

Reliable One-Dimensional Model of Two-Dimensional Insulated Oval Duct Considering Heat Radiation

King-Leung Wong, Wen-Lih Chen, and Yu-feng Chang

Abstract—The reliable results of an insulated oval duct considering heat radiation are obtained basing on accurate oval perimeter obtained by integral method as well as one-dimensional Plane Wedge Thermal Resistance (PWTR) model. This is an extension study of former paper of insulated oval duct neglecting heat radiation. It is found that in the practical situations with long-short-axes ratio $a/b \leq 5/1$, heat transfer rate errors are within 1.2 % by comparing with accurate two-dimensional numerical solutions for most practical dimensionless insulated thickness ($t/R_2 \leq 0.5$). On the contrary, neglecting the heat radiation effect is likely to produce very big heat transfer rate errors of non-insulated ($E > 43\%$ at $t/R_2 = 0$) and thin-insulated ($E > 4.5\%$ while $t/R_2 \leq 0.1$) oval ducts in situations of ambient air with lower external convection heat coefficients and larger surface emissivity.

Keywords—Heat convection, heat radiation, oval duct, PWTR model.

I. INTRODUCTION

THERE has been interest in heat transfer and flow characteristics of an oval duct. Schulenberg [1] presented the applications of finned elliptical tubes in air-cooled heat exchanger, Ota et al.[2] investigated the heat transfer and flow characteristics around an elliptic cylinder, Chen et al.[3] studied the heat transfer enhancement of finned oval tubes with staggered punched longitudinal vortex generators, Kim et al.[4] presented the heat transfer and pressure drop characteristics during R22 evaporation in an oval micro-fin tube, Leu et al.[5] investigated the louvered fin-and-tube heat exchangers having circular and oval tube configurations, Tiwari et al.[6] investigated the heat transfer enhancement in cross-flow heat exchangers using oval tubes and multiple delta winglets, Hasan and Siren[7] studied the performance of plain circular and oval tube evaporatively cooled heat exchangers, O'Brien et al.[8] presented the local heat transfer and pressure drop for finned-tube heat exchangers using oval tubes and vortex

generators, Wong et al.[9] studied the critical heat transfer characteristics of an insulated oval duct.

Additionally, insulation of hot and cold duct has been a very important engineering problem. Since the use of insulation is an essential step in refrigeration and air-conditioning systems, yet the constant surface area Plate Thermal Resistance (PTR) model has been the main conventionally used method to analyze the heat transfer characteristics of insulated polygonal ducts. The model assumes that the insulated surface area of the external insulation is the same as that of a bare duct. Definitely neglecting increasing surface area caused by increasing insulated thickness will significantly reduce the results of heat transfer rate. Recently, Chou and Wong [10] proposed a Plane Wedge Thermal Resistance (PWTR) model aimed to investigate the heat-transfer characteristics associated with insulated polygonal ducts. In this model, the thermal resistance due to the inner convection term and the duct conduction term was not considered, hence theoretically, its results were only sufficient when applied to ducts with high inner convection coefficient h_i and high wall conductivity K , such as condensers and evaporators. However, Wong et al. [11] proved that the PWTR model can be used in cases involving low to medium values of h_i and K , extensively enlarging the model's applicability to practical situations. In general if the one-dimensional PWTR model can apply to an insulated circular duct to obtain accurate solutions. And while it is used to analyze the two-dimensional heat transfer problems of insulated polygonal ducts, lower accuracy is obtained with decreasing numbers of edges. Wong et al. [12] studied the heat-transfer characteristics of an insulated long rectangular or square duct using the one-dimensional PWTR model and PTR model as well as numerical methods. They found that the errors generated by the PWTR model are all positive and those generated by the PTR model are all negative. Thus, the 64-CPWTR model (combined 60% PWTR and 40% PTR models) can balance the positive and negative errors and obtain reasonably accurate results in comparison with a two-dimensional numerical solution. Wong et al. [13] extended the various CPWTR models to obtain the reliable one-dimensional heat-transfer results associated with insulated regular polygonal ducts. Later, Chen et al. [14] investigated an insulated oval duct neglecting heat radiation based on a very accurate oval perimeter obtained by integral method, and the one-dimensional results of PWTR and 91-CPWTR models are compared with those of numerical results for the insulated oval duct in various situations

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neglecting heat radiation with different long-short-axes ratios. They found that the acceptable results of heat transfer rate can be obtained by the one-dimensional PWTR model long-short-axes ratios a/b is less than or equal to 3/1, when the long to short axes ratio is greater than 3/1, better solutions can be achieved by the one-dimensional 91-CPWTR model. For most practical insulated thickness situations ($t/R_2 < 0.5$), applying the one-dimensional PWTR model to insulated oval ducts neglecting heat radiation with the long to short axes ratio a/b less than or equal to 5/1 results in errors within $\pm 1\%$.

