THE HEAT EXCHANGE CHARACTERISTICS OF A CIRCULAR DUCT CONSIDERING AND NEGLECTING THE INFLUENCE OF HEAT RADIATION

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ABSTRACT

A heat radiation equation contains 4th exponential order of temperature which makes mathematics analysis very complicated, 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 duct surfaces and surrounding to simplify analysis. Due to this reason, examples shown in most heat transfer, air conditioning and refrigeration text books, and commercial design of heat exchanger, commonly ignore the effect of heat radiation. This paper studies in detail for complete heat transfer characteristics of a circular duct with heat radiation effect taken into account. It is found that, in some practical conditions, the heat radiation effect can not be ignored especially in cases of lower ambient convection heat coefficients and larger surface emissivities, as well as the smaller the duct size or the greater the duct conductivity or internal fluid convection coefficients, even though the temperature-difference between inner and outer fluid-temperatures of duct is very low (such as 1 ℃). In most situations, ignoring heat radiation is likely to produce large errors and degrade the design quality of a heat exchanger. It is also proved that a larger surface emissivity can greatly improve the performance of a heat exchanger.

Keywords：Circular duct, heat exchanger, heat radiation, heat convection, emissivity

Nomenclature

ε= emissivity
A_1= inner surface area of a duct
A_2 = outer surface area of a bare duct
h_i = internal heat convection coefficient
h_o = external heat convection coefficient
h_r = outer radiation heat convection coefficient
HR= percentage of convection heat coefficient ratio
K_A= conductivity of duct
L = length of oval duct
N_u =Nusselt Number, 2r2ho/KA
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 =convection heat transfer rate
q_r =radioactive heat transfer rate
QR =error of heat transfer rate generated by without considering heat radiation
r_1 =inner radius of circular duct
r_2 =outer radius of circular duct
SR=error of surface temperature generated by neglecting heat radiation
TR=absolute temperature ratio between inside and outside fluids of duct
t_1 = thickness of duct
T_2 = surface temperature generated by neglecting heat radiation
T_{2a} = 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 surrounding

Introduction

The heat exchange of cold/hot ducts is commonly encountered in industry and building 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 duct surfaces and surrounding. From this aspect, examples demonstrated in most heat transfer text books [1-8], air conditioning and refrigeration text books [9-12], and in many research papers or even in the practices of most commercial heat exchanger design 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: Moawed [13] completed the experimental investigation of natural convection from vertical and horizontal helicoidal pipes in HVAC applications, Kasayapanand [14] considered Electrode arrangement effect on natural convection, Wang et. al [15] studied the performance of a new gas to gas heat exchanger with strip fin, Bhowmik and Tou [16] finished the experimental study of transient natural convection heat transfer from simulated electronic chips, Desrayaud et. al [17] researched the natural convection air-cooling of a substrate-mounted protruding heat source in a stack of parallel boards,. Kasayapanand [18] did the numerical modeling of natural convection in partially open square cavities under electric field, and Stickland et. al [19] completed an experimental investigation of natural convection with solidification in a differentially heated cavity.

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

Problem Formulation

Fig. 1 shows that a circular duct with duct thickness ***t***, outer and inner radii $\boldsymbol{r_2}$ and $\boldsymbol{r_1}$, duct length ***L***, internal and external fluids with convection heat transfer coefficients $\boldsymbol{h_i}$ and $\boldsymbol{h_o}$ and internal and external fluid-temperatures $\boldsymbol{T_i}$ and $\boldsymbol{T_o}$, respectively, wall conductivity $\boldsymbol{K_A}$ and with surface emissivity $\boldsymbol{\varepsilon}$, exposed to the outside surrounding temperature $\boldsymbol{T_{sur}}$.

