Multi Heat Source Collaborative Heating Scheme and CFD Assisted Air Distribution Design for a Large-Space Factory Building

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Abstract: With the increase in the number of large space buildings, especially those far away from the centralized Heat-supply network, how to design the indoor heat and humidity environment under the dual-carbon national strategy has become a matter of concern. This article focuses on a large space warehousing and logistics building in Harbin Logistics Park. Based on the calculation and analysis of typical annual heating and cooling load data, combined with the location of the building and the combination of cold and heat sources, air-cooled heat pump units are selected for annual cooling. CFD simulation is used to simulate the air distribution design, determine the stratified air conditioning mode with ejector nozzle air supply outlets and radiator, the paper provides a reference for the indoor environment creation of Large-space Factory Building under the condition of dual carbon background distributed heat sources.

Keywords: Large-space Factory Building; Collaborative Heating; CFD Simulation; Air Distribution

1. Introduction

At present, the number of logistics parks and other large-scale buildings are increasing year by year, heating energy consumption problems attract attention, such buildings are often far away from the city centralized heating pipe network, long-distance heating in winter will produce unnecessary heat loss, resulting in energy waste, Regarding this issue, it can be based on the characteristics of the building load and the local climatic conditions, to consider the use of distributed heat source in order to solve the problem of heat loss in the long-distance heat transfer. In addition, logistics park buildings belong to the high space, in addition to heating energy issues, the appropriate indoor airflow organization can meet the requirements of the working environment, airflow organization design focus not only needs to ensure that personnel activities as far as possible in the reflux zone, personnel activities in the area of wind speed to maintain the appropriate range of common requirements of the general building, but also need to pay special attention to the loss of energy, the temperature of the vertical stratification. Different air supply methods, air supply parameters, enclosure structure conditions and outdoor meteorological parameters and other factors will have a direct impact on the temperature vertical stratification, energy loss, which is precisely the air conditioning design of such buildings is difficult.

Article for Harbin Logistics Park, a large space warehousing logistics building, in the calculation and analysis of the typical annual supply (cold) heating load data, based on the location of the building cold and heat sources, the use of solar-assisted air-cooled heat pump unit to provide year-round heating (cold). Determine the heating end form of nozzle air supply and radiator, and apply FLUENT software to simulate the airflow organization of the building space, and the design provides a reference for the indoor environment creation of large space warehouse logistics building under the condition of distributed heat source with double carbon background.

2. Building Load Calculation

The building as shown in Figure 1 for the Harbin Logistics Park, a single building, length 157.44m, width 60.74m, height 9.2m, winter indoor design temperature of 10 ℃, summer indoor design temperature of 16 ℃, the calculation of the wall heat transfer coefficient of 1.58, the roof heat transfer coefficient of 0.95, the ground is divided into four areas, according to the degree of proximity to the outdoors using different heat transfer coefficients. The floor is divided into four zones and different heat transfer coefficients are used according to the distance from the outdoor.

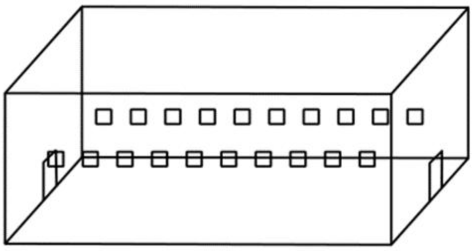

Figure 1. Physical Model of the Building

2.1 Calculation of Building Heat Load

The composition of winter heat load mainly contains the calculation of the basic heat consumption of the enclosure structure, the calculation of the ground heat transfer, the calculation of the additional heat consumption, and the heat consumption of cold air infiltration through the gaps of doors and windows [1].

After calculation, the whole heating season (November 1 ~ March 31) time-by-time heat load change curve shown in Figure 2, from the figure can be seen in the heating season in the early and late heating load is small, about 270kw. part of the heat load appeared in the negative value, through Figure 1 time-by-time temperature change curve can be learned, the heat load appeared in the negative value of the reason is because of in the heating season in the early and late at a certain time outdoor temperature is higher than the indoor design temperature. The maximum value of heat load occurs in the middle of the heating season (December and January), which is about 710kw, and the outdoor temperature corresponding to the maximum heat load can be seen in Figure 1, which is about -30℃.

