RESEARCH ON INDOOR ENVIRONMENT FOR THE TERMINAL 1 OF CHENGDU SHUANGLIU INTERNATIONAL AIRPORT

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ABSTRACT

In this paper, indoor thermal environment of the Terminal 1 of Chengdu Shuangliu International Airport (CSIA) is taken as study object, by means of investigating, measuring and simulating based on computational fluid dynamics (CFD) codes. A total of 569 participants provided 569 sets of physical data and subjective questionnaires, and an indoor thermal comfort meter was used to collect the measured parameters of the indoor environment and the predicted mean vote (PMV). The accepted thermal comfort environment range is obtained. Results considering personal mobility obtained from measuring a number of points have been compared and analyzed, which provides some drawbacks for airflow distributions. Main characteristics of vertical temperature distributions in winter and summer conditions are discussed, measured by the equipment named scaling ladder. The comparison held measuring and CFD results generally shows a good agreement, which confirms that the veracity of the CFD simulation can be achieved. The results are of great importance for designing indoor thermal environment and air conditioning systems of the Terminal 2 of CSIA.

INTRODUCTION

The condition of indoor thermal environment in terminal buildings has attracted extensive attention increasingly, as the uncertain factor, people's density, people's mobility, people's stay time, tending to be influenced for weather and so on. At present, there are few researches about indoor thermal environment for large space building such as airport building, and fewer measurements especially due to the special safety. Airport terminals are among the major transportation facilities. These types of buildings have distinct operational and architectural characteristics.

Balaras (2003) performed detailed analysis using thermal simulations for assessing specific measures to reduce energy use without compromising comfort in Hellenic airports, and to identify possible actions for improving indoor environmental quality.

A number of studies of simulating the air distribution and thermal comfort environment in the terminal buildings have been carried out so far by Computational fluid dynamics (CFD) codes. Yau Raymond (1991) used CFD and reduced-scale physical modeling to optimize the air distribution design in a large airport terminal. Meng and Chen (2007) simulated the potential of natural ventilation and air modulation by mechanical fans in different plans under typical meteorological conditions of Sanya in summer. Neofytou et al. (2006) investigated the wind environment around a large airport terminal building, in which CFD simulations have been performed. Vrachopoulos (2006) develop a model based on the numerical solution of the three-dimensional flow field of the Athens-Greece airport train station using CFD.

In the last two decades, a significant number or scientific papers exists dealing with the application of CFD models in various indoor environments, especially large space buildings, such as museums, lecture theatres, gymnasium, large industrial premises. Papakonstantinou (2000) assessed the environmental conditions inside the main Hall of the National Archaeological Museum of Athens with external conditions, during summer and winter days in large enclosures. Cheong (2003) evaluated the current thermal comfort conditions of an air-conditioned lecture theatre in a tertiary institution using objective lecture theatre in the tropics measurement, CFD models and subjective assessment. Rohdin (2007) predicted the flow pattern and temperature distribution in this large industrial facility.

Thermal stratification is common phenomenon in many large space buildings, particularly atria, gymnasium and so on. During the heating season, the presence of a warm air layer below the ceiling will increase the roof heat losses and may increase the stack effect. The latter will increase air infiltration and exfiltration. More heat would be required to maintain comfort temperature in the working zones. The higher temperature near the ceiling may also affect the performance of some lighting fixtures and optical fire detectors (1996). Huang (2003) provided a series of numerical simulation result based on the site measurement and discusses thermal environment in a large space building. Gao (2006) constructed a zonal model to predict the vertical temperature profiles of large enclosures under such a system combined stratificated air conditioning and natural ventilation. Huang (2007) carried out the site-measurement in a Stadium during summer, winter, and the transitional seasons, including outdoor environment and indoor air temperature distribution, and the heat balance of air-conditioning system, etc.

In this paper, indoor thermal environment of the Terminal 1 of Chengdu Shuangliu International Airport (CSIA) in winter and summer conditions is taken as study object, by means of investigating, measuring and simulating based on computational fluid dynamics (CFD) codes.

