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Abstract: Extra cooling pads on the sidewalls are needed for larger poultry houses using tunnel ventilation system. Preliminary study showed that the airflow velocity going through different aisles varies greatly when the extra pads are installed at the end of sidewalls, making a "["-shape air inlet. Combined with field tests, the CFD (computational fluid dynamics) technology was used to study the uniformity of airflow distribution in a tunnel-ventilated laying-hen house. The air distribution was first monitored in a layer house to find the main reason resulting in the variations of airflows in different aisles. Then CFD simulations were carried out with different distances (D=2 m, 3 m or 4 m) between the pads on end-wall and the extra pads on side walls. The field test showed that airflow streams from the different groups of cooling pads collided vertically at the house corners, mixed with each other, then flew towards the center of the house. This was the main reason that the wind speed in the middle aisle was much higher than in other aisles, leaving large zones of lower ventilation in the aisles adjacent to the sidewalls. The results of CFD simulations indicated that air distributions could be significantly improved when the extra pieces of pads were moved away for an appropriate distance from the end coolingpads. As far as conventional poultry house with a span of 12 m, the air speeds in different aisles were more uniform when this distance was about 3 m.

Keywords: Pad cooling system, air distribution, air speed, laying-hen house; CFD
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1 Introduction

Dominated by continental climates, high ambient temperature in most regions of China can easily exceed 30°C for several months in a year[1,2], presenting a big challenge for poultry production in summer. Nowadays as the number of birds per house continuously keeping increasing in China, heat stress problem for poultry production tends to be more serious. To alleviate heat stress, tunnel ventilation and evaporative pad cooling systems have been widely used in poultry houses[3-8]. Based on tunnel ventilation mechanism, much more pads should be equipped to improve the cooling effect for a larger house. While constrained by the house design, the end wall is often not big enough to allow sufficient cooling pads to be installed. The most common practice in China to deal with this issue is to install extra cooling pads at the end of both sidewalls, thus making a "["-shape end-wall air inlet. A preliminary study showed that the

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airflow velocities in different aisles of such houses vary greatly. Dead zones, where there is little or no air movement, could be found in the aisles adjacent to the sidewalls[9]. Large variation of air movement in different aisles affects thermal comfort of theanimals. Caged birds under low air speed environment with serious heat stress are easily observed, leading to higher mortalities and lower production performance[9,10].

An ideal ventilation system should be designed not onlyto meet air exchange requirement but also to provide good air distribution in bird occupied zone (BOZ) to ensure production performance[11]. Effective air distribution is characterized by thoroughly mixing the indoor air with fresh air before being exhausted out[12]. Generally, it is considered that a tunnel-ventilation system can provide relative uniformity of air distribution at the cross-sections of the house[2-4]. But the intensive production and highly mechanized operation of commercial laying-hen houses make the ventilation a very complex issue. Previous studies indicated that air speed varied significantly along the cross-section of a tunnel ventilated house, and the air velocity tended be higher in the center of the house than along the side walls, and higher at ceiling than in area where birds resided[13,15]. Several factors, such as “smoothness” of sidewalls, feeding facilities, and so on, affect air distributions. Whole column of cages, which are filled with birds, dramatically impede air diffusion between aisles in BOZ, due to the fact that air tends to take the path of least resistance. It is important to keep in mind that the air speed distribution can be improved by designing the proper inlets and deflectors, as air inlet is critical to the success of the ventilation system and responsible for providing good air distribution throughout the house[11,13,14]. However, recent studies mainly focused on ventilation rate and effectiveness[12-19] in commercial laying-hen houses; little work has been carried out on the air distribution in these houses[20].

The objective of this research was to study the air distribution in a tunnel-ventilated poultry house with a “[”-shape cooling pads, identify key factors that affect the uniformity of airflow in different aisles, and optimize the installation extra cooling pads to improve the indoor air distribution using CFD simulation.

