Available online www.jocpr.com

Journal of Chemical and Pharmaceutical Research, 2013, 5(10):363-368

Research Article

ISSN : 0975-7384
CODEN(USA) : JCPRC5

Effect of ventilation tube spacing on refrigerated warehouse floor antifreezing mechanical ventilation system

Wei HE and Jing-fu JIA

Ocean College of Hebei Agricultural University, Qinhuangdao, Hebei, China

---

ABSTRACT

To investigate the effect of ventilation tube spacing on refrigerated warehouse floor antifreezing mechanical ventilation system, a steady three-dimensional mathematical model of heat transfer is built in this paper, and the calculational conditions of the heat transfer model are defined according to the heat transfer process and reasonable simplification. The temperature fields of this system are simulated and calculated by CFD software PHOENICS under different kinds of working conditions, such as the mechanical ventilation system isn't circulating, and the mechanical ventilation system is circulating with different ventilation tube spacing. The results show the advantages of the refrigerated warehouse floor antifreezing mechanical ventilation system. They also indicate the greater effect of ventilation tube spacing on thermal performance of this system. It is pointed that the proper tube spacing is 2.0m.

Key words: Refrigerated warehouse floor, Antifreezing mechanical ventilation, Ventilation tube spacing

---

INTRODUCTION

Antifreezing of refrigerated warehouse floor is very important, and it shouldn't be neglected during designing, constructing and running. Thermal insulating layer is built in refrigerated warehouse floor to match the demand of refrigerated warehouse temperature, but the insulating layer couldn't prevent the freezing of soil under the floor. It can only postpone the freezing time [1]. A greater temperature difference will produce between refrigerated warehouse temperature and soil under the floor when refrigerated warehouse temperature is reduced. So the cold energy should transfer from refrigerated warehouse to soil. The temperature of soil layer will fall down. The refrigerated warehouse temperature is always below 0℃ throughout the year. The moisture contained in the soil under the floor would be frozen into ice if without heat supplement. The expansive force generated by the frozen ice will cause the floor or foundation humped. It is injurious to the safety of building and construction [2]. Therefore, measures of ground antifreezing should be adopted when the design temperature of first floor in refrigerated warehouse is below 0℃, to maintain the soil under refrigerated warehouse floor at a temperature above 0℃.

Ventilation is the most common measure of refrigerated warehouse floor antifreezing. It is realized by providing heat to refrigerated warehouse floor through mechanical or natural ventilation using buried pipes. Some empirical values are often used to design and build refrigerated warehouse floor antifreezing ventilation system for years. Deeper researches on this system are not commonly known, unless in document [1, 3]. So a systematic research is of practical significance to direct design, construction and operation better.

The refrigerated warehouse floor with antifreezing ventilation system is regarded as research object in this paper, and a steady state three-dimensional mathematical model of heat transfer is built. The heat-transfer model is simplified reasonably, and the calculational conditions are defined according to the heat-transfer process. The temperature fields of this system are simulated and calculated using CFD software

PHOENICS[4,5].under different kinds of working conditions, such as without circulating the ventilation system, with different tube spacing. The effect of ventilation tube spacing on thermal performance of refrigerated warehouse floor antifreezing ventilation system is investigated.

EXPERIMENTAL SECTION

ARITHMETIC ELEMENT OF HEAT-TRANSFER MODEL

A tiny part of the cold-storage room floor and soil layer is picked out and used as an arithmetic element of the heat-transfer model, as shown in Fig.1. The size of the model is defined as:

X×Y×Z=1500mm×3900mm×1000mm.

The top-down structure layers of the refrigerated warehouse floor are [7]: reinforced concrete surface course, cement mortar protection course, asphalt felt damp-proof course, rigid polyurethane foam heat insulating layer, asphalt felts vapour barrier, cement mortar leveling course, precast concrete board, medium sand packing layer(buried concrete ventilation tube within). Soil layer is below structure layers of floor.

