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# The Heat Transfer Characteristics of an Insulated Circular Duct Considering and Neglecting the Influence of Heat Radiation

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***Abstract*—Heat radiation equation contains 4th exponential order of temperature which makes mathematics analysis very complicated and troublesome. Therefore, most heat transfer experts and scholars believe, based on their own experiences, that the heat radiation effect can be ignored due to small temperature difference between insulated surfaces and surrounding to simplify analysis. Due to this reason, insulation examples shown in most heat transfer, air conditioning and refrigeration text books, and some commercial packages for the design of insulations, commonly ignore the effect of heat radiation. However, this paper demonstrates that the heat radiation effect can not be ignored in the situations of thin insulation with low ambient convective heat coefficient and a large surface emissivity. In those situations, ignoring the heat radiation will result in an inaccurate insulation design. The paper also shows that a smaller insulated surface emissivity can largely promote the insulation effect.**

***Index Terms*—Insulation, heat radiation, circular duct, emissivity, insulation effect.**

## I. INTRODUCTION

The insulations of cold/hot ducts are commonly encountered in industry and house applications; hence, they have been important research objects for many decades. Conventionally heat radiation effects are normally neglected because a heat radiation equation contains 4th exponential order of temperature which in turn makes analysis complicated. The negligence is seemingly harmless for cases with small temperature differences between insulated surfaces and surrounding. From this aspect, insulation examples demonstrated in most heat transfer and air conditioning and refrigeration text books, and in many research papers or even in the practices of most commercial insulation design packages commonly ignore the influence of heat radiation even in situations involving low convection coefficients. Such examples can be found in the following papers, which all neglected the influence of heat radiation: Wong et al. [1] figured out the complete heat transfer solutions of an insulated regular polygonal pipe by using a PWTR model, Lee et al. [2] investigated the complete heat transfer solutions of an insulated regular polyhedron by using a RPSWT model, Wong et. al [3] applied the reliable simple one-dimensional 64-CPWTR model to the two-dimensional heat transfer problem of an insulated rectangular duct in an air conditioning or refrigeration system, Wong et al. [4] used a reliable one-dimensional method to solve heat-transfer problems associated with insulated rectangular tanks in refrigeration systems, Wong et al. [5] applied the reliable one-dimensional CPWTR models to two-dimensional insulated polygonal ducts, Chen and Wong [6] applied a reliable analytical method to heat transfer problems associated with insulated cylindrical tanks.

In the present investigation, the influence of heat radiation is taken into account when computing heat-transfer characteristics of an insulated circular duct and the results are compared with those neglecting heat radiation. This highlights the inaccuracy of the heat transfer characteristics of an insulated circular duct obtained by neglecting the influence of heat radiation.

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## II. PROBLEM FORMULATION

Fig. 1(a) shows that an insulated circular duct with duct thickness $t_1$, duct length $L$, wall conductivity $K_A$, duct surface emissivity $\varepsilon_0$, insulation thickness $t$ and conductivity $K_s$, internal and external fluids with convection heat transfer coefficients $h_i$ and $h_o$ temperatures $T_i$ and $T_o$, respectively, and with its surface emissivity ε, exposed to an outside surrounding temperature of $T_{sur}$.

### A. *Cases with the influence of heat radiation being neglected*

While the influence of outside surface heat radiation is not considered, it can be seen from Fig. 1 that the total thermal resistance is:

$$\sum R_{th} = \frac{1}{h_i 2\pi r_1 L} + \frac{\ln r_2/r_1}{2\pi K_A L} + \frac{\ln r_3/r_2}{2\pi K_S L} + \frac{1}{h_o 2\pi r_3 L} \tag{1}$$

And the relative heat transfer rate with heat convection and heat conduction terms is:

$$q = \frac{T_i - T_o}{\sum R_{th}} = \frac{T_3 - T_0}{\dfrac{1}{h_o 2\pi r_3 L}} \tag{2}$$

Equation (2) implies that heat transfer rate q and insulted surface temperature $T_3$ of situations without considering the influence of outside surface heat radiation can be readily obtained. From Fig. 2, the heat transfer rate of bare circular duct of situations without considering the influence of outside surface heat radiation can be written as:

$$q_0 = \frac{T_i - T_o}{\dfrac{1}{h_i 2\pi r_1 L} + \dfrac{\ln r_2/r_1}{2\pi K_A L} + \dfrac{1}{h_0 2\pi r_2 L}} \tag{3}$$