DESCRIPTION OF THE AIRPORT TERMINAL BUILDING

The plane section of terminal 1 building of CSIA is shown in Figure 1. The building can hold 3500 persons at rush hour, which has the height of 30.8 m, the length of 60 m. The passengers can buy air ticket and check-in, as well as take a rest in check-in hall. The indoor scene of the check-in hall is shown Figure 2. The glazing system is used in the front windows and less than 20% of the roof. The application of external sunshade devices over windows can ameliorate the indoor environment and reduce the energy consumption of air-conditioning in summer, which can greatly decrease solar radiation.

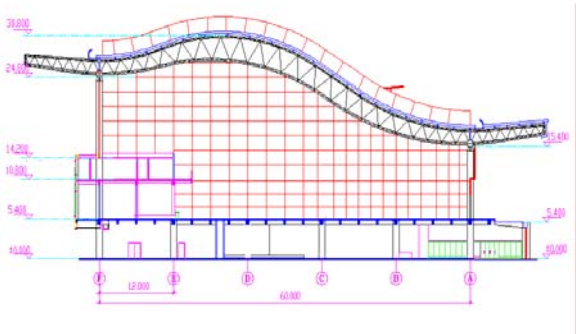


Figure 1 Plane section of the building

Figure 2 shows the indoor scene of the check-in hall. The air is delivered from upper-side nozzles installed at the wall with the height of 4 m. The return air goes out mainly through the return intakes fixed near floor and below nozzles. There are 130 supplying nozzles with an inclination of 15° downwards to supply air for passenger zone, fixed according to the distance of 3.5 m and 5.5m

Figure 2 Indoor scene of the check-in hall

OUTLINE OF SURVEY AND NUMERICAL METHOD

Questionnaire of Thermal and Humid Sensation

Subjects were asked to complete a questionnaire at the same time as the environmental variables (air temperature, globe temperature, surface temperature, air velocity and relative humidity) were being recorded. Details of clothing and activities were noted for each subject. The subjective study involved collecting data using questionnaires (Ealiwaa, 2001). The questionnaire is based on six sections: background and personal information, social interaction, thermal environment and personal influences, passengers' perceptions of the environmental conditions in the whole building, passengers' thermal comfort, people's general feeling and personal well being.

For the purpose of this paper, only the passengers' thermal sensation data have been presented including data about the sensation of air velocity, air relative humidity and thermal comfort. The passengers' thermal comfort has been tested using 7-point ASHRAE sensation scale, ranges from −3 as cold to +3 as hot and 0 as neutral. In addition, preference and satisfaction scales have been used. The subjects were selected randomly from different groups of people indoor to represent typical range of samples. Care was taken to minimize the risks of misunderstandings arising from translations of the words describing the points on the various scales by interviewing the respondents and assisting them with completion of the questionnaires.

Thermal Comfort Measurement

Unlike questionnaire, the environmental variables (air temperature, globe temperature, surface temperature, air velocity and relative humidity) were recorded automatically, using an equipment named indoor thermal comfort meter. Air temperature and air velocity were measured at the 1.7 level, representing the immediate environment of occupants' head. Relative humidity and radiant asymmetry were measured at the 1.2 level. The distributions of the site-measurements indoor are shown in Figure 3.

The data calculated every minutes use Fanger's predicted mean value (PMV) index to describe the expected degree of thermal comfort. This is because thermal comfort is influenced by many variables such as, temperature, relative humidity, air velocity, environment radiation, activity level and cloths insulation. The equipment is shown in Figure 5. The activity level and cloths insulation may set up four different values to calculate the corresponding PMV index, showing different passengers' thermal sensation. Table 1 (Matzarakis, 2007) shows the relationship between PMV, thermal sensation and physiological stress level.

Vertical Temperature Measurement

The vertical air temperatures were measured, using a more sensitive globe thermometer in order to record more data that are accurate. These values were logged every 3 minutes and every 0.5 meter, whose scale of vertical direction ranges from 0 to 10 meters (away from the ceiling). Scaling ladder, which can manual controlled easily, was used to test different temperature at given height. The equipment is shown in Figure 5. For safety consideration, the measure sites were away from the passenger lounge area as far as possible. The distributions of the site-measurements indoor are shown in Figure 4.