Recently, Hsien et al. [15] studied the inaccuracy of heat transfer characteristics of insulated and non-insulated circular duct while neglecting the influence of heat radiation. They found that neglecting the heat radiation effect is likely to produce large errors of non-insulated oxidized metal ducts with larger surface emissivity in situations of ambient air with low external convection heat coefficients, especially while the ambient air temperature, T_o , is different from surroundings temperature, T_{sur} , and greater internal fluid convection coefficients, h_i . Wong et al. [16] applied this finding to developed the log mean heat transfer rate method of heat exchanger considering the influence of heat radiation. Later, Wong et al. [17] obtained the inaccuracy of heat transfer characteristics for non-insulated and insulated spherical containers neglecting the influence of heat radiation. Thus, the same phenomena will also occur in the situations of insulated and non-insulated oval ducts while neglecting the influence of heat radiation.

The purpose of this study is extended the above theory to obtained reliable one-dimensional numerical solutions of two-dimensional insulated and non-insulated oval duct considering heat radiation.

II. PROBLEM FORMULATION

Fig. 1 shows that an insulated oval duct with the outside half-long-axis-length a as well as half-short-axis-length b of an bare cross-sectional profile, duct thickness t_1 , duct length L , wall conductivity K , and duct surface emissivity ϵ_0 ; the insulated layer with surface emissivity ϵ , and with thickness t , is wrapped around the duct with conductivity K_s , internal fluid and external ambient air with convection heat transfer coefficients h_i and h_o , temperatures T_i and T_o , respectively. The duct is exposed to the outside surroundings temperature T_{sur} , and the insulated duct surface area is very small compared with that of surroundings. In addition, the bare oval ducts with the various long-short-axes ratios a/b of 3/2, 2/1, 3/1, 4/1 and 5/1, transformed from a bare circular duct with the same perimeter (the radius R_2 of the bare circular duct is treated as the equivalent radius of oval duct) are analyzed in this study. The equivalent radius R_2 is also used as a base of the dimensionless thickness t/R_2 and duct size $R_2 h_o / K_s$.

A. The Oval Perimeter

The following approximate oval perimeter equation edited by William [18] has been conventionally used to analyze the characteristics of oval duct:

$$S = 2\pi \sqrt{\frac{a^2 + b^2}{2}} \quad (1)$$

The oval equation of the oval cross section profile shown in Fig. 2 is:

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1 \quad (2)$$

The perimeter of the oval cross-sectional profile can then be divided into n line segments, as shown in Fig. 2, the two end points for the i^{th} line segment can be expressed as:

$$x_i = a \cos(2i\pi/n) \quad (3)$$

$$y_i = b \sin(2i\pi/n) \quad (4)$$

$$x_{i+1} = a \cos[2(i+1)\pi/n] \quad (5)$$

$$y_{i+1} = b \sin[2(i+1)\pi/n] \quad (6)$$

From the definition of the line segment, the perimeter of the oval profile shown in Fig. 2 can be obtained by the following summation:

$$P_{E2} = \sum_{\theta=0}^{2\pi} \sqrt{(x_{i+1} - x_i)^2 + (y_{i+1} - y_i)^2} \quad (7)$$

$$A_2 = LP_{E2} \quad (8)$$

Similarly, accurate internal surface area of oval duct, A_1 , can be obtained by replacing the values of a and b by $(a-t_1)$ and $(b-t_1)$, respectively in equations (3) to (8); meanwhile, accurate external insulation surface area, A_3 , can be obtained by replacing the values of a and b by $(a+t)$ and $(b+t)$ in equations (3) to (8), respectively.