Equations (2) and (3) lead to an insulated effect ***EF*** of an insulated circular duct neglecting heat radiation being:

$$EF = (1 - \frac{q}{q_0}) * 100\% \tag{4}$$

### B. Cases with the influence of heat radiation being considered

While the influence of outside surface heat radiation is considered, the complete heat transfer rate of an insulated circular can be expressed as:

$$q_a = \frac{T_i - T_{3a}}{\dfrac{1}{h_i 2\pi r_1 L} + \dfrac{\ell_n \frac{r_2}{r_1}}{2\pi K_A L} + \dfrac{\ell_n \frac{r_3}{r_2}}{2\pi K_s L}} \tag{5}$$

Where $T_{3a}$ is the actual surface temperature after considering heat radiation. Then the surface convective heat transfer rate becomes:

$$q_c = h_0 2\pi r_3 L\left(T_{3a} - T_0\right) \tag{6}$$

The surface radiation heat transfer rate is:

$$q_r = \sigma\varepsilon 2\pi r_3 L\left(T_{3a}^4 - T_{sur}^4\right) \tag{7}$$

From Eqs. (6)and (7), under the conditions of $\varepsilon \neq 0$, $T_{3a} \neq T_3$, $q_c \neq q = h_o 2\pi r_2 L(T_3 - T_o)$.

And the heat balance of $\boldsymbol{q_a}$, $\boldsymbol{q_c}$ and $\boldsymbol{q_r}$ can be written as:

$$q_a = q_c + q_r \tag{8}$$

Therefore, the complete heat transfer rate $\boldsymbol{q_a}$ and the surface temperature $\boldsymbol{T_{3a}}$ can be readily deduced from equations. (5)~(8).

The external radiation heat convection coefficient is defined from equation (7) as:

$$h_r = \sigma\varepsilon\left(T_{3a}^2 + T_{sur}^2\right)\left(T_{3a} + T_{sur}\right) \tag{9}$$

Here, $\boldsymbol{h_r}$ is normally used to compare with the heat convection coefficient $\boldsymbol{h_o}$ which in turn shows how significant the effect of radiation is. With both $\boldsymbol{q}$ and $\boldsymbol{q_a}$ being available, we can now further define the error of heat transfer rate generated by neglecting heat radiation effect as:

$$QR = \left(1 - \frac{q}{q_a}\right) \times 100\% \tag{10}$$

The ratio between radiation- and convection- heat transfer coefficients is defined as:

$$HR = \frac{h_r}{h_0} \times 100\% \cong \frac{2\pi r_2 L(T_{2a} - T_{sur})h_r}{2\pi r_2 L(T_{2a} - T_o)h_0} \times 100\% = \frac{q_r}{q_c} \times 100\% \tag{11}$$

Where $\boldsymbol{T_{sur}}$ is always close or equal to $\boldsymbol{T_o}$

Similar to equation (10), the error of surface temperature generated by neglecting heat radiation effect, based on $T_i > T_o$ and using Celsius temperature scale, is defined as:

$$SR = (1 - \frac{T_3}{T_{3a}}) \times 100\% \tag{12}$$

In the case of an insulated cold duct, condensed water can be formed if the insulated surface temperature is less than the outside ambient dew point temperature. Therefore, predicting the insulated surface temperature is very important. Dimensionless SR tends to reduce the magnitude of error; hence it is not suitable to be used to demonstrate the heat characteristic of cold duct. In stead, the surface temperature difference is redefined as:

$$TD = T_{3a} - T_3 \tag{13}$$

While the influence of outside surface heat radiation is considered, the complete heat transfer rate of a bare circular duct can be written as:

$$q_{a0} = \frac{T_i - T_{2a}}{\dfrac{1}{h_i 2\pi r_1 L} + \dfrac{\ell_n \frac{r_2}{r_1}}{2\pi K_A L}} \tag{14}$$