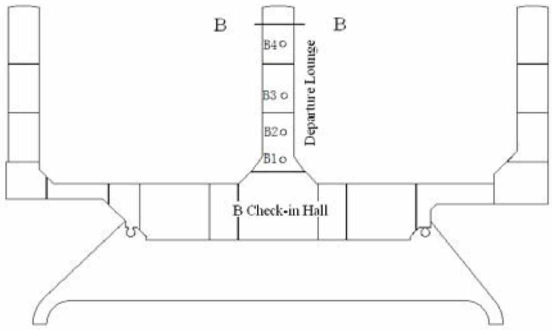


Figure 3 Indoor Measurement Points of Terminal Building

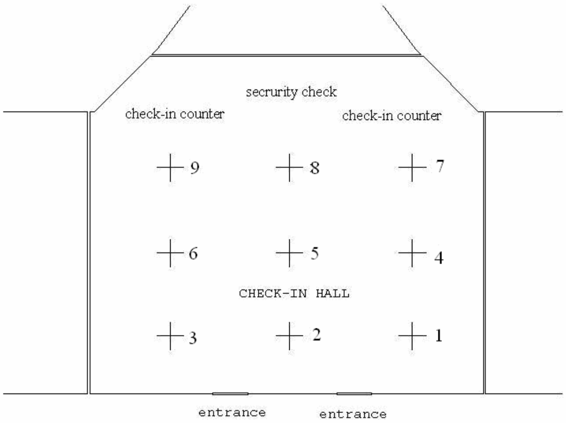


Figure 4 Distributions of the site-measurements of Check-in Hall

Table 1 Relationship between PMV and thermal sensation

PMV	Thermal sensation	Physiological stress level
	Very cold	Extreme cold stress
-3.5 -----------		
	Cold	Strong cold stress
-2.5 -----------		
	Cool	Moderate cold stress
-1.5 -----------		
	Slightly cool	Slight cold stress
-0.5 -----------		
	Comfortable	No thermal stress
0.5 -----------		
	Slightly warm	Slight heat stress
1.5 -----------		
	Warm	Moderate heat stress
2.5 -----------		
	Hot	Strong heat stress
3.5 -----------		
	Very hot	Extreme heat stress

Figure 5 Thermal comfort meter

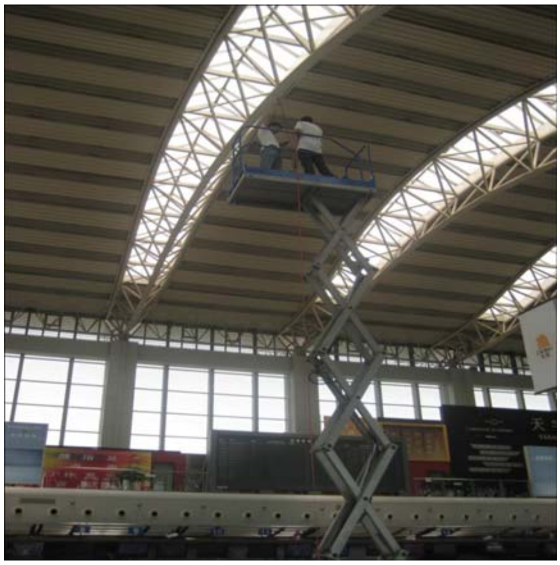


Figure 6 Scaling ladder

Numerical Simulation Method

Numerous investigators have found numerical simulations to be valuable tool for room ventilation design and with the ever increasing computational power of computers, reliance on simulations for indoor air flow prediction will increase. Two different representations of the fluid flow in the room are often used, turbulent flow using the standard $k-\varepsilon$ turbulence model, and turbulent flow using the renormalization group (RNG) $k-\varepsilon$ turbulence model.

