A Comparative Study of Performance Between Two Combined Ventilation Systems and Their Effect On Indoor Air Quality and Thermal Comfort Inside Office Rooms

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Abstract. The most important problems that must be solved and improved in ventilation research are indoor air quality and thermal comfort. The Iraqi climate is characterized by being very hot and dry, so fresh cold air must be provided to the occupants of offices, factories, workshops and others to improve thermal comfort and increase their efficiency. This study focused on improving the quality of inhaled air by using two types of ventilation systems (mixing and displacement ventilation) in combination with a personal ventilation system. This study was a set of numerical tests that were conducted to predict the temperature distribution, air movement and speed inside a closed office room subject to the Iraqi climate in terms of weather conditions. This numerical study was carried out using (AIRPAK3.0.16) which is used to solve turbulence equations, Naiver stock, energy equations, and use of (FVM). A thermally isolated room was simulated with a personal ventilation system that supplies air at temperatures from 21 to 23 °C for case I (personal and mixing) and supplies air at a temperature from 18 to 20 °C for case II (personal and displacement) the air velocity for both cases was 0.6 m/s. It was found that there was a positive effect of the personal ventilation system in both cases, but its effect was more pronounced in the first case, through numerical calculations for each of the effectiveness of heat removal (Ɛt) and air distribution performance index (ADPI), where (74.361) and (1.549) were found for the first case, and (68.321) and (1.71) for the second case.

Keywords: Thermal comfort, AIRPAK, Computational fluid-dynamics (CFD), Mixing ventilation (MV), Displacement ventilation(DV), Indoor air quality (IAQ), Personalized ventilation (PV)

1. Introduction

Nowadays, with industrial progress and technological development, most people spend 80% of their time in closed buildings and environments full of pollutants and emissions, so the need for innovations for ventilation systems has emerged. It is very necessary to have indoor environments that are distinguished by providing thermal comfort to their occupants and are suitable in terms of recreation and health for them. Recent research on building ventilation issues has focused on both thermal comfort and indoor air quality, which are considered among the most important factors affecting the quality of the indoor environment and human comfort.

There are many types of mechanical ventilation systems, but there are two types, the most common, which are mixing ventilation and displacement ventilation [1]. Mixing ventilation, It is a ventilation system that mixes the supply air with the old air of the room so that the air inside the room becomes at the same design temperature. In the cooling mode, the supply air is provided at a temperature of 13 °C. Displacement ventilation system (DV) is defined as the displacement of the room’s old air and its replacement with new air coming from the outside of the room [2]. Many problems arise when using ventilation systems, so solutions must be found to them as one of the most common of these problems is the mixing of the cold and fresh air supplies coming from the diffusers with the hot room air full of pollutants before it reaches the breathing zone [3]. Hence the need to use personal ventilation systems, which supply fresh cold air directly to the breathing zone. Personal ventilation systems increase the quality of inhaled air by continually...
preparing and supplying the breathing zone with fresh, cool air[4]. Personal ventilation systems (PV) are preventive systems as they work to protect people from airborne diseases, and cold air supplies from personal ventilation systems reduce the percentage of pollutants in the breathing area, and this feature is one of the most important advantages of personal ventilation systems. [5]. Personal ventilation systems (PV) were used in addition to mixing ventilation systems, and displacement ventilation systems because personal ventilation systems are distinguished by their ability to give full freedom to occupants of the space by controlling the surrounding climate. [6]. The addition of personal ventilation systems inside buildings improves thermal comfort, increases and improves indoor air quality, as well as reduces pollutants and eliminates symptoms of SBS [7, 8, 9].

2. Related Work

Al-ssaad et al., 2017 [10] investigated how to improve mechanical ventilation systems by adding and combining a personal ventilation system and a mixed ventilation system. This study was conducted numerically and experimentally in laboratories under standard conditions. And it was found that when using the personal ventilation system, the thermal comfort was increased and it was about 0.95, as for the efficiency of ventilation it was 77%, and the use of a personal ventilation system led to a decrease in energy costs by about 21.34%.

