Study of the Heating Load of a Manufactured Space with a Gas-fired Radiant Heating System

Xuejing Zheng
Doctoral Candidate
Department of Building Environment and Equipment Engineering, Tianjin University
Tianjin, PR China
xuejinghappy@yahoo.com.cn

Zongcheng Dong
Professor
School of Municipal & Environmental Engineering,
Harbin Institute of Technology,
Harbin, Heilongjiang, PR China
dzc1214@163.com

Abstract: A thermal balance mathematics model of a manufactured space with a gas-fired radiant heating system is established to calculate the heating load. Computer programs are used to solve the model. Envelope internal surface temperatures under different outdoor temperatures are obtained, and the heating load of the manufactured space is analyzed. The relationship between the envelope internal surface temperature and the workspace temperature is also analyzed in this paper. CFD simulation software is used to simulate the temperature field and the envelope's internal surface temperature of the manufacture space with hot-air heating system. Comparison and analysis of heating loads are done between the manufactured spaces with convection heating and radiant heating systems.

Key words: gas-fired radiant heating system; manufacture Space; internal surface temperature; heating load

1. INTRODUCTION

Gas-fired infrared radiant heating system is one of the advanced heating equipments in the world at present. Infrared heating is the transmission of the energy by means of low-intensity electromagnetic waves. This system consists of burner, burner housing, emitter tube, reflector, and vacuum fan. Radiant heat warms up cold bodies directly without heating the whole building. Radiant heat offers considerable energy savings over warm air systems, especially under otherwise difficult conditions. Consequently, radiant heating responds more quickly than warm air to the needs of a manufacture Space. The shorter warm up periods save fuel and allow the heating system to match flexible, modern working arrangements.

Under the same thermal comfort, the indoor air temperature in the room with radiant heating system is 2-3℃ lower than that with convection heating system.[1]

Heat transferred though envelope is always calculated as follow:

$$Q = KA(t_{in} - t_{out}) \tag{1}$$

Where K——total heat-transfer coefficient between indoor air and outdoor air, (W/(m^2 • ℃))

A——area of the envelope, (m^2)

t_{in}——temperature of indoor air, (℃)

t_{out}——temperature of outdoor air, (℃) .

In fact, heat transferred though the envelope is exactly calculated as follow:

$$Q = \frac{A\left(t_i - t_{out}\right)}{R_i} \tag{2}$$

Where R_i——thermal resistance between envelope internal surface and outdoor air, ((m^2 • ℃)/W)

t_i —— temperature of envelope internal surface, (℃) .

As shown in Equation (2), heating load is determined by temperature of envelope internal surface, no matter what heating system is applied. With the same envelope, temperature difference between envelope internal surface and out door air is the major factor to determine heating load of the room.

As the difference in heat transfer principle, in rooms with convection heating system and radiant heating system, the temperature of envelope internal surface will be different. Under the same thermal comfort, though the indoor air temperature in the room with radiant heating system is lower, it is uncertainty that the temperature of envelope internal surface is lower. So there is difference in heating load when Manufacture Space applied gas-fired radiant heating system and hot-air heating system. This paper compared the heating load.

2. PHYSICAL MODEL OF THERMAL BALANCE

The figure of Manufacture Space horizontal is always rectangle, L form, or two-L form. The Manufacture Space in this paper is one with rectangle figure. Gas-fired radiant heating equipments are hung symmetrically in a horizontal, as shown in Fig.1 and Fig. 2.

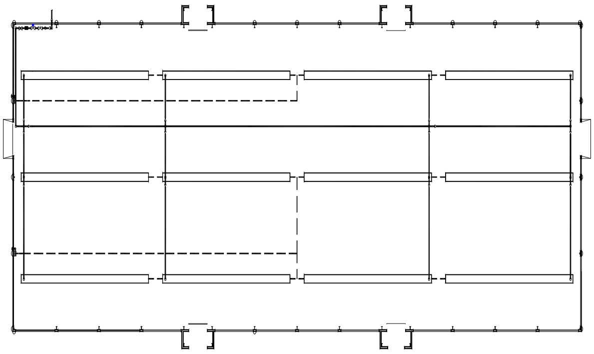


Fig. 1 Horizontal sketch of gas-fired radiant heating equipments

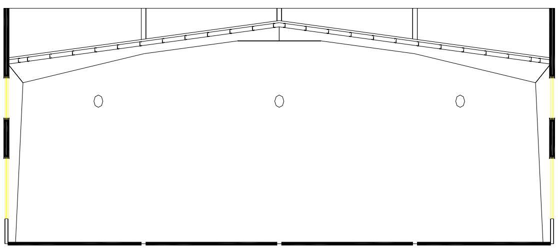


Fig. 2 Vertical sketch of gas-fired radiant heating equipments

3. MATHEMATICAL MODEL OF THERMAL BALANCE

Heat transfer processes in Manufacture Space: conduction forced by temperature difference between envelope internal surface and outdoor air, radiation between the heating equipments and envelope internal surface, convection between the heating equipments and indoor air, radiation among envelope internal surfaces, convection between envelope internal surfaces and indoor air. Thermal balance in this paper consists of thermal balance of envelope internal surfaces and thermal balance of indoor air. Temperatures of envelope internal surfaces are calculated by solving a series of equation.