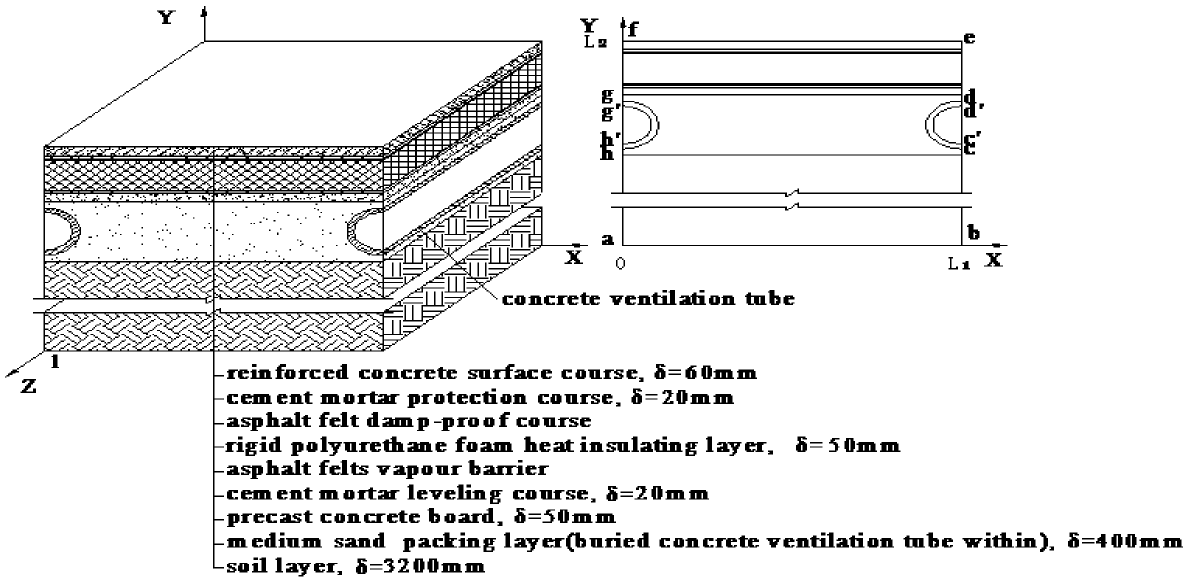

Fig.1 The calculating unit of heat transfer model for refrigerated warehouse floor

MODEL ASSUMPTIONS

The heat transfer process of the refrigerated warehouse floor is very complex in fact. But to facilitate solving and analyzing, the heat-transfer model is simplified and assumed reasonably. These assumptions are as follows: Firstly, the heat transfer process of the refrigerated warehouse floor is assumed to be a steady state heat conduction process of three-dimensional. Secondly, the materials in the same structural layer of the shipboard are assumed to be homogeneous, isotropic and with constant physical properties. Lastly, all the thermal contact resistance between each layer, the thermal resistance of the asphalt felt layer and the moisture transfer are assumed to be negligible.

EQUATION AND BOUNDARY CONDITIONS

The research object is the refrigerated warehouse floor in Tianjin area of China in winter in this paper. There is not heat source in the refrigerated warehouse floor, so the steady state three-dimensional differential equation of heat conduction of the arithmetic element of the heat-transfer model may be described as follow:

$$\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}+\frac{\partial^2 t}{\partial z^2}=0. \quad (1)$$

The boundary conditions of the arithmetic element are defined as follows:

The design temperature of the cold-storage room floor is defined as: t_n =-20℃. There are fan blowers in the cold-storage room, therefore the heat convective coefficient of the upper surface of the cold-storage room floor is defined as: α_n = 12W/m^2·℃ [2]. So the boundary condition of the upper surface of the cold-storage room floor is:

$$-\lambda\frac{\partial t}{\partial y}\Big|_{y=L_2}=\alpha_n\left(t_n-t_{nb}\right)=12\times\left(-20-t_{nb}\right). \tag{2}$$

Where, λ is the heat conduction coefficient of each kind of material (W/m·℃), t_{nb} is the temperature of the upper surface of the cold-storage room floor (℃).