Where $\boldsymbol{T_{2a}}$ is the actual surface temperature of a bare circular duct with the consideration of the influence of heat radiation. Additionally, its surface convective heat transfer rate is:

$$q_{c0} = h_0 2\pi r_2 L\left(T_{2a} - T_0\right) \tag{15}$$

and the surface radiation heat transfer rate is:

$$q_{r0} = \sigma\varepsilon 2\pi r_2 L\left(T_{2a}^4 - T_{sur}^4\right) \tag{16}$$

Similarly, the heat balance can be expressed as:

$$q_{a0} = q_{c0} + q_{r0} \tag{17}$$

To this end, the complete heat transfer rate $\boldsymbol{q_{a0}}$ and surface temperature $\boldsymbol{T_{2a}}$ of bare circular duct in situations considering the influence of outside surface heat radiation can be readily obtained from equations (14)~(17). And finally, the insulated effect $\boldsymbol{EF_0}$ of an insulated circular duct while considering heat radiation can be obtained by:

$$EF_a = (1 - \frac{q_a}{q_{a0}}) * 100\% \tag{18}$$

## III. Numerical Heat Transfer Results

There exists a one-dimensional exact solution for the energy equation of an insulated circular duct. In this paper, the exact solution is obtained by a one-dimensional LabVIEW programming. According to the emissivities shown in Table 1, values of $\varepsilon = \varepsilon_0 = 0.8$ and $\varepsilon = \varepsilon_0 = 0.2$ are adopted to respectively represent high and low surface emissivity cases. In Table 2, it can be seen that most of the natural convection coefficients of air are below 10 $W/m^2$-K; even in the cases of high air velocities, the forced convection coefficients are less than 100 $W/m^2$-K. Between the two extremes, a convection coefficient of 30 $W/m^2$-K can represent the convective heat-transfer effect from medium wind speed. Therefore, $\boldsymbol{h}$=10 and 30 $W/m^2$-K are selected to represent low and medium convection coefficients of air**.** On the other hand, even the natural convection coefficients of water are normally larger than 890 $W/m^2$-K, and most forced convection coefficients of water are over 5000 W/m$^2$-K. Therefore, $\boldsymbol{h}$=5000 W/m$^2$-K is assumed to represent medium forced convection coefficients of water. In most industrial applications, carbon steel, with a conductivity of $\boldsymbol{K_A}$=77 W/m$^2$-K, is common used as the duct material.

In order to check if the computer results are reliable, the following measures are adopted:

(1). Let surface emissivity $\varepsilon$=0, check for the resulted ***QR*** being zero.

(2). Let surface emissivity $\varepsilon$=0.8 and external convection coefficient $\boldsymbol{h_o}$=5000W/m$^2$-K, check for the resulted QR being close to zero.

## IV. Results and Discussions

For the current results to be applicable to more general cases, all the results are shown in dimensionless parameters except for TD, the surface temperature difference due to neglecting heat radiation effect for insulated cold duct.

We first examine the case of an insulated hot duct with $\boldsymbol{T_i}$=100℃, $\boldsymbol{T_o}$=$\boldsymbol{T_{sur}}$=30℃, $\boldsymbol{K_A}$=77W/m-K, $\boldsymbol{K_s}$=0.035W/m-K, $\boldsymbol{r_1}$=195mm, $\boldsymbol{r_2}$=200mm, and $\boldsymbol{h_o}$=10W/m$^2$-K, and the results are shown in Figs. 3-6. These figures show that the heat transfer rate errors QR, convective coefficients ratio HR, the error of surface temperature SR, and insulated effect EF are affected by the dimensionless insulated thickness $\boldsymbol{t/r_2}$, internal convection coefficient $\boldsymbol{h_i}$, and the surface emisivity $\boldsymbol{\varepsilon}$. Among them, QR and HR decrease while EF increases as $\boldsymbol{t/r_2}$ increases. The higher the $\varepsilon$ and $\boldsymbol{h_i}$ are, the larger the QR, HR and EF will be. Fig. 5 also shows that SR reaches its minimum value near $\boldsymbol{t/r_2}$=0.025 and the absolute value of SR becomes higher as $\varepsilon$ increases. But if the value of $\varepsilon$ is fixed, smaller $\boldsymbol{h_i}$ tends to result in larger absolute value of SR before some point near $\boldsymbol{t/r_2}$=0.025, but the trend reverses after that point. Fig. 6 shows that for a fixed $\boldsymbol{h_i}$, EF value for $\boldsymbol{\varepsilon=\varepsilon_0}\neq$0 is greater than that that for $\boldsymbol{\varepsilon=\varepsilon_0}$=0.