3.1 Thermal Balance of Envelope Internal Surfaces

As the blackness of envelope internal surfaces is generally larger than 0.9[2], reradiate is neglected in this paper. For each internal surface, equation of thermal balance is formed. (Equation 3)

$$Q_{tran,i} + Q_{con,i} + Q_{rad1,i} + Q_{rad2,i} = 0 \qquad (3)$$

Where $Q_{tran,i}$ ——conduction though envelope, (W)

$Q_{con,i}$ —— convection between envelope internal surfaces and indoor air , (W)

$Q_{rad1,i}$ ——radiation between equipments and envelop internal surface, (W)

$Q_{rad2,i}$ ——radiation among envelope internal surfaces, (W)

Conduction though envelope $Q_{tran,I}$ is calculated as Equation (4):

$$Q_{tran,i} = \frac{1}{\frac{1}{k_w} + \frac{1}{\alpha_0}} \left(t_i - t_{out} \right) A_i \qquad (4)$$

Where α_0 ——convective heat-transfer coefficient of envelope external surface, W/(m^2·℃)

k_w ——conductivity coefficient of envelop, W/(m^2·℃)

A_i —— area of the i^{th} envelope internal surface, m^2

Convection between the i^{th} envelope internal surface and indoor air $Q_{con,i}$ is calculated as Equation

(5):

$$Q_{con,i} = \alpha_i (t_i - t_{in}) A_i \qquad (5)$$

Where α_i ——convective heat-transfer coefficient of the i^{th} envelope internal surface, W/(m^2·℃)

Radiation between the heating equipments and envelope internal surface $Q_{rad1,j}$ is calculated as Equation(6):

$$Q_{rad1,i} = \sum C_b \varepsilon_{ie} \varphi_{ie} [(\frac{T_i}{100})^4 - (\frac{T_e}{100})^4] A_i \qquad (6)$$

Where C_b —— Stefan-Boltzman Constant, (5.67W • m^{-2} • K^{-4})

ε_{ie} ——coefficient of blackness, equal to product of two coefficients of blackness of surfaces, $\varepsilon_{ie} = \varepsilon_i \varepsilon_e$

φ_{ie} ——angle factor of the i^{th} envelop to heating equipments,

T_i —— absolute temperature of the i^{th} envelope internal surface, K

T_e —— average absolute temperature of heating equipments surfaces, K

Radiation among envelope internal surfaces $Q_{rad2,j}$ is calculated as Equation (7):

$$Q_{rad2,i} = \sum C_b \varepsilon_{ik} \varphi_{ik} [(\frac{T_i}{100})^4 - (\frac{T_k}{100})^4] A_i \qquad (7)$$

Where ε_{ik} ——coefficient of blackness, equal to product of two coefficients of blackness of surfaces, $\varepsilon_{ik} = \varepsilon_i \varepsilon_k$

T_k —— absolute temperature of the k^{th} envelope internal surface, K

3.2 Thermal Balance of Indoor Air

To study the vertical gradient of temperature, the Manufacture Space in this paper is divided into sections, in which temperature distribution is level off degree. In each section, thermal balance of air is shown in Equation (8).

$$Q_{econ,i} + Q_c + Q_{con,i} + Q_{sf,i} = 0 \qquad (8)$$

Where $Q_{econ,i}$ ——convection between the heating equipments and indoor air, W

Q_c ——heat transfer among sections, W

$Q_{sf,i}$ ——heat loss of the i^{th} section by infiltration, W

Convection between the heating equipments and indoor air is calculated by using equation (9):

$$Q_{econ,j} = \alpha_e (t_e - t_{in}) A_e \qquad (9)$$

Where α_e ——convective heat transfer coefficient of heating equipment surface, W/(m^2·℃)

t_e ——temperature of heating equipment surface, ℃;

A_e ——area of heating equipment, m^2

Convective heat transfer coefficient of surface is calculated as Equation (10):

$$\alpha = A_c \sqrt[3]{\Delta t} \qquad (10)$$

Where Δt —— temperature difference between surface and ambience air, ℃

A_c ——coefficient, A_c is given in Ref. [2]

Angle factor is calculated as method in Ref [3] and Ref. [4], air change method is applied to calculate heat loss by infiltration. Floor in the model is uninsulated. Method of weighted mean is applied to calculate conductivity coefficient of envelop with windows.

There is temperature gradient in the space because of mass flow between sections. In each section, mass exchange at the boundary surface must follow mass conservation. Mass flux is calculated as method given in Ref. [5].

To solve the nonlinear equations (3)-(10), Monte Carlo method is used in this paper.