In order to check the accuracy of Eq. (7), let $a=b=1$ to make Eq. (2) become a circle; if let $n=10^6$, the error generated from Eq. (7) compared with accurate solution is about 10^{-6} ; if let $n=10^7$, the error generated from Eq. (7) is about 10^{-7} ; for obtaining more convincing results, $n=10^7$ is applied in the present investigation, it takes only about 2 seconds computing time in every result in a one-dimensional programming. Thus, the results obtained from Eq. (7) can be treated as an accurate oval perimeter. Therefore, the error of oval perimeter generated by Eq. (1) can be expressed as:

$$E_S = \frac{S - L_{E2}}{L_{E2}} \times 100\% \quad (9)$$

The results of Eq. (9) are shown in Fig. 3. Fig. 3 shows that the greater the long-short-axes ratio a/b , the bigger the E_S will be. For example, $a/b=2$, $E_S=2.2\%$; $a/b=5$, $E_S=7.8\%$. It can be proved from Fig. 3 that the results based on Eq. (1) are not reliable.

B. The Equivalent Circular Duct Based On Accurate Oval Perimeter

Conventionally, to find out the heat transfer characteristics of an insulated oval by using the equivalent circular duct based on same oval perimeter. The radius R_2 of equivalent of circular duct based on accurate oval perimeter is:

$$R_2 = \frac{L_{E2}}{2\pi} \quad (9)$$

R_2 , shown in Eq. (9), can also be used as a base for the dimensionless thickness t/R_2 or dimensionless size $R_2 h_o/K_s$ of an insulated oval duct.

III. THE HEAT TRANSFER RATES OF INSULATED OVAL DUCTS

It has been found that the PWTR model can also be applied to the circular duct since the PWTR model can be simplified to the circular duct thermal resistance model as follows:

$$R_{thw} = \frac{t \times \ln(A_3/A_2)}{K_s(A_3 - A_2)} = \frac{t \times \ln(2\pi R_3 L)/(2\pi R_2 L)}{2\pi(R_3 - R_2)K_s L} = \frac{\ln(R_3/R_2)}{2\pi K_s L} \quad (10)$$

Eq. (10) shows that the thermal resistance of an insulated circular duct is a special case of the PWTR model for an insulated regular polygonal duct, since the circular duct can be obtained by letting the number of edges approach infinity. Since the circular duct solution is an accurate analytic result, with no approximations required, it follows that greater accuracy is obtained with regular insulated polygonal ducts with high numbers of edges when applying the PWTR model. While the PWTR applying to an insulated oval ducts, the smaller the value of long-short-axes ratio a/b is, the more accurate results will be obtained.

A. Situations Neglecting the Influence of Heat Radiation

The heat transfer rate of an insulated oval duct with PWTR model and based on accurate oval perimeter is:

$$Q = \frac{T_i - T_o}{\frac{1}{h_i A_1} + \frac{t \ln \frac{A_2}{A_1}}{K(A_2 - A_1)} + \frac{t \ln \frac{A_3}{A_2}}{K_s(A_3 - A_2)} + \frac{1}{h_o A_3}} = \frac{T_i - T_o}{\frac{1}{h_o A_3}} \quad (11)$$

Thus, the heat transfer rate neglecting heat radiation, Q , and insulated surface, T_s , can be obtained from Eq. (11).

Heat transfer rate error compared with accurate two-dimensional numerical heat transfer rate considering heat radiation, Q_n is:

$$E = \left(\frac{Q - Q_n}{Q_n} \right) \times 100 \quad (12)$$

And surface temperature error compared with average

numerical surface temperature considering heat radiation, T_n is:

$$ET = \left(\frac{T_s - T_n}{T_n} \right) \times 100 \quad (13)$$

B. Situations Considering the Influence of Heat Radiation

The heat transfer rate of an insulated oval duct considering heat radiation as shown in Fig. 1 with PWTR model and based on accurate oval perimeter can be written as :

$$Q_a = \frac{T_i - T_{sa}}{\frac{1}{h_i A_1} + \frac{t \ln \frac{A_2}{A_1}}{K(A_2 - A_1)} + \frac{t \ln \frac{A_3}{A_2}}{K_s(A_3 - A_2)}} \quad (14)$$

The heat convection transfer rate outside the insulated surface of an insulated oval duct as shown in Fig. 1 is:

$$Q_c = h_o A_3 (T_{sa} - T_o) \quad (15)$$

The surface heat radiation transfer rate as shown in Fig. 1 is:

$$Q_r = \epsilon \sigma A_3 (T_{sa}^4 - T_{sur}^4) \quad (16)$$

The following equation is obtained from heat balance:

$$Q_a = Q_c + Q_r \quad (17)$$

Thus, the heat transfer rate Q_a , and surface temperature T_{sa} of situations considering the influence of external surface heat radiation can be obtained from Eqs. (14)-(17).