The second case examined in this paper is an insulated cold duct with $\boldsymbol{T_i}$= - 20℃, $\boldsymbol{T_o}$=$\boldsymbol{T_{sur}}$=30℃, $\boldsymbol{K_A}$=77 W/m-K, $\boldsymbol{K_s}$=0.035 W/m-K, $\boldsymbol{r_1}$=195mm, $\boldsymbol{r_2}$=200mm and $\boldsymbol{h_o}$=8.3W/m$^2$-K, and the results are shown in Figs. 7-10. Those figures again show that ***QR***, ***HR***, ***TD***, and ***EF*** are affected by factors $\boldsymbol{t/r_2}$, $\boldsymbol{h_i}$, and $\varepsilon$. Here, ***QR*** and ***HR*** decrease, but ***EF*** increases with an increase of $\boldsymbol{t/r_2}$. The higher the values of $\varepsilon$ and $\boldsymbol{h_i}$ are, the larger the values of ***QR***, ***HR***, and ***EF*** will be. Fig. 9 shows that the minimum value of ***TD*** is also near **t/r$_2$**=0.025. Meanwhile the value of ***TD*** is also proportional to the value of $\varepsilon$. In the case of fixed $\varepsilon$, ***TD*** is reversely proportional to $\boldsymbol{h_i}$ before $\boldsymbol{t/r_2}$=0.025, but again the tendency reverses after that point. Fig. 10 shows that for the same $\boldsymbol{h_i}$, ***EF*** with $\boldsymbol{\varepsilon=\varepsilon_0}\neq$0 is greater than that with $\boldsymbol{\varepsilon=\varepsilon_0}$=0. This can be found in Figs. 3 and 7 for cases of $\boldsymbol{\varepsilon}$=0.8 and $\boldsymbol{t/R_2}\leqq$ 0.1, ***QR***$\geqq$ 5%.

In order to demonstrate the main differences between the heat transfer characteristics of cases with and without considering heat radiation, the detailed data of an insulated hot duct and an insulated cold duct are listed in Tables 4 and 5, respectively. It can be seen from Tables 4 and 5 that the values of $\boldsymbol{q_c}$ are different from those of $\boldsymbol{q}$. For example, the data of $\boldsymbol{t/r_2}$=0.005 and $\boldsymbol{\varepsilon=\varepsilon_0}$ =0.8 listed in Table 4 show that $\boldsymbol{q_a}$=703.86W/m (>$\boldsymbol{q}$) and $\boldsymbol{q_r}$ =263.72W/m; $\boldsymbol{q_c}$ (=440.14W/m) is not equal to $\boldsymbol{q}$ (=542.13W/m), since $\boldsymbol{T_{2a}}$ (=64.8℃), which is smaller than $\boldsymbol{T_2}$ (=72.9℃); thus ***QR*** (=22.9%) and ***HR***=59.9% are quite large,. It can be explained from equations (1) and (3) that if $\varepsilon\neq$0 and $\boldsymbol{T_{3a}\neq T_3}$, then $\boldsymbol{q_c=h_o 2\pi r_3 L(T_{3a}-T_o)}$, which is not equal to $\boldsymbol{q=h_o 2\pi r_3 L(T_3-T_o)}$ with $\boldsymbol{\varepsilon=\varepsilon_0}$=0. It can be found from above that the heat transfer rate $\boldsymbol{q}$ obtained by neglecting the heat radiation is not accurate. Therefore, the effect of heat radiation need to be taken into account, especially in the situations of thin insulated thickness $\boldsymbol{t}$, low ambient convective coefficient $\boldsymbol{h_o}$, and large surface emissivity $\boldsymbol{\varepsilon}$.

## V. Conclusion

From the cases investigated in this study, it has been demonstrated that neglecting the influence of heat radiation effect, especially in the situations of low external ambient air convection coefficients, thin insulated thickness and large surface emissivity are likely to produce inaccurate results. Meanwhile, to incorporate heat radiation into a computer program, such as the LabVIEW program in this study, does not require significant efforts. Therefore, taking heat radiation effect into account is a small effort worth investing to ensure the accuracy of the heat transfer results of an insulated circular duct.