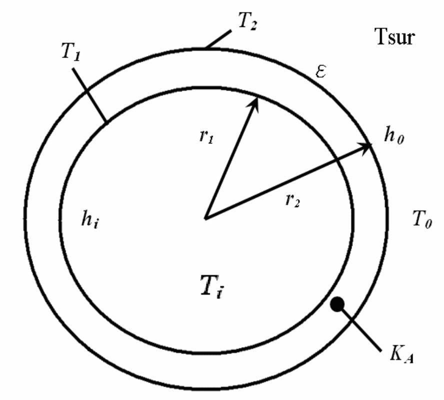

Fig. 1、The circular duct and its relative parameters

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 heat transfer rate of a circular duct can be written as:

$$q=\frac{T_i-T_0}{\frac{1}{h_i 2\pi r_1 L}+\frac{\ell_n \frac{r_2}{r_1}}{2\pi K_A L}+\frac{1}{h_0 2\pi r_2 L}}=\frac{T_2-T_0}{\frac{1}{h_0 2\pi r_2 L}} \tag{1}$$

That is, heat transfer rate ***q*** and surface temperature $\boldsymbol{T_2}$ for situations without considering the influence of outside surface heat radiation can be obtained from equation (1).

Cases with the influence of heat radiation being considered

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

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

where $\boldsymbol{T_{2a}}$ is the actual surface temperature in situations of considering the influence of heat radiation. The surface convection heat transfer rate is:

$$q_c=h_0 2\pi r_2 L\left(T_{2a}-T_0\right) \tag{3}$$

The surface radiation heat transfer rate is:

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

The energy balance reads:

$$q_a=q_c+q_r \tag{5}$$

Thus, the total heat transfer rate $\boldsymbol{q_a}$ and actual surface temperature $\boldsymbol{T_{2a}}$ for situations considering the influence of external surface heat radiation can be obtained from equations (2)~(5).

Comparing Eqs. (1) and (3), we obtain

If $\boldsymbol{\varepsilon\neq 0}$, then $\boldsymbol{T_{2a}\neq T_2}$, then $\boldsymbol{q_c \neq q = h_o 2\pi r_2 L(T_2-T_o)}$.

The outer radiation heat convection coefficient can be derived from Eq. (4):

$$h_r=\sigma\varepsilon\left(T_{2a}^2+T_{sur}^2\right)\left(T_{2a}+T_{sur}\right) \tag{6}$$

where $\boldsymbol{h_r}$ is conventionally used to compare with the heat convection coefficient $\boldsymbol{h_o}$ and show the significant effect of radiation. The error of heat transfer rate generated by neglecting heat radiation effect can then be defined as:

$$QR=\left(1-\frac{q}{q_a}\right)\times 100\% \tag{7}$$

The ratio between the radiation and convection coefficients is defined as:

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

(8)

Here, ***HR*** also is equivalent to $\boldsymbol{q_r/q_c}\times 100\%$ under the condition $\boldsymbol{T_o=T_{sur}}$. A high value of ***HR*** signifies more importance the influence of heat radiation.

For a circular duct, Nusselt number is defined as:

$$N_u = \frac{h_0 \left(2r_2\right)}{K_A} \quad (9)$$

The ratio between the absolute temperatures of inside and outside fluids of duct is defined as:

$$TR = \frac{T_i}{T_0} \quad (10)$$

The error of surface temperature produced by neglecting heat radiation effect is defined as:

$$SR = (1 - \frac{T_2}{T_{2a}}) \times 100\% \quad (11)$$

Numerical Heat Transfer Results

There exists a one-dimensional exact solution for the energy equation of a circular duct. In this paper, the exact solution is obtained by a one-dimensional LabVIEW programming. According to the emissivities shown in Table 1, ε=0.8 or 0.9 and ε=0.1 or 0.2 are adopted to represent the high and low surface emissivity cases, respectively. 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. Thus, a convection coefficient of 30 W/m^2-K can represent the convective heat-transfer effect from medium wind speed. Therefore, ***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, ***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 shown in Table 3, is used as the duct material in this study.

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

(1).Let surface emissivity ε=0, check for the resulted QR being zero.

