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An under-aisle air distribution system facilitating humidification of commercial aircraft cabins

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A B S T R A C T

Air environment in aircraft cabins has long been criticized especially for the dryness of the air within. Low moisture content in cabins is known to be responsible for headache, tiredness and many other non-specific symptoms. In addition, current widely used air distribution systems on airplanes dilute internally generated pollutants by promoting air mixing and thus impose risks of infectious airborne disease transmission. To boost air humidity level while simultaneously restricting air mixing, this investigation uses a validated computational fluid dynamics (CFD) program to design a new under-aisle air distribution system for wide-body aircraft cabins. The new system supplies fully outside, dry air at low momentum through a narrow channel passage along both side cabin walls to middle height of the cabin just beneath the stowage bins, while simultaneously humidified air is supplied through both perforated under aisles. By comparing with the current mixing air distribution system in terms of distribution of relative humidity, CO₂ concentration, velocity, temperature and draught risk, the new system is found being able to improve the relative humidity from the existent 10% to the new level of 20% and lessen the inhaled CO₂ concentration by 30%, without causing moisture condensation on cabin interior and inducing draught risks for passengers. The water consumption rate in air humidification is only around 0.05 kg/h per person, which should be affordable by airliners.

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1. Introduction

Commercial airplanes cruise at a typical altitude of 11,000 m where the outside temperature is about −55 °C (−67 °F), the atmospheric pressure is only about one-fifth of that at the sea level and the air is nearly dry [1]. Under such an extreme ambient environment, human beings cannot survive without protection of an environmental control system (ECS). As a part of the ECS, an air distribution system plays roles including distributing air appropriately, cleaning up contaminated air, and minimizing cross airborne disease transmission, towards to creating a comfortable, healthy and safe cabin environment for passengers and crews. However, current widely-used mixing air distribution systems promote air mixing by supplying high-momentum air at the ceiling level and exhausting contaminated air at the deck level. The extensive air mixing is supposed to dilute occupant-generated pollutants, but may also impose risks of infectious airborne disease transmission because of easy access of the exhaled pollutants released by some infected passengers in the mixing context. The outbreaks of airborne communicable diseases, such as the severe acute respiratory syndrome (SARS) in 2003 [2], the later on possible multi-drug resistant tuberculosis transmission [3], and the recent pandemic outbreak of the influenza A (H1N1) [4], etc., confirm such risks are not hypothetical but may come to reality. A new air distribution system that can reduce air mixing on airplanes is extremely necessary.

In addition, significant attention should be paid to the numerous long complaints about cabin air quality and thermal comfort [5]. Since the current mixing air distribution system mixes the cabin air very well, temperature inside is highly uniform, hence it should not be the major reason leading to thermal discomfort. The mostly criticized in aircraft cabin is air dryness, which is known to be responsible for headache, tiredness, eye irritation, nasal allergy and other non-specific syndrome symptoms [6,7]. During flight, the relative humidity of cabin air ranges averagely from 10 to 20%, as similar to typical indoor levels during wintertime [8]. The intercontinental flights may be even dryer, where measurements show 6–8% of relative humidity may last for long periods [9]. Low humidity in cabins is attributed to the frequent renewal (20–30 ACH, air change per hour) of cabin air with the outside air that is substantially dry during cruise. Such large air change rate is to dilute internally generated pollutants.

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and also to maintain thermal comfort, though current cabin air quality is still being criticized. The other major reason without humidifying cabin air is to avoid moisture condensation on cabin walls which typically include three layers, i.e., from the outside to the inside the fuselage, insulation panel and lining. The cabin walls experience large temperature difference because of low temperature of aircraft shell at cruising. In addition, low humidity is thought to be helpful for inhibiting fungal and bacterial growth. Therefore, until a new air distribution system has been proposed that can prevent moisture condensation, and at the same time a method has been found to better remove the internal pollutants but without sacrificing thermal comfort so that the air change rate can be lowered, the air dryness and air quality problem in commercial cabins may not be well resolved.

