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The log mean heat transfer rate method of heat exchanger considering the influence of heat radiation

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ABSTRACT

The log mean temperature difference (LMTD) method is conventionally used to calculate the total heat transfer rate of heat exchangers. Because the heat radiation equation contains the 4th order exponential of temperature which is very complicate in calculations, thus LMTD method neglects the influence of heat radiation. From the recent investigation of a circular duct in some practical situations, it is found that even in the situation of the temperature difference between outer duct surface and surrounding is low to 1 °C, the heat radiation effect can not be ignored in the situations of lower ambient convective heat coefficient and greater surface emissivities. In this investigation, the log mean heat transfer rate (LMHTR) method which considering the influence of heat radiation, is developed to calculate the total heat transfer rate of heat exchangers.

1. Introduction

Heat exchangers are widely applied to the industries and living surroundings. The log mean temperature difference (LMTD) method which introduced in most heat transfer text books [1–8] as well as air conditioning and refrigeration text books [9–12], is conventionally used to calculate the total heat transfer rate of heat exchangers. Because the heat radiation equation contains the 4th order exponential of temperature which is very troublesome in calculations, thus LMTD method neglects the influence of heat radiation. Recently, Hsien et al. [13] studied about the complete heat transfer characteristics of a circular duct considering the heat radiation effect while applying to heat exchanging. From the simulations in some practical situations, it is found that the heat radiation effect can not be ignored in situations of lower ambient convection heat coefficients and greater surface emissivities; even in situations of temperature difference between inner fluid and surrounding ambient air low to 1 °C, the errors generated by neglecting the heat radiation are still very big and unacceptable. In most situations, ignoring the heat radiation will generate big errors and affect the design quality of heat exchanger. Hsien et al. [13] also proved that using greater surface emissivity can greatly improve the performance of heat exchanger.

In this present investigation, the log mean heat transfer rate (LMHTR) method which considering the influence of heat radiation, is developed to calculate the total heat transfer rate of heat exchangers.

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2. Problem formulation

Fig. 1 shows that an circular duct with inner radius r_1, outer radius r_2, duct thickness t_1, duct length L, wall conductivity K_A, surface emissivity ε, is exposed to internal and external fluids with convection heat transfer coefficients h_{i1} and h_{o1} as well as temperatures T_{i1} and T_{o1} at entrance section of the duct, respectively; and it is exposed to internal and external fluids with convection heat transfer coefficients h_{i2} and h_{o2} as well as temperatures T_{i2} and T_{o2} at exit section of the duct, respectively.

2.1. LMTD method neglecting the influence of heat radiation

While the influence of outside surface heat radiation is not considered, the log mean temperature difference (LMTD) method [1–12] is conventionally used to calculate the total heat transfer rate of heat exchangers. From the relative temperatures of entrance and exit sections as shown in Fig. 1, LMTD can be expressed as:

$$\mathrm{LMTD} = \frac{(T_{i1} - T_{o1}) - (T_{i2} - T_{o2})}{\ln\frac{(T_{i1} - T_{o1})}{(T_{i2} - T_{o2})}} \tag{1}$$

The total thermal resistance of the circular duct shown in Fig. 1 can be written as:

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

Nomenclature

ε	emissivity
A_1	inner surface area of a duct
A_2	outer surface area of a bare duct
h_{i1}	inner heat convection coefficient at entrance section
h_{i2}	inner heat convection coefficient at exit section
h_{o1}	outer heat convection coefficient at entrance section
h_{o2}	outer heat convection coefficient at exit section
K_A	conductivity of duct
L	duct length
Q	total heat transfer rate without considering heat radiation
Q_a	total heat transfer rate considering heat radiation
q_1	unit length heat transfer rate neglecting heat radiation at entrance section
q_2	unit length heat transfer rate neglecting heat radiation at exit section
q_{a1}	unit length heat transfer rate considering heat radiation at entrance section
q_{a2}	unit length heat transfer rate considering heat radiation at exit section
q_{c1}	unit length convective heat transfer rate at entrance section
q_{c2}	unit length convective heat transfer rate at exit section
q_{r1}	unit length radioactive heat transfer rate at entrance section
q_{r2}	unit length radioactive heat transfer rate at exit section
QR	error of heat transfer rate generated by neglecting heat radiation
r_1	inner radius of circular duct
r_2	outer radius of circular duct
t_1	thickness of duct
T_{21}	the average surface temperature at the entrance section in situation of considering heat radiation
T_{22}	the average surface temperature at the exit section in situation of considering heat radiation
T_{i1}	temperature of the fluid inside the duct at entrance section
T_{i2}	temperature of the fluid inside the duct at exit section
T_{o1}	temperature of the fluid outside the duct at entrance section
T_{o2}	temperature of the fluid outside the duct at exit section
T_{s1}	the average surface temperature at the entrance section in situation of neglecting heat radiation
T_{s2}	the average surface temperature at the exit section in situation of neglecting heat radiation
T_{sur}	surrounding temperature
TR_1	error of average surface temperature at the entrance section generated by neglecting heat radiation
TR_2	error of average surface temperature at the exit section generated by neglecting heat radiation