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

Table 1、The emissivities ε of various substances from the manual of infrared temperature demonstrator [20]

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 0.03mm=0.27 film 0.13mm=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/m2-K)	
Mode	h (W/m^2-K)
Natural 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	12

plate	
Air flow at 35 m/s over 0.75-m square plate	75
Forced convection	h (W/m^2-K)
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 [3]

Thermal conductivity of various materials at 20 ℃	
Material	K (W/m-K)
Metals	
Copper (pure)	386
Aluminum (pure)	204
Carbon steel, l % 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

Results and Discussions

For the current results to be applicable to more general cases, all the results are shown in dimensionless parameters. At first, we examine the cases with conditions of K_A=77W/m-K, r_1 =195mm, r_2 =200mm and fixed T_i =573K with varying T_o=T_{sur} from 286.5K to 572K (**TR**=T_i / T_o=1.0017 to 2), and the results are shown in Figs. 2~4. Figs. 2 and 3 show that the heat transfer rate errors **QR** and percentage of convection coefficients ratio **HR** are strongly affected by the absolute temperature ratio **TR**, external convection coefficient h_o, the surface emisivity ε and internal convection coefficient h_i. As **TR** increases, **QR** and **HR** decrease. Conditions of smaller values of **TR**, ho and higher values of ε and hi tend to result in higher values of **QR** and **HR** (a high value of **HR** signifies more importance the influence of heat radiation).

It is surprising that even in situations of very small **TR** (for example, **TR** =1.0017 from T_i =573 K and T_o=T_{sur} =572 K), the resulted **QR** and **HR** are quite large. We first suspected that this might be caused by the high level of T_i =573 K (=300 ℃) which in turn gives rise to higher hr and q_r. Therefore, investigate the second set of cases with conditions of K_A=77 W/m-K, r_1 =195mm, r_2 =200mm and fixed T_o=T_{sur} =300 K with varying T_i from 301 K to 600 K as shown in Figs. 5~7.

Figs. 5 and 6 also show that **QR** and **HR** are strongly affected by **TR**, h_o, ε and h_i. Comparing with the previous set of cases, the main difference is that **QR** and **HR** increase as **TR** increases, and smaller ho and higher **TR**, ε and hi tend to result in higher **QR** and **HR**. Even in the situations of smaller **TR** (for example, **TR**= 1.0033 from T_i=301K and T_o=T_{sur}=300K), the resulted values of **QR** and **HR** are still quite high.

Figs. 4 and 7 show that the errors of surface temperature **SR**, are strongly affected by **TR**, ε, ho and h_i. The negative value of **SR** is caused by T_{2a} less than T_2. The greater the **TR** or ε or h_i, or the smaller the h_o is, the greater the absolute value of **SR** will be.

The third set of cases are with conditions of K_A=1.7W/m-K, r_1=165mm, r_2=170mm, T_i=100℃ and T_o=T_{sur}=30℃, and the results are shown in Figs. 8~10. Figs. 8~10 show that **QR**, **HR** and **SQ** are strongly affected by the Nusselt number **Nu**, *ε* and hi. Here, **QR** and **HR** decrease but **SR** increases as Nu increases. The smaller the N_u ($N_u=2r_2h_o/K_A$, ie., the smaller the r_2 or h_o or the greater the K_A), the higher the *ε* and the smaller the hi are, the higher the **QR**, **HR** and the absolute value of **SR** become. Fig. 10 shows that the absolute value of **SR** become smaller in situations of h_i=5000 W/m^2-K, because higher h_i causes smaller internal convection thermal resistance.

In order to demonstrate the main differences between the heat transfer characteristics of situations with and without considering heat radiation, the detail data of cases with h_o=10 W/m^2-K, h_i=30 W/m^2-K, K_A=77W/m-K, r_1=195mm, r_2=200mm, T_o=T_{sur} =300K with T_i=301K~600K are listed in Table 4. It can be found in Table 4 that the values of qc are different from those of q. For example, the data for **TR**=1.1 and *ε*=0.9 in Table4(a) show that q_a=390.65W/m (>q=270.81W/m) and q_r=147.47W/m, q_c(=243.18W/m)≠q(=270.81W/m),

$\boldsymbol{T_{2a}}$(=46.20℃)<$\boldsymbol{T_2}$(=49.20℃). The error ***QR***=28.12%, ***HR***=60.64% and ***SR***=-6.48% are quite large. This can be explained from equations (1) and (3) that if $\varepsilon \neq 0$, $T_{2a} \neq T_2$, qc will not equal to q (ε=0). As a result, even in a very small temperature difference of T_i=301K and T_o=T_{sur} =300K (***TR***=1.0033 and ε=0.9 in Table 4(a)), q_a=12.83W/m(>***q***=9.36W/m)andq_r=4.53W/m,q_c(=8.30 W/m)≠***q***(=9.36W/m).The error ***QR***=24.69% and ***HR***=55.26% are quite large. 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 can not be ignored, especially in situations of lower ho and higher ε.