In a relatively recent study, the standard $k-\varepsilon$ model has proven very successful for numerous engineering applications. However, certain characteristics of indoor airflow, such as the creation of regions with very low velocities and thus low Reynolds numbers, especially near the wall boundaries, require the use of more effective models (Stamous, 2006) Although the standard and RNG

$k-\varepsilon$ models are similar, there are some important differences besides the constant values used in them, as discussed from (Chen, 1995, Posner, 2003, Yakhot, 1986). These potential improvements and freedom from the need of calibration make the RNG $k-\varepsilon$ model attractive for engineering applications. Chen (1995) compared the performance of five different models for simulating simple indoor airflows and found that the standard and RNG $k-\varepsilon$ predicted actual flow patterns best. The RNG $k-\varepsilon$ was found to perform slightly better than the standard model in some situations.

The code solves the following conservation equations for mass, momentum and enthalpy for steady, incompressible turbulent flow:

$$\frac{\partial u_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j u_i\right) = \frac{\partial}{\partial x_j}\left(-p_0\delta_{ij} + \left(\mu+\mu_t\right)\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right)+\rho g_i \tag{2}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j He\right) - \frac{\partial}{\partial x_j}\left(\left(\frac{\lambda}{C_P}+\frac{\mu_t}{\delta_{He}}\right)\frac{\partial He}{\partial x_i}\right) = 0 \tag{3}$$

The Boussinesq approximation is applied in which the density in momentum equation is written as:

$$\rho = \rho_0\left[1-\beta\left(T-T_0\right)\right] \tag{4}$$

The turbulent viscosity μ_t is determined using a two-equation $k-\varepsilon$ model. Based on the findings of previous work (Cook, 1998, 2000, 2005), the RNG $k-\varepsilon$ turbulence model was employed.

Gambit (2006), an automatic and unstructured three-dimensional mesh generator with an adaptive mesh refinement algorithm is employed. The numerical grid consists of approximately 3100 000 unstructured elements (tetrahedrons and hexahedrons) with grid refinement in the inlet and outlet regions. During the detailed measurements, the boundary temperatures were measured continuously to ensure that the temperature was stable during the measurements. The temperature of the floor, the ceiling, the northern, the southern walls are set equal to 19.2℃, 18.4℃, 20.6℃, 24.1℃ in winter condition, respectively. Additionally, no radiation model is used. Heat fluxes are modelled to represent the actual amount of heat generated by the types of heat sources indoor.

There are various efficient computer codes, which are frequently used for indoor CFD calculations. In the present work, the latest version of the commercial CFD package Fluent (2006) is employed for the prediction of air stratification. The SIMPLEC algorithm is employed as the pressure-velocity coupling method. The convergence criteria have the default set in Fluent code. To research converge as far as possible and avoid vibrating, under-relaxation factors can appropriately be decreased.

DISCUSSION AND RESULT ANALYSIS

Indoor Environment Measurement

The surveys were accompanied by measurements of the relevant thermal comfort parameters at the height of 1.5 meter using indoor thermal comfort meter during the time the outdoor data for the site were recorded continuously, from January 7th to 10th and July 23rd to 30th. Additionally, the measuring period was concentrated on two hours at noon, as the typical outdoor environment is significant for analytical results. During this survey, the outdoor temperature ranges, which were only 6℃ difference from 2.0℃ to 8.0℃in winter, and 8℃ difference from 24℃ to 32℃, show small difference for measuring.

Table 2 shows the indoor environment parameters of measuring points against air velocity, air temperature and air humidity in check-in hall and departure lounge in winter and summer. It is obvious that the resulting mean air temperature, velocity, humidity or measurement point in check-in hall is higher than departure lounge in winter and lower in summer, as a great many passengers in check-in hall and wind infiltration from outdoor in two seasons.

It also shows that resulting mean air velocity of point 9 close to check-in counters is about 0.22 m/s in winter, and the air temperature varies between 22.4℃ and 23.2℃. It is obvious somewhat uncomfortable for the passengers who were waiting for check-in, as their PMV index computed according to above data was merely 0.5. The passengers had to take coat off in order to keep comfortable. This drawback of airflow mainly attributes to inaccurate design, e.g. the height of air supplying nozzles, the parameters of supply air.

