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The inaccuracy of heat transfer characteristics of insulated and non-insulated circular duct while neglecting the influence of heat radiation

Tsung-Lieh Hsien [b], King-Leung Wong [a,*], Shiue-Jie Yu [a]

[a] Department of Mechanical Engineering, Kun-Shan University of Technology, 949, Da-Wan Road, Yung-Kang City, Tainan County, Taiwan, ROC
[b] Department of Electrical Engineering, Kun-Shan University of Technology, 949, Da-Wan Road, Yung-Kang City, Tainan County, Taiwan, ROC

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ABSTRACT

The non-insulated and insulated ducts are commonly applied in the industries and various buildings, because the heat radiation equation contains the 4th order exponential of temperature which is very complicate in calculations. Most heat transfer experts recognized from their own experiences that the heat radiation effect can be ignored due to the small temperature difference between insulated and non-insulated surface and surroundings. This paper studies in detail to check the inaccuracies of heat transfer characteristics non-insulated and insulated duct by comparing the results between considering and neglecting heat radiation effect. It is found that neglecting the heat radiation effect is likely to produce large errors of non-insulated and thin-insulated ducts in situations of ambient air with low external convection heat coefficients and larger surface emissivity, especially while the ambient air temperature is different from that of surroundings and greater internal fluid convection coefficients. It is also found in this paper that using greater duct surface emissivity can greatly improve the heat exchanger effect and using smaller insulated surface emissivity can obtain better insulation.

1. Introduction

The insulated and non-insulated cold/hot ducts are commonly applied in the industries and various buildings, because the heat radiation equation contains the 4th order exponential of temperature which is very complicate in calculations. Most heat transfer experts recognized from their own experiences that the heat radiation effect can be ignored due to the small temperature difference between insulated and non-insulated surface and surroundings. From this view point, the demonstrated insulation examples shown in most heat transfer text books (such as [1–8]), air conditioning and refrigeration text books (such as [9–12]), most practical commercial designs of insulation, and many research papers commonly ignore the influence of heat radiation even in the situation of very low heat convective coefficient.

Since the insulation of hot and cold ducts and containers has been one of the most important engineering problems. Levinson et al. [13] considered the effects of airflow infiltration on the thermal performance of internally insulated ducts, Wong and Chou [14] studied the heat transfer characteristics of an insulated regular cubic box by using a solid wedge thermal-resistance model, Chou [15] used the optimum interior area thermal-resistance model to analyze the heat transfer characteristics of an insulated pipe with arbitrary shape, Wong and Chou [16] investigated heat transfer characteristics of an insulated regular cubic box by using a regular polygonal top solid wedge thermal-resistance model, Wong et al. [17] figured out the complete heat transfer solutions of an insulated regular polygonal pipe by using a PWTR model, Wong et al. [18] studied the complete heat transfer solutions of an insulated regular cubic tank with a SSWT model, Lee et al. [19] investigated the complete heat transfer solutions of an insulated regular polyhedron by using a RPSWT model, Wong et al. [20] 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. [21] used a reliable one-dimensional method to solve heat-transfer problems associated with insulated rectangular tanks in refrigeration systems, Wong et al. [22] applied the reliable one-dimensional CPWTR models to two-dimensional insulated polygonal ducts, Chen and Wong [23] applied a reliable analytical method to heat-transfer problems associated with insulated cylindrical tanks. The heat radiation effect was ignored in above papers; it should be acceptable in situations of very thick insulation.

In this present investigation, the influence of heat radiation of insulated and non-insulated ducts is considered and the results of complete heat transfer characteristics are compared with those of neglecting the heat radiation effect. Then the errors of heat transfer characteristics for insulated and non-insulated ducts with neglecting the influence of heat radiation can be found.

* Corresponding author. Tel.: +886 62057121; fax: +886 62050509.
E-mail address: klwong@mail.ksu.edu.tw (K.-L. Wong).

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Nomenclature

ε	emissivity of insulation surface
ε_0	emissivity of duct surface
A_1	inner surface area of a duct
A_2	external surface area of a bare duct
A_3	external surface area of a insulated duct
EF	insulated effect
h_i	inner heat convection coefficient
h_o	external heat convection coefficient
h_r	external radiation heat convection coefficient
h_o	external heat convection coefficient
HR	convective heat coefficient ratio
K_A	conductivity of duct material
K_s	conductivity of insulated material
L	length of oval duct
q	total heat transfer rate without considering heat radiation
q_a	total heat transfer rate considering heat radiation; $q_a = q_c + q_r$
q_c	convective heat transfer rate
q_r	radiation heat transfer rate
QR	error of heat transfer rate generated by without considering heat radiation
r_1	inner radius of circular duct
r_2	external radius of bare circular duct
r_3	external radius of insulated circular duct
SR	error of surface temperature generated by without considering heat radiation
TD	the surface temperature difference generated by neglecting heat radiation effect
t_1	thickness of duct wall
t	thickness of insulated material
t/r_2	dimensionless insulated thickness
T_2	bare duct surface temperature generated by neglecting heat radiation
T_{2a}	bare duct surface temperature generated by considering heat radiation
T_3	insulated surface temperature generated by neglecting heat radiation
T_{3a}	insulated surface temperature generated by considering heat radiation
T_i	temperature of the fluid inside the duct
T_o	temperature of the fluid outside the duct
T_{sur}	temperature of the outside surroundings

2. Problem formulation

Fig. 1 shows that the non-insulated or insulated circular duct with thickness t_1, duct length L, wall conductivity K_A, duct surface emissivity ε_0; the insulated layer with surface emissivity ε, insulation thickness t and conductivity K_s; internal and external fluids with convection heat transfer coefficients h_i and h_o as well as temperatures T_i and T_o, respectively; the duct is exposed to the outside surroundings temperature T_{sur}, and the duct surface area is very small compared with that of surroundings.