2 Materials and Methods

2.1 Experimental poultry house and ventilation system

The field study was conducted for two months (from July to September 2014) in a tunnel-ventilated layer house (90 m × 12 m × 3.6 m), located in Hebei Province, China. Fourteen thousand laying hens (W-36), half-year-old (onset age), were confined in conventional cages with four rows and three tiers. As shown in Figure 1, eight 1.38-m diameter exhaust fans (TUHE-S1, Tuhe Equipment Industrial Co., Ltd., Foshan, China) were located on one end wall of the house. Three groups of cooling pads were installed at the other end of the building and make the inlet like a “[” shape. The dimensions of the end pads and the extra pads were 11.4 m × 2.4 m × 0.15 m and 1.5 m × 2.4 m ×0.15 m, respectively. An air deflector was placed at 45º angle towards the cages, and the horizontal distance from the bottom of the deflector to the end pads was 0.75 m. The micro-climate inside the building was controlled by an AC-2000 Environmental controller (ROTEM Control and Management, Petach-Tikva, Israel).

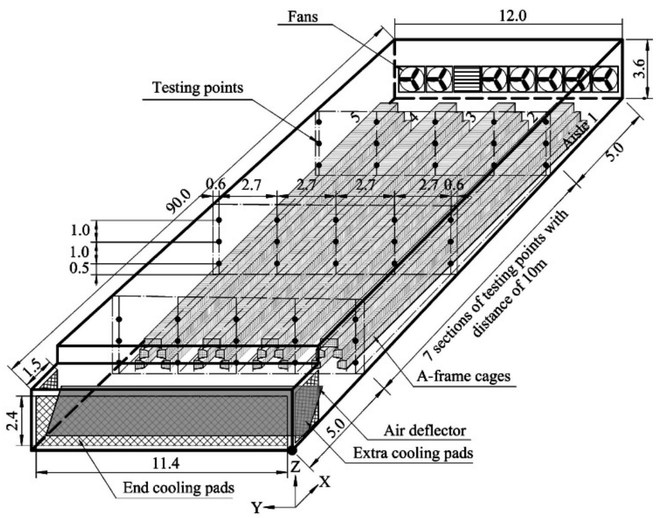

Figure 1 Layout of experimental poultry house and the location of testing points (unit: m)

2.2 Experimental test plan

In order to find the cause of the undesirable airflow distribution in different aisles, air velocities at the test points were measured with (1) all of the pads in operations and (2) only end wall pads in operations. When test 2 was conducted, thin plastic films were covered on the outer surface of the extra pads on side walls, so that the fresh air can enter house only through the pad at the gable wall.

2.3 Air velocity measurement

A hot-wire anemometer (Model KA41L, KANOMAX, Osaka, Japan) with ±0.01 m/s sensitivity was used to measure the airflow in the house. Seven single-speed fans were operating during the experiment. Ten air velocities were logged every second and the averaged air velocity was used as the air velocity of each testing point.

2.3.1 Face velocity of pad

Average face velocity after pad was used to calculate the ventilation rate of the building. More than 80 points were tested at 10-15 cm away from the inside surface of pad. Average air velocity was calculated as the face velocity of pad by using air velocities at different points.

2.3.2 Air velocity in the building

As shown in Figure 1, nine cross sections were selected at X direction and there were 15 test points on each cross section. The 15 test points on each cross section were divided into 5 groups and located at the middle of each aisle (at Y direction). Three points in each group were set vertically at a height of 0.5 m, 1.5 m and 2.5 m (at Z direction), respectively. The coordinates of each point were shown in Figure 1.

2.4 Numerical simulation

2.4.1 Model set-up and general settings

The numerical simulations were carried out using CFD under steady-state conditions. The interior space of the house was considered as the flow domain, and full scale model was developed based on real geometry of the poultry house using UG (Unigraphics NX 8.0, Siemens PLM Software, Germany). An unstructured mesh was built in ANSYS (ANSYS FLUENT 14.0, FLUENT, America), mainly composed by tetrahedral cells, the mash of edge was refined (Figure 2). The commercial code ANSYS 14.0 was utilized to calculate on three-dimensional simulation of airflow in the poultry house. The standard k-ε turbulence model was used in this work.

The exhaust fans were modelled as circular surface with 1.3 m in diameter. Cooling pads were modelled as porous medium[21]. Due to computational limitations, the CFD model cannot generally contain all the microscopic details of the house system and some form of simplification was needed[22]. In this study, systems such as feeding, drinking, egg trough and manure collection were not modelled. Cages with birds inside were simplified as solid cube.