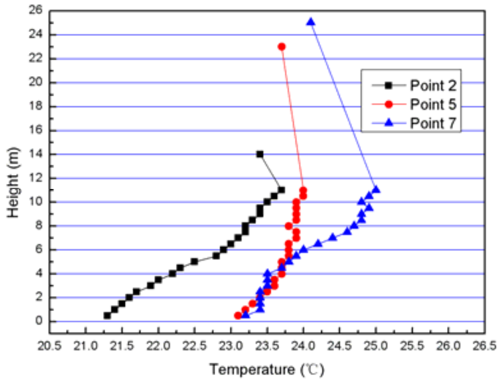
Table 2 Indoor environment parameters

		Check-in Hall		Departure Lounge	
		Summer	Winter	Summer	Winter
Air Temperature (℃)	Maximum	28.6	23.8	28.0	25.1
	Minimum	24.5	19.1	25.4	20.2
	Average	26.7	20.5	26.3	22.4
	Standard Deviation	0.23	0.25	0.13	0.22
Air Velocity (m/s)	Maximum	0.61	0.53	0.94	0.47
	Minimum	0.12	0.08	0.02	0.01
	Average	0.39	0.18	0.33	0.16
	Standard Deviation	0.09	0.15	0.21	0.18
Air Humidity (%)	Maximum	78.0	68.9	69.5	67.2
	Minimum	55.1	53.3	53.1	52.8
	Average	62.5	59.7	58.6	57.4
	Standard Deviation	5.21	3.94	2.16	2.36

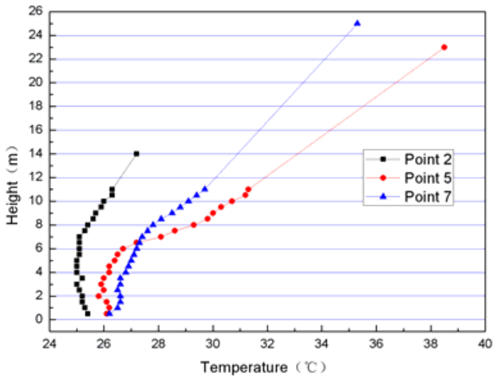
In addition, indoor thermal comfort can be acceptant, according to PMV index ranging from -0.5 to 0.5, but air diffusion performance index (ADPI) is no more than 0.6 in winter and summer, mainly owing to airflow characteristic resulting from personal mobility and higher supply air velocity and lower supply air altitude. As a result, the indoor airflow is unsteady, and the values of ADPI change with time. However, as mentioned from Liu (2007), considering variation of the airflow field was small in the temperature preserving phase, both values can achieve stabilization (vary slightly) at one time. Here, this time is titled as "stabilization time".

Indoor Vertical Temperature Distributions

The measured results indicated that vertical temperature distribution of the measurement point 2, 4, 5, 6 and 7 in Check-in Hall and Departure Lounge were fairly familiar. The vertical temperature curves of these points varying with height in two seasons are shown in Figure 7, Figure 8 and Figure 9.