4. SIMULATION OF MANUFACTURE SPACE WITH HOT-AIR HEATING SYSTEM

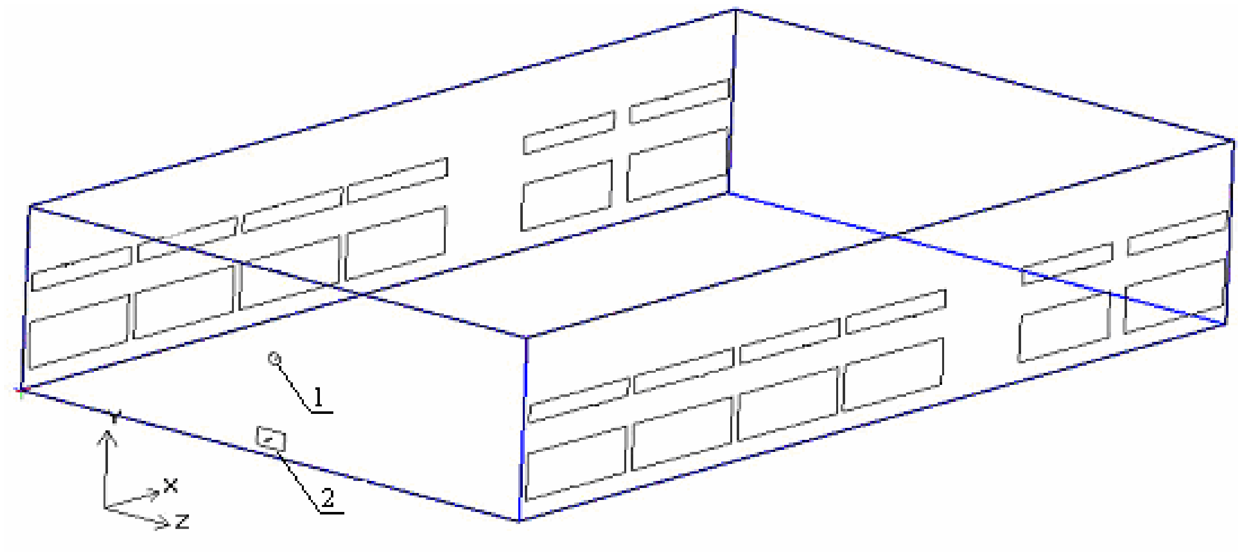
Radiator heating system is always used in cold area in winter. This system does not adapt to Manufacture Space which is higher than 10m for its large heat loss. As warm air rise by buoyancy force to areas where it is not beneficial, heating loss also rise. Therefore the roof of a Manufacture Space heated with warm air will be substantially hotter than the working area beneath. Poor roof insulation will allow this heat to escape, further increasing fuel costs. Hot-air stratified heating system is usually used, which blast inlet is set at the middle to separate space into heated space and unheated space. The dimensions of blast inlet and return intake, the supply air speed and temperature are calculated by the method given in Ref. [1].

In this paper, field distributions of temperature in a Manufacture Space with hot-air stratified heating system is simulated using a commercial CFD package Airpak. For the geometrically symmetry of Manufacture Space, half in length of the space is selected to be simulation cell model (Fig. 3). Among various turbulent models featured in the software package, the two-equation (standard $k-\varepsilon$) turbulent model is applied to simulation. Because structured grids have higher quality than unstructured grids, the space is divided into structured grids (hexahedral cell topology) using multigrid method. Parameters of the objects in simulation model are shown in Tab. 1.

5. RESULTS

This paper numerically simulated two Manufacture Spaces, one with hot-air heating system, and the other with gas-fired radiant heating system. With different room temperature, envelope internal surface temperatures are shown in Tab. 2.

When using radiant heating system, the envelop internal surface temperature of the model is about 0.4-0.9℃ lower than room temperature, and about 0.6-1.1℃ higher than the envelop internal surface temperature of the model with hot-air system. The difference increased as the room temperature increased.



1-blast inlet 2-return intake

Fig. 3 Simulation cell model

Tab. 1 Parameters in simulation model

Object		Value
Room		dimension 120.4×36.4×11.4m^3
Envelope	Wall	heat-transfer coefficient k=0.399W/(m^2 • ℃)
	Window	heat-transfer coefficient k_w=2.5 W/(m^2 • ℃)
Room temperature (Working area) (℃)		12～15℃
Outdoor air temperature (℃)		-26℃

Tab. 2 Envelop internal surface temperatures of models (℃)

Heating system \ Room temperature (℃)	12	12.5	13	13.5	14	14.5	15	15.5	16
Hot-air heating system	10.56	11.04	11.52	12.00	12.48	12.96	13.45	13.93	14.41
Gas-fired radiant heating system	11.12	11.66	12.21	12.76	13.31	13.87	14.42	14.98	15.55

Under the same thermal comfort, the indoor air temperature in the room with radiant heating system is 2℃-3℃ lower than that with convection heating system. But the envelop internal surface temperature decreased only 1.4-2.4℃. Calculated by Equation (2), heating load of the model with radiant system in this paper is 94%-97% of the load of the model with hot-air system.

6. CONCLUSION

Under the same thermal comfort, the indoor air temperature in the room with radiant heating system is 2℃-3℃ lower than that with convection heating system. But the difference between the envelope internal surface is lower. The decrease in heating load is smaller. It increased while the heat transfer coefficient of envelope decreased. Heating load of the model with radiant system in this paper is 94%～97% of the load of the model with hot-air system.

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