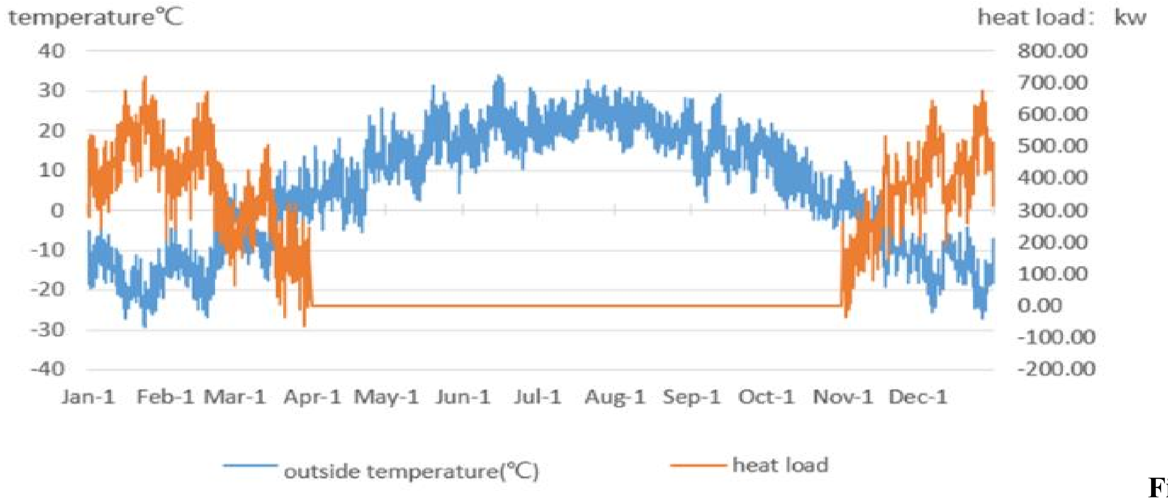

Figure 2. Heat Load Change Curve by Hour during the Heating Season

2.2 Building Cold Load Calculation

A large number of studies have shown that the temperature of stratified air-conditioning presents a step-like distribution in the longitudinal gradient, and the temperature stratification phenomenon is obvious [2, 3], and this feature is closely related to the calculation of indoor cooling load. Unlike the winter heat load calculation, the stratified air conditioning concept is applied in the calculation of the summer cold load [4], as shown in Figure 3. Note: Q_1w is the cold load of the enclosure of the air-conditioned area, and Q_2w is the cold load of the enclosure of the non-air-conditioned area. Qx is the cold load formed by the outdoor fresh or infiltration air, Q_f is the cold load formed by the indoor heat source, q_d is the cold load formed by convective heat transfer from the non-air-conditioned to the air-conditioned area, and q_f is the cold load formed by radiant heat transfer from the non-air-conditioned to the air-conditioned area.

Conventional cooling load calculations can be solved step by step in the above manner to add up the total cooling load, i.e.:

$$Q = Q_{1w} + Q_{2w} + Q_f + Q_x + q_d + q_f \quad (1)$$

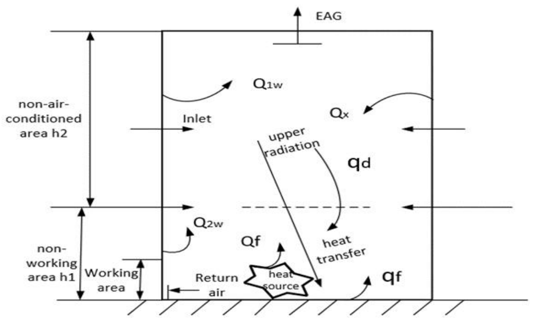

Figure 3. Schematic Diagram of Stratified Air-conditioning Cold Load Composition

In the design of the general calculation of the external wall roof heat transfer cold load, the external window temperature difference heat transfer cold load, the human body sensible heat cold load, the human body moisture dissipation latent heat formation of the cold load. After calculation, the empirical coefficient method is further used. That is, all the cold load multiplied by the empirical coefficient a, where a = air-conditioning district layer air-conditioning cold load / air-conditioning cold load of the whole room, when the lack of data, can be taken as a = 0.7. Finally obtained the logistics park typical day summer hour-by-hour cold load shown in Figure 4, the maximum cold load of 603.19Kw.