Fortunately, concepts of new air distribution systems for aircraft cabins have been emerged, such as an under-floor displacement air distribution system and a personalized air distribution system. A typical under-floor displacement system supplies conditioned air from the low level at a low velocity and small temperature difference and extracts air at the ceiling level. The personalized air distribution system supplies conditioned air directly to the breathing region of occupants. Zhang and Chen [10] proposed the under-floor air distribution system with perforated-aisle air supply, and the personalized air system by supplying air from a seatback-embedded diffuser in front of a passenger in wide-body aircraft cabins, where both systems were found being capable of providing much better air quality than the current mixing air system. Schmidt et al. [11] applies computational fluid dynamics (CFD) to study the under-floor air distribution system for a single-aisle aircraft cabin and pointed out it may have draught risks for inside passengers. Again, with CFD Gao and Niu [12] investigated a personalized air system by supplying air through a flexible nozzle just beneath the nose and mouth of a passenger and claimed 60% of the pollutants can be shielded up from inhalation. Similarly, Jacobs and de Gids [13] proposed to supply conditioned outside air via the headrest of the seat, where each passenger can individually control their preferred condition. In addition, Gao and Niu [14] further explored humidifying the personalized supply air to relative humidity of 40–50%, but results show only the facial region may have higher moisture content whereas the whole cabin is still maintained very dry. It should be noted that a personalized air distribution system usually requires a mixing or displacement air distribution system as a background system, and moreover our preliminary survey shows not all passengers accept such system mainly due to draught complain, even though some of current airplanes are equipped with personal overhead gaspers.

The above review reveals none of the proposed air system can humidify the whole cabin efficiently while simultaneously removing internal pollutants appropriately. A personalized air distribution system supplying air at high velocity may impose draught risk during cruise, which prevents it from being widely accepted. This investigation therefore proposes a new under-aisle air distribution system, by supplying air both from under-aisles and from the two cabin sides below the stowage bins, and thus to enhance moisture content level and improve air distribution efficiency. Performance of the new system in terms of humidity distribution, moisture condensation risk on cabin linings, cabin air quality and thermal comfort are thoroughly evaluated.

2. The new under-aisle air distribution system

A new air distribution system in a section of a double-aisle aircraft cabin during cruise flight, as shown in Fig. 1(a), was proposed. The cabin contains five rows of seats with all seats well occupied. Four strips of heat sources mounted on the ceiling simulate lighting to provide illuminantion for the cabin. The box-shape manikins mimic the seated passengers inside. The small squares on each manikin are where moisture and carbon dioxide (CO₂) is released to represent human exhalation effects.

The new system supplies fully conditioned outside air that is substantially dry and at low velocity (around 0.2 m/s), through a narrow channel passage along both side walls to middle height of the inside cabin just beneath the stowage bins. Fig. 1(b) shows the schematics of the system from the sectional view. The air channel is designed to run the length of the aircraft, which is actually a curved air passage formed by the cabin insulation panel and the interior lining with warm air running upward after entering from the underdeck. Such design is in purpose of directly supplying conditioned, outside dry air to the upper region of the cabin, as well as warming the cabin interior linings to avoid possible moisture condensation on them, due to relatively low temperature of the airplane fuselage at cruising. Simultaneously, the other part of outside air mixed together with the recirculated air after humidification is supplied through both perforated under aisles (Fig. 1(a) and (b)). Such under-aisle air supply is to ensure thermally comfortable cabin environment and also to elevate the cabin humidity level. Finally, the contaminated air is extracted through the both ceiling exhausts.

To show the extent of the new system in improving cabin humidity and air quality, a mixing air distribution system was also studied and compared with the new system. As shown in Fig. 2, the mixing air system supplies conditioned mixed air (outside air together with recirculated air) from both linear slot inlets on the ceiling and extracts air at the deck level. Other settings are the same with the under-aisle system. Carbon dioxide (CO₂) was applied as an indicator to quantify air distribution efficiency. To be comparable, the new system does not
change the total outside air supply rate and air recirculation ratio as compared with the current mixing air system.  