Heat transfer rate error compared with accurate two-dimensional numerical heat transfer rate considering heat radiation, Q_n is:

$$E_a = \left(\frac{Q_a - Q_n}{Q_n} \right) \times 100 \quad (18)$$

And surface temperature error compared with average two-dimensional numerical surface temperature considering heat radiation, T_n is:

$$ET_a = \left(\frac{T_{sa} - T_n}{T_n} \right) \times 100 \quad (19)$$

IV. ENERGY EQUATION AND BOUNDARY CONDITIONS

The heat conduction equation for a two-dimensional insulated oval duct is:

$$\frac{\partial}{\partial x} (K \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (K \frac{\partial T}{\partial y}) = 0 \quad (20)$$

With the boundary conditions:

$$h_i(T_i - T_s)_{A_1} = -K \left(\frac{\partial T}{\partial N} \right)_{A_1} \quad (21)$$

$$-K_s \left(\frac{\partial T}{\partial N} \right)_{A_3} = h_o(T_s - T_o)_{A_3} + \varepsilon\sigma(T_s^4 - T_{sur}^4)_{A_3} \quad (22)$$

$$\left(\frac{\partial T}{\partial N} \right)_{\text{symmetrical boundary}} = 0 \quad (23)$$

where N is the normal direction at a boundary and T_s represented the surface temperature of insulated layer.

V. NUMERICAL TWO-DIMENSIONAL HEAT TRANSFER RESULTS

The two-dimensional numerical heat transfer results of an insulated oval duct are obtained FORTRAN programming. In order to check if the numerical results are reliable, an insulated circular duct is analyzed to determine how many quadrilateral cells are needed to obtain a satisfactory result. It was found that a model of an insulated circular duct, which used 10738 quadrilateral cells, gave a satisfactory solution of heat transfer rate within $\pm 0.01\%$ of the accurate analytic solution. Thus, using a greater number of quadrilateral cells to analyze an insulated oval duct can be expected to provide reliable results. The other judgment to prove the numerical results are very reliable by checking the errors are almost zero in the situations of non-insulated oval duct ($t/R_2=0$).

VI. RESULTS AND DISCUSSIONS

Table I shows that most surface emissivity of oxidized metals and insulated materials are quite high, $\varepsilon=\varepsilon_0=0.8$ is adopted to represent the emissivity in show-cases. It is shown in Table II that most natural convection coefficients of air are below $10 \text{ Wm}^{-2}\text{K}^{-1}$, the value of $8.3 \text{ Wm}^{-2}\text{K}^{-1}$ is always adopted to represent the ambient air in air-condition applications, thus $h_o=8.3 \text{ Wm}^{-2}\text{K}^{-1}$ is adopted in this study. Table II also shows that forced convection coefficients of water are most over $5000 \text{ Wm}^{-2}\text{K}^{-1}$, and the bigger value of h_i intends to obtained bigger heat transfer rate errors, thus $h_i=5000 \text{ Wm}^{-2}\text{K}^{-1}$ is adopted in this study. In the practical application, carbon steel with heat conductivity $K=77 \text{ Wm}^{-1}\text{K}^{-1}$, is common applied as a duct material, and it is also adopted in this study.

The heat transfer rate and surface temperature errors with $h_i=5000 \text{ Wm}^{-2}\text{K}^{-1}$, $t_1=1 \text{ mm}$, $K=77 \text{ Wm}^{-2}\text{K}^{-1}$, $K_s=0.035 \text{ Wm}^{-2}\text{K}^{-1}$, $h_o=8.3 \text{ Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $\varepsilon_0=\varepsilon=0.8$ with $a/b=2/3$, $2/1$, $3/1$, $4/1$, $5/1$ and $R_2h_o/K_s=36.56\sim 64.75$ as well as three combination of $T_o=T_{sur}=0^\circ\text{C}$, $T_o=0^\circ\text{C}$ and $T_{sur}=2^\circ\text{C}$, $T_o=0^\circ\text{C}$ and $T_{sur}=5^\circ\text{C}$ are shown in Figs. 4-9, respectively. It can be seen from Figs.4a, 6a and 8a that the heat transfer rate error of an insulated oval duct considering heat radiation with PWTR model and based on accurate oval perimeter, E_a , are acceptable; while $t/R_2 \geq 0.2$, the bigger the a/b is, the bigger E_a will be; $E_a \leq 4.3\%$ with $t/R_2 \leq 2.0$ even at $a/b=5/1$; Hsien et al.[15] proved that the optimum