Al-ssaad et al., 2018 [11] studied the effectiveness of adding a personal ventilation system to a mixing ventilation system and achieving the best performance. This study was simulated with available numerical methods to predict air flows at a suitable supply frequency for space occupants. This study was conducted under certain conditions where the room air temperature was considered to be 25 °C, while the air temperature for supplying personal ventilation systems was 22 °C. This study found that by increasing the frequency at a constant flow rate, the thermal comfort will increase by about 15.2%, and when the average flow rate increases at a constant frequency, the thermal comfort will decrease at a low frequency of 0.3 Hz, but it remains acceptable at higher. 0.5 Hz frequency.

Yakoob et al., 2019 [12] investigated increasing and improving the efficiency of air exchange inside office rooms by using the displacement ventilation system with the mixing ventilation system. This study was conducted by experimental methods under a laboratory, as well as the distribution of air temperatures and air flows by numerical methods. The RNG disturbance model was adopted. This study has demonstrated, that the presence of a combined system of personal ventilation systems with displacement ventilation will lead to an increase in the thermal comfort of a person, depending on the values of (Ɛt) and (ADPI) which are (71%) and (1.8), respectively. Using a personal ventilation system with a displacement ventilation system will increase the air quality and thermal comfort as well as increase the quality of the air inhaled in the breathing zone.

Most of the studies in the past and present time focused on studying air movement and temperature distribution inside the room, as well as the locations of diffusers and their geometric shapes.

This study took another side as it focused on the following

• This study aims to find out which ventilation system is suitable for use with a personal ventilation system inside office rooms.
• Study how to increase the quality of inhaled air and thermal comfort by choosing the most appropriate system inside the office rooms.
• Investigating both indoor air quality and thermal comfort by studying and analyzing the values of two important indicators, namely air diffusion performance index(ADPI) and heat removal effectiveness (Ɛt).

In this numerical work, it was used the ANSYS Fluent AIRPAK 3.0.16 for the purpose of simulating a thermally isolated room and predicting the air flows inside it, as well as simulating the temperature distribution inside the room and around its occupants.
3. Description Room Configuration and Assumptions

An office room was simulated by numerical methods and its dimensions were (3×2.5×2.5) m as shown in Fig (1), and the conditions under which the study was conducted were where the 3D computational model of the steady-state. The room was numerically simulated under the Iraqi climate and the room was thermally insulated. Two types of ventilation systems were simulated, which are the mixing ventilation system and the displacement ventilation system, as shown in Fig (1). The personal ventilation system supplies air at a temperature of 22 °C at a speed of 0.6 m / s at a distance of 0.5 m from the face of the simulated seated person. A person sitting in front of a computer is simulated and there is a personal ventilation system in front of the sitting person that draws air from a height of 0.4 m from the floor of the room and supplies it at a height of 1.1 m. The mixing ventilation system supplies the air at a speed of 2.5 m/s from a diffuser just below the ceiling onto the north wall of the room. As for the displacement ventilation system, the air diffuser installed near the floor of the room supplies the air at a speed of 0.25 m/s. The heat gain (only sensible heat) for a person is 75 W, computer 60 W, and lump 100 W. It was used (Cooling Load Temperature Difference Method (CLTD)) [13].

4. Computational Fluid Modeling

The governing equations in this numerical study are [14].

4.1 Governing Equations:

\[
\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w) = 0
\]

\[
\frac{\partial}{\partial x}(\rho uu) + \frac{\partial}{\partial y}(\rho uv) + \frac{\partial}{\partial z}(\rho uw) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\mu \frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu \frac{\partial u}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu \frac{\partial u}{\partial z}\right) + \frac{\partial}{\partial x}\left(\rho u'w'\right) + \frac{\partial}{\partial y}\left(-\rho u'v'\right) + \frac{\partial}{\partial z}\left(-\rho u'w'\right) + \rho g_x
\]

\[
\frac{\partial}{\partial x}(\rho vw) + \frac{\partial}{\partial y}(\rho vv) + \frac{\partial}{\partial z}(\rho vw) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\mu \frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu \frac{\partial v}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu \frac{\partial v}{\partial z}\right) + \frac{\partial}{\partial x}\left(-\rho v'u'\right) + \frac{\partial}{\partial y}\left(-\rho v'v'\right) + \frac{\partial}{\partial z}\left(-\rho v'u'\right) + \rho g_y
\]