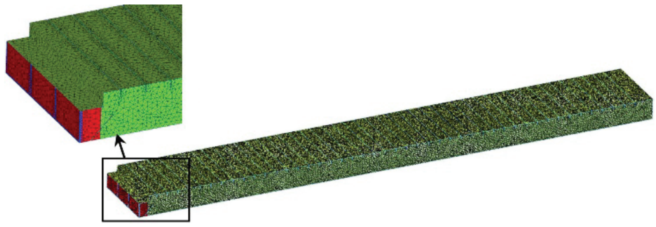

Figure 2 Model of the experimental house and surface mesh

2.4.2 Boundary conditions and assumptions

All walls, floor, roof and cages were specified as no-slip wall boundary. Face velocity of the pads (the value was 1.32 m/s) was measured and used as the inlet boundary condition. Considering the volumetric flow, the designed ventilation rate could be calculated using Equation (1)[23]:

$$AER = \frac{\sum v_i A_i}{V} = \frac{\sum v_o A_o}{V} \tag{1}$$

where, AER is the air exchange rate, s^{-1}; v_i and v_o are air velocities of inlet and outlet, respectively, m/s; A_i and A_o are vent opening area of inlet and outlet, respectively, m^2; and V is the inner volume of the poultry house, m^3.

The house was simplified as an absolutely tight system because it had good air tightness. The volume of exhaust air was considered as equal to the volume of incoming fresh air. The exhaust fans had only one speed level and seven fans were operating during the testing. Rewrite Equation (1) and the air speed of the exhaust fans could be expressed as:

$$\overline{v}_o = \frac{\overline{v}_i A_i}{7A_o} \tag{2}$$

where, v_i is face velocity of cooling pad, m/s; v_o is average air velocity of exhaust fans, m/s; A_i and A_o are vent opening areas of pad and single fan, respectively, m^2. Average air velocity of exhaust fans calculated using Equation (2) was 5.31 m/s and used as outlet boundary condition.

2.4.3 CFD validation

The quality of CFD simulation was highly dependent on the turbulence model. Because some simplification were made, experimental validation of model was necessary[24]. The relative error (E) analysis were carried out to determine the differences between measured results from field experiments and those obtained from the CFD simulation. The E was defined as[25]:

$$E = \frac{V_{CFD} - V_m}{V_m} \times 100\% \tag{3}$$

where, V_m is the measured air velocity at each testing point, m/s; V_{CFD} is the air velocity obtained from the CFD simulation results at each testing point on the same coordinate, m/s.

Normalized mean square error ($NMSE$) was also used to examine the validation using Equation (4). Values of $NMSE$ less than 0.25 were accepted as indications of good agreement[23].

$$NMSE = \frac{(V_{CFD} - V_m)^2}{(V_{op} - V_{om})} \tag{4}$$

where, V_{op} is the average air velocity obtained from the CFD simulation results, m/s; V_{om} is the average measured air velocity at each testing point, m/s.

CFD simulation was used to improve airflow distribution in the experimental poultry house. The air velocity of measurement points at positions of height Z=1.5 m in aisle 2, aisle 3 and aisle 4 were chose to validate the CFD model. Predicted velocities at same chosen points were taken from the simulation using four kinds of grid cells (1.32 million, 2.20 million, 3.38 million and 4.08 million).

3 Results and discussion

3.1 Air velocity distribution in the experimental poultry house

The air velocity profiles at 135 sampling points with extra cooling pads measured in different aisles along the length at Z=0.5 m, Z=1.5 m, Z=2.5 m, are shown in Figure 3. There was a greater variability among different aisles and the airflow distribution was unsatisfactory. Air speed at all points in aisle 3 was higher than others and the highest velocity was measured at X=10 m, Z=2.5 m. Air velocity in aisle 1 and aisle 5 was relative lower and the lowest velocity was measured in aisle 5 at X=10 m, Z=0.5 m. Air deflector installed inner side of pad (Figure 1) was used to not only increase the air speed, but also change the airflow direction. This could explain the notable variations of air velocity at different height within 30 m distance from the cooling pads at the end of house.