Table 1 lists the design parameters for the two air systems. The designed average air temperature for both systems inside the cabin is controlled at 24 °C. Each passenger is provided 10 l/s of conditioned air, in which 5 l/s is from the outside. A passenger is assumed to generate 0.005 l/s of CO2 and 0.05 kg/h of moisture through respiration. Since in the new system, both channel inlets are only supplied with fully outside air, the supply CO2 concentration is assumed to be 350 ppm without moisture. The aisle inlets supply the mixture of the outside air (3.572 l/s per person) with the recirculated air (5 l/s per person), so CO2 concentration is around 933 ppm and has been humidified into 25% relative humidity (at the saturation temperature of 22 °C). The mixing system supplies the mixed air directly and therefore the supply CO2 concentration is around 850 ppm. Because the mixing system mixes the inside air very well, air supply temperature should be lower than the under-aisle system to ensure thermal comfort. This investigation does not consider heat dissipation from electronics devices and other heat sources. Moisture content in the mixing air is only from air recirculation generated by inside passengers, where the corresponding relative humidity is around 6%.

The cabin interior temperature in Table 1 for the mixing system was taken from an on-site measurement with more details in our previously published work [15]. The surface temperatures for the under-aisle system are estimated values. The criterion to estimate the surface temperatures in the under-aisle air system is that one shall obtain a cooling load comparable to that in the mixing system since the designed average air temperatures are the same in the two systems. Note in the under-aisle system, due to the warm air running through the channel passage, the side lining and window temperature is much higher than those in the mixing system from our rough heat transfer analysis of the channel flow in the passage.

3. CFD modeling principles

As most researchers who investigated air distribution system for aircraft cabins used CFD as the tool, due to its efficiency, flexibility and relatively low cost, this study also adopted CFD to evaluate the proposed new system.

CFD solves a set of partial differential governing equations that are usually casted into the general scalar format according to the Reynolds-averaged Navier-Stokes (RANS) CFD approach as

\[
\frac{\partial}{\partial t} \left( \rho \phi \right) + \nabla \cdot \left( \rho \mathbf{u} \phi \right) = \nabla \cdot \left( \Gamma_{\phi,eff} \nabla \phi \right) + S_\phi
\]

where \( \rho \) is the air density, \( \phi \) is a scalar variable, \( t \) is time, \( \mathbf{u} \) is the velocity component in three directions \( x, y, z \) of a Cartesian coordinate system, \( \Gamma_{\phi,eff} \) is the effective diffusion coefficient, \( S_\phi \) is the source term. With different values for \( \phi \), the above equation can represent continuity, momentum, energy, turbulence, water vapor content and contaminant concentration equations. The turbulence model, involved in this study, is the RNG k-\( \varepsilon \) equations model proposed by Yakhot et al. [16]. Chen [17] and Zhang et al. [18] have compared a couple of eddy-viscosity models in modeling indoor airflow and concluded the RNG k-\( \varepsilon \) model behaves generally best.

CO2 and moisture are assumed as a passive tracer gas for simplicity to model their dispersion in the cabin. Water vapor is assumed as an ideal gas, so concentration of moisture content (density) can be easily converted into its partial pressure by following the ideal gas law,

\[
p_w = \rho_wRT
\]

where \( p_w \) is the partial pressure of water vapor, \( \rho_w \) is the moisture density, \( kg/m^3 \); \( R \) is the gas constant, 462 J/(kg K); and \( T \) is the absolute temperature, K. The water saturation pressure with respect to temperature can be determined with the ASHRAE recommended formula [19] as

\[
\ln p_{ws} = C_8/T + C_9 + C_{10} T + C_{11} T^2 + C_{12} T^3 + C_{13} \ln T
\]

where \( p_{ws} \) is the water saturation pressure, \( Pa \); \( T \) is the temperature, \( K \); and

\[
C_8 = -5.8002206E + 03;
\]

\[
C_9 = 1.3914993E + 00;
\]

\[
C_{10} = -4.8640239E - 02;
\]

\[
C_{11} = 4.1764768E - 05;
\]

\[
C_{12} = -1.4452093E - 08;
\]

\[
C_{13} = 6.5459673E + 00.
\]