\[
\frac{\partial}{\partial x}(\rho uw) + \frac{\partial}{\partial y}(\rho vw) + \frac{\partial}{\partial z}(\rho vw) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\mu \frac{\partial w}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu \frac{\partial w}{\partial y}\right) + \frac{\partial}{\partial z}\left(\mu \frac{\partial w}{\partial z}\right) + \frac{\partial}{\partial x}\left(-\rho w'u'\right) + \frac{\partial}{\partial y}\left(-\rho w'v'\right) + \frac{\partial}{\partial z}\left(-\rho w'u'\right) + \rho g_z
\]

4.2 Models of Turbulence:

The simulation was done with the use of 4 types of disturbance models, and the disturbance model was the best in terms of speed of performance, the accuracy of results, and its proximity to realism was the type disturbance model the turbulent model RNG K-\( \varepsilon \). This was fully compatible with Chen, [15] investigated eight K-\( \varepsilon \) models for mixed convection flow and found that the Renormalization Group (RNG K-\( \varepsilon \)) model was the best, closest to reality and acceptable results. Yakhot [16], the Renormalization Group (RNG K-\( \varepsilon \)) better kind of turbulence models have tested. Yuan, [17] studied predicting the distribution of indoor pollutants in the displacement ventilation chamber using the model (RNG K-\( \varepsilon \)), the equation for the model of turbulent used are [18]:
\[ \rho U_i \frac{\partial k}{\partial x_i} = \mu_t S^2 + \frac{\partial}{\partial x_i} \left[ \alpha_i \mu_{eff} \frac{\partial k}{\partial x_i} \right] - p e \]  
\[ \text{Where } S = \sqrt{\sum_{i,j} S_{ij}^2} \quad S_{ij} = \frac{1}{2} \left( \frac{\partial U_j}{\partial x_i} + \frac{\partial U_i}{\partial x_j} \right) \]  
\[ \rho u_i \frac{\partial e}{\partial x_i} = C_{1e} \mu_t S^2 + \frac{\partial}{\partial x_i} \left[ \alpha_i \mu_{eff} \frac{\partial e}{\partial x_i} \right] - C_{2e} \rho e^2 - R \]  

4.3 Numerical Solution and Mesh Generation:

The values of \((\varepsilon_t)\) and \((ADPI)\) were calculated through numerical simulation by AIRPAK3.0.16. In this work, a method was used the finite volume method. The turbulence model (RNG, K-\(\varepsilon\)) used was, after several attempts, to reach the most acceptable results and the lowest error rates. Hexa unstructured geometry to discretize. For this function, all types of the element were used to fit the grid to the geometry. After reaching the error rate \(10^{-3}\), the results became more acceptable and the solutions were completely stable, as shown in Fig (3). In this numerical simulation (finer mesh) is used to reduce the error rate. After testing several types of mesh, coarser, coarse, medium, and fine type were generated, to perform the test and the most appropriate for this study was the type (coarser). The AIRPAK3.0.16 software used to create the model and meshed study cases based on many testing meshes as shown in Figs (4 and 5), the dimensions of edges of the room in \((x, y, z)\) meshed as interval size of \((0.04)\), mesh parameter normal, with Max side ratio \((2)\).

4.4 Boundary Conditions

Boundary conditions are one of the most important factors that greatly and directly affect the success and accuracy of CFD simulation, and they affect the reliability of results for problems that are solved numerically. The boundary conditions were assumed as consisting of one type of inlet and flow velocity as shown in Table (3). Also, the walls were considered fixed, non-slip, insulating, not allowing heat to transfer through them (thermally insulated. Reaching accurate solutions to the end is impossible, so it is necessary to calculate and determine the residual error acceptable for different conditions such as continuity, speed components, and energy. To ensure the validity of the program and the results in this study, this was verified after comparing the experimental results of another study with the numerical results obtained with this numerical simulation by (AIRPAK3.0.16) where the validity of the RNG K-model was verified as shown in Fig (4) . This comparison gave a convergence between the numerical data of the study and the experimental data \([19]\), where the perturbed RNG K-epsilon model was used, and it was found that the percentage of deviation is about 5.37%.