The soil temperature is defined as: $t_s = 10.4$ ℃. The soil temperature is derived from the minimum mean soil temperature of 3.2 meters deep in Tianjin city in two months, March and April, over the years [5]. So the boundary condition of the lower surface of the heat-transfer model is:

$$t\Big|_{y=0}=t_s=10.4\text{℃}. \tag{3}$$

In the refrigerated warehouse floor, ventilation tubes are symmetrically placed in terms of certain spacing in X-direction, so in the arithmetic element as shown in Fig.1, the two boundary surfaces of the model in X-direction are approximately seemed as adiabatic surfaces and the boundaries are:

$$\frac{\partial t}{\partial x}\Big|_{x=0}=\frac{\partial t}{\partial x}\Big|_{x=L_1}=0 \tag{4}$$

Similarly, the two boundary surfaces of the model in Z-direction are also approximately seemed as adiabatic surfaces and the boundaries are:

$$\frac{\partial t}{\partial z}\Big|_{z=0}=\frac{\partial t}{\partial z}\Big|_{z=1}=0 \tag{5}$$

The air supply velocity and temperature of ventilation tubes in the refrigerated warehouse floor are defined as: v = 1.5m/s, t_{in} = 10℃. The inner diameter and outer diameter are: d_1=250mm, d_2=316mm. The boundary condition of outer surface of ventilation tube is:

$$t=t_{w2}=t_f+\frac{V\cdot\rho\cdot c_p\cdot\left(t_{out}-t_{in}\right)}{K\cdot l}\left(1+\frac{1}{2\pi\lambda}\ln\frac{d_2}{d_1}\cdot K\cdot l\right)\quad x,y\in c\wedge d、g\wedge h，z\in(0,1) \tag{6}$$

Where, t_{w2} is the temperature of outer surface of ventilation tube (℃), $t_f=\frac{t_{in}+t_{out}}{2}$ is mean air temperature of ventilation tube (℃), V is the air flow of the system (m^3/s), ρ is the air density (kg/m^3), c_p is specific heat at constant pressure of air (J/kg·℃), t_{out} is the mean temperature of air at outlet of the system (℃), K is the coefficient of heat-transfer between air and tube wall in ventilation tube on unit length (W/m·℃), l is total length of ventilation tube.

RESULTS AND DISCUSSION

SIMULATION RESULTS OF HEAT-TRANSFER MODEL

When the ventilation system isn't circulating, or the ventilation system is circulating with different tube[8,9]spacing of 1.5m and 2.0m, and the other simulation conditions are not changed, numerical simulations of the heat-transfer model of the refrigerated warehouse floor are performed using PHOENICS software. The obtained temperature distributions of the refrigerated warehouse floor of the model in each case are shown in Fig.2, Fig.3and Fig.4. The temperature distribution curves of points at Z＝1m on upper, middle and lower surface for heating layer with 1.5m or 2.0m tube spacing are shown in Fig.5 and Fig.6.