Table 1

| Item                          | Under-aisle system | Mixing system |
|-------------------------------|--------------------|---------------|
| Supply air flow rate (l/s) per person | 1.428 | 8.572 | 10 |
| Supply air velocity (m/s)     | 0.2083 | 0.09375 | 2.129 |
| Supply air temperature (°C)   | 24.0 | 22.0 | 19.5 |
| Supply CO2 concentration (ppm)| 350 | 933 | 850 |
| Supply water vapor content (kg/m³) | 0 | 0.004848 (RH: 25%) | 0.001007 (RH: 6%) |
| Central ceiling temperature (°C) | 25.5 | 25.5 | 25 |
| Side ceiling temperature (°C) | 25.5 | 22.0 | 23 |
| Side wall temperature (°C)    | 23 | 23 | 18 |
| Window temperature (°C)       | 20 | 20 | 13 |
| Central deck temperature (°C) | 23 | 23 | 24 |
| Side deck temperature (°C)    | 22 | 22 | 23 |
| Passenger surface temperature (°C) | 30.3 | 30.3 | 30.3 |
| Lighting heat generation rate (W/m²) | 12.5 W/row | 12.5 W/row |
| Seats                         | adiabatic | adiabatic |
| Front and rear cross sections  | periodic | periodic |
Hence relative humidity can be easily calculated in the post processing once the moisture density is solved from CFD modeling.

Three distinct numerical solution techniques are available for CFD: finite difference, finite element, and finite volume method. The finite volume method is the most well-established and widely-used in commercial CFD codes due to a clear relationship between the numerical algorithm and the underlying physical conservation principle. This investigation has thus used the finite volume method. The finite volume method divides the domain into many CFD cells, and then integrates the governing equations over all CFD cells (control volumes). The integral equations are then discretized with a variety of finite-difference-type approximation converting the integral equations into a system of algebraic equations. As for boundary cells, the standard wall function method is used to simplify the solution. Finally, these algebraic equations are solved with suitable solvers via the SIMPLE method.

This study used a commercial CFD software, GAMBIT, to build the geometry domain of the cases and generate the cells for simulation in the solver, FLUENT. Both combined structured (hexahedral grids) and unstructured (tetrahedral grids) meshes were created in both cases using the “Tet/Hybrid” scheme with “Hex core”. The grid size near the air supply region is around 0.01 m, while in the other regions grid distance is about 0.08 m. A size function was applied to change grid size gradually therein. The grid number for both air systems are 525,388. To check if the grid-independent results have been obtained, the finer grids with a total number of 993,644 were also tested but did not show meaningful differences. The flow and temperature distribution were solved first and then the flow was frozen for CO₂ and moisture content solution. The continuity and momentum equations were thought to reach convergence when the ratio of the sum of the mass gain and loss on all boundaries to the overall mass gain in the cabin was less than 1.0e-6. In a similar method the convergent ratio limit for energy was 3.0e-3, and 1.0e-6 for CO₂ and moisture.

4. Validation of the CFD program

The RANS CFD modeling employs a significant amount of assumptions, so it is crucial to validate the CFD program and also the users to ensure reliable results have been obtained [20]. Due to lack of published quality data for displacement ventilation in aircraft cabins, this validation process uses the air flow, temperature, and contaminant concentration data obtained from a small workshop served by under-floor displacement ventilation as an alternative. The flow characteristics in the enclosed workshop and aircraft cabins should be similar.