2.1. Situations neglecting the influence of heat radiation

The heat transfer rate of bare circular duct of situations without considering the influence of external surface heat radiation can be obtained from Fig. 1a is:

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

While the influence of external surface heat radiation is not considered, thus from Fig. 1b that the total thermal resistance is:

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

And relative heat equation with heat convection and heat conduction terms is:

$$q = \frac{T_i - T_o}{\sum R_{th}} = \frac{T_3 - T_o}{\frac{1}{h_o 2\pi r_3 L}} \tag{3}$$

Thus, heat transfer rate q and insulted surface temperature T_3 of situations without considering the influence of external surface heat radiation can be obtained from Eq. (3).

The insulated effect EF of an insulated circular duct while neglecting heat radiation can be obtained:

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

2.2. Situations neglecting the influence of heat radiation

While the influence of external surface heat radiation is considered, the surface temperature of the bare duct shown in Fig. 1a becomes T_{2a} rather than T_2 of the neglecting heat radiation situation, and the complete heat transfer rate of a bare circular duct is:

$$q_{a_0} = \frac{T_i - T_{2a}}{\frac{1}{h_i 2\pi r_1 L} + \frac{\ell n \frac{r_2}{r_1}}{2\pi K_A L}} \tag{5}$$

The surface convective heat transfer rate of a bare circular duct is:

$$q_{c0} = h_0 2\pi r_2 L (T_{2a} - T_o) \tag{6}$$

Since the duct surface area is very small compared with that of surroundings, thus the effect of emissivity of surroundings can be neglected. Form this, the surface radiation heat transfer rate of a bare circular duct is:

$$q_{r0} = \sigma \varepsilon 2\pi r_2 L (T_{2a}^4 - T_{sur}^4) \tag{7}$$

The following equation of a bare circular duct is obtained from heat balance:

$$q_{a0} = q_{c0} + q_{r0} \tag{8}$$

Thus, the complete heat transfer rate q_{a0} and surface temperature T_{2a} of bare circular duct in situations considering the influence of external surface heat radiation can be obtained from Eqs. (5)–(8).

While the influence of external surface heat radiation is considered, the insulated surface temperature of the bare duct shown in Fig. 1b becomes T_{3a} rather than T_3 of the neglecting heat radiation situation, and the complete heat transfer rate of an insulated circular duct is:

$$q_a = \frac{T_i - T_{3a}}{\frac{1}{h_i 2\pi r_1 L} + \frac{\ell n \frac{r_2}{r_1}}{2\pi K_A L} + \frac{\ell n \frac{r_3}{r_2}}{2\pi K_s L}} \tag{9}$$

The surface convective heat transfer rate is:

$$q_c = h_0 2\pi r_3 L (T_{3a} - T_o) \tag{10}$$

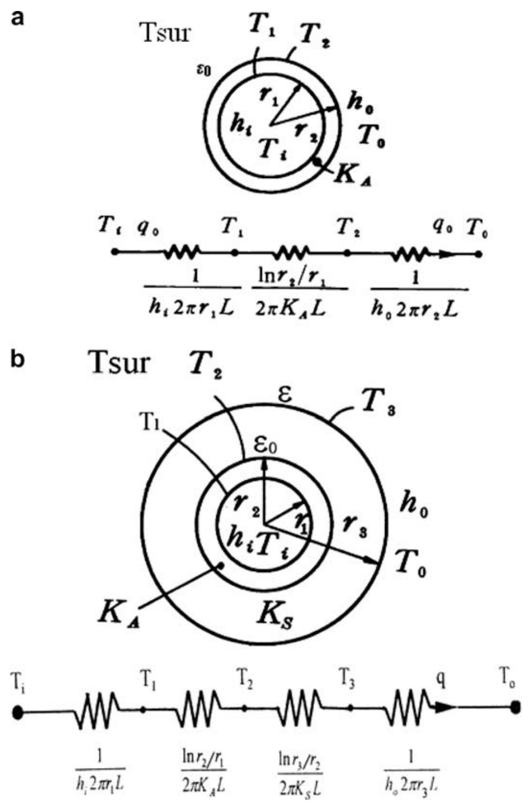


Fig. 1. (a) A non-insulated circular duct and its thermal resistance diagram and (b) an insulated circular duct and relative parameters while neglecting the heat radiation.

If $\varepsilon \neq 0$, $T_{3a} \neq T_3$, $q_c \neq q = h_o 2\pi r_2 L(T_3 - T_o)$.

Since the insulated surface area is very small compared with that of surroundings, thus the effect of emissivity of surroundings can be neglected. Form this, the surface radiation heat transfer rate is:

$$q_r = \sigma\varepsilon 2\pi r_3 L(T_{3a}^4 - T_{sur}^4) \tag{11}$$

The following equation is obtained from heat balance:

$$q_a = q_c + q_r \tag{12}$$

Thus, the complete heat transfer rate q_a and surface temperature T_{3a} of situations considering the influence of external surface heat radiation can be obtained from Eqs. (9)–(12).

The external radiation heat convection coefficient is conventionally defined from Eq. (11)

$$h_r = \sigma\varepsilon(T_{3a}^2 + T_{sur}^2)(T_{3a} + T_{sur}) \tag{13}$$

where h_r is conventionally used to compare to the heat convection coefficient h_o and show the significant effect of radiation.