The air velocity in the aisles without the extra cooling pads was tested and the results were shown in Figure 4. The air velocity profiles showed that the air distribution could be improved obviously without the two extra pads at the end of the sidewalls. In negative pressure ventilation system, the direction of airflow was perpendicular to the surface of the cooling pads. When the layout of cooling pads was like a "[" shape, the airflow from the pads hit each other at the corners, and the direction of the mixed airflow was changed to the center of the house. It was the main reason that the wind speed in the middle aisle was significantly higher than that was in other aisles. However, in most times, it was often difficult to meet the cooling demand of layers in summer when the cooling pads were only installed in one of both end wall, extra cooling pads were necessary. The remained question is whether thelayout of cooling pads could be further optimized to improve the airflow distribution in the room space.

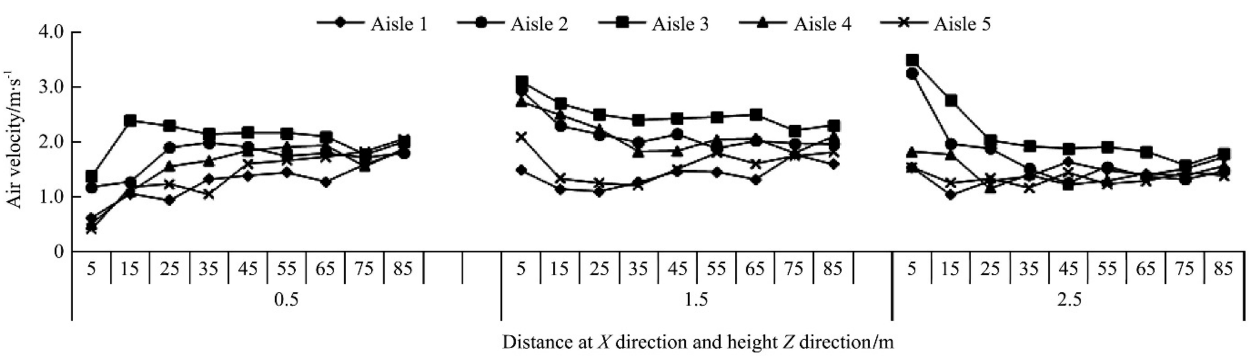

Figure 3 Air distribution in different aisles in the room space with extra cooling pads

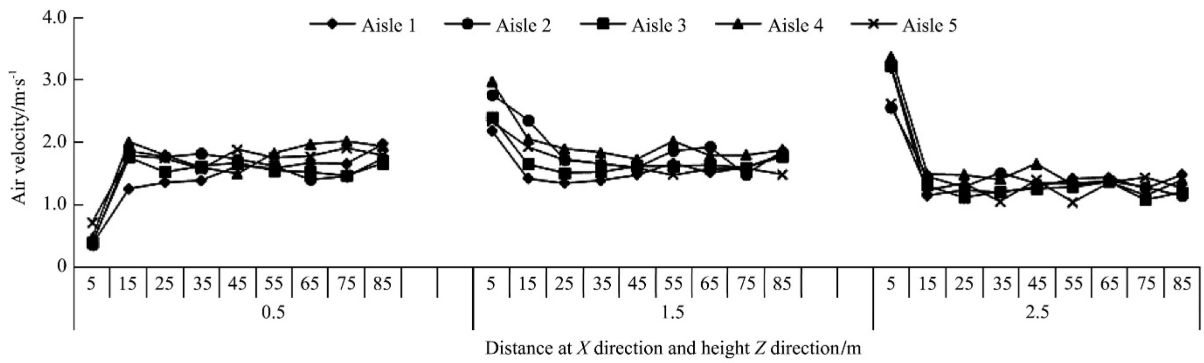

Figure 4 Air distribution in different aisles in the room space without the extra cooling pads

3.2 CFD Model validation and grid-independent

The values of *E* and *NMSE* calculated using the simulation and measurement values at the chosen points were shown in Table 1. According to the values of *E*, the average percentage difference between measured and predicted velocities for the four grids was more than 20% and not very satisfactory[26], but the difference among the four grids was insignificantly small. Except for the grid for 1.32 million, the values of *NMSE* for other three grids were mostly less than 0.25, which is an indicator of good agreement[23]. Then, the predicted velocities for these three grids were averaged, respectively, shown in Figure 5. As the number of grid cells increased from 3.38 million to 4.08 million, little improvement of the simulated results was gained. This indicated that the results from the CFD simulation in most of the domain could be assumed to be grid-independent and the grid cells of 3.38 million was used in this study.