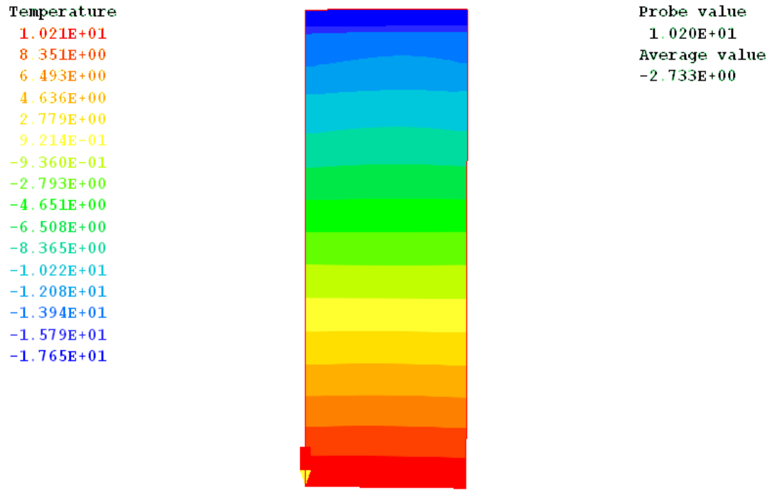

Fig.2 The temperature field of refrigerated warehouse floor that ventilation system isn't circulating

The obtained numerical result is shown in Fig.2 when ventilation system isn't circulating. The average temperature of upper surface of heating layer is -12.92℃. The average temperature of lower surface of heating layer is -10.32℃.

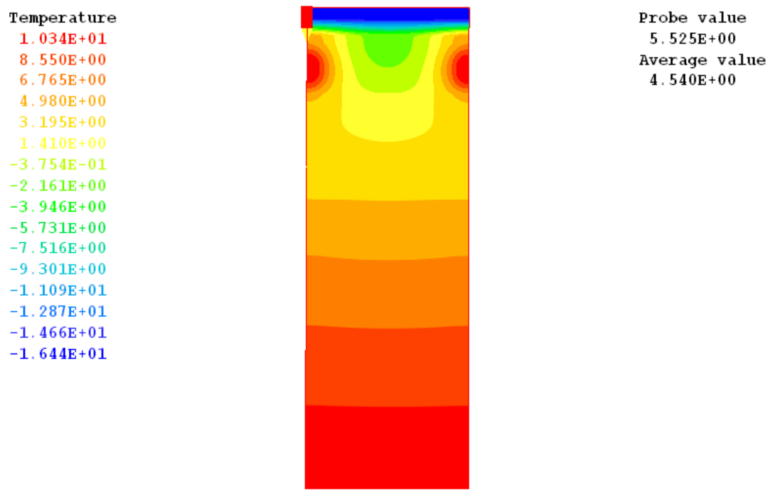

Fig.3 The temperature field of refrigerated warehouse floor that ventilation system is circulating with 1.5m tube spacing

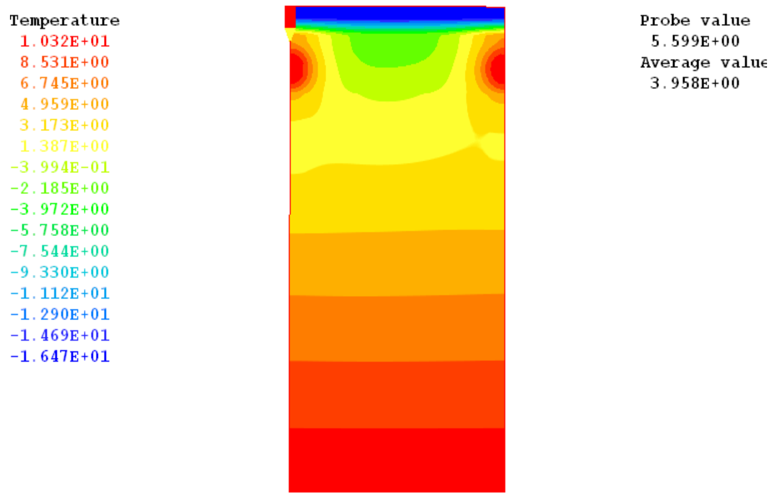

Fig.4 The temperature field of refrigerated warehouse floor that ventilation system is circulating with 2.0m tube spacing

The obtained numerical results are shown in Fig.3 and Fig.5 when ventilation system is circulating with tube spacing of 1.5m. The average temperature of upper surface of heating layer is 1.290℃. The average temperature of lower surface of heating layer is 2.322℃. The average temperature of the surface at Z＝1m is 4.54℃. The temperature fluctuation amplitudes of each point at Z＝1m on upper surface of heat layer are 7.723℃(from -2.186 to 5.537℃). The temperature fluctuation amplitudes of each point at Z＝1m on lower surface of heat layer are 4.424℃(from 0.387 to 4.811℃).