5. Results and discussion

The results will be discussed numerically in this part of this work, for the two cases in light of the Iraqi climate. The air distribution inside the office room and the temperature distribution inside the room was studied.

- **Case I (Personal and Mixing Ventilation System)**

  The contour of the temperature of air \((z\)-plane at \(z=1.25m)\) is shown in Fig. (4). This figure shows the entry of air from the diffuser installed in the upper part of the walls, where the air enters at a temperature of 17 °C and then begins to rise due to the thermal loads and heat sources inside the room. Fig. (5) indicates the contour of the speed of air. The air enters at a high velocity of 2.5 m/s mixed with the air in the room and there is a uniformity in the distribution of temperature.

  Personal ventilation devices (PV) draw air from a height of 0.4 m. At a height of 0.4 m from the floor of the room, the temperature for case I is within (21-23) °C, which provides suitable cool air to the breathing zone for the occupant of the space as shown in Fig. (6). Therefore, the air is prepared to the breathing zone for the occupant of the space at an appropriate and acceptable temperature as shown in Fig. (7).
• **Case II (Personal and Displacement Ventilation System)**

It is noticed that the temperature distribution in case I is more uniform than the temperature distribution in case II, and as shown in Figs. (8 and 9).

Also, the use of mixed ventilation systems leads to a high velocity of air supply, which allows for uniformity in the velocity of air inside the room. The use of personal ventilation systems does not affect the air currents and its movement within the room, as shown in Fig (9). As for the use of personal ventilation systems with displacement ventilation systems, it was found that there was a non-uniform in the distribution of the velocity of air inside the room, and the temperature was non uniform, as it was the lowest possible on the floor and the highest possible near the ceiling of the room and this affected the thermal comfort and the air quality inside the room.

The contour of the mean age of air is shown in Fig. (10). By calculating the mean age of the air in the breathing zone, it was found that the age of the air in the breathing zone for case I which is shorter than the age of the air in the breathing zone for the case II, as shown in Fig. (11). This means the availability of fresh, cold air as soon as possible, which leads to a decrease in the concentration of pollutants in the breathing area, thus increasing thermal comfort for humans and an increase in the quality of their inhaled air.

The distribution of temperatures inside the laboratory room at six different altitudes, where it is noticed that the temperatures in the case I rise as the height from the floor of the room decreases. As for case II, the air temperatures rise with the increase in the height from the floor of the room, as it is noticed that the difference in temperatures is for the first case. It is very little, which means a uniform and homogeneous distribution of air. As for the temperature distribution of case II, there will be a clear difference in the temperature difference at different heights from the room floor, and this leads to the mixing ventilation system being considered better in terms of temperature distribution and air velocity inside the room as well as the concentration Pollutants.

The temperatures were calculated in three poles (x,y,z), pole 1(2,0,1.25), pole2(1,0,0.5) and pole 3 (1,0,2) as shown in the Figs. (12). It is noticed that the temperatures are uniform and there is very little difference in temperature at different altitudes from the floor of the room, and this means a uniform distribution of temperatures inside the whole room.

In the case II, as shown in Fig (13) when measuring the temperatures in the three poles, (x,y,z), pole 1(2,0,1.25), pole2(1,0,0.5) and pole 3 (1,0,2) it is noticed that the temperature distribution is less than possible near the floor of the room and increases as the height from the floor increases until it becomes more as possible near the ceiling.

When making a comparison between I and II case, the following was found:

- The mean age of the air inside the room for case II is less than the average age of the air for case II, which means an increase in the supply of cold fresh air continuously for case I.
- In case I, the cold air coming from the diffuser mixes with the room air before it is withdrawn by the personal ventilation systems, which means that the air is supplied from (21-23) °C at a speed suitable for breathing.
- In-case II, the air is supplied from a diffuser near the floor of the room and at a low speed, so mixing of the air does not occur, and the air is drawn in by personal ventilation systems at low temperatures that may be unsuitable for human breathing.
- The effect of using personal ventilation systems is widely seen with the displacement ventilation system because the air velocity of the air diffuser is lower for the case II compared to the air velocity for case I.