The total heat transfer rate of the long circular duct neglecting the heat radiation by LMTD method is:

$$Q = \frac{\text{LMTD}}{R_{th}} \tag{3}$$

The unit length heat transfer rate, q_1, at the entrance section is:

$$q_1 = \frac{T_{i1} - T_{o1}}{R_{th}L} = \frac{T_{S1} - T_{o1}}{\frac{1}{h_o 2\pi r_2}} \tag{4}$$

The unit length heat transfer rate, q_2, at the exit section is:

$$q_2 = \frac{T_{i2} - T_{o2}}{R_{th}L} = \frac{T_{S2} - T_{o2}}{\frac{1}{h_o 2\pi r_2}}, \tag{5}$$

The values of total heat transfer rate Q, the average surface temperature at the entrance section T_{s1}, the average surface temperature at the exit section T_{s2}, can obtained from Eqs. (1)–(5) under the given values of h_i, ($h_i = h_{i1} = h_{i2}$), h_o($h_o = h_{o1} = h_{o2}$), r_1, r_2, K_A, L, T_{i1},T_{o1}, T_{i2}, T_{o2} and L.

2.2. Situations considering the influence of heat radiation

While the influence of outside surface heat radiation is considered, the complete unit length heat transfer rate at the entrance section is:

$$q_{a1} = \frac{T_{i1} - T_{o_1}}{\frac{1}{h_{i1}2\pi r_1} + \frac{\ln\frac{r_2}{r_1}}{2\pi K_A}} \tag{6}$$

The unit length surface convective heat transfer rate at the entrance section is:

$$q_{c1} = h_{o1}2\pi r_2(T_{21} - T_{o_1}) \tag{7}$$

The unit length surface radiation heat transfer rate at the entrance section is:

$$q_{r1} = \sigma\varepsilon 2\pi r_2(T_{21}^4 - T_{sur}^4) \tag{8}$$

The following equation is obtained from heat balance at the entrance section:

$$q_{a1} = q_{c1} + q_{r1} \tag{9}$$

The values of q_{a1}, q_{r1}, q_{c1} and T_{21} can obtained from Eqs. (6)–(9) under the given values of h_{i1}, h_{o1}, r_1, r_2, K_A, L, T_{i1},T_{o1}, ε and T_{sur}.

Similarly, the complete unit length heat transfer rate at the exit section is:

$$q_{a2} = \frac{T_{i2} - T_{o_2}}{\frac{1}{h_{i2}2\pi r_1} + \frac{\ln\frac{r_2}{r_1}}{2\pi K_A}} \tag{10}$$

The unit length surface convective heat transfer rate at the exit section is:

$$q_{c2} = h_{o2}2\pi r_2(T_{2_2} - T_{o_2}) \tag{11}$$

The unit length surface radiation heat transfer rate at the exit section is:

$$q_{r2} = \sigma\varepsilon 2\pi r_2(T_{22}^4 - T_{sur}^4) \tag{12}$$

The following equation is obtained from heat balance at the exit section:

$$q_{a2} = q_{c2} + q_{r2} \tag{13}$$

The values of q_{a2}, q_{r2}, q_{c2} and T_{22} can obtained from Eqs. (10)–(14) under the given values of h_{i2}, h_{o2}, r_1, r_2, K_A, L, T_{i2}, T_{o2}, ε and T_{sur}.

The total heat transfer rate of the long circular duct considering the heat radiation by log mean heat transfer rate (LMHTR) method is:

$$Q_a = \frac{q_{a1} - q_{a_2}}{\ln\frac{q_{a1}}{q_{a2}}}L \tag{14}$$

The above LMHTR method (considering heat radiation) under the same concept as LMTD method (neglecting heat radiation) is developed in this study. While the heat radiation is not considered, assume the temperatures T_{i1} and T_{o1} keep constant at the entrance