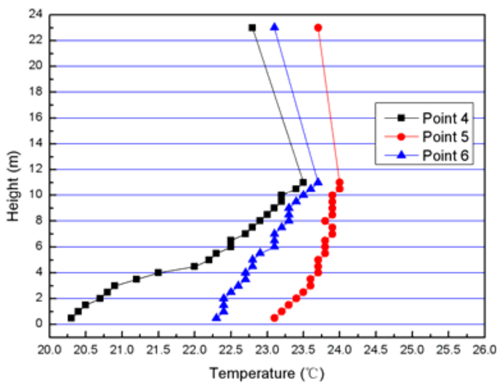


(a) Winter

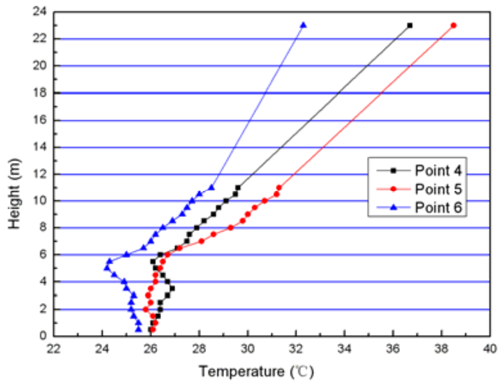


(b) Summer

Figure 7 Vertical temperature of measurement Point 2, 5 and 7 in Check-in Hall

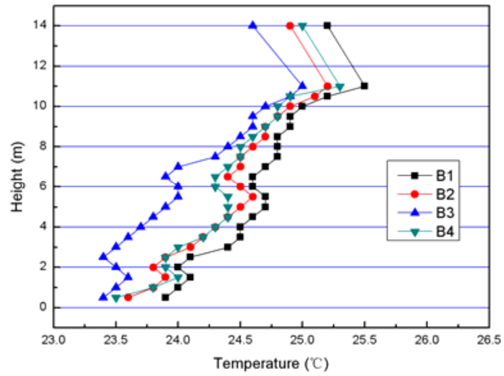


(a) Winter

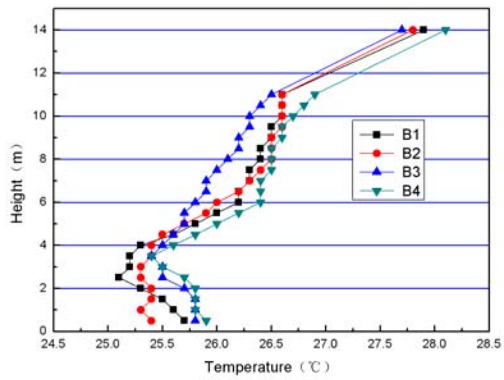


(b) Summer

Figure 8 Vertical temperature of measurement point 4, 5 and 6 in Check-in Hall



(a) Winter



(b) Summer

Figure 9 Vertical temperature of measurement Point 2, 5 and 7 in Check-in Hall

In Figure 7, the vertical temperature of point 2 at the entrance of check-in hall was easily affected by outdoor environment in winter and summer, as the automatic control door may be always open because of a great many passengers. Porch can be used to solve problems due to its function of obstructing from storm, resisting chilliness. Whereas the point 5 locates in the centre of Check-in Hall, there is hardly hot air in occupant zone.

In Figure 8, the vertical temperature of Point 4 and Point 6 was hardly affected by supply air, as the height of supply air is too high to reach lower zone close to supply nozzles, and there are a great deal of machines in occupant zone.

As shown in Figure 9, it was the most obvious temperature variation in different region of Departure Lounge, which results from lower supply air altitude at the height of 3.0 meter, and upper large accumulated heat loads from the height of 6.5 meter to ceiling. In addition, there is another reason of no upper opening at the ceiling.

In addition, the ceiling temperature is extremely high, especially in noon in summer. The solar radiation has great influence to indoor environment, like as the wall temperature near inside wall. The ceiling temperature of Point 4, Point 5 and Point 7 was higher than other point for solar radiation.

In winter, the temperature gradient was larger along with the height, so that the supplied hot air was hard to reach the occupant zone because of buoyancy. Therefore, in thermal environment and air-conditioning design of large space building, special care should be taken to achieve efficient energy-saving, easy-to-fix airflow, which the availability in both winter and summer(Huang, 2007)

Questionnaire of Indoor Thermal Environment

During the study, which were carried out in January 2008 and July 2008 over a period of two week, the results of the field study on thermal comfort in CSIA were analyzed and summarized. Thermal sensations for different temperatures were discussed and their thermal neutral temperatures were given. 569 sets of questionnaire responses were obtained from passengers and employees in winter and summer. 95.8% of the passengers considered their thermal conditions can be acceptable. The high acceptance rate for thermal environment shows the indoor thermal conditions are highly appropriate and the passenger's adaptive ability is very powerful.

