NUMERICAL MODELING OF TRANSIENT HEAT TRANSFER IN MICROSYSTEM OF PROTECTIVE CLOTHING

by

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Original scientific paper
DOI: 10.2298/TSCI1603945D

Heat protective clothing is always being treated as a main personal protective equipment to shield robust flame injection and high temperature, therefore, it is significant and essential to investigate transient heat transfer and heat insulation ability of heat protective clothing. In this paper, a novel co-operative model composed of heat protective clothing, air gap, and test sensor was established under the convection and radiation heat source, and the temperature and heat flux were numerically solved by finite element method. The results showed an acceptable agreement between the experimental data and numerical prediction.

Key words: *heat protective clothing, air gap, finite element method*

Introduction

A wide variety of emergencies, *e. g.* fire hazard and inflammable gas leakage, have led to serious burn of firefighters and workers during industrial manufacturing. Heat protective clothing is commonly deemed as the significant shielding layer which is fabricated by some flame retardant and heatproof fibers, such as Nomex, Kevlar, and PTFE, *etc*. [1-6]. Experimental and numerical investigation of the protective clothing have already been carried out during past several decades, theoretical methods could not be readily applied in protective clothing owing to the complex boundary condition and changeable heat transfer models, luckily, the numerical methods are widely utilized to solve the problems in this field. In details, Torvi *et al*. [7] established a heat transfer model of one-layer fabric when subjected to the high heat fluxes used in bench top tests. Also, the apparent heat capacity method was utilized to represent thermo-chemical reactions in fabric. Song *et al*. [8] treated the fabric matrix as the porous media and the effect of fabric configuration was also simulated. Zhu *et al*. [9] proposed a heat-transfer model considering thermal degradation of heat-resistant fabrics when subjected to the radiant heat flux. In most mentioned research, the heat transfer in the air gap was always simplified and assumed that the heat conduction and radiation in the air gap were decoupled and the heat transfer was treated

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at the steady-state condition while the radiation was denoted as the surface radiation meaning that the air gap could not absorb and emit any radiation during the entire heat transfer. Therefore, Ghazy *et al.* [10] proposed a more reasonable model considering the accurate condition in the air gap, but the calculation process was seemed to be complex and cannot been handled easily. In this paper, a readily numerical finite element method was employed to deal with the transient heat transfer of heat protective clothing via finite element method, the resultant temperature and heat flux located at the surface of the fabric and the sensor were elucidated and compared with the experimental data from the previous literature.

Modeling of the transient heat transfer

The original experimental apparatus of this model based on the bench top (fabric-air gap-test sensor) system was illustrated in fig. 1. The cone heater located at bottom could release constant robust heat flux more than 80 kW/m^2, such heat flux transferred to the fabric surface through convection and radiation portion, the convection part from the burner's hot gas to the fabric outside was denoted as q_{cnv} while the radiation portion exchanged between the burner, fabric and ambient was named as q_{rad}. In the fabric section, the heat transfer was mainly through heat conduction, but, considering the in-depth absorption of radiation. In the air gap section, the thick air gap ($\delta < 6$ mm) could transfer the heat flux through heat conduction, q_{cnd}, and heat radiation, q_{rad}, to test senor. Test sensor was utilized to record the temperature and further assessed the skin burn degree. The backside of test sensor was set at the constant ambient temperature, T_{amb}.

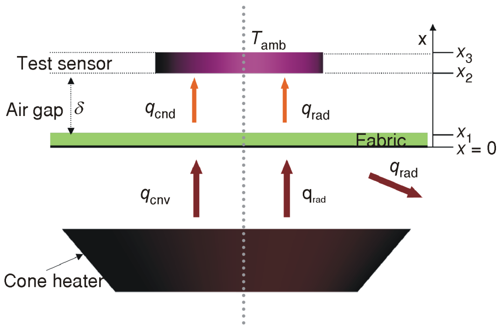

Figure 1. The fabric-air gap-test senor system

Before establishing the model of the experimental system, some assumptions should be defined first: (1) The 1-D heat transfer was employed in this model in order to simplify the problem, (2) the fabric was treated as a homogeneous slab with the effective thermal properties of the actual complex fibrous structure, (3) the moisture transfer was neglected in this model, (4) only infrared radiation was considered here as the marginal amount of ultraviolet radiation was present, and (5) the apparent heat capacity method was incorporated into the fabric heat transfer equation.