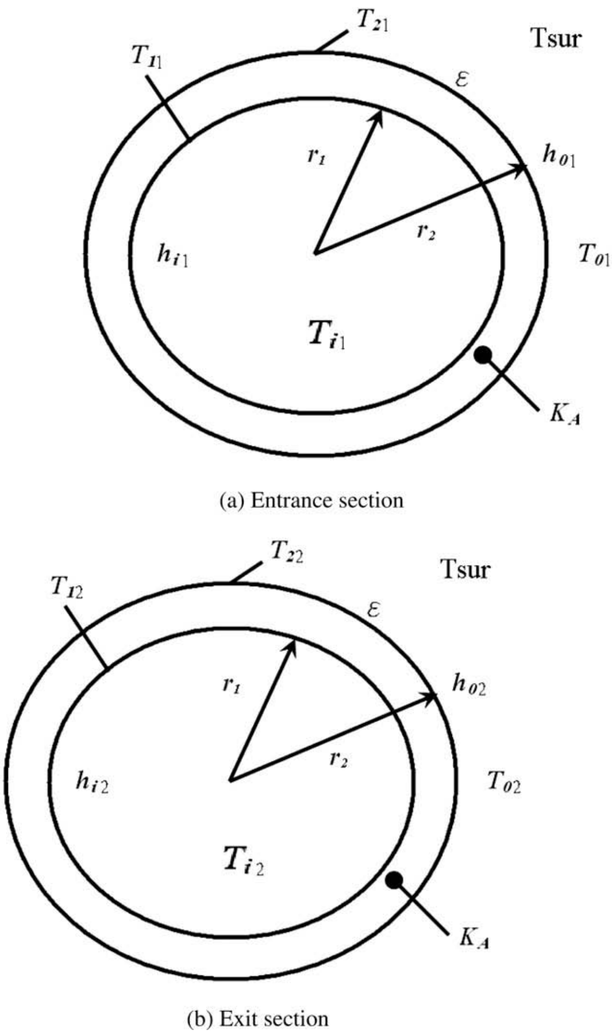

Fig. 1. A long circular heat exchanger and its relative parameters.

section, and T_{i2} and T_{o2} keep constant at the exit section, then the following relations can be obtained in the situation of neglecting heat radiation (or $\varepsilon = 0$):

$$q_{a1} = \frac{(T_{i1} - T_{o1})}{R_{th}L} \tag{15}$$

$$q_{a2} = \frac{(T_{i2} - T_{o2})}{R_{th}L} \tag{16}$$

Then the following relation between LMTD method and LMHTR method in the situation of neglecting heat radiation (or $\varepsilon = 0$) can be proven:

$$Q_a = \frac{q_{a1} - q_{a2}}{\ln\frac{q_{a1}}{q_{a2}}} L = \frac{\frac{(T_{i1}-T_{o1})}{R_{th}L} - \frac{(T_{i2}-T_{o2})}{R_{th}L}}{\ln\frac{\frac{(T_{i1}-T_{o1})}{R_{th}L}}{\frac{(T_{i2}-T_{o2})}{R_{th}L}}} L = \frac{\frac{(T_{i1}-T_{o1})-(T_{i2}-T_{o2})}{\ln\frac{(T_{i1}-T_{o1})}{(T_{i2}-T_{o2})}}}{R_{th}} = \frac{\text{LMTD}}{R_{th}} \tag{17}$$

It can be proved from Eq. (17) that the results obtained from LMTD method and LMHTR method in the situation of neglecting heat radiation (or $\varepsilon = 0$) are the same value.

The error of total heat transfer rate generated by neglecting heat radiation effect (while $\varepsilon \neq 0$) is defined as:

$$QR = \left(\frac{Q - Q_a}{Q_a}\right) \times 100\% \tag{18}$$

Table 1
Referred approximate values of thermal conductivities [1].

Thermal conductivity of various materials at 20 °C	
Metals	
Material	K (W/m K)
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

Table 2
The emissivities ε of various substances from the manual of infrared temperature demonstrator [13].

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

The error of average surface temperature at the entrance section generated by neglecting heat radiation is:

$$TR_1 = \left(\frac{T_{S_1} - T_{21}}{T_{21}}\right) \times 100\% \tag{19}$$

Table 3
Referred approximate values of convection heat transfer [1].

Approximate values of convection heat transfer, h (W/m^2 K)	
Mode	h (W/m^2 K)
Free convection, DT = 30 °C horizontal plate 0.3 in high in air	4.5
Free convection, DT = 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 DT = 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 flowing 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

The error of average surface temperature at the exit section generated by neglecting heat radiation is:

$$TR_2 = \left(\frac{T_{S2} - T_{22}}{T_{22}}\right) \times 100\% \qquad (20)$$

3. The reliability of the numerical results

The nature of results of heat equation for a circular duct is one-dimension exact solution. The exact numerical heat transfer results of a circular duct can be obtained by any one-dimensional computer code (such as LabVIEW programming in this study). The computer aid results are obtained within one second computing time by a common PC.