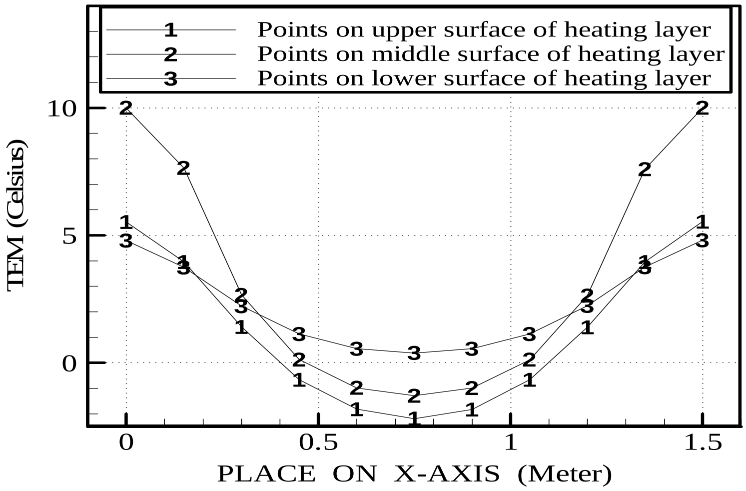

Fig.5 The temperature distribution curves of points at Z＝1m on upper, middle and lower surface for heating layer with 1.5m tube spacing

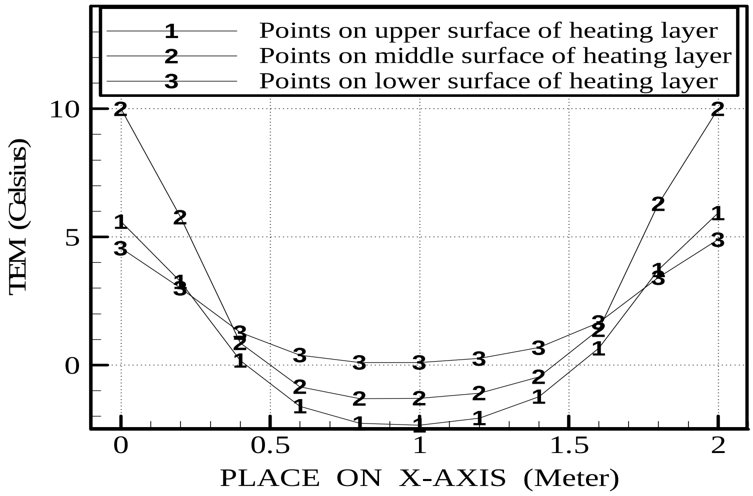

Fig.6 The temperature distribution curves of points at Z＝1m on upper, middle and lower surface for heating layer with 2.0m tube spacing

The obtained numerical results are shown in Fig.4 and Fig.6 when ventilation system is circulating with tube spacing of 2.0m. The average temperature of upper surface of heating layer is 0.642℃. The average temperature of lower surface of heating layer is 1.73℃. The average temperature of the surface at Z＝1m is 3.958℃. The temperature fluctuation amplitudes of each point at Z＝1m on upper surface of heat layer are 8.282℃(from -2.346 to 5.936℃). The temperature fluctuation amplitudes of each point at Z＝1m on lower surface of heat layer are 4.795℃(from 0.104 to 4.899℃).