The results show that 78.33% of passengers were generally satisfied with indoor thermal environment. Mean thermal sensation (MTS) predicted model based on regression analysis was made to predict thermal sensation, and to calculate thermal neutral temperatures. The linear regression equations are fitted for these binned thermal sensations against operative temperature:

$$\text{Winter: } MTS=0.211t_0-4.513 \quad (5)$$

$$\text{Summer: } MTS=0.502t_0-12.952 \quad (6)$$

Where t_0 is the operative temperature.

The neutrality is derived by solving the equations for the MTS of zero, and the neutral operative temperatures in winter for the passengers is 21.4℃, and the comfort zone is 19.2~23.1℃; the neutral operative temperatures in summer for the passengers is 25.6℃, and the comfort zone is 23.9~27.3℃. The correlation coefficient between the MTS votes and the operative temperature for Eq. (5) and (6) are 0.5706 and 0.6376. Figure.5 and Figure.6 shows calculation of thermal neutral temperatures in winter and summer. The results are little difference with the indoor comfort zone (ASHRAE Standard 55-1992) and (ISO Standard 7730) for winter and summer conditions.

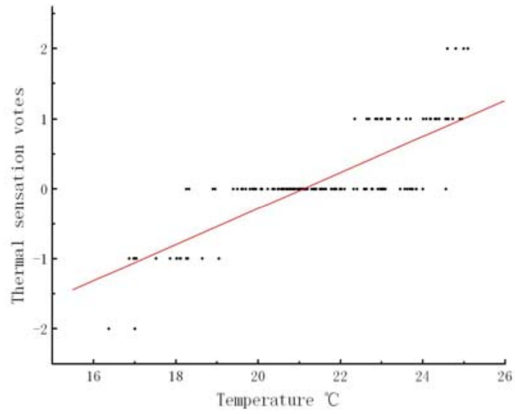


Figure 10 Calculation of thermal neutral temperatures in winter conditions

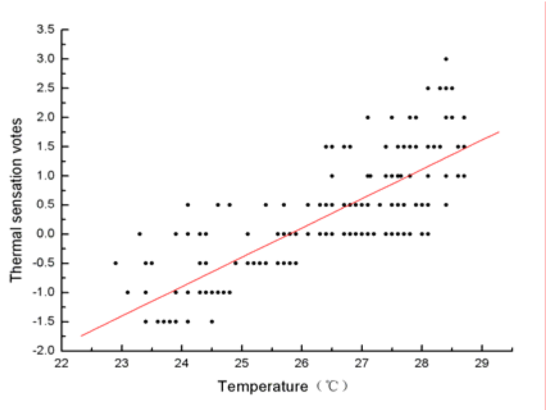
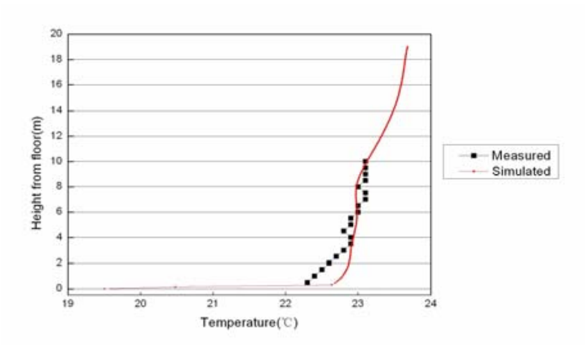


Figure 11 Calculation of thermal neutral temperatures in summer conditions

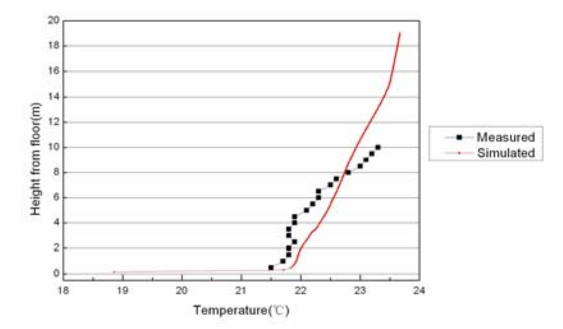
Compared Measurement and CFD Simulation of Vertical Temperature in Check-in Hall

This section presents a comparison between the measured vertical temperatures and the results of the simulations with the RNG turbulence model. The aim is not only to validate the CFD simulation but also to provide guidance to the project Terminal 2.