The convective heat coefficient ratio percentage between radiation and convection is defined as:

$$HR = \frac{h_r}{h_0} \times 100\% = \frac{2\pi r_2 L(T_{2a} - T_{sur})h_r}{2\pi r_2 L(T_{2a} - T_o)h_0} \times 100\% \simeq \frac{q_r}{q_c} \times 100\% \tag{14}$$

Actually, T_o should not be the same value as T_{sur}, but in most situations quite close to T_{sur}, thus HR should be approximate equivalent to $q_r/q_c \times 100\%$. In this paper, we let $T_o = T_{sur}$ just for the purpose to compare the results between considering and neglecting heat radiation. Thus in this paper, value of HR is actually the same as $q_r/q_c \times 100\%$ only in situation of $T_o = T_{sur}$. A high value of HR indicates that the influence of heat radiation cannot be neglected.

The error of heat transfer rate generated by neglecting heat radiation effect is defined as:

$$QR = \left(1 - \frac{q}{q_a}\right) \times 100\% \tag{15}$$

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:

$$SR = \left(1 - \frac{T_3}{T_{3a}}\right) \times 100\% \tag{16}$$

For the insulated cold duct, the condensed water is occurred if the insulated surface temperature less than the outside ambient dew point. Thus, the insulated surface temperature is very important; dimensionless SR will reduce the magnitude of error thus is not suitable to demonstrate the heat characteristic of cold duct.

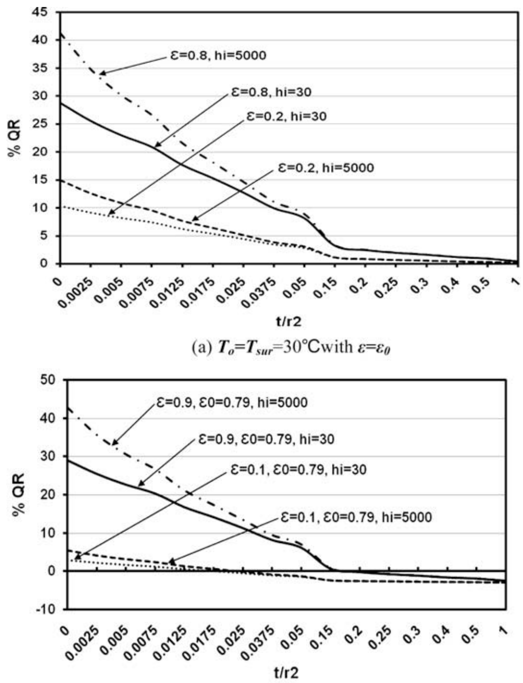
Table 1
The emissivities ε of various substances from the manual of infrared temperature demonstrator [24].

Substance	Emissivity
Human Skin	0.98
Gold	0.02
Silver	0.02
Aluminum	Weathered = 0.83; foil (bright) = 0.04 disk, rough = 0.96
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
Oil, lubricant	Film 0.03 mm = 0.27, film 0.13 mm = 0.72 thick = 0.82
Soil	Dry = 0.92, saturated water = 0.95
Water	Distilled = 0.96, frost = 0.98 snow = 0.85

Table 2
Referred approximate values of convection heat transfer [3].

Approximate values of convection heat transfer, h (W m^{-2} K^{-1})	
Natural convection	
Temp. diff. = 30 °C, horizontal plate 0.3 in high in air	4.5
Temp. diff. = 30 °C, 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 °C	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 (=36 km/h)	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 (=180 km/h)	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

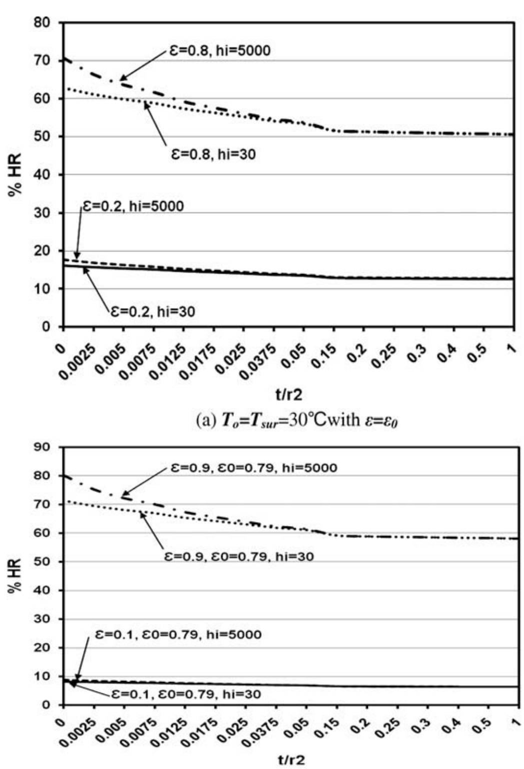
In steadily, the surface temperature difference generated by neglecting heat radiation effect based on $T_i < T_o$ and using Celsius temperature scale is defined as:



(a) T_o=T_{sur}=30°C with ε=ε_0

(b) T_o=30°C, T_{sur}=32°C with $\varepsilon \neq \varepsilon_0$

Fig. 2. The relations between QR and t/r_2 in the situation of T_i = 100 °C, K_A = 77 W m^{-1} K^{-1}, K_s = 0.035 W m^{-1} K^{-1}, r_1 = 195 mm, r_2 = 200 mm and h_o = 10 W m^{-2} K^{-1} (a) T_o = T_{sur} = 30 °C with $\varepsilon = \varepsilon_0$ and (b) T_o = 30 °C, T_{sur} = 32 °C with $\varepsilon \neq \varepsilon_0$.