3.3 Optimization of airflow distribution

It could be seen from the above analysis that the "[" shape air inlet of the cooling pad was the root reason why airflow distribution in different aisles is uneven. The air velocity profiles showed that the air distribution could be improved obviously without the two extra pads, but the shorter building may be needed to match the cooling demand. Someone may think that it could be done by increasing the face velocity via the end pad only. However, high face velocity means lower cooling efficiency and higher ventilation resistance. An increase in pad thickness directly can increase the contact time of air traversing the pad, the cooling efficiency can be improved in some ways, but the resistance to airflow also is greatly increased. So it seems to be a better choice to install more pads. That the main reason why the extra cool pads were installed in house. The remained question is whether thelayout of cooling pads could be further optimized to improve the airflow distribution in the room space. Research showed airflow distribution could be improved in different aisles when the extra cooling pad in side walls were moved away from the end wall for a certain distance[9]. However, there were few research articles about the suitable distance (*D*) between the cooling pads in the end wall and the extra cooling pads in the side walls (Figure 6).

To determine the influence of different distances on the airflow distribution, simulated calculation was carried out with distances of 2 m, 3 m and 4 m, respectively, and the airflow distribution is shown in Figure 7.

Figure 7 appeared to show that it was conducive to improving the uniformity of airflow inside the house when the extra cooling pads in the sidewalls away from the end wall for a certain distance. When *D*=2 m, the airflow in the aisle close to the sidewalls was increased, and the dead zone of airflow was obviously reduced. With the advancing of airflow to the exhaust fans, the difference of air velocities in different aisles was significantly narrowed, but the air speed in the middle aisle was still greater than that in the other aisles. When *D*=3 m, the airflow velocities in all aisles was relatively uniform, with the development of the airflow, the uniformity of the aisle was good. When *D*=4 m, the airflow in the house was obviously inclined to the aisle 1 and aisle 5, the wind speed in the center aisles has declined. The air velocity in aisle 3 was changed from the highest to the lowest and always lower than the other aisles.

Table 1 The difference of *E* and *NMSE* for four grid cells[1,2]