There are four occupants mimicked by box-shaped manikins doing assembling work in front of two desks in this workshop as shown in Fig. 3. The heat boxes dissipate heat to the air simulating heat generation from the assembling line. Six fluorescent lamps are mounted on the ceiling to provide illumination. Conditioned cool air is supplied through four under-floor perforated panels, and then extracted to the outside through a square ceiling exhaust. A contaminant source, mimicked by a tracer gas, sulphur hexafluoride (SF₆), is introduced to the head level of one occupant to quantify performance of the displacement ventilation in removing the pollutants. More details on the measurement can be found in the literature [21].

The validation employed the RNG k-ε turbulence model. Due to limited space available for this paper, the quantitative comparison for air velocity, air temperature and contaminant concentration was only presented at the center of the room on a vertical pole as shown in Fig. 4. Air velocity in the room was maintained at very low level but reaching around 0.2 m/s at the top of the room because the central pole is very close to the exhaust, where air motion is strong. The simulated air temperature is generally lower than the measurement 1 °C due to unknown reason, but the simulated concentration matches the experimental data very well. In general, the agreement between the CFD results and the experimental data is excellent both for air velocity and concentration, and acceptable for temperature. Note the accuracy of the anemometers for velocity measurement is 0.02 m/s with 1% error, for air temperature is 0.2 °C with 1% error, and the resolution of
the tracer-gas measurement is 0.01 ppm and the accuracy is 1% of the measured values.

The quantitative comparison of results in other positions is similar, and hence results are not listed in this paper for brevity. As the test data for moisture distribution are not available, the moisture modeling has not been carried out. However, because both contaminant and moisture can be modeled as a passive tracer gas, the transport characteristics should be similar. This validation therefore concludes the CFD program together with the users is capable to provide reasonable results for enclosed environment served by under-floor displacement ventilation.

5. Results and discussions

Similar to the validation process, after solving the cases the new air system in terms of moisture content distribution, moisture condensation risk on cabin linings, capability in removing internally generated pollutants, and thermal comfort was analyzed.
5.1. Distribution of water vapor content

Fig. 5 presents the profiles of inhaled water vapor content in terms of relative humidity on five vertical poles for the passengers seated in one side of the cabin in the third row. As highlighted from the top view of the cabin in Fig. 5, poles P1, P2, P4 and P5 are across the passenger thighs and seats in directly front of the passenger torsos with 5 cm, while P3 is in the aisle. Since P1 and P2 are at the window side, the both poles are shorter. When calculating the inhaled moisture content, we have assumed a passenger does not inhale the moisture released by himself or herself. That means the inhaled water vapor content profiles for a passenger were plotted when temporarily turning off the moisture generation of that passenger. Generally, as shown in Fig. 5 the mixing system holds uniform relative humidity of around 10%, which falls within the range of the measured values on airplanes [9]. Such low level of humidity may lead to discomfort complain. However in the new system, after humidification the humidity level is much elevated.

Fig. 8. Comparison of inhaled CO₂ concentration between the under-aisle system and the mixing system at the five vertical poles: red solid lines for the under-aisle system and black dashed lines for the mixing system.

Fig. 9. Distribution of CO₂ concentration in the under-aisle air system: (a) in a cross section across the passengers in the third row, (b) in the mid longitudinal section.

Fig. 10. Distribution of CO₂ concentration in the mixing air system: (a) in a cross section across the passengers in the third row, (b) in the mid longitudinal section.
reaching 20% on average. In the aisle along P3, relative humidity is slightly higher because the dry bulb temperature of air is lower in the aisle and thus a little smaller for the saturation pressure. Note, the study only considers moisture generation from human respiration and neglects moisture generated from human skins during sweating or water evaporation from drinks.