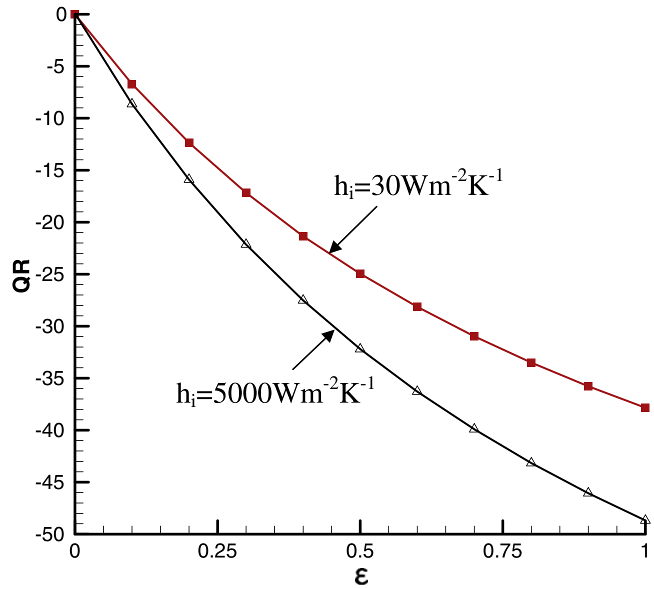

Fig. 2. The relations between heat transfer rate error, QR, and emissivity, ε, for a heater.

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

(1) Let surface emissivity $\varepsilon = 0$, make sure if the results of $QR = 0$.
(2) Let surface emissivity $\varepsilon = 1$, outer convection coefficient $h_{o1} = h_{o2} = 100{,}000\ \mathrm{Wm^{-2}\,K^{-1}}$, make sure if the results of QR close to zero.

4. Results and discussions

The following practical examples is use to demonstrate what the differences and relationships of the results between the LMTD method and LMHTR method are:

Table 4
A heater constructed by a circular duct with $K_A = 77\ \mathrm{Wm^{-1}\,K^{-1}}$, $L = 10$ m, $r_1 = 198$ mm, $r_2 = 200$ mm, $T_{i1} = 65$ °C, $T_{i2} = 60$ °C, $T_{o1} = 22$ °C and $h_{o2} = h_{o1} = 8\ \mathrm{Wm^{-2}\,K^{-1}}$; $T_{o2} = 24$ °C and $T_{sur} = 20$ °C.

ε	Q (Watt)	Q_a (Watt)	QR (%)	T_{S1} (°C)	T_{21} (°C)	TR_1 (%)	T_{S2} (°C)	T_{22} (°C)	TR_2 (%)
(a) $h_{i2} = h_{i1} = 30\ Wm^{-2}\,K^{-1}$									
0.0	3119.6	3119.6	0.0	55.9	55.9	0.0	52.4	52.4	0.0
0.1	3119.6	3343.9	−6.7	55.9	55.2	1.2	52.4	51.8	1.1
0.2	3119.6	3559.2	−12.4	55.9	54.6	2.3	52.4	51.2	2.2
0.3	3119.6	3766.1	−17.2	55.9	54.0	3.4	52.4	50.7	3.2
0.4	3119.6	3965.1	−21.4	55.9	53.5	4.5	52.4	50.2	4.3
0.5	3119.6	4156.8	−24.9	55.9	52. 9	5.6	52.4	49.7	5.3
0.6	3119.6	4341.5	−28.1	55.9	52.4	6.6	52.4	49.3	6.3
0.7	3119.6	4519.6	−31.0	55.9	51.9	7.7	52.4	48.8	7.3
0.8	3119.6	4691.6	−33.5	55.9	51.4	8.7	52.4	48.4	8.2
0.9	3119.6	4857.9	−35.8	55.9	50.9	9.7	52.4	48.0	9.2
1.0	3119.6	5018.6	−37.8	55.9	50.5	10.7	52.4	47.6	10.1
(b) $h_{i2} = h_{i1} = 5000\ Wm^{-2}\,K^{-1}$									
0.0	3953.3	3953.4	0.0	64.922	64.922	0.000	59.935	59.935	0.000
0.1	3953.3	4327.1	−8.6	64.922	64.914	0.011	59.935	59.928	0.011
0.2	3953.3	4701.0	−15.9	64.922	64.907	0.022	59.935	59.922	0.021
0.3	3953.3	5077.8	−22.1	64.922	64.900	0.034	59.935	59.915	0.032
0.4	3953.3	5454.4	−27.5	64.922	64.892	0.045	59.935	59.909	0.042
0.5	3953.3	5831.2	−32.2	64.922	64.885	0.056	59.935	59.903	0.053
0.6	3953.3	6205.1	−36.3	64.922	64.878	0.068	59.935	59.896	0.064
0.7	3953.3	6579.0	−39.9	64.922	64.871	0.079	59.935	59.890	0.074
0.8	3953.3	6955.9	−43.2	64.922	64.856	0.090	59.935	59.884	0.085
0.9	3953.3	7329.8	−46.1	64.922	64.863	0.101	59.935	59.877	0.095
1.0	3953.3	7703.7	−48.7	64.922	64.849	0.113	59.935	59.871	0.106