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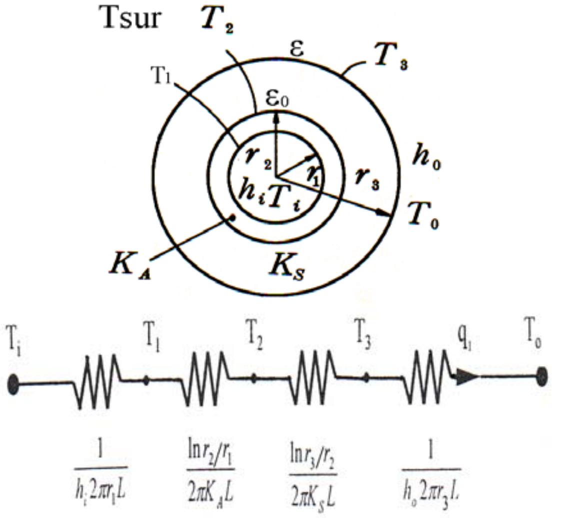


Fig. 1 An insulated circular duct and relative parameters and its relative thermal resistance diagram while neglecting the heat radiation

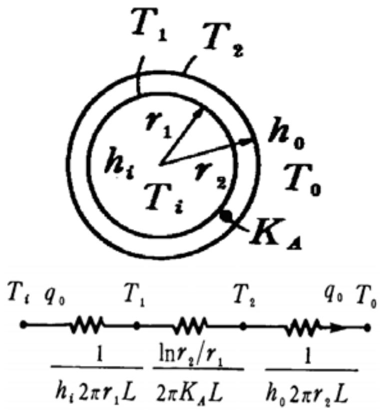


Fig. 2 A bare circular duct and its thermal resistance diagram while neglecting the heat radiation

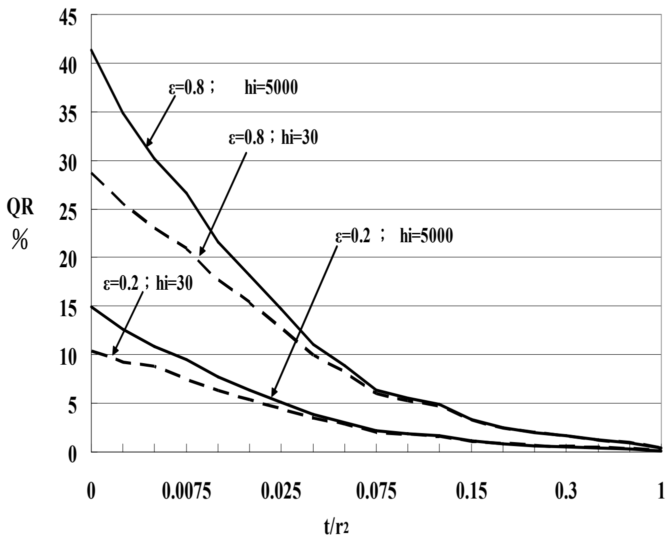


Fig. 3 The relations between QR and t/R$_2$ in the situation of $T_i$=100℃, $T_o$=$T_{sur}$=30℃, $K_A$=77W/m-K, $K_s$=0.035W/m-K, $r_1$=195mm, $r_2$=200mm and $h_o$=10W/m$^2$-K