– Under these assumptions, the governing equation of fabric is listed:

$$\rho C^{A}(T)\frac{\partial T}{\partial t} \quad \frac{\partial}{\partial x}\, k_{\mathrm{fab}}(T)\frac{\partial T}{\partial x} \quad \frac{\partial q_{\mathrm{rad}}(x)}{\partial x} \tag{1}$$

where $q_{\mathrm{rad}}(x)$ is the portion heat flux due to thermal radiation from the heat source, according to the [4], this term can be derive: $q_{\mathrm{rad}}(x) = q_{\mathrm{rad}}(x=0)[1-\exp(-\gamma x)] = \sigma\varepsilon_{\mathrm{g}}T_{g}^{4}[1-\exp(-\gamma x)]$, T_{g} – the temperature of hot gas (2000 K), $C^{\mathrm{A}}(T)$ – the fabric apparent heat capacity that accounts for the evaporation of the moisture and the energy released from the thermo-chemical reactions, and ρ, k, T, and x have their usual meanings.

The fabric boundary conditions at two surface ($x = 0$ and x_1) are:

$$-k_{\mathrm{fab}}(T)\frac{\partial T_{\mathrm{fab}}}{\partial x}\bigg|_{x\ \ 0} \quad h_{\mathrm{fab}}(T_{\mathrm{g}} \quad T_{\mathrm{fab}}|_{x\ \ 0}) \quad \sigma\varepsilon_{\mathrm{fab}}(1 \quad \varepsilon_{\mathrm{g}})(T_{\mathrm{fab}}^{4}|_{x\ \ 0} \quad T_{\mathrm{amb}}^{4}) \tag{2}$$

$$k_{\mathrm{fab}}(T)\frac{\partial T_{\mathrm{fab}}}{\partial x}\bigg|_{x_1} \qquad q_x\Big|_{x_1} \qquad k_{\mathrm{air}}(T)\frac{\partial T_{\mathrm{air}}}{\partial x}\bigg|_{x_1} \tag{3}$$

where h_{fab} and $\varepsilon_{\mathrm{fab}}$ are the convection heat transfer coefficient between the heat source and the fabric surface, respectively, $q_x|_{x_1}$ is the emitted radiation from the fabric backside surface, which can be derived from $q_x|_{x_1} = (1-\varepsilon_{\mathrm{fab}})(G_{\mathrm{m}} + G_{\mathrm{ext}} + G_{\mathrm{amb}}) + \varepsilon_{\mathrm{fab}}n^2\sigma T^4$.

– The governing equation of the air gap is listed:

$$\rho C_{\mathrm{air}}(T)\frac{\partial T}{\partial t} \qquad \frac{\partial}{\partial x} \quad k_{\mathrm{air}}(T)\frac{\partial T}{\partial x} \qquad \frac{\partial q_{\mathrm{air}}(x)}{\partial x} \tag{4}$$

The air gap boundary conditions at two surfaces (x_1 and x_2) are:

$$T_{\mathrm{air}}|_{x_1} \quad T_{\mathrm{fab}}|_{x_1}, \quad T_{\mathrm{air}}|_{x_2} \quad T_{\mathrm{sen}}|_{x_2} \tag{5}$$

– The governing equation of test sensor is listed:

$$\rho C_{sen}(T)\frac{\partial T}{\partial t} \qquad \frac{\partial}{\partial x} \quad k_{\mathrm{sen}}\frac{\partial T}{\partial x} \tag{6}$$

The test sensor boundary conditions at two surfaces (x_2 and x_3) are:

$$k_{\mathrm{air}}(T)\frac{\partial T_{\mathrm{air}}}{\partial x}\bigg|_{x_2} \qquad q_x\Big|_{x_2}$$
$$k_{\mathrm{sen}}\frac{\partial T_{\mathrm{sen}}}{\partial x}\bigg|_{x_2}, \quad T_{\mathrm{sen}}\Big|_{x_3} \quad T_{\mathrm{amb}} \tag{7}$$

The initial condition of this system is:

$$T_{\mathrm{fab}}\Big|_{t\ \ 0} \quad T_{\mathrm{air}}\Big|_{t\ \ 0} \quad T_{\mathrm{sen}}\Big|_{t\ \ 0} \quad T_{\mathrm{amb}} \tag{8}$$

The model was programmed under COMSOL Multiphysics and the default physical model heat transfer in solids module considering surface to surface radiation was employed based on the eqs. (1)-(8). The calculation parameters were all cited in [4].