Table 4 Demonstrated example of h_o=10 W/m2-K, h_i=30 W/m2-K, K_A=77W/m-K, t_1=5mm, r_2=200mm, T_o=Tsur =300K with T_i=301K~600K

(a) ε=0.9

TR	qa w/m	qr w/m	qc w/m	q w/m	QR %	HR %	T_{2a} ℃	T_2 ℃	SR %
1.0033	12.83	4.53	8.30	9.36	26.49	55.26	27.50	27.59	-0.33
1.1	390.65	147.47	243.18	270.81	28.12	60.64	46.20	49.20	-6.48
1.2	799.51	319.39	480.12	561.61	29.76	66.52	65.01	71.54	-9.97
1.3	1227.23	516.63	710.60	842.41	31.36	72.70	83.40	93.89	-12.58
1.4	1674.19	739.70	934.49	1123.22	32.91	79.15	101.22	116.23	-14.84
1.5	2140.56	988.82	1151.74	1404.02	34.41	85.85	118.50	138.57	-16.94
1.6	2626.31	1263.97	1362.34	1684.83	35.85	92.78	135.26	160.92	-18.97
1.7	3131.24	1564.87	1556.37	1965.63	37.23	99.90	151.50	183.27	-20.97
1.8	3655.03	1891.08	1763.95	2246.44	38.54	107.21	167.22	205.62	-22.96
1.9	4197.20	2241.98	1955.22	2527.24	39.79	114.67	182.44	227.96	-24.95
2.0	4757.24	2616.87	2140.37	2808.05	40.97	122.26	197.18	250.31	-26.95

(b) ε=0.1

TR	qa w/m	qr w/m	qc w/m	q w/m	QR %	HR %	T_{2a} ℃	T_2 ℃	SR %
1.0033	9.79	0.57	9.22	9.36	4.30	6.14	27.58	27.60	-0.04
1.1	294.82	18.83	275.99	280.80	4.76	6.83	48.81	49.20	-0.78
1.2	592.78	41.85	550.93	561.61	5.26	7.60	70.69	71.54	-1.20
1.3	894.24	69.58	824.66	842.41	5.80	8.44	92.47	93.89	-1.53
1.4	1199.61	102.55	1097.06	1123.22	6.37	9.35	114.15	116.23	-1.82
1.5	1509.28	141.31	1367.97	1404.02	6.97	10.33	135.71	138.58	-2.11
1.6	1823.69	186.43	1637.26	1684.83	7.61	11.39	157.14	160.92	-2.41
1.7	2143.24	238.44	1904.80	1965.63	8.29	12.52	178.43	183.27	-2.71
1.8	2468.34	297.91	2170.43	2246.44	8.99	13.73	199.57	205.62	-3.03
1.9	2799.38	365.35	2434.03	2527.24	9.27	15.01	220.54	227.96	-3.36
2.0	3136.73	441.29	2695.46	2808.04	10.48	16.37	241.35	250.31	-3.71

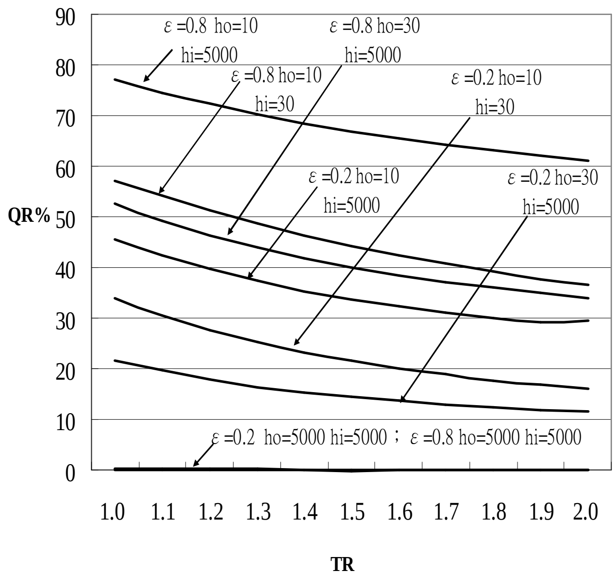

Fig. 2 The relations between QR and TR in situation of K_A=77W/m-K , r_1=195mm, r_2=200mm, T_i=573K with T_o=T_{sur} =286.5K~572K

