 m/s.

Measurement and apparatus

All surfaces were assumed to be diffuse and gray radiation. Employed the following equations based on irradiation, total radiation incident on surface per unit time and per unit area, the radiation exchange could be calculated.

\[
\sum_{i=1}^{n} J_i F_{j-i} - \frac{J_j}{1 - \varepsilon_j} = \sigma_b T_j^4
\]  

(3)

\[
R = \sum_{i=1}^{n} \frac{I_p - I_i}{1/A_p/A_{p-i}}
\]  

(4)

Where \( J \) was irradiation of wall surface, indicated the view factor on two arbitrary wall surfaces from surface \( i \) to surface \( j \), \( \varepsilon \) is the emissivity of surface, \( \sigma \) is Stenfen-Boltzmann coefficient, \( 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K} \), subscript p represented the manikin of our experiment, \( A_p \) was the manikin’s surface area exposed in the air. According to the radiant formulation based on the mean radiant temperature,  

\[
R = \varepsilon_p \cdot \sigma \cdot A_{\text{eff}} \cdot (\bar{T}_s^4 - \bar{T}_r^4)
\]  

(5)

\( \bar{T}_r \), the mean radiant temperature, would be solved if some radiant quantities were worked out such as the effective radiant area of manikin and the mean skin temperature. Assuming that there was no conduction between the human and its contact surface, little radiant exchange between dummy and chair, the convection heat loss could be expressed by the following equation,
\[ C = M - R \]  \hspace{1cm} (6)

where \( M \) was the total sensible heat loss, 75 watts.

All apparatuses were calibrated before the test. During the measurement of radiation, the temperature acquisition system of Pt1000 was used for the measurement of mean skin temperature, air temperature and wall temperature distributions whose accuracy was \( \pm 0.3^\circ C \). Platinum temperature probes were mounted at three kinds of levels: ankle level, abandon level and head level according to ASHRAE Standard 55 (2004) and positioned at a variation distance from 10cm to 15cm near the manikin to avoid the strong perturbation produced by the upward thermal plume. In order to obtain highly accurate with less test points, the Winslow-14V approach was adopted for measuring the mean skin temperature (J.K. Choi, 1997).

The effective radiant surface area could be derived by the following equation base on the meaning of mean radiant temperature:

\[
A_{eff} = \sum_{i} \frac{1}{\varepsilon_p} \frac{1}{A_{i\varepsilon_p}} A_{i\varepsilon_p F_{i\varepsilon_p - wall}} \hspace{1cm} (7)
\]

Where \( A_i \) represented the divided skin surface area of segment \( i \) exposed in the air ensuring that \( F_{i\varepsilon_p - i} \) was nearly approached to zero, m\(^2\); \( F_{i\varepsilon_p - wall} \) was the view factor between the seated manikin and its enclosure.

**Validation of a CFD program for occupant’s heat loss in a confined space**

In order to verify the accuracy of the measurement of radiant component separated from the total dry heat flux rejection, CFD was selected as a research tool to predict the convective heat loss. Chen (1995, 1996) proved that the Re-Normalization Group (RNG) \( \kappa - \varepsilon \) model performed best of the eddy-viscosity models in indoor environment. However, there also existed some uncertainty computational code when predicted heat fluxes, so it needed a case with an detailed and accurate measurement data to validate that capability. Håkan O. Nilsson, et al. (2007) had a benchmark test focused on the heat loss from the manikin, and provided detailed test data of heat loss so as to the comparison with CFD predictions. The noticeable thing was that the manikins adopted in CFD displayed in Figure 1 and experiment was different with respect to size, posture. The radiation was calculated by assuming that all the surfaces were supposed to be diffuse and gray which had an emissivity of 0.9. And view factors between segments and walls surrounding the manikin were obtained by a CFD program.

Workbench was used to generate meshes of the fluid domain, the average size meshed on manikin was 0.004m ensuring that \( y^+ \) values at the first grid of skin surface were within 5.
During the process of simulation, RNG $\kappa - \varepsilon$ model with Enhanced Wall Treatment was applied, the solution variables were discretized with second order except the pressure term with standard. When the tolerant residual was met and quantity of the mass balance decreased into $10^{-8}$, net heat flux decreased into $10^{-1}$, the calculation was considered to be converged.