Results and discussion

Figure 2(a) shows the comparison between the curves of the numerical simulated results in our paper and the experimental data from [4], the predicted fabric temperature are close to those measured data from the bench top test, indicating the numerical simulation proposed in this paper is suitable and accurate to predict the thermal response of heat protective clothing under bench top test conditions. Furthermore, the convection and radiation heat flux on the fabric outside surface could also be extracted from our

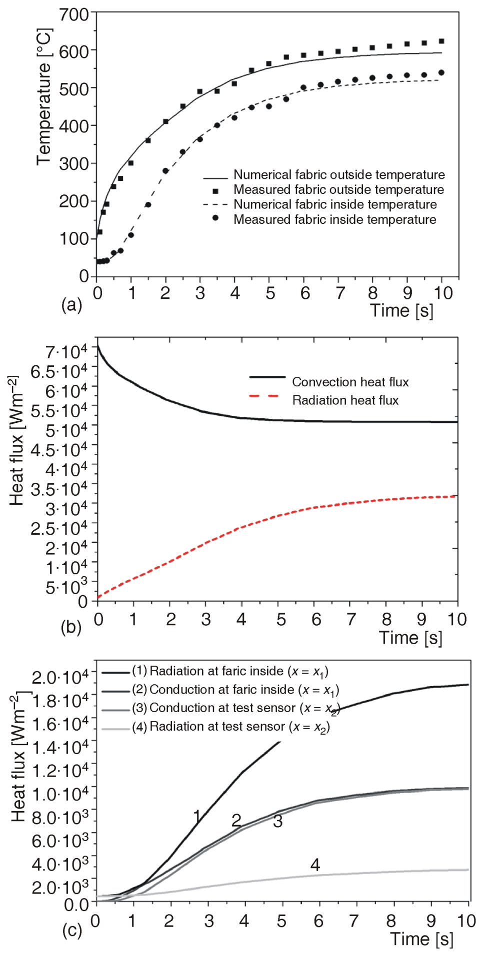

Figure 2. (a) comparison between experimental and numerical results, (b) convection and radiation heat flux on the fabric outside surface, (c) the heat flux at fabric inside surfasurface and test sensor

numerical results as shown in fig. 2(b). Convection heat flux from the hot gas dominates the initial heat transfer period while the radiation emitting from the fabric to ambient condition gradually increases with the exposure time. In addition, the heat transfers at fabric inside surface and test sensor are also illustrated in fig. 2(c). The radiation and convection at fabric inside surface are both higher than that of the test sensor demonstrating that the air gap, as the significant shielding layer, could absorb part of heat flux emitting from the fabric, especially in the radiant heat. Therefore, air gap acted as the significant role in thermal insulation of the protective system.

Conclusion

A transient heat transfer model of heat protective clothing is established and the radiation heat transfer in the air gap is reasonably considered. By the aid of numerical simulation using finite element method, the numerical solution of a heat protective clothing system is solved; a good agreement between the results of numerical simulation and experimental data could be concluded. Furthermore, the heat transfer type from the heat source is clarified and the portion of radiation and conduction in the air gap is also calculated. Therefore, this readily model could be utilized in the application of heat protective system.

Acknowledgment

Financial supported by Natural Science Foundation of China Nos. 51306095, 51273097, and 11402155, China Postdoctoral Science Foundation No. 2014M561887, No. 2015T80697, Natural Science Foundation of Jiangsu Province (Grant Nos. BK20140396 and BK20140399), Nantong Science and Technology Planning Project (Grant No. GY12015013), and Qingdao Postdoctoral Application Research Funded Project and Qingdao Application Basic Research Funded Project (14-2-4-1-JCH, 15-9-1-41-JCH).

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Paper submitted: February 10, 2015
Paper revised: February 3, 2016
Paper accepted: February 3, 2016