Grid Cells		1.32 million	2.20 million	3.38 million	4.08 million	1.32 million	2.20 million	3.38 million	4.08 million
	X	Relative Error (*E*)/%				Normalized Mean Square Error (*NMSE*)			
Aisle 1	25	−36.17	−57.33	−34.29	−66.02	−0.41	0.24	0.09	0.32
	35	−67.97	−47.22	−45.96	−67.43	−1.94	0.22	0.21	0.45
	45	−40.13	−42.43	−51.03	−57.33	−0.91	0.24	0.34	0.44
	55	−67.65	−43.39	−37.55	−45.91	−2.55	0.25	0.18	0.28
	65	−22.50	−25.22	−35.28	−19.75	−0.23	0.07	0.13	0.04
	75	−32.69	−43.56	−36.31	−43.55	−0.89	0.37	0.26	0.37
	85	−14.30	−18.02	−27.27	−24.47	−0.14	0.05	0.12	0.09
Aisle 2	25	−24.11	−13.76	−17.02	−16.31	−0.70	0.05	0.08	0.08
	35	−4.53	−12.69	−12.19	−9.81	−0.02	0.04	0.04	0.02
	45	−26.01	−24.47	−14.15	−16.19	−0.82	0.17	0.06	0.08
	55	−2.22	−11.68	−5.94	1.00	0.00	0.03	0.01	0.00
	65	−13.69	−17.34	−12.12	−3.34	−0.20	0.08	0.04	0.00
	75	−4.23	−12.92	−22.62	−11.16	−0.02	0.04	0.12	0.03
	85	−20.91	−22.87	−17.56	−17.99	−0.44	0.12	0.07	0.08
Aisle 3	25	13.73	9.26	10.32	6.44	−0.31	0.03	0.04	0.02
	35	21.00	14.18	14.95	6.67	−0.67	0.07	0.08	0.02
	45	−1.80	0.31	6.29	1.74	−0.01	0.00	0.01	0.00
	55	10.34	−0.58	3.81	1.90	−0.17	0.00	0.01	0.00
	65	−1.32	2.01	−0.14	−12.51	0.00	0.00	0.00	0.06
	75	5.86	8.54	17.82	13.99	−0.04	0.02	0.10	0.06
	85	−0.74	−3.65	−7.84	−19.10	0.00	0.00	0.02	0.12
Aisle 4	25	−20.83	−27.65	−17.12	−19.45	−0.57	0.24	0.09	0.12
	35	−9.14	−18.53	−2.17	7.90	−0.07	0.07	0.00	0.01
	45	5.64	8.17	−1.18	11.04	−0.03	0.01	0.00	0.03
	55	−9.04	1.34	−9.32	−7.86	−0.09	0.00	0.02	0.02
	65	−21.63	−20.70	−4.40	−30.00	−0.53	0.11	0.01	0.24
	75	−3.60	10.14	−11.22	6.25	−0.01	0.02	0.03	0.01
	85	−25.30	−24.83	−1.27	−4.82	−0.75	0.17	0.00	0.01
Aisle 5	25	−77.62	−67.81	−74.53	−74.94	−2.49	0.45	0.54	0.55
	35	−71.22	−47.67	−66.48	−49.73	−1.97	0.21	0.40	0.23
	45	−50.29	−68.04	−63.20	−58.81	−1.51	0.65	0.56	0.49
	55	−60.58	−59.56	−55.98	−59.87	−3.15	0.71	0.63	0.73
	65	−63.23	−30.58	−39.42	−31.96	−2.71	0.15	0.25	0.16
	75	−41.66	−34.45	−40.51	−51.09	−1.39	0.22	0.31	0.50
	85	−16.64	−24.30	−38.18	−23.41	−0.24	0.12	0.30	0.11
Average		−22.72	−21.92	−21.40	−22.45	−0.74	0.15	0.15	0.16
Maximum		21.00	14.18	17.82	13.99	0.00	0.71	0.63	0.73
Minimum		−77.62	−68.04	−74.53	−74.94	−3.15	0.00	0.00	0.00
SD		26.34	22.86	23.81	25.63	0.90	0.17	0.17	0.20