Figure 4 depicted the comparison between the experimental data and CFD prediction on the temperature and velocity distribution of the flow field. Green solid line stands for the simulated value and green square for the experimental data. CFD simulations showed good agreement with experimental data of temperature and velocity distribution. Moreover, the simulated convection from manikin was 122.63W, and calculated radiation was 70.58W which means there 7.3% error existed for the whole body compared to experimental heat loss of 180.09 watts. It is likely the difference of two thermal manikins that caused the relative error. Nevertheless, the approximate heat loss of whole body between CFD result and experimental value has proven that CFD has the capacity of predicting occupant’s heat loss with highly accuracy in a confined space.

![Figure 4. Comparison between the experimental data and CFD prediction](image)

**RESULTS AND DISCUSSION**

Measurement showed that the manikin had mean skin temperature of 302.48K who was surrounded by air with temperature of 297.30K. After theoretical calculation based on radiant equation(3), equation(4) and computational simulation of convection, results showed that radiant heat transfer from the thermal manikin was 50.61W, and simulated convection was 24.29W. In other words, the sum of radiation and convection was 74.9W which was nearly the same as the electricity consumption of the heating power, 75W. The percentage of radiation accounts for 67.48% of the total heat loss while convection occupied 32.52%. Therefore, the ratio of radiation to convection was 67:33. The whole-body convective heat transfer coefficient, $h_c$, indicated 3.27W/m² per K derived from the convective equation(1). According to equation(7), effective surface area, $A_{eff}$, was calculated for 1.029 m² accounting for 71.81% of the whole-body surface area, 1.433m². As a result, mean radiant temperature was 293.68K employed equation(5), and the overall radiant heat transfer coefficient, $h_r$, showed value of 4.01W/m² per K.
This research was dedicated to splitting convection/radiation from human’s heat release in a similar cabin environment, and found that C:R ratio was 33:67. Srebric et al. (2008) recommended 30:70 of C:R ratio for human simulator. ASHRAE (2001) demonstrated that 40:60 ratio was suitable for the numerical simulations of commercial and residential building. From the analysis of result, C:R ratio distributed among 30:70 and 40:60 which was identical to that in office buildings although aircraft cabin was typically characterized with highly occupant density and limited space.

As an assessment of heat transfer intensity, $h_c$ and $h_r$ were of great importance for reflecting the features of occupant’s surroundings. Convective heat transfer coefficient was mainly affected by air velocity around human body. The general formation of $h_c$ depending on ambient velocity, $v$, was as follows:

$$h_c = B v^n$$  \hspace{1cm} (8)

Where $n$ varied in the region of 0.5 to 0.6. Therefore, large velocity around human body means large convective heat transfer coefficient. Radiative heat transfer coefficient was determined by three factors for an unclothed manikin: ratio $A_{eff}/A_D$ where $A_D$ represented the whole-body surface area, $\bar{T}_s$, $\bar{T}_r$. Ratio $A_{eff}/A_D$ was affected by the posture of human body like seating or standing and mean radiant temperature embodies the real wall temperature distribution surrounding occupants although its imaginary black enclosure factor.

Experimental and numerical research on overall $h_c$ and $h_r$ had been done by previous researchers in commercial and residential buildings. Owing to the space limitation, some research results would not list in this paper. For natural air flow, typically the air speed, $v$, less than 0.2m/s according to the definition from de deur et al. (1997) or even limit to 0.1m/s, $h_c$ varies in the region of 3.13 W/m$^2$·K to 3.90 W/m$^2$·K. Radiative heat transfer coefficient does not vary much compared to $h_c$, and floats around the widely accepted value of 4.7 W/m$^2$·K. In the similar cabin environment, the convective heat transfer coefficient, 3.27 W/m$^2$·K, within in the scope of natural convective heat transfer coefficient, indicated that the flow surrounding the manikin was natural. However, $h_r$ of this cabin is obviously lower than that in commercial and residential buildings, which may show the difference of heat release between cabin environment and office building. Probably this characteristic is caused by the minor mutual radiant size due to higher occupant’s density in confined space.

CONCLUSIONS

The ratio of convection to radiation from a passenger, 33:67, was validated by both the experiment and the CFD simulation. And it is reasonable by comparing to that in the published literature. Lower radiative heat transfer coefficient and natural convection heat transfer coefficient for the whole body was compared with that in residential constructions due to the characteristics of the cabin itself.
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