commercial dimensionless insulated thickness $t/R_2=0.2$ for an insulated circular duct considering heat radiation, because the insulated effect does only increase with small percentage after $t/R_2 > 0.2$; thus for the most practical insulated thickness situations: $t/R_2 \leq 0.2$, $E_a < 0.3\%$ and $t/R_2 \leq 0.5$, $E_a < 1.2\%$ with $a/b \leq 5/1$; E_a is almost independent of the three combination of T_o and T_{sur} ; it is amazing that $E_a \approx 0.0\%$ at $t/R_2=0$, thus the PWTR model and based on accurate oval perimeter can be used to solve the practical heat transfer characteristics of oval-pipe heat exchanger applying in ambient air. On the contrary, it can be seen from Figs. 4b, 6b and 8b that the heat transfer errors neglecting heat radiation E, are not acceptable; neglecting the heat radiation effect is likely to produce very big errors of non-insulated ($E > 43\%$ at $t/R_2=0$) and thin-insulated ($E > 4.5\%$ while $t/R_2 \leq 0.1$) ducts in situations of ambient air with low external convection heat coefficients and larger surface emissivity; E is a little bit relative to the three combination of T_o and T_{sur} , the bigger the difference between T_o and T_{sur} is, higher E may be obtained, especially at $t/R_2=0$.

It also can be seen from Figs. 5, 7 and 9 that the surface temperature errors of an insulated oval duct considering heat radiation with PWTR model and based on accurate oval perimeter, ET_a , are quite small and smaller than those neglecting heat radiation in the same situations, i.e., $ET_a < ET$; thus, the surface temperatures of an insulated oval duct obtained with PWTR model and based on accurate oval perimeter and considering heat radiation are acceptable.

VII. CONCLUSION

The conventional approximate oval perimeter is not suitable to apply to the insulated oval duct to obtain the reliable results. Heat radiation must be considered for the non-insulated and thin-insulated oval ducts in situations of ambient air with low external convection heat coefficients and larger surface emissivity. The reliable one-dimensional approximate solution of an insulated and non-insulated oval duct considering heat radiation basing on accurate oval perimeter as well as one-dimensional PWTR model can be obtained by simple one-dimensional programming within two seconds computing time. Thus, it is more convenient in practical engineering applications.

VIII. ACKNOWLEDGMENT

The authors would like to acknowledge the National Science Council of Taiwan, R.O.C. This investigation is completed under the support of the project NSC-101-2221-E-168-025-.