(a) T_o=T_{sur}=30°C with ε=ε_0

(b) T_o=30°C, T_{sur}=32°C with $\varepsilon \neq \varepsilon_0$

Fig. 3. The relations between HR and t/r_2 in the situation of T_i = 100 °C, K_A = 77 W m^{-1} K^{-1}, K_s = 0.035 W m^{-1} K^{-1}, r_1 = 195 mm, r_2 = 200 mm and h_o = 10 W m^{-2} K^{-1} (a) T_o = T_{sur} = 30 °C with $\varepsilon = \varepsilon_0$; (b) T_o = 30 °C, T_{sur} = 32 °C with $\varepsilon \neq \varepsilon_0$.

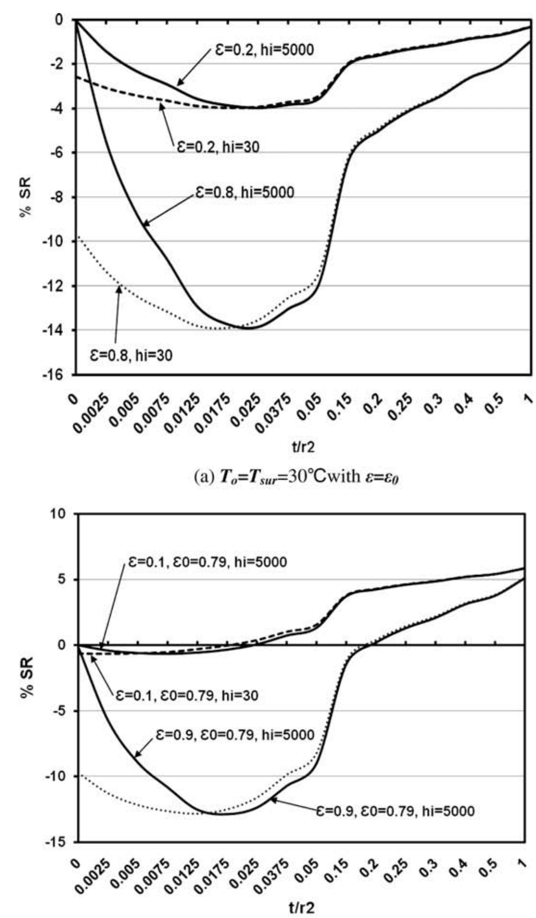
$$TD = T_{3a} - T_3 \tag{17}$$

The insulated effect EF_a of an insulated circular duct while considering heat radiation can be obtained:

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

3. Numerical heat transfer results

The nature of results of heat equation of non-insulated and insulated circular duct are one-dimensional exact solution. The exact numerical heat transfer results of an insulated circular duct are obtained by one-dimensional LabVIEW programming in this paper. According to the emissivities shown in Table 1, $\varepsilon = \varepsilon_0 = 0.8$ and $\varepsilon = \varepsilon_0 = 0.2$ are adopted to represent the high and low surface emissivity cases, respectively. It is shown in Table 2 that most natural convection coefficients of air are below 10 W m^{-2} K^{-1}; even in the situations of quite a high air speed, most convection coefficients are less than 100 W m^{-2} K^{-1}; thus 30 W m^{-2} K^{-1} can be used to represent the situation of medium wind speed of air; thus, h = 10 and 30 W m^{-2} K^{-1} are chosen to represent the low and medium convection coefficients cases of air/gas; meanwhile, even the natural convection coefficient of water is about 890 W m^{-2} K^{-1}, most forced convection coefficients of water are higher than 5000 W m^{-2} K^{-1}; thus, h = 5000 W m^{-2} K^{-1} is chosen to represent

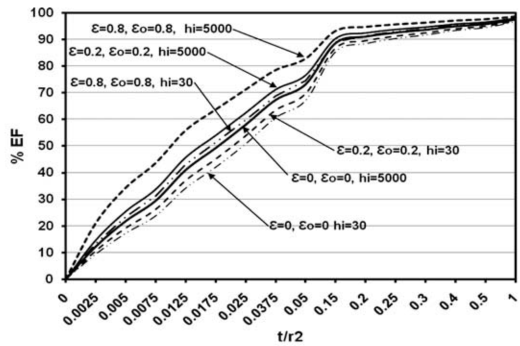


(a) T_o=T_{sur}=30°C with ε=ε_0

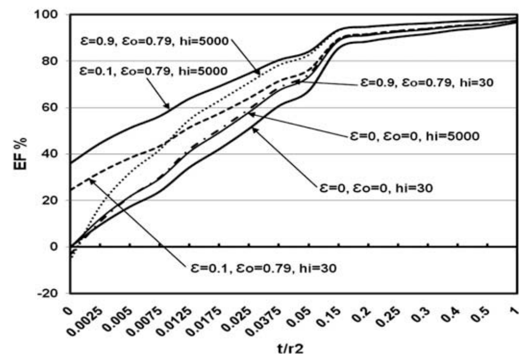
(b) T_o=30°C, T_{sur}=32°C with $\varepsilon \neq \varepsilon_0$

Fig. 4. The relations between SR and t/r_2 in the situation of T_i = 100 °C, K_A = 77 W m^{-1} K^{-1}, K_s = 0.035 W m^{-1} K^{-1}, r_1 = 195 mm, r_2 = 200 mm and h_o = 10 W m^{-2} K^{-1} (a) T_o = T_{sur} = 30 °C with $\varepsilon = \varepsilon_0$; (b) T_o = 30 °C, T_{sur} = 32 °C with $\varepsilon \neq \varepsilon_0$.

the medium forced convection coefficients cases of water/liquid. In the practical application, carbon steel duct is common applied to duct material, with heat conductivity $K_A = 77\ \mathrm{W\,m^{-1}\,K^{-1}}$, is used as the duct conductivity.