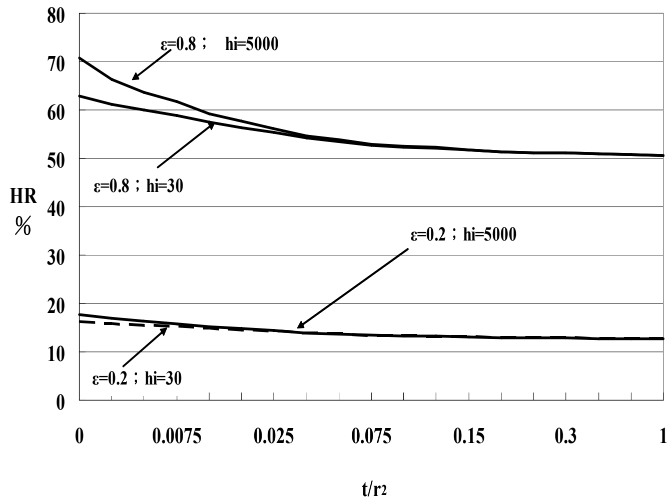


Fig. 4 The relations between HR and t/R$_2$ in the situation of $T_i$=100℃, $T_o$=$T_{sur}$=30℃, $K_A$=77W/m-K, $K_s$=0.035W/m-K, $r_1$=195mm, $r_2$=200mm and $h_o$=10W/m$^2$-K

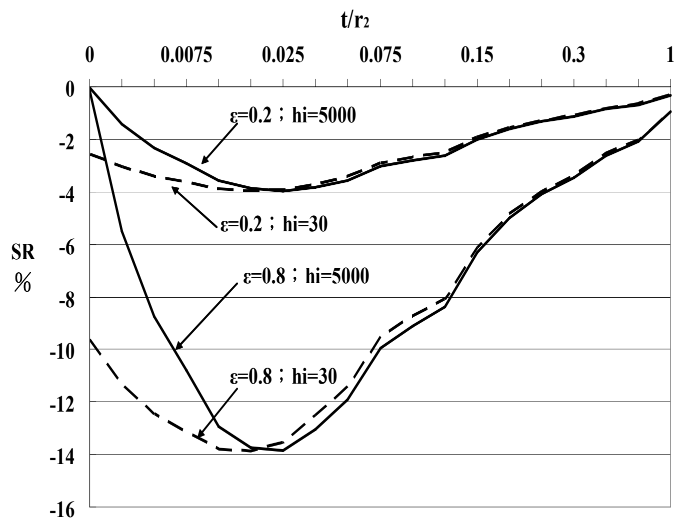


Fig. 5 The relations between SR and t/R$_2$ in the situation of $T_i$=100℃, $T_o$=$T_{sur}$=30℃, $K_A$=77W/m-K, $K_s$=0.035W/m-K, $r_1$=195mm, $r_2$=200mm and $h_o$=10W/m$^2$-K

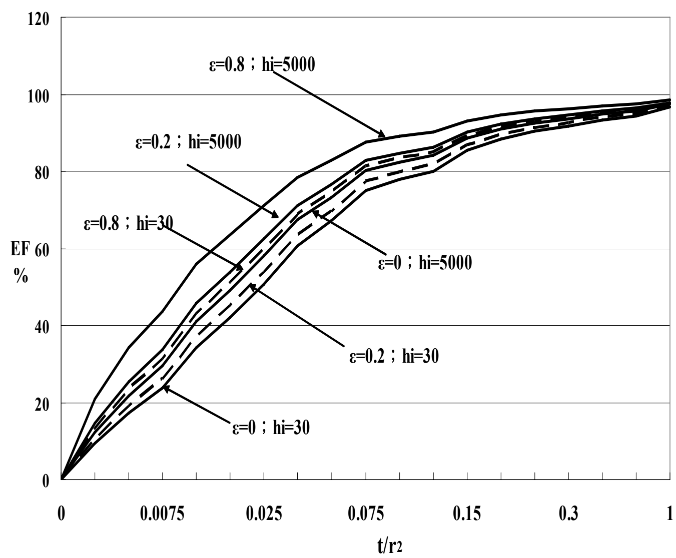


Fig. 6 The relations between EF and t/R$_2$ in the situation of

$T_i$=100℃, $T_o$=$T_{sur}$=30℃, $K_A$=77W/m-K, $K_s$=0.035W/m-K, $r_1$=195mm, $r_2$=200m and $h_o$=10W/m$^2$-K, $\varepsilon$=$\varepsilon_0$