NUMERICAL RESULTS ANALYZING

In terms of the obtained temperature distribution figures, the numerical results are analyzed. When ventilation system isn't circulating, the temperature distribution of each point on Y plane of each structure layer of the refrigerated warehouse floor is more uniform, and the average temperature of each Y plane is far below 0℃. Therefore, if the ventilation system isn't circulating, soil under the heating layer would be frozen into ice, and the expansive force generated by the frozen ice will cause the floor or foundation humped.

When ventilation system is circulating, the temperature distribution of each point on Y plane of each structure layer of the refrigerated warehouse floor is nonuniform, but the average temperature of lower surface of heating layer is above 0℃. When tube spacing is smaller, the temperature fluctuation amplitudes of points on upper and lower surface of heating layer are both smaller, and the average temperatures of the upper and lower surface are both higher. When tube spacing is smaller, the heat-exchange amount is bigger, the cold energy transferred from refrigerated warehouse floor to soil layer reduces substantially. For the higher temperature of the lower surface of heating layer, the moisture of soil can prevent freezing effectively. So the refrigerated warehouse floor can avoid being frozen and humped.

CONCLUSION

Based on the developed model of the refrigerated warehouse floor, performing numerical simulations and analyzing the obtained temperature distributions, some conclusions are summarized as follows:

If no measure of refrigerated warehouse floor antifreezing is adopted, the average temperature of lower surface of refrigerated warehouse floor will be far below 0℃. For a long time, soil layer under the refrigerated warehouse floor would be frozen, and the expansive force generated by the frozen ice will cause the floor humped.

If antifreezing mechanical ventilation measure is adopted to refrigerated warehouse floor on the basis of document [4], under the simulation conditions of this paper, the average temperature of lower surface of heating layer will be always above 0℃. The effectiveness of antifreezing is better.

Ventilation tube spacing has great influence both on the average temperature of upper and lower surface of heating layer and on the temperature distribution in the refrigerated warehouse floor antifreezing ventilation system. When the other simulation conditions are not changed, if tube spacing is smaller, the temperature fluctuation amplitudes of points in heating layer are smaller, and the effectiveness of refrigerated warehouse floor antifreezing is better. But these can cause the cost of tube to increase, and keep the temperature of the heating layer higher. Then, the cold load of refrigerated warehouse will be bound to increase, and the energy waste will be inevitable. Therefore, based on an overall consideration of various factors, the proper tube spacing is 2.0m under the simulation conditions of this paper.

Acknowledgement
We kindly acknowledge financial support by the science and technology research and development Program of Qinhuangdao city (No.2012022A010). It is also important to note that these two authors contributed to this paper equally and should be regarded as co-first authors.

REFERENCES

[1] Zukun Gao. *Heating Ventilation* & *Air Conditioning*, **1996**, 26(4), 68-71.
[2] Jianhua Li, Chun Wang. Refrigerating House Design, 1st Edition, China Machine Press, Beijing, **2003**; 1-42.
[3] Ran Zhou, Jinyuan Chen, Suilin Wang, et al. *Cold Storage Technology*, **2007**, 30(4), 37-40.
[4]Zhao Xi, Zhang Zhimin, Zhang Baohong. *Journal of Chemical and Pharmaceutical Research*, **2013**, 5(9):549-554.
[5] Liu Jie-qun, Chen Lu-wang, Liu Jin-long. *Journal of Chemical and Pharmaceutical Research*, **2013,** 5(9):362-371
[6] The original State Administration for Domestic Trade of the People's Republic of China. Code for Design of Cold Store, 1st Edition, China Planning Press, Beijing, **2001**; 1-64.
[7] Chibin Yu. Practical Handbook of Refrigeration and Air-Conditioning Engineering, 1st Edition, China Machine Press, Beijing, **2001**; 297-314.
[8] Lingfeng LI, Changhui XU, Yunxia Chen, *et al. Journal of Chemical and Pharmaceutical Research*, **2013**, 5(9):555-562
[9]Srashti Dwivedi, Priyanka Jain. *Journal of Chemical and Pharmaceutical Research*, **2013,** 5(6):12-13.