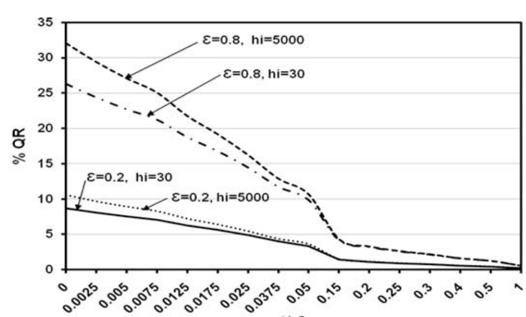


(a) $T_o=T_{sur}$=30°C with $\varepsilon=\varepsilon_0$

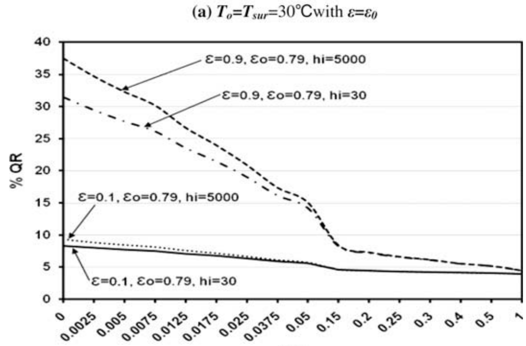


(b) T_o=30°C, T_{sur}=32°C with $\varepsilon\neq\varepsilon_0$

Fig. 5. The relations between *EF* and t/r_2 in the situation of $T_i = 100\ °\mathrm{C}$, $K_A = 77\ \mathrm{W\,m^{-1}\,K^{-1}}$, $K_s = 0.035\ \mathrm{W\,m^{-1}\,K^{-1}}$, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 10\ \mathrm{W\,m^{-2}\,K^{-1}}$ (a) $T_o = T_{\mathrm{sur}} = 30\ °\mathrm{C}$ with $\varepsilon = \varepsilon_0$; (b) $T_o = 30\ °\mathrm{C}$, $T_{\mathrm{sur}} = 32\ °\mathrm{C}$ with $\varepsilon \neq \varepsilon_0$.



(a) $T_o=T_{sur}$=30°C with $\varepsilon=\varepsilon_0$



(b) T_o=30°C, T_{sur}=32°C with $\varepsilon\neq\varepsilon_0$

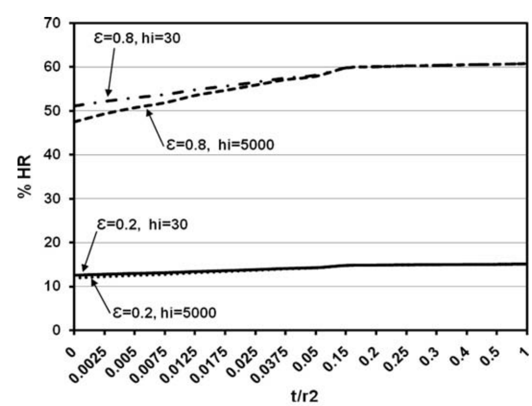
Fig. 6. The relations between *QR* and t/r_2 in the situation of $T_i = -20\ °\mathrm{C}$, $K_A = 77\ \mathrm{W\,m^{-1}\,K^{-1}}$, $K_s = 0.035\ \mathrm{W\,m^{-1}\,K^{-1}}$, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 8.3\ \mathrm{W\,m^{-2}\,K^{-1}}$ (a) $T_o = T_{\mathrm{sur}} = 30\ °\mathrm{C}$ with $\varepsilon = \varepsilon_0$; (b) $T_o = 30\ °\mathrm{C}$, $T_{\mathrm{sur}} = 32\ °\mathrm{C}$ with $\varepsilon \neq \varepsilon_0$.

In order to check if the numerical results are reliable, the following checking methods are used:

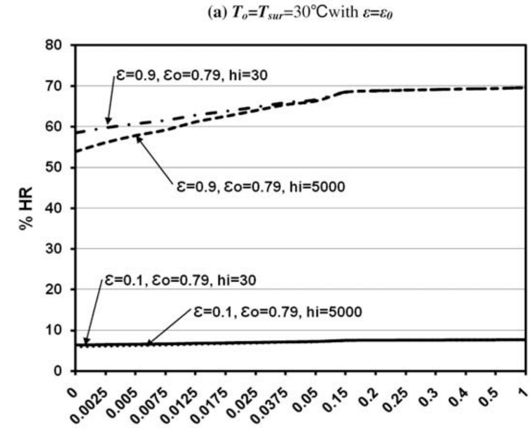
(1) Let surface emissivity $\varepsilon = \varepsilon_0 = 0$, make sure the results of $QR = 0$.
(2) Let surface emissivity $\varepsilon = \varepsilon_0 = 1$, external convection coefficient $h_o = 10{,}000\ \mathrm{W\,m^{-2}\,K^{-1}}$, make sure the results of *QR* close to zero.