6. **Conclusions**

This numerical work dealt with indoor air quality and how to improve it, using personal ventilation systems with mixing and displacement ventilation systems as two different cases. The most important conclusions of this study are:

1- The use of a personal ventilation system will improve and increase the indoor air quality and thermal comfort of the occupants.
2- The best heat removal effectiveness obtained in case II, while the best Air Diffusion Performance Index (ADPI) obtained in case I.

3- The personal ventilation system is effective for both cases, but the effect is more positive in case I, depending on the temperatures of the air being supplied.

4- The study showed that the use of personal ventilation systems increases the quality of inhaled air in the breathing zone and case I is the best under the same conditions.

5- The limits of thermal comfort found in case I (personal and mixing ventilation), due to depending on the numerical results obtained where the values of (ADPI) and ($\varepsilon_t$) are (74.361) and (1.549) respectively for case I.

The primary suggestions for future works are as follows:
1. Studying different models and calculating the numerical results of the unsteady state turbulent flow.
2. Studying the conditions of thermal comfort, and the quality of the indoor air for one or more residents in parking and other activities inside the office room (walking, standing, and other activities) with different metabolic rates, as well as thermal insulation of clothing and conditions of relative humidity and others.
3. Studying the different shapes and types of air supply diffusers, such as the circular diffuser, the swirl inlet, and the four-way diffuser in addition to the ceiling diffuser.
4. Studying the effectiveness and capacity of different ATD PV devices to reduce and mitigate the transmission of aerosol disease to healthy occupants in the event of an infected agent in an adjacent office or room.

| Nomenclature | Description | Unit |
|--------------|-------------|------|
| A            | Surface Area for Wall | (m²) |
| $C_p$        | Specific heat of the air at pressure constant | (kJ/kg.K) |
| $C_1\varepsilon$, $C_2\varepsilon$ | Coefficient in the specific dissipation rate | |
| D            | Diameter | (m) |
| dx dy dz     | Control volume | (m) |
| G            | Gravitational acceleration | (m²/s²) |
| N            | Total number of draft temperature points measured in occupied zone | |
| $N_0$        | Number of points of draft temperature measured in occupied zone | (N/ m²) |
| P            | Pressure | |
| Q            | Heat transfer through the wall. | (W) |
| $St$         | Source term for the rate of thermal energy Production | (J/kg) |
| T            | Temperature | (°C) |
| $u_x,y,w$    | component of speed in x,y and z-directions | (m/s) |
| $u_{i,j,k}$  | Speed at cell (i,j,k) | (m/s) |
| $x,y,z$      | Cartesian coordinates | (m) |
| Vx           | Local air speed | (m/s) |
| Vroom        | Volume of room (m³) | |
| $Q_s$        | design flow rate of the supply air | (l/s) |
| $q_{ex}$     | Cooling load for the heat conduction through the walls and transmitted solar radiation | (W) |
| $q_l$        | Cooling load for the overhead lighting | (W) |
| $q_{oc}$     | Cooling load for occupant, desk lamp and equipments | (W) |
| $Q_t$        | Total heat transfer | (W) |
| $T_{av}$     | average room temperature | (°C) |
| $T_e$        | Exhaust Air Temperature | (°C) |
| $T_{oc}$     | outlet design temperature | (°C) |
| $T_{o}$      | outlet temperature | (°C) |
| $T_x$        | local temperature | (°C) |
| $T_{sp}$     | Setup (design) temperature. | (°C) |
Greek Symbols

\( \rho \)  Air density \( (\text{kg/m}^3) \)

\( \varepsilon \)  Turbulent energy dissipation rate. \( (\text{J/kg .s}) \)

\( \sigma \)  Prandtl or Schmidt number

\( \Sigma k \)  Model constant

\( \Sigma \varepsilon \)  Model constant

\( \Omega \)  Specific dissipation rate \( (1/\text{s}) \)

\( \Gamma \)  Diffusion coefficient \( (\text{m}^2/\text{s}) \)

\( \mu_t \)  Turbulent viscosity \( (\text{N.s/m}^2) \)

\( \Omega \)  Rotation speed \( (\text{rad/s}) \)