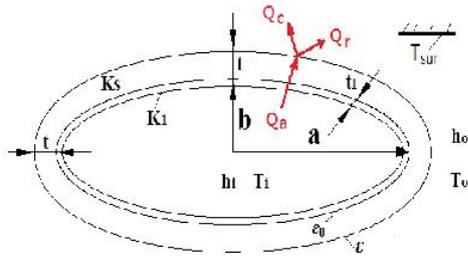


Fig.1 The parameters of an insulation oval duct

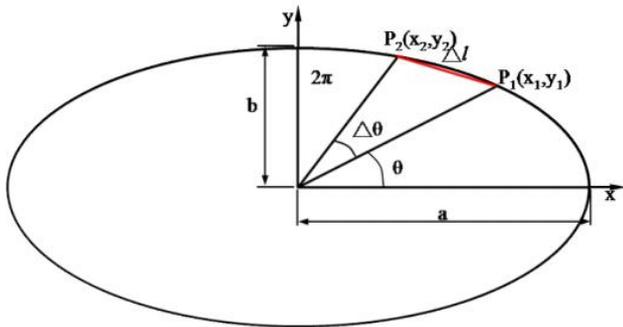


Fig. 2 The perimeter of the oval cross-sectional profile

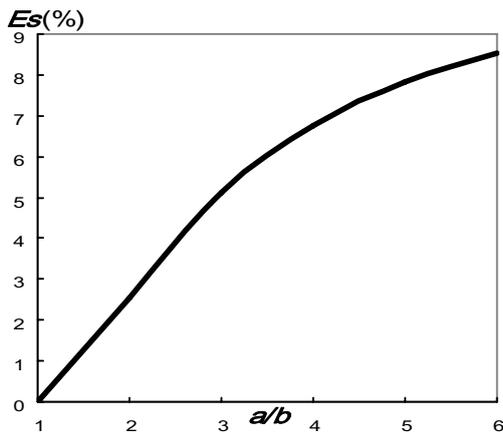
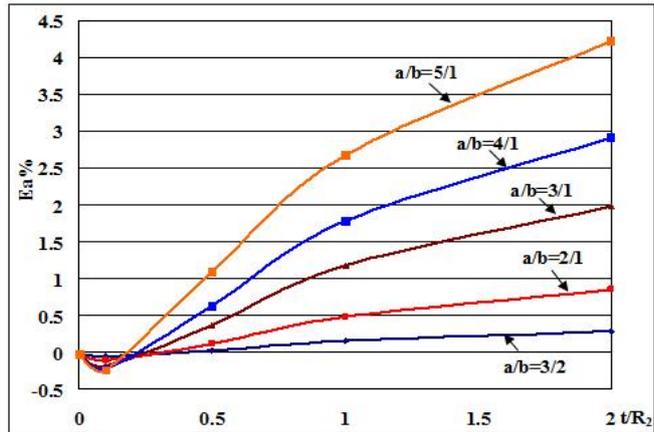
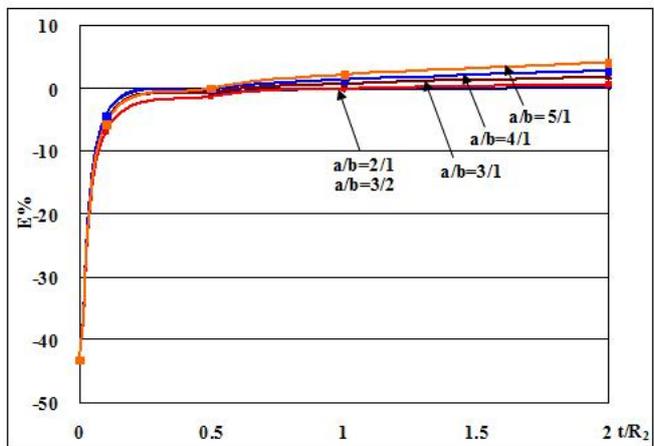


Fig. 3 The error generated by approximate oval perimeter along with long-short-axes ratio a/b

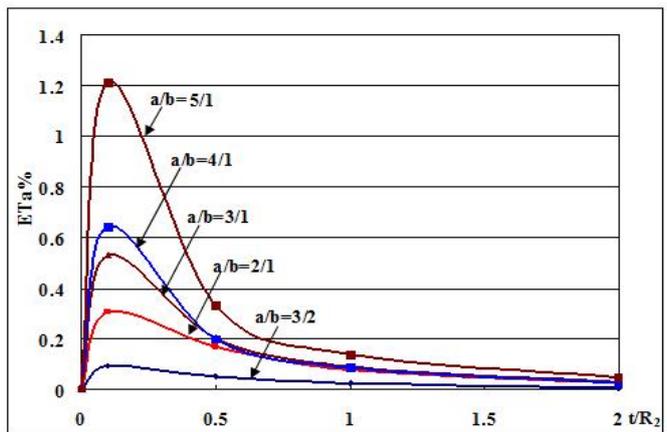


(a) Considering heat radiation

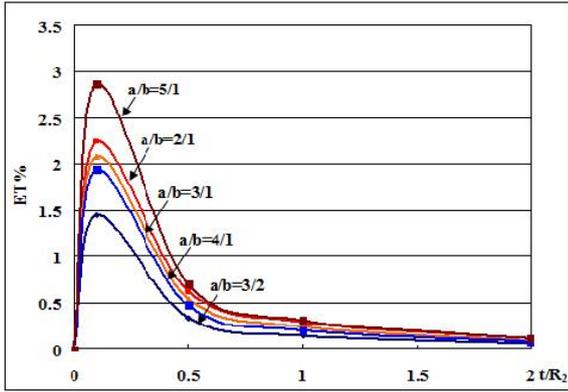


(b) Neglecting heat radiation

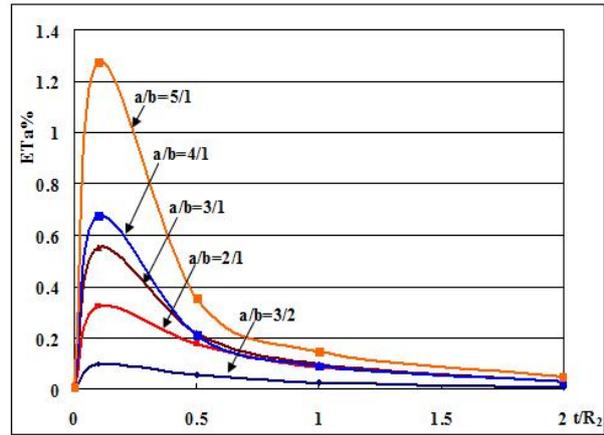
Fig. 4 The heat transfer rate errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $T_o=T_{\text{sur}}=0^\circ\text{C}$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.75$