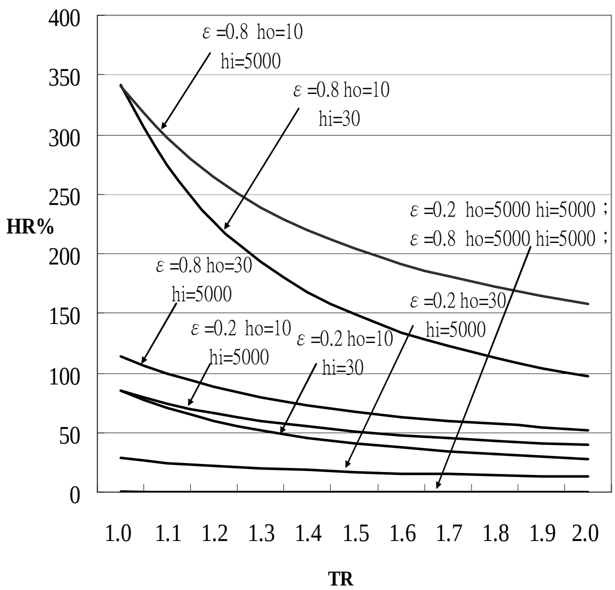

Fig.3 The relations between HR and TR in situation of in situation of K_A=77W/m-K , r_1=190mm, r_2=200mm, T_i=573K with T_o=T_{sur} =286.5K~572K

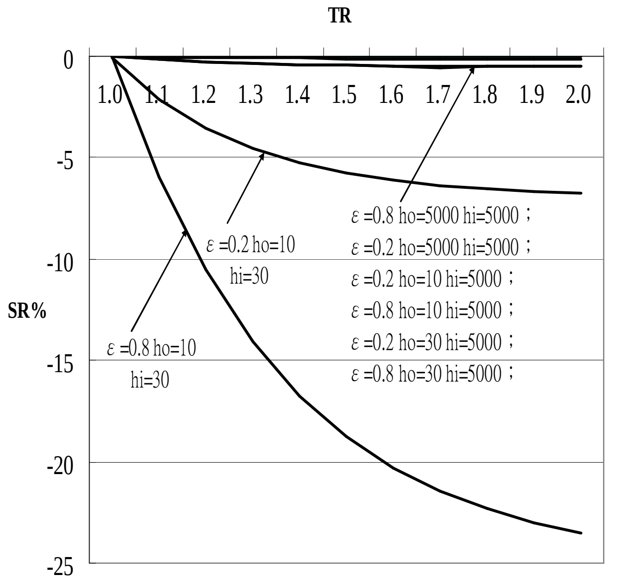

Fig. 4 The relations between SR and TR in situation of K_A=77W/m-K, r_1=195mm, r_2=200mm, T_i=573K with $T_o=T_{sur}$ =286.5K~572 K