4. Results and discussion

In order to increase the generalization of the results, all the results are shown in dimensionless parameters except *TD*, the surface temperature difference generated by neglecting heat radiation effect for insulated cold duct. An insulated hot duct in situations of $T_i = 100\ °\mathrm{C}$, $K_A = 77\ \mathrm{W\,m^{-1}\,K^{-1}}$, $K_s = 0.035\ \mathrm{W\,m^{-1}\,K^{-1}}$, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 10\ \mathrm{W\,m^{-2}\,K^{-1}}$ and divided into two groups. Group (a): $T_o = T_{\mathrm{sur}} = 30\ °\mathrm{C}$ with $\varepsilon = \varepsilon_0$, and group (b): $T_o = 30°\mathrm{C}$, $T_{\mathrm{sur}} = 32\ °\mathrm{C}$ with $\varepsilon \neq \varepsilon_0$, are applied as the first case study and the results are shown in Figs. 2–5. The group (a), with $T_o = T_{\mathrm{sur}} = 30\ °\mathrm{C}$ and $\varepsilon = \varepsilon_0$, is a rare happened special case just using for more convenient to compare the differences between $\varepsilon = \varepsilon_0 \neq 0$ and $\varepsilon = \varepsilon_0 = 0$. On the other hand, the group (b), with $T_o = 30\ °\mathrm{C}$, $T_{\mathrm{sur}} = 32\ °\mathrm{C}$ associated to $\varepsilon = 0.9$ and 0.1 with $\varepsilon_0 = 0.79$, are some practical situations may actually occur. It can be found in Table 1 that $\varepsilon_0 = 0.79$ represents for oxidized steel and $\varepsilon = 0.9$ may represent average value of insulated material and $\varepsilon = 0.04$ represents aluminum foil, thus $\varepsilon = 0.1$ may represent the situation



(a) $T_o=T_{sur}$=30°C with $\varepsilon=\varepsilon_0$



(b) T_o=30°C, T_{sur}=32°C with $\varepsilon\neq\varepsilon_0$

Fig. 7. The relations between *HR* and t/r_2 in the situation of $T_i = -20\ °\mathrm{C}$, $K_A = 77\ \mathrm{W\,m^{-1}\,K^{-1}}$, $K_s = 0.035\ \mathrm{W\,m^{-1}\,K^{-1}}$, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 8.3\ \mathrm{W\,m^{-2}\,K^{-1}}$ (a) $T_o = T_{\mathrm{sur}} = 30\ °\mathrm{C}$ with $\varepsilon = \varepsilon_0$; (b) $T_o = 30\ °\mathrm{C}$, $T_{\mathrm{sur}} = 32\ °\mathrm{C}$ with $\varepsilon \neq \varepsilon_0$.

while an insulation enveloped with a wrinkle aluminum foil. From group (b), it can figure out the actual heat transfer characteristics in some practical situations of $\varepsilon \neq \varepsilon_0$ and $T_0 \neq T_{sur}$.

Figs. 2–5 show that the heat transfer rate errors QR, convective coefficients ratio percentage HR, the error of surface temperature SR and insulated effect EF are affected by the dimensionless insulated thickness t/r_2, internal convection coefficient h_i and the surface emissivity ε; QR and HR decrease other than EF increases along with increasing t/r_2; the higher the ε and h_i are, the higher the QR, HR and EF will be. It is found that from Fig. 2a that all QR are greater than zero, indicating $q_a > q$, and QR gets close to zero when t/r_2 approaches to one. On the contrary, Fig. 2b shows that $QR > 0$ before some point of small t/r_2, and reverse ($QR < 0$) after that point, indicating q_a may $\geqq$ or $\leqq$ q; and $QR \fallingdotseq -2\%$ even at $t/r_2 = 1$. Fig. 4 shows that absolute value of SR increases before some point near $t/r_2 = 0.02$ then decreases after that point for higher values of emissivity ($\varepsilon \geqq 0.2$); this critical phenomenon is quite interesting, it may explain that heat radiation effect with high emissivity values causes lowest surface temperature compared with that of neglecting heat radiation effect at some thin insulated thickness in situations of hot duct; the higher the ε is, the greater the absolute value of SR will be; but for the same ε, the smaller the h_i is, the greater the absolute value of SR will be before some point of small t/r_2, and reverse after that point. Fig. 4a shows that all $SR < 0$, indicating $T_{3a} < T_3$ for hot duct, and SR tends getting close from -0.5% to -1% at $t/r_2 = 1$. On the contrary, Fig. 4b shows that $SR < 0$ before some point of small t/r_2, and reverse ($SR > 0$) after that point, indicating T_{3a} may $\leqq$ or $\geqq$ T_3; and $SR > 5\%$ even at $t/r_2 = 1$. Fig. 5a shows that for the same h_i, the EF of the $\varepsilon = \varepsilon_0 > 0$ are greater than that of $\varepsilon = \varepsilon_0 = 0$; Relatively, Fig. 5b shows that for the same h_i, the EF of the $\varepsilon < \varepsilon_0$ are all greater than that of $\varepsilon = \varepsilon_0 = 0$ but not for those of $\varepsilon > \varepsilon_0$.

Meanwhile, the situations of an insulated cold duct with $T_i = -20$ °C, $K_A = 77$ W m^{-1} K^{-1}, $K_s = 0.035$ W m^{-1} K^{-1}, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 8.3$ W m^{-2} K^{-1} and divided into two groups: (a) $T_o = T_{sur} = 30$ °C with $\varepsilon = \varepsilon_0$ (b) $T_o = 30$°C, $T_{sur} = 32$ °C with $\varepsilon \neq \varepsilon_0$ is applied as the second case study and the results are shown in Figs. 6–9.