(a) Considering heat radiation

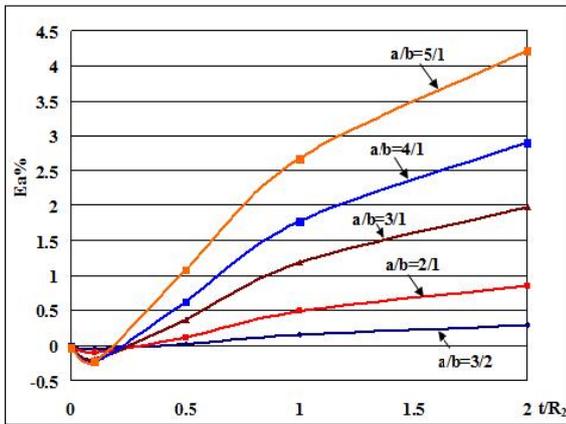


(b) Neglecting heat radiation

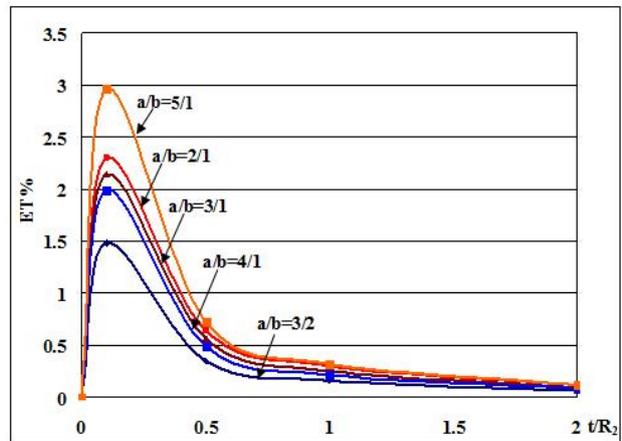


(a) Considering heat radiation

Fig. 5 The surface temperature errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100$, $T_o=T_{\text{sur}}=0$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.75$

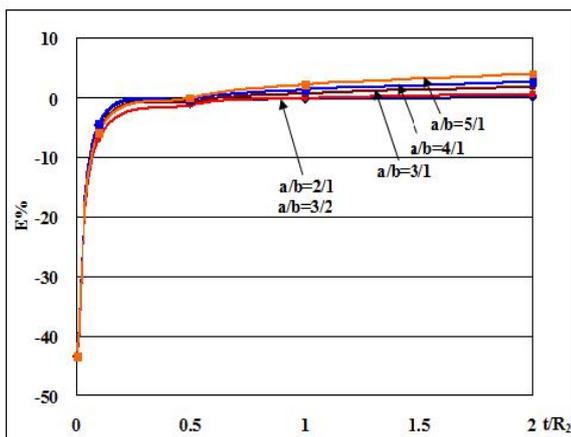


(a) Considering heat radiation



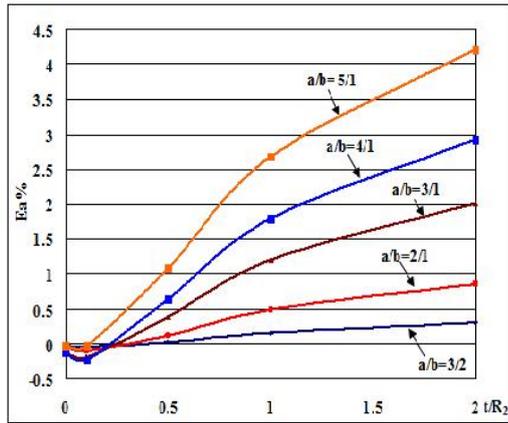
(b) Neglecting heat radiation

Fig. 7 The surface temperature errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $T_o=0^\circ\text{C}$, $T_{\text{sur}}=2^\circ\text{C}$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.75$