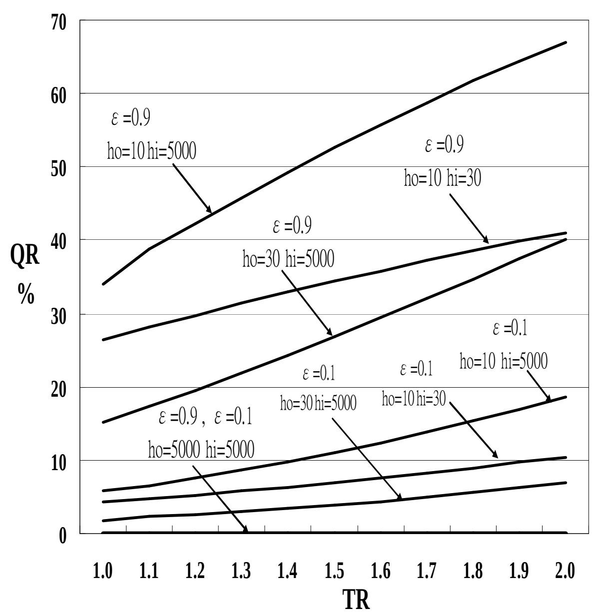

Fig. 5 The relations between QR and TR in situation of K_A=77W/m-K, r_1=195mm, r_2=200mm, $T_o=T_{sur}$ =300K with T_i=301K~600K

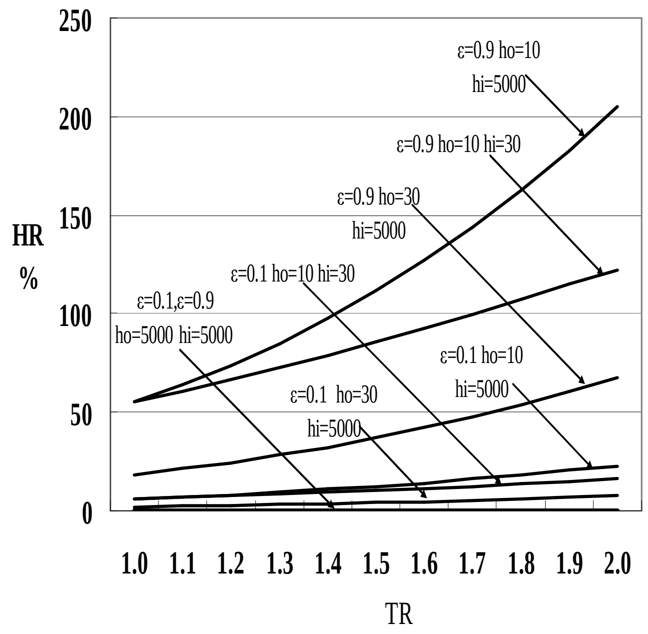

Fig. 6 The relations between HR and TR in situation of K_A=77W/m-K ,r_1=195mm, r_2=200mm, T_o=Tsur =300K with T_i=301K~600K

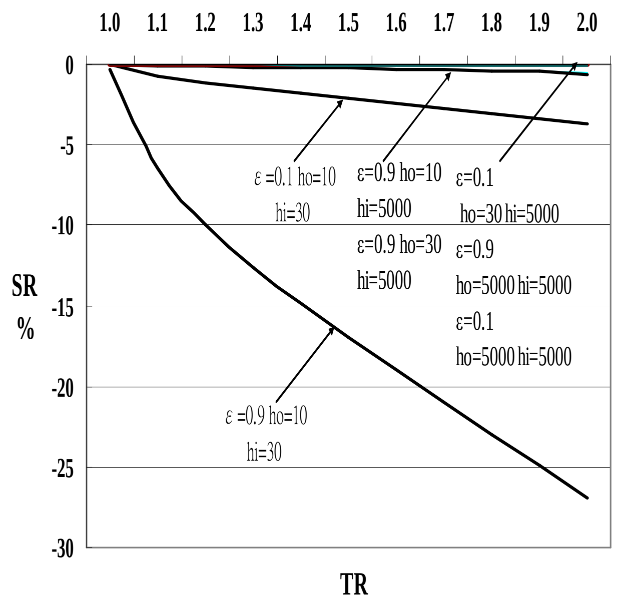

Fig. 7 The relations between SR and TR in situation of K_A=77W/m-K, r_1=195mm, r_2=200mm, $T_o=T_{sur}$ =300K with T_i=301K~600K