To show if there is any condensation on the cabin interior, Fig. 6 presents the distribution of relative humidity in the whole cabin for the under-aisle system. The region below the seats generally holds relative humidity of 25% because humid air is supplied from the both under-aisles. Above the passenger thighs, relative humidity reduces due to increasing air temperature with height in the displacement ventilation system, and thus elevating the saturated water pressure. Above the heads, because of respiration and the associated moisture generation, relative humidity increases. By observing the moisture content along the interior surfaces, none of locations reach saturation, so there should be no risk of moisture condensation on cabin linings. The highest relative humidity (less than 50%) is on the cabin ceiling as the moisture rises with air streams to the ceiling. The dry air supplied from the both channel inlets also helps reduce the moisture condensation as shown in Fig. 6(a). Note at the both sides of the cabin and even on the windows that are usually at low temperature, the air is far from reaching saturation. This is in benefit of the elevated temperature on the airplane interior linings due to warm air running through both channel passages as that shown in Fig. 1(b).

As a comparison, Fig. 7 also presents the distribution of relative humidity in the mixing system. The moisture distribution in the mixing air system is very uniform and the whole cabin is maintained very dry. However, against the current air system, the boost of relative humidity in the under-aisle air system is around 10%. Nagda and Hodgson [7] strongly recommend to increase 5–10% of air humidity on airplanes to alleviate the dryness related symptoms, and claim such enhancement of humidity is unlikely to induce microbial growth because there is no moisture condensation in the cabin interior. On the other hand, by calculating the moisture balance according to the design parameters provided from Table 1, the water consumption rate used for humidification is about 0.05 kg/h per person, which should be affordable by the airliners. For example, for a commercial airplane carrying 300 passengers/crews for an 8-hour flight, an extra weight of 120 kg water is needed. The water load corresponds to one additional passenger weight (60 kg) since the water is continuously consumed during the flight. This research therefore concludes the humidification of aircraft cabins with the new under-aisle system is feasible.

5.2. Distribution of CO2 concentration

In order to evaluate the under-aisle system in removing the occupant-generated pollutants, the distribution of CO2 concentration was analyzed. Fig. 8 shows the inhaled CO2 concentration profiles at the same five poles. Again, when calculating the inhaled CO2 concentration, the CO2 source corresponding to each passenger was turned off to account for the impact of CO2 generation only by the other passengers. The CO2 concentration in the inhaling regions (Z = 1.1 m) for the new system is around 930 ppm, which is much lower than the mixing system that reaches around 1500 ppm. The inhaled concentration in the new system is close to that at the under-aisle air supply, indicating that cross air motion among passengers is very weak, so that the exhaled CO2 from one passenger is not transported to the inhalation region of the other passengers. The concentration values for the four passengers in either respective system are more or less the same. Even for P3 in the aisle, the inhaled CO2 at Z = 1.7 m in the under-aisle system is
around 1200 ppm, whereas the mixing system approaches 1600 ppm. So the under-aisle system still can maintain good air quality for the crew members who occasionally walk around in the aisles. It therefore concludes under the same outside air rate and recirculation ratio, the new under-aisle system can lessen the inhaled CO₂ concentration by about 30% as compared with the current mixing system.

To better illustrate the dispersion of exhaled pollutants in the whole cabin, Fig. 9 shows the CO₂ distribution in the same two sections as those in Fig. 6. Similar to the moisture content distribution, with the rising air streams the plumes of CO₂ generated by each passenger goes up to the ceiling and are diluted by the surrounding air. Below the exhalation height, the CO₂ is maintained at a relatively low level without being contaminated by the inside passengers. Above the head level, CO₂ concentration rises rapidly. The cross mixing among the passengers in the same row (Fig. 9 (a)), or for passengers seated in different rows (Fig. 9 (b)) is very weak. The conditioned, outside air supply from both channel inlets helps somewhat dilute the CO₂ concentration at the both sides of the cabin.