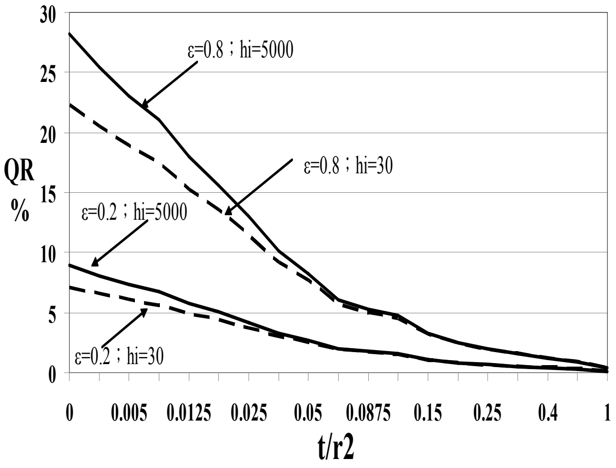


Fig. 7 The relations between QR and t/$R_2$ in the situation of $T_i$= -20℃, $T_o$=$T_{sur}$=30℃, $K_A$=77 W/m-K, $K_s$=0.035 W/m-K, r1=195mm, $r_2$=200mm and $h_o$=8.3W/m$^2$-K

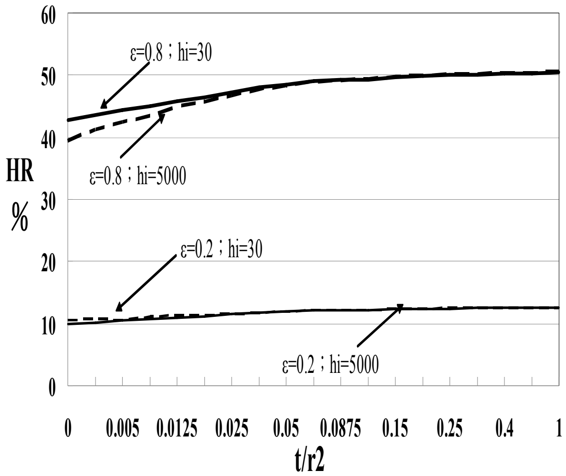


Fig.8 The relations between HR and t/$R_2$ in the situation of $T_i$= -20℃, $T_o$=$T_{sur}$=30℃, $K_A$=77 W/m-K, $K_s$=0.035 W/m-K, r1=195mm, $r_2$=200mm and $h_o$=8.3W/m$^2$-K

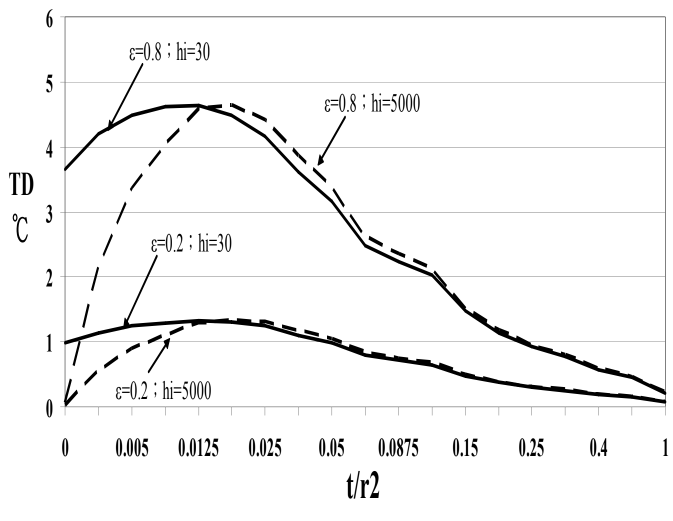


Fig. 9 The relations between TD and t/$R_2$ in the situation of $T_i$= -20℃, $T_o$=$T_{sur}$=30℃, $K_A$=77 W/m-K, $K_s$=0.035 W/m-K, r1=195mm, $r_2$=200mm and $h_o$=8.3W/m$^2$-K

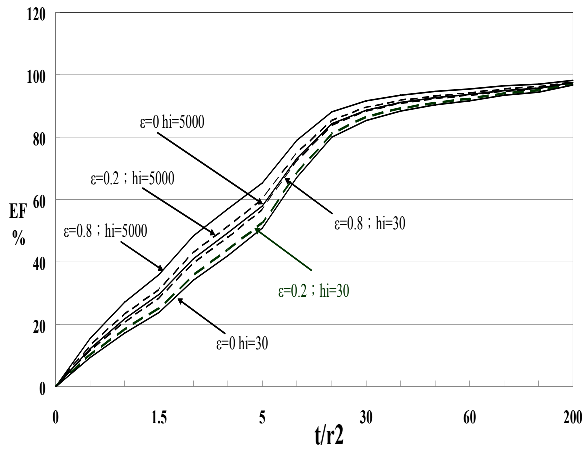


Fig. 10 The relations between EF and t/$R_2$ in the situation of $T_i$= -20℃, $T_o$=$T_{sur}$=30℃,$K_A$=77 W/m-K, $K_s$=0.035 W/m-K, r1=195mm, $r_2$=200mm and $h_o$=8.3W/m$^2$-K, $\varepsilon$=$\varepsilon_0$