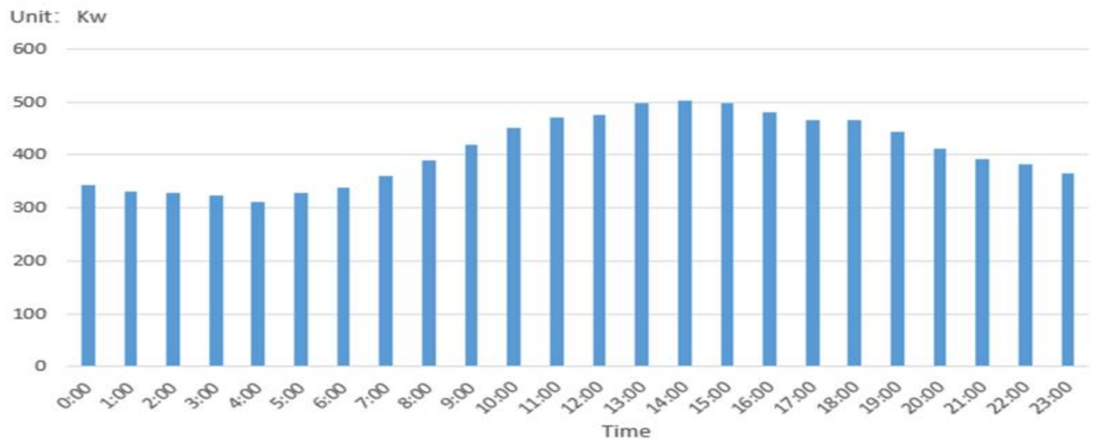

Figure 4. Hour-by-hour Cooling Load Map for a Typical Summer Day

3. Distributed Heating Determination

The logistics park is far away from the city heating pipe network, long distance heating in winter is easy to cause unnecessary heat loss, while the air-cooled heat pump unit uses air renewable energy as a source of cooling and heating, electric drive to achieve cooling and heating, does not take up the effective floor space, eliminating the cooling tower arrangement, with reliable operation, installation and ease of use and other advantages [5]. Considering the attenuation of heating effect under extreme weather conditions, 3 sets of F300 series air-cooled heat pump units produced by a certain factory are selected to calibrate the summer cold load according to their rated cooling capacity, and the rated cooling capacity of the air-cooled heat pump units is calibrated to meet the summer cold load. In addition, Harbin is a solar energy resource class II area with abundant solar energy resources, which is a renewable energy source. By GB50495-2009 *solar heating and heating engineering technical specifications* [6] and the spare area of the roof, to determine the total area of solar collector panels, and ultimately determine the solar-assisted air-cooled heat pump unit heating.

The solar collector area shall be selected in accordance with the following provisions:

$$A_c = \frac{86400 Q_h f}{J_\tau \eta_{cd}(1-\eta_L)} \quad (2)$$

Where:

Q_h——building heat load during the heating season, W;

f——Solar guarantee rate, %;

J_τ——annual average daily solar radiation, J/(m^2 -d);

η_{cd}——average collector efficiency based on total area, %;

η_{cd}——heat loss from piping and heat storage devices, %.

Solar energy installation needs to choose favorable installation angle and installation angle needs to be greater than the local latitude 10 °, select 1000mm * 2000mm * 80mm solar collector panels, from the angle of calculation of the collector panels projected area of $1.29m^2$, according to the "Solar Heating Engineering Technical Specification" appendix to select the appropriate parameters, combined with the roof of the actual usable area of $2000m^3$ calculations of solar collectors heating The heat capacity is 350kw, and it can be seen from Figure 1 that solar energy can bear 80% of the heat load in the pre-heating period and the post-heating period. Therefore, it is considered that solar energy is the main heating source in the early and late stages of heating.

4 Simulation Results and Analysis

Heating system end cooling methods mainly include radiator heating, low temperature hot water floor radiation heating, hot air heating and other methods [7]. After comprehensive consideration, it is determined that the building is heated by nozzle air supply through forced convection heat transfer plus radiator duty heating [8].

Due to the symmetry of the indoor airflow organization, the two sides of the corresponding air supply nozzles as a group, each group of airflow organization characteristics are basically the same, so this paper is only one group of nozzles during the work of the airflow organization simulation calculations, the model size of 60m × 16m × 9.2m, the room heat load of 105kw, the air supply mode for the upper send lower return [9, 10], the physical model shown in Figure 5.