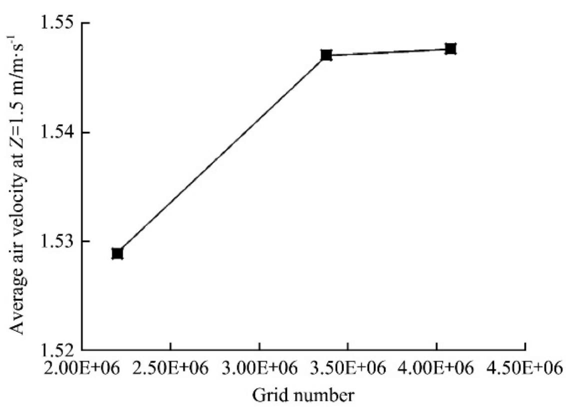

Figure 5 Grid number independent test

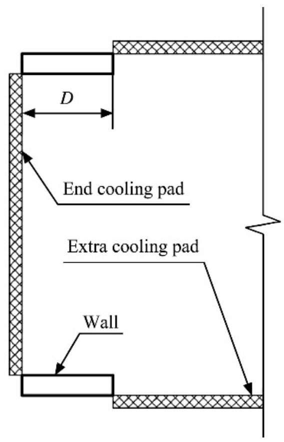

Figure 6 Simulated optimization scheme

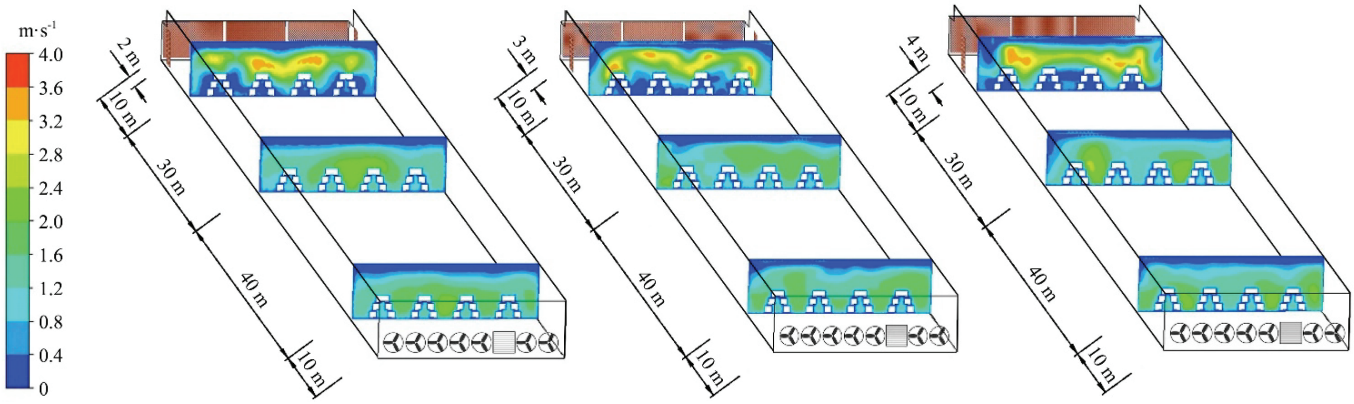

Figure 7 Airflow speed distribution at different values of *D*

Analysis showed, when the extra cooling pads in the sidewalls were needed in confined poultry house, it was necessary to keep a certain distance from the cooling pads on the end wall. When the distance was 3 m, the airflow distribution in the bird occupied zone was good, except near the air deflector, as shown in Figure 8. The air velocity at animal-level was more than 1.5 m/s in most regions. The air speed at *Z* direction was stable, and the airflow velocity in different aisles was relatively uniform along *X* direction.

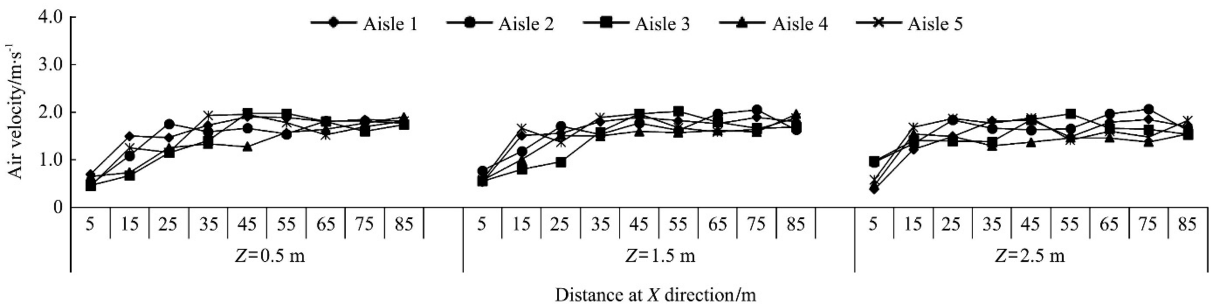

Figure 8 Air velocities in animal occupied zone in different aisles when *D* was 3 m

4 Conclusions

The following conclusions were drawn from the field measurements, CFD simulation, and CFD model validation:

1) In the tunnel ventilation house, when the cooling pads were installed in a "[" shape, different airflow streams collided vertically at the house corners, mixed with each other, then flew towards the center of the house. This was the main reason that the wind speed in the middle aisle was much higher than that in other aisles, leaving large zones of lower ventilation in the aisles adjacent to the sidewalls.

2) The CFD simulation predicted that air velocity in the aisles close to the sidewalls can be significantly improved when the extra cooling pads are moved away for a certain distance from the end wall. For houses with 12 m width, the uniformity of airflow in all aisles can be significantly improved when this distance is about 3 m and the air velocity in different aisles is 1.5-2.0 m/s in most locations.

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