Figs. 6–9 show that QR, HR, the temperature difference TD and insulated effect EF are affected by t/r_2, h_i and ε; QR and HR decrease other than EF increases along with increasing t/r_2; the higher the ε and h_i are, the higher the QR, HR and EF will be. Fig. 8 shows that TD decreases before some point near $t/r_2 = 0.02$ then increases after that point for the higher value of emissivity ($\varepsilon \geqq 0.2$); the critical phenomenon is quite interesting, it may explain that heat radiation effect with high emissivity value causes highest surface temperature compared with that of neglecting heat radiation effect at some thin insulated thickness in situations of cold duct; the higher the ε is, the greater the TD will be; but for the same ε, the smaller the h_i is, the greater the TD will be before some point of small t/r_2, and reverse after that point. Fig. 8 shows that all $TD > 0$, indicating $T_{3a} > T_3$ for cold duct. It is found that TD tends getting close from 0.1 to 0.3 °C at $t/r_2 = 1$ in Fig. 8a; on the contrary, $TD > 2°$ C at $t/r_2 = 1$ in Fig. 8b. Fig. 9a shows that for the same h_i, the EF of the $\varepsilon = \varepsilon_0 > 0$ are greater than that of $\varepsilon = \varepsilon_0 = 0$; Relatively, Fig. 9b shows that for the same h_i, the EF of the $\varepsilon < \varepsilon_0$ are all greater than that of $\varepsilon = \varepsilon_0 = 0$ but not for those of $\varepsilon > \varepsilon_0$.

In this paper, we let group (a) with $T_o = T_{sur}$ just for the purpose to compare the results between considering and neglecting heat radiation, thus the values of HR of group (a) shown in Figs. 3a

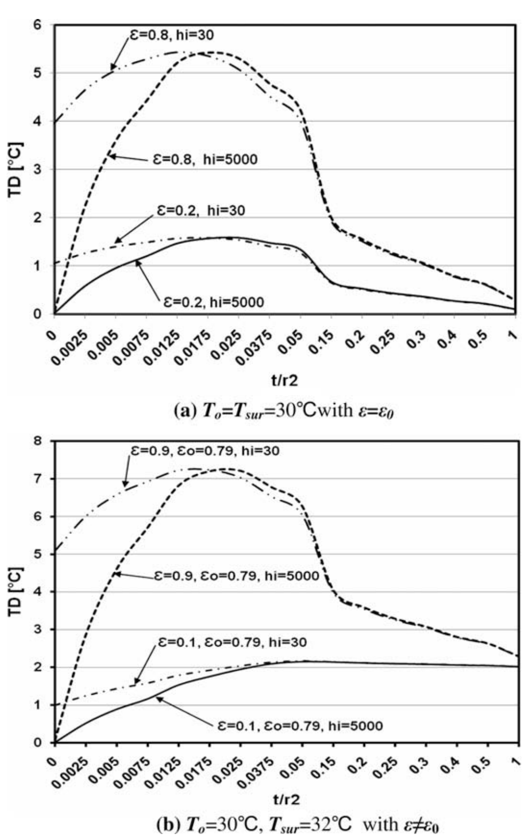


Fig. 8. The relations between SR and t/r_2 in the situation of $T_i = -20$ °C, $K_A = 77$ W m^{-1} K^{-1}, $K_s = 0.035$ W m^{-1} K^{-1}, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 8.3$ W m^{-2} K^{-1} (a) $T_o = T_{sur} = 30$ °C with $\varepsilon = \varepsilon_0$; (b) $T_o = 30$ °C, $T_{sur} = 32$ °C with $\varepsilon \neq \varepsilon_0$.

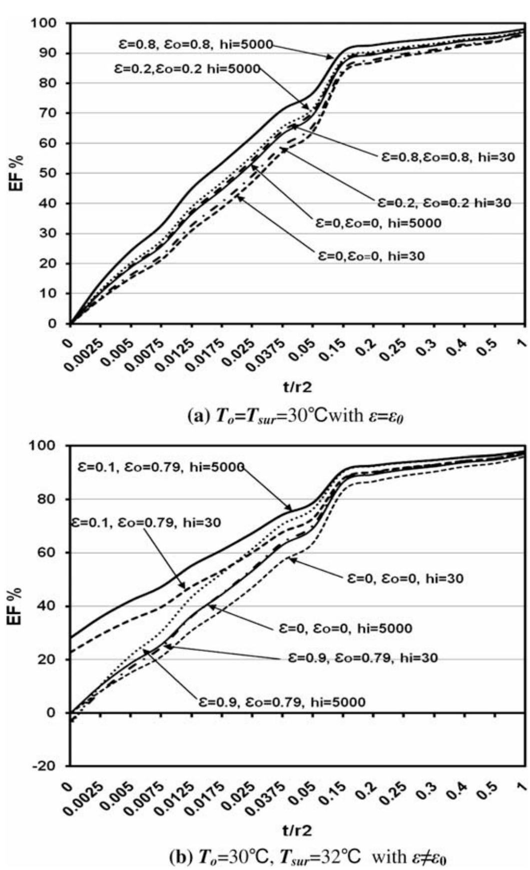


Fig. 9. The relations between EF and t/r_2 in the situation of $T_i = -20$ °C, $K_A = 77$ W m^{-1} K^{-1}, $K_s = 0.035$ W m^{-1} K^{-1}, $r_1 = 195$ mm, $r_2 = 200$ mm and $h_o = 8.3$ W m^{-2} K^{-1} (a) $T_o = T_{sur} = 30$ °C with $\varepsilon = \varepsilon_0$; (b) $T_o = 30$ °C, $T_{sur} = 32$ °C with $\varepsilon \neq \varepsilon_0$.