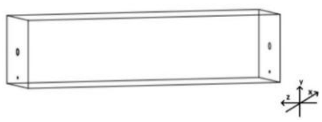

Figure 5. Schematic Diagram of Nozzle Air Supply

4.1 Simulation Results and Analysis

The following assumptions need to be made when simulating indoor airflow organization:

(1) The room is considered to be airtight and the effect of air leakage is not considered;

(2) The indoor airflow velocity is low and is considered an incompressible fluid, in accordance with the Boussinesq assumption;

(3) The airflow parameters at the nozzle are uniform and steady state flow;

(4) Since the window wall is relatively small, the window is considered as a wall and heat transfer is treated according to the wall temperature difference.

4.2 Boundary Conditions

Inlet and outlet boundary conditions: the outlet using pressure outlet, the inlet using velocity inlet, after a number of groups of simulation when the air velocity of 9.573m / s, the air temperature of 16 ℃, the air effect is more satisfactory.

Wall boundary conditions: Because the model verification is one of a group of nozzle airflow organization, so the model around the four sides of which there are two surfaces in contact with the indoor air is not involved in the enclosure temperature difference heat transfer, set the temperature conditions for 10 ℃ (constant temperature, uniform), the other two sides of the wall heat transfer coefficient of 1.58, the top heat transfer coefficient of 0.93, the outdoor temperature is -13.6 ℃. The rest of the wall was set as an adiabatic boundary.

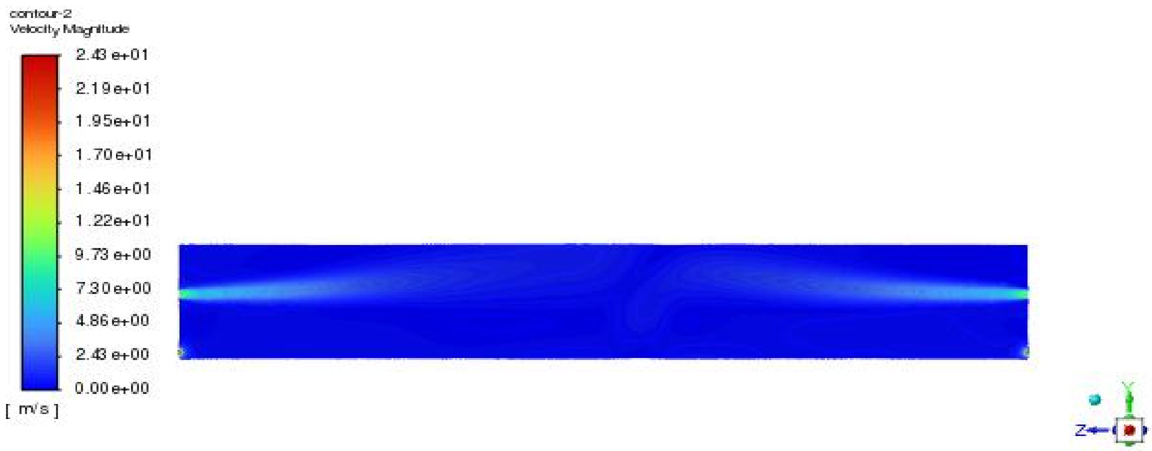

Figure 6. Vertical Fluid Velocity Field at the Center of the Nozzle

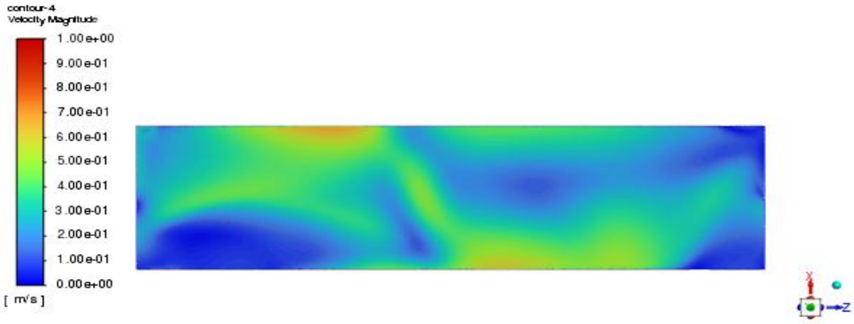

Figure 7. y=2 Plane Fluid Velocity Field

From Figure 6, it can be seen that the nozzle air velocity along the direction of air supply shows a gradual attenuation trend and air velocity change is getting slower and slower, in the center of the two sides of the nozzle air velocity of about 0m / s, meaning that the airflow in this convergence. Velocity change is not along the horizontal direction of air supply decay to 0 but along an upward curve decay to 0, by analyzing the hot air floating phenomenon, the airflow after leaving the nozzle, along the direction of air supply gradually upward movement. From Figure 7 can be seen, in the upper region of the entire work area that is the height of 2m when the maximum speed of airflow is about 0.07m / s, most of the region for the 0.05m / s, by the logistics park work attributes can be seen, 0.05m / s in line with the requirements of non-comfort air conditioning, the human body will not produce a strong sense of blowing, the speed of air supply to meet the requirements.