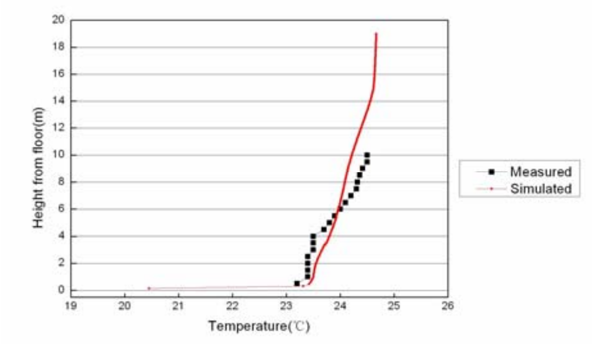
The CFD results and the experimental results of vertical air temperature profiles in winter conditions at three measurement points (point 5, 8, and 9) are compared in Figure 12. The figures show that the CFD results agree well with the measurement results although the maximal error reaches 2.4℃ at the entrance, which mainly because heat load from passengers is uncertain and fluctuant.



(a) Point 5



(b) Point 8



(c) Point 9

Figure 12 Compared CFD results with measurement results of vertical air temperature profiles

Except that the temperature near the floor in winter was slightly different, the other simulated temperature was similar to that of the occupant zone measured. As shown in Figure 12, the measured temperature varies from 19.5℃ at the height of 0.5 meter to about 24.5℃ at the height of 10 meter upper. The vertical temperate is approximately linear distribution, but changes gently, which is because the ceiling and upper wall in winter conditions was in the state of decalescence, and the air temperature upper zones is not liable to increase. There is a main character of vertical temperature distributions, which is no obvious thermal stratification at the height of nozzle in winter conditions.

However, the high ceilings and upper-walls in the present building are not a successful design, because of the standards that regulate ventilation in this type of facility often use the concept of air changes per hour, which results in the use for large quantities of supply air. In addition, large amount of heat upper zones radiate lots of thermal, which is harmful to human health. These in turn lead to a substantial energy use and health risk, as was discussed (Rohdin, et. al, 2007). Therefore, ceiling openings or upper-wall openings should be taken into account for the Terminal 2 of CSIA.

CONCLUSION

1. The result analyzed by means of measuring and investigating is of great importance for designing thermal environment and air conditioning in the Terminal 2 of Chengdu Shuangliu International Airport.
2. Measurement and simulation on thermal environment is important methods to design an air-conditioning system in large space with better thermal performance and energy saving. In this study, CFD results agree well with the measurement results, so it can be confirmed that the veracity of the CFD simulation can be achieved.
3. The result shows that 95.8% of the passengers considered their thermal conditions can be acceptable. The neutral operative temperatures in winter for the passengers is 21.4℃, and the comfort zone ranges from 19.2℃ to 23.1℃; the neutral operative temperatures in summer for the passengers is 25.6℃, and the comfort zone is 23.9~27.3℃ which is a little difference with the indoor comfort zone (ASHRAE Standard 55-1992) and (ISO Standard 7730). So the criterion in the form of above standard cannot be used without modifications, for predicting the indoor thermal comfort of the passengers in this Terminal 1 building.
4. As a whole, indoor thermal comfort can be acceptant, but there are some drawbacks for airflow distributions. Additionally, porch can be used to improve thermal environment at the entrance due to its function of obstructing from storm, resisting chilliness. What's more, ADPI is no more than 0.6, mainly owing to airflow air characteristic resulting from personal mobility.

ACKNOWLEDGEMENT

This study was supported by leaders and engineers of China Southwest Architectural Design and Research Institute Corporation Limited and Sichuan Province Airport Group Cooperation Limited and students from Southwest Jiaotong University. Authors acknowledge people who supported this study.

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