Table 3
T_i = 100 °C, $T_o = T_{sur}$ = 30 °C, K_A = 77 W m^{-1} K^{-1}, K_s = 0.035 W m^{-1} K^{-1}, r_1 = 195 mm, r_2 = 200 mm, h_o = 10 W m^{-2} K^{-1} and h_i = 30 W m^{-2} K^{-1}.

t (mm)	t/r_2	q_a (W m^{-1})	q_r (W m^{-1})	q_c (W m^{-1})	q (W m^{-1})	QR (%)	HR (%)	T_{3a} (°C)	T_3 (°C)	SR (%)
(a) $\varepsilon = \varepsilon_0$ = 0.8										
0	0	919.73	355.14	564.61	655.21	28.7	62.9	74.9	82.1	−9.6
2.5	0.0125	523.68	191.05	332.63	431.19	17.6	57.4	56.1	63.8	−13.8
5	0.025	369.34	131.55	237.79	322.31	12.7	55.3	48.4	55.0	−13.5
10	0.05	234.51	81.72	152.79	215.44	8.1	53.4	41.5	46.3	−11.4
50	0.25	64.58	21.87	42.71	63.31	1.9	51.2	32.7	34.0	−4.0
100	0.5	36.72	12.38	24.34	36.38	0.9	50.8	31.2	31.9	−2.0
200	1.0	21.83	7.34	14.49	21.75	0.4	50.6	30.5	30.8	−0.9
(b) $\varepsilon = \varepsilon_0$ = 0.1										
0	0	693.9	51.98	641.9	655.2	5.58	8.1	81.1	82.1	−1.30
2.5	0.0125	446.1	30.77	415.3	431.2	3.34	7.41	62.6	63.9	−1.99
5	0.025	330.1	21.89	308.2	322.3	2.37	7.1	53.9	55	−2.02
10	0.05	218.7	13.96	204.7	215.4	1.49	6.82	45.5	46.3	−1.78
50	0.25	63.53	3.841	59.69	63.31	0.35	6.43	33.8	34	−0.68
100	0.5	36.44	2.183	34.26	36.38	0.17	6.37	31.8	31.9	−0.35
200	1.0	21.76	1.298	20.47	21.75	0.07	6.34	30.8	30.9	−0.17

and 7a are actually the same as $q_r/q_c \times 100\%$; relatively, the values of HR of group (b) shown in Figs. 3b and 7b are only close to $q_r/q_c \times 100\%$. Anyway, high value of HR indicates that the influence of heat radiation cannot be neglected. It can be found in Figs. 3 and 7 that the values of HR keep with quite high values at all t/r_2, even at the very thick insulation of $t/r_2 = 1$, $HR \fallingdotseq 70\%$ for $\varepsilon = 0.98$ and $HR \fallingdotseq 51\%$ for $\varepsilon = 0.8$, $HR \fallingdotseq 13\%$ for $\varepsilon = 0.2$ and $HR \fallingdotseq 7.5\%$ for $\varepsilon = 0.1$. It can be seen in Figs. 2b and 6b that QR is very big at small t/r_2, $QR > 3\%$ even in very thick insulation of $t/r_2 = 1$ for group (b). The group (b) can represent some situations may happen in our common living surroundings. Therefore, heat radiation effect should not be neglected for non-insulated or insulated duct in practical application with ambient air.

In order to demonstrate the main differences between the heat transfer characteristics of situations with and without considering heat radiation, the detail data of an insulated hot duct with $\varepsilon = \varepsilon_0 = 0.8$ and 0.1 are listed in Table 3a and b, respectively. For example, the data of $t/r_2 = 0.05$ and $\varepsilon = \varepsilon_0 = 0.8$ listed in Table 3a show that $q_a = 234.51$ W m^{-1} ($>q = 215.44$) and $q_r = 81.72$ W m^{-1}, $q_c = 152.79$ W m^{-1} $\neq q = 215.44$ W m^{-1}; $T_{3a} = 41.5$ °C $< T_3 = 46.3$ °C; thus $HR = q_r/q_c \times 100\% = 53.4\%$ causes quite a big $QR = 8.1\%$. It can be explained from Eqs. (10) and (3) that if $\varepsilon \neq 0$, $T_{3a} \neq T_3$, $q_c = h_o 2\pi r_3 L(T_{3a} - T_o)$ is not equal to $q = h_o 2\pi r_3 L(T_3 - T_o)$ with $\varepsilon = \varepsilon_0 = 0$. It can be found from above information that the heat transfer rate, q, in situations neglecting the heat radiation, is not the real heat convection, q_c. It can be concluded from above findings that the heat transfer rate calculated by neglecting the heat radiation is not satisfactorily accurate. Therefore, the heat radiation effect of non-insulated and insulated duct can not be ignored in situations of lower h_o and higher ε. It is shown in Table 3 that the values of q_a with $\varepsilon = \varepsilon_0 = 0.8$ are much greater than those with $\varepsilon = \varepsilon_0 = 0.1$ in the same non-insulated situations ($t/r_2 = 0$); thus, it is proved that a larger surface emissivity can greatly improve the performance of a heat exchanger. Meanwhile, It can be also found in Table 3 that the values of q_a with $\varepsilon = \varepsilon_0 = 0.1$ are much less than those with $\varepsilon = \varepsilon_0 = 0.8$ in the same insulated situations; thus, it is also proved that a smaller surface emissivity can obtain better insulation.

5. Conclusion

From the practical numerical results of this study, it demonstrates that one will take a big risk to neglect the influence of heat radiation for the non-insulated and thinner insulated duct in the situations of ambient air with low external convection coefficients and greater surface emissivity, especially while the ambient air temperature is different from that of surroundings and greater internal fluid convection coefficients. Since the emissivity of duct surface should always be different from that of insulated surface, and practically the temperature of external air should not be the same value as surroundings. Therefore, it becomes more complicate in real situations of considering heat radiation effect. Fortunately, it is no difficult to calculate the heat transfer results in such actual situations with considering heat radiation; this only requires a PC incorporated with a small computer code in any language (such as LabVIEW in this study). It is also found in this paper that using greater duct surface emissivity can greatly improve the heat exchanger effect and using smaller insulated surface emissivity can obtain better insulation.

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