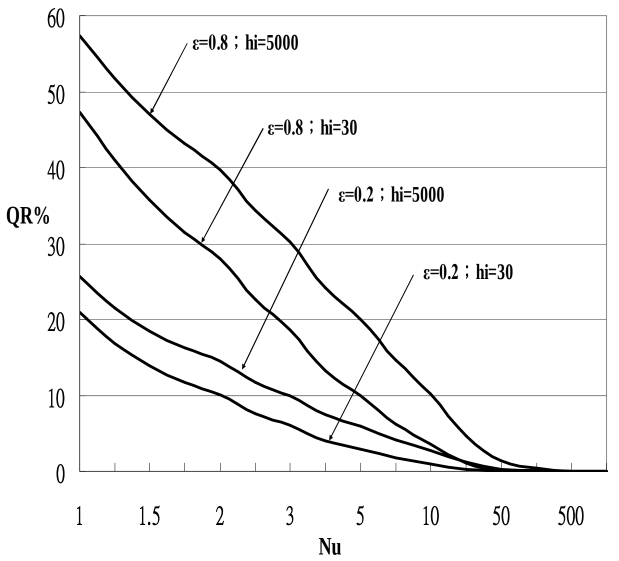

Fig. 8 The relations between QR and N_u in situation of K_A=1.7W/m-K , r_1=165mm, r_2=170mm; T_i=100℃, T_o=T_{sur} =30℃

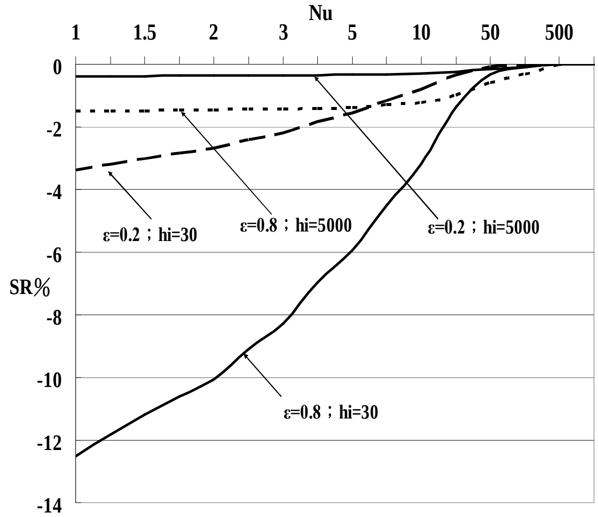

Fig. 10 The relations between SR and Nu in situation of K_A=1.7W/m-K , r_1=165mm, r_2=170mm; T_i=100℃, T_o=T_{sur} =30℃

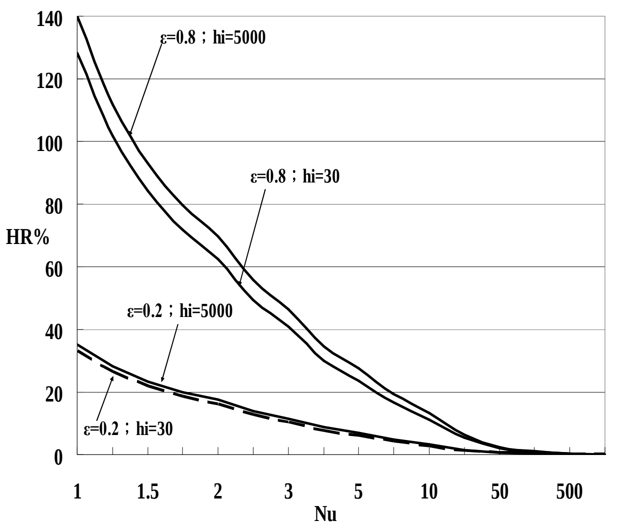

Fig. 9 The relations between HR and Nu in situation of K_A=1.7W/m-K , r_1=165mm, r_2=170mm; T_i=100℃, T_o=T_{sur} =30℃

Conclusion

From the results of this study, it is evident that even the temperature difference between internal and external fluids of circular duct is very small, neglecting the influence of heat radiation, especially in situations of lower external ambient air convection coefficients and greater surface emissivity, as well as the smaller the duct size or the greater the duct conductivity or internal fluid convection coefficients, tends to result in inaccurate solutions. It is also demonstrated that a greater surface emissivity can greatly improve the performance of a heat exchanger. Since incorporating the effect of heat radiation into computer software, such as LabVIEW, used in the present study, only requires little effort, heat radiation effect should be taken into account in any study concerning heat transfer of a duct.

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