Tables

Table 1、The emissivities ε of various substances from the manual of infrared temperature demonstrator [7]

| | |
|---|---|
| Human Skin | 0.98 |
| Gold | 0.02 |
| Silver | 0.02 |
| Aluminum | Weathered=0.83; Foil (bright)=0.04 |
| Copper | polished=0.05 oxidized=0.78 |
| Iron | cast(ox)=0.64 sheet, rusted=0.69 |
| Stainless steel | polished=0.16 Oxidized=0.85 |
| Steel | polished=0.07 Oxidized=0.79 |
| Nickel | Electro pole=0.05 |
| Brick | 0.81 |
| Carbon | 0.95 |
| Concrete | 0.95 |
| Glass | 0.84~0.97 |
| Paint oil | 0.94 |
| Paper, white | 0.70 |
| Paper, | 0.89 |
| Plaster | 0.86 |
| Rubber, black | 0.95 |
| Wood, oak | 0.90 |
| White ceramic | 0.91 |
| Black painting | 0.96 |
| Soil | dry=0.92 saturated water=0.95 |

Table 2 Referred approximate values of convection heat transfer

| **Approximate values of convection heat transfer, h(W/m$^2$-K)** | |
|---|---|
| **Mode** | **h W/m$^2$-K** |
| **Free convection** | |
| Temp. Diff. =30 ℃ Vertical plate 0.3 in high in air | 4.5 |
| Temp. Diff. =30 ℃ Vertical plate 0.3 in high in air | 6.5 |
| Horizontal cylinder, 2 cm diameter, in water | 890 |
| Heat transfer across 1.5 cm vertical air gap with Temp. Diff. =60 ℃ | 2.64 |
| **Forced convection** | |
| Air flow at 2 m/s over 0.2-m square plate | 12 |
| Air flow at 35 m/s over 0.75-m square plate | 75 |
| Air at 2 atm flowing in 2.5 cm diameter tube at 10 m/s (=36km/hr) | 65 |
| Water at 0.5 kg/s flow in 2.5 cm diameter tube | 3500 |
| Air flow across 5 cm diameter cylinder with velocity of 50 m/s(=180km/hr) | 180 |
| **Boiling water** | |
| In a pool or container | 2500–35,000 |
| Flowing in a tube | 5000-100,000 |
| **Condensation of water vapor, 1 atm** | |
| Vertical surfaces | 4000–11,300 |
| Outside horizontal tubes | 9500–25,000 |

Table 3 Referred approximate values of thermal conductivities

| Thermal conductivity of various materials at 20 ℃ | |
|---|---|
| **Metals** | |
| Material | K (W/m-K) |
| Copper (pure) | 386 |
| Aluminum (pure) | 204 |
| Carbon steel, 1 % C | 73-77 |
| Carbon steel (18%Cr, 8%Ni) | 43 |
| Cast iron | 16 |
| **Nonmetallic solids** | |
| Glass, window | 0.78 |
| Plaster, gypsum | 0.48 |
| Metal lath | 0.4 |
| Woof lath | 0.28 |
| Teflon | 0.35 |
| Asphalt | 0.7 |
| Wood fiber sheet | 0.047 |
| wool | 0.038 |
| Glass fiber | 0.035 |
| Building brick common | 0.69 |
| Building brick face | 1.32 |
| Concrete, cinder | 0.76 |
| Stone, 1–2–4 mix | 1.37 |
| **Graphite, pyrolytic** | |
| Perpendicular to layers | 5.6 |
| Polyethylene | 0.33 |
| Polypropylene | 0.16 |
| Polyvinylchloride | 0.09 |
| Rubber, hard | 0.1 |