Figure 8. Fluid Temperature Field in the x=0 Plane

From Figure 8, it can be seen that the airflow in the nozzle at the highest temperature of the air supply, when the airflow away from the nozzle, the temperature of the air supply gradually decay, but due to the upward movement of hot air leads to the temperature decay rate is not along the horizontal direction of the air supply to decay to 0, this feature with the characteristics of the decay of air velocity in Figure 6 coincides with the characteristics of the airflow.

Figure 9. Fluid Temperature Field in the x=0 Plane

Figure 8 is only a part of the vertical direction of the intercepted plane, and can not completely explain the entire working area to meet the indoor design temperature, so intercept the horizontal plane that Figure 9 (y = 2 plane), combined with Figure 9 can be found in the entire working area indoor temperature of 10 ℃, to meet the indoor temperature design requirements.

Range:

$$X = 0.93S = 0.93(B - E) \quad (3)$$

Layered height:

$$h_1 = h + Y \quad (4)$$

h - height of the working area, m.

Round nozzle diameter:

$$d_0 = 0.064\left(\frac{T}{\Delta t_0}\right)^{0.615} X^{-0.302} Y^{0.687} u_x^{1.23} \quad (5)$$

T - absolute temperature of air in the air-conditioned zone, k;

△t0 - temperature difference of air supply, ℃.

Round nozzle air supply velocity:

$$u_0 = 4.925\left(\frac{T}{\Delta t_0}\right)^{-0.951} X^{1.124} Y^{-0.553} u_x^{-0.182} (6)$$

Air volume per air supply outlet:

$$l = \frac{\pi}{4} d_0^2 v_0 \times 3600 \quad m^3/h \quad (7)$$

Total airflow:

$$\mathrm{L} = \frac{3600Q}{1.01 \times \rho \times \Delta t_0} \quad m^3/h \quad (8)$$

Number of nozzles:

$$\mathrm{n} = \frac{L}{l}$$

Take n to be an integer and find the actual wind speed

$$u_0 = \frac{4L}{\pi d_0^2 n \times 3600} \quad m/s \quad (9)$$

$$Ar = \frac{g \cdot \Delta t_0 \cdot d_0}{u_0^2 \cdot T} \quad (10)$$

g - acceleration of gravity, m^2 /s.

Find the actual drop Y' and the axial velocity ux' at the end of the jet:

Axial trajectory equations:

$$\frac{Y}{d_0} = 0.812 Ar^{1.158}\left(\frac{X}{d_0}\right)^{2.5} \quad (11)$$

Axial velocity decay equation:

$$\frac{u_x}{u_0} = 3.347 Ar^{-0.147}\left(\frac{X}{d_0}\right)^{-1.51} \quad (12)$$

Bringing in the relevant data to get $Ar=1.27\times10^{-3}$, Y'=5.28m, verify △Y=|Y'-Y|=0.08≤0.2m, meet the requirements, the jet axial velocity u_x =1.03m/s, the return area wind speed u_p =0.5×1.03=0.515m/s≈0.5m/s, meet the relevant specification requirements.

After the above calculations to determine the final height of the work area h for 2m, drop height Y for 3.2m, circular nozzle diameter of 0.57mm, nozzle wind speed of 9.57m, the number of nozzles for the 20, will be the results of the calculations and simulation results compared to the diameter of the nozzle and the speed of the air supply is basically the same, it can be approximated that the design is more reasonable.

6. Conclusion

By calculating the cooling and heating load of the building, the heating mode of solar-assisted air-cooled heat pump unit is determined in the case of distributed energy. The combination of simulation and calculation was used to determine the air supply speed and temperature of the heating system end nozzle, and the simulation found that the overall pattern of indoor airflow organization and distribution is basically consistent with the design concept to meet the requirements of the indoor design temperature.

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