As a comparison, Fig. 10 presents temperature profiles at the five positions. Since these vertical poles are very close to the passengers (in front of the passengers with a distance of 5 cm), temperature profiles between the two systems are similar with the dominant impact of metabolic heat dissipation. Combined with velocity distribution in Fig. 11, the draught risk for the passenger seated near the window in the mixing system both in the head and ankle level is very high. This was also illustrated in the distribution of percentage of dissatisfied (PD) resulted from draught as shown in Fig. 13(a), where the PD reaches around 30% at the head level and around 20% at the ankle level.

5.3. Distribution of air velocity, temperature and draught risk

An air distribution system shall create a thermally comfortable environment for the passengers and crews. As mentioned in the previous sections, the draught risk has been a major obstacle for a new air distribution system being accepted by the passengers. The following details our analysis of the under-aisle air system in terms of thermal comfort and draught risk.

Fig. 11 compares air velocity profiles on the five poles between the two systems. In general, velocity magnitude in the new system is maintained at levels less than 0.2 m/s. This is because the driving force for air motion in the under-aisle air system is buoyancy generated by heat sources. Velocity increases slightly with height until $Z = 1.3$ m on P1, P2, P4 and P5, due to the continuous heating of the surround air by warm human bodies. In the aisle, air motion is even weaker since there is no heat source. However, air motion in the mixing system is much stronger, especially near the ceiling and deck level. As P5 is in the middle of the cabin, which falls in the air recirculation zone, the air motion is not so evident. The high velocity on P1 and the associated turbulence (with intensity of around 9%) at the head level may impose draught risk for the passenger seated near the window. More detailed analysis of the draught risk is in the latter part.

Fig. 12 presents temperature profiles at the five positions. Since these vertical poles are very close to the passengers (in front of the passengers with a distance of 5 cm), temperature profiles between the two systems are similar with the dominant impact of metabolic heat dissipation. Combined with velocity distribution in Fig. 11, the draught risk for the passenger seated near the window in the mixing system both in the head and ankle level is very high. This was also illustrated in the distribution of percentage of dissatisfied (PD) resulted from draught as shown in Fig. 13(a), where the PD reaches around 30% at the head level and around 20% at the ankle level.
level. The PD was calculated using the formulas recommended by the ASHRAE Fundamentals [19]. It may also have small draught risk on P2, P3 and P5 at the ankle level. However, in the under-aisle system, due to very weak air motion and also very low level of turbulence (generally less than 6%) inside the cabin, the draught risk is very minimal as can be observed in Fig. 13(b), though slightly cool air (22 °C) is supplied through both under aisles. This analysis therefore concludes there is no draught risk in the new under-aisle air distribution system.

Before concluding this paper, the authors would like to remind the readers that this preliminary study only presents the concept of a new air system from the insight of cabin air quality and thermal comfort. More research efforts, such as on the integration of the new system with aircraft structure and payload as well as the related economic analysis, are extremely required. With respect to the cabin wall, the insulation panel and the fuselage hold lower temperature than the lining during cruise, which may impose greater risk of moisture condensation. The study of condensation within the cabin wall itself is out of the scope of this paper. The modeling of flow and heat transfer in the channel passages also awaits more exploration.

6. Conclusions

In order to enhance cabin humidity level and reduce air mixing, a new under-aisle air distribution system was proposed by simultaneously supplying humidified air from the both under aisles and warm dry air through the channel passages below the stowage bins. CFD was applied to investigate the performance of the new system after successfully validating it in a workshop environment with under-floor displacement ventilation. The new system was found being capable of boosting the relative humidity from 10% to 20% and can lessen the inhaled CO2 concentration by 30% as compared with the current mixing air system. There should be no condensation on cabin interior linings because of the elevation of cabin side temperature with warm air running through the channel passage, and the dry air supplied from the both channels also helps prevent condensation. The draught risks are minimal on benefit of conditioned air supply from both under aisles and weak air motion created inside the cabin. The water consumption in air humidification is only around 0.05 kg/h per person, which should be affordable for airliners. Therefore, with comprehensively good performance based on this analysis, the new under-aisle air system is recommended for possible use on future airplanes.

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