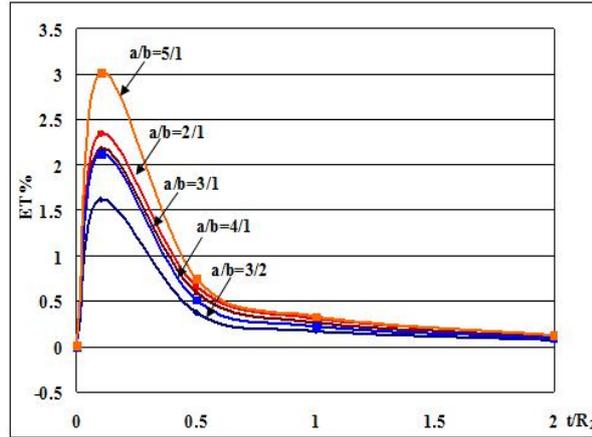


(b) Neglecting heat radiation

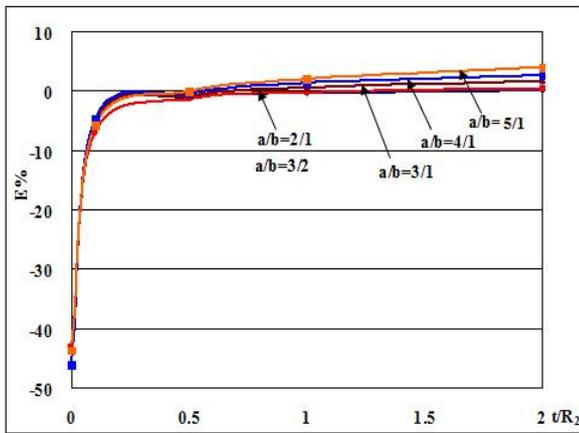
Fig. 6 The heat transfer rate errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $T_o=0^\circ\text{C}$, $T_{\text{sur}}=2^\circ\text{C}$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.75$



(a) Considering heat radiation

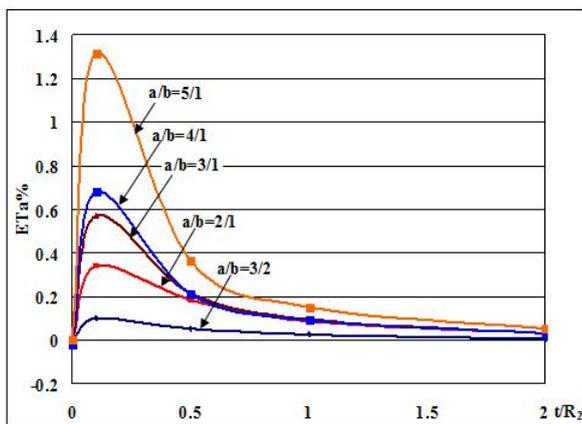


(b) Neglecting heat radiation



(b) Neglecting heat radiation

Fig. 8 The heat transfer rate errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $T_o=0^\circ\text{C}$, $T_{\text{sur}}=5^\circ\text{C}$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.7$



(a) Considering heat radiation

Fig. 9 The surface temperature errors with $h_i=5000\text{Wm}^{-2}\text{K}^{-1}$, $t_1=1\text{mm}$, $K=77\text{Wm}^{-2}\text{K}^{-1}$, $K_s=0.035\text{Wm}^{-2}\text{K}^{-1}$, $h_o=8.3\text{Wm}^{-2}\text{K}^{-1}$, $T_i=100^\circ\text{C}$, $T_o=0^\circ\text{C}$, $T_{\text{sur}}=5^\circ\text{C}$, $\epsilon_0=\epsilon=0.8$ with various a/b and $R_2h_o/K_s=36.56\sim 64.7$

TABLE I
THE EMISSIVITY ϵ OF VARIOUS SUBSTANCES FROM TEXT BOOK EDITED BY HOLMAN [19]

Substance	emissivity
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

TABLE II
REFERRED APPROXIMATE VALUES OF CONVECTION HEAT TRANSFER FROM TEXT BOOK EDITED BY HOLMAN [19]

Approximate convection heat values	
Free convection	h ($\text{W/m}^2\text{-K}$)
DT =30 °C Vertical plate 0.3 in high in air	4.5
DT =30 °C Vertical plate 0.3 in high in air	6.5
Horizontal cylinder, 2 cm diameter, in water	890
Across 1.5 cm vertical air gap with DT =60 °C	2.64
Forced convection	
Air flow at 2 m/s over	12

0.2-m square plate	
Air flow at 35 m/s over 0.75-m square plate	75
Air flowing in 2.5 cm diameter tube at 10 m/s=36km/hr	65
Water at 0.5 kg/s flowing in 2.5 cm diameter tube	3500
Air flow across 5 cm diameter cylinder with velocity of 50 m/s	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

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