\( \Delta T_{hf} \)  The difference in temperature from head to foot level. \( (^{\circ}\text{C}) \)

\( \alpha_k, \alpha_{\varepsilon} \)  Coefficient in the specific dissipation rate

\( \xi_t \)  Effectiveness temperature

\( \Theta_t \)  Turbulent Reynolds stress

Sub-Scripts

av. Average
c Correct
e Exhaust Air
ex External
f Floor
hf Head to foot level.
H Hydraulic
i Inside
i,j,k Location of point in a Cartesian grid
l Overhead light

Abbreviations

ACH Change of air per hour
ADPI Air Distribution Performance Index
ASHRAE American Society for Heating, Refrigeration, and Air Conditioning Engineers
CFD Computational Fluid Dynamics
PV Personalized ventilation
PMV Personal-mixing Ventilation
FVM Finite Volume Method
EDT Effective Draft Temperature
IAQ Indoor air quality
MV Mixing Ventilation
RNG Re-Normalization Group
SBS Sick Building Syndrome
ATD Air terminal device
PAQ Perceived air quality
DV Displacement Ventilation

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| Item             | Dimensions (m) | Heat (W) |
|------------------|----------------|----------|
| Office Room      | X-start | Y-start | Z-start |          |
| Mixing device    | 0.05    | 0.1     | 0.16    |          |
| Person no.1      | 0.2     | 1.1     | 0.2     | 75       |
| Computer_no.1    | 0.3     | 0.3     | 0.3     | 60       |
| Light            | 0.1     | 0.05    | 0.2     | 100      |
| Computer table1  | 0.4     | 0.76    | 1       |          |
Table 1: Room configuration for (Case-II)

| Item                  | Dimensions (m) | Heat(W) |
|-----------------------|----------------|---------|
| X-start               | Y-start        | Z-start |
| Office Room           | 3              | 2.5     | 2.5    |
| Displacement device   | 0.05           | 0.2     | 0.75   |
| Person no.1           | 0.2            | 1.1     | 0.2    |
| Light                 | 0.3            | 0.3     | 0.3    |
| Computer_no.1         | 0.1            | 0.05    | 0.2    |
| Computer table1       | 0.4            | 0.76    | 1      |

Table 2: The required conditions inside the office buildings in the Iraqi climate during the summer season, [11].

| Property item                  | DBT  | RH   | Recommended air velocity inside the office room |
|--------------------------------|------|------|-------------------------------------------------|
| The conditions required within the office buildings | 23 ºC – 26 ºC | 40% – 50% | 0.25m/s_0.13m/s |

Table 3: Boundary Conditions

| Part                        | Types | Condition of momentum | Shear condition |
|-----------------------------|-------|-----------------------|-----------------|
| Person, wall                | Motion of wall | Stationary | Not Slipping |
| computer wall               | Wall   | Stationary            | Not Slipping   |
| side walls, floor, ceiling | Wall   | Stationary            | Not Slipping   |
| tables and lights           | Wall   | Stationary            | Not Slipping   |
| Supply air diffuser          | Speed inlet | Static Pressure | Backflow direction specification |
| Extract Grill               | Pressure outlet | 0 Pa                | Method: (Normal to Boundary) |

Table 4: Numerical values of ADPI and effectiveness (Ɛt) temp. for each ventilation type

| Ventilation devices used   | CASE | Case I | Case II |
|----------------------------|------|--------|---------|
| ADPI%                      | 74.361 | 68.321 |
| Effectiveness              | 1.549  | 1.71   |

Fig.1. Model of the room
Fig. 2. A part from meshed model

Fig. 3. Validation between the numerical data of this study and an experimental data. [20].
Fig. 4. Contour of air temperature (z-plane)

Fig. 5. Contour of air speed (z-plane)

Fig. 6. Contour of air temperature (x-plane)

Fig. 7. Contour of air temperature case I (z-plane)

Fig. 8. Contour of air temperature case II (z-plane)

Fig. 9. Contour of air speed case II (z-plane)
Fig. 10. Contour of mean age of air(°C) (z-plane)

Fig. 11. Contour of mean age of air(z-plane)

Fig. 12. Air temperature distribution for case I

Fig. 13. Air temperature distribution for case II