Table 5
A chiller constructed by a circular duct with K_A = 77 Wm^{-1} K^{-1}, L = 10 m, r_1 = 198 mm, r_2 = 200 mm, T_{i1} = 7 °C, T_{i2} = 12 °C, T_{o1} = 24 °C and h_{o2} = h_{o1} = 8 Wm^{-2} K^{-1}; T_{o2} = 26 °C and T_{sur} = 30 °C.

ε	Q (Watt)	Q_a (Watt)	QR (%)	T_{S1} (°C)	T_{21} (°C)	TR_1 (%)	T_{S2} (°C)	T_{22} (°C)	TR_2 (%)
(a) h_{i2} = h_{i1} = 30 Wm^{-2} K^{-1}									
0.0	−1223.5	−1223.5	0.0	10.6	10.6	0.0	14.9	14.9	0.0
0.1	−1223.5	−1320.4	−7.3	10.6	10.9	−2.67	14.9	15.2	−1.5
0.2	−1223.5	−1414.5	−13.5	10.6	11.2	−5.14	14.9	15.4	−2.9
0.3	−1223.5	−1506.1	−18.8	10.6	11.5	−7.42	14.9	15.6	−4.3
0.4	−1223.5	−1595.3	−23.3	10.6	11.7	−9.54	14.9	15.9	−5.6
0.5	−1223.5	−1682.0	−27.3	10.6	12.0	−11.5	14.9	16.1	−6.8
0.6	−1223.5	−1766.5	−30.7	10.6	12.2	−13.4	14.9	16.3	−7.9
0.7	−1223.5	−1848.7	−33.8	10.6	12.5	−15.1	14.9	16.5	−9.0
0.8	−1223.5	−1928.9	−36.6	10.6	12.7	−16.7	14.9	16.6	−10.1
0.9	−1223.5	−2006.9	−39.0	10.6	13.0	−18.2	14.9	16.8	−11.1
1.0	−1223.5	−2083.1	−41.3	10.6	13.2	−19.6	14.9	17.0	−12.0
(b) h_{i2} = h_{i1} = 5000 Wm^{-2} K^{-1}									
0.0	−1550.5	−1550.8	−0.0	7.030	7.030	0.000	12.026	12.026	0.000
0.1	−1550.5	−1696.4	−8.6	7.030	7.034	−0.042	12.026	12.028	−0.020
0.2	−1550.5	−1841.8	−15.8	7.030	7.037	−0.084	12.026	12.030	−0.039
0.3	−1550.5	−1987.8	−22.0	7.030	7.040	−0.126	12.026	12.033	−0.059
0.4	−1550.5	−2133.2	−27.3	7.030	7.043	−0.167	12.026	12.035	−0.079
0.5	−1550.5	−2278.8	−31.9	7.030	7.046	−0.209	12.026	12.037	−0.098
0.6	−1550.5	−2424.2	−36.0	7.030	7.049	−0.251	12.026	12.040	−0.118
0.7	−1550.5	−2569.6	−39.7	7.030	7.052	−0.292	12.026	12.042	−0.137
0.8	−1550.5	−2715.0	−42.9	7.030	7.055	−0.334	12.026	12.044	−0.157
0.9	−1550.5	−2860.4	−45.8	7.030	7.057	−0.375	12.026	12.047	−0.177
1.0	−1550.5	−3005.8	−48.4	7.030	7.060	−0.417	12.026	12.049	−0.196

4.1. Example 1

A heater constructed by a circular duct with conductivity K_A = 77 Wm^{-1} K^{-1} (refer Table 1) and various surface emissivities ε = 0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9 and 1 (refer Table 2), length L = 10 m, inner radius r_1 = 198 mm, outer radius r_2 = 200 mm, the hot water is flowing inside the heater with entrance temperature T_{i1} = 65 °C and convective heat coefficient h_{i1} = 30 and 5000 Wm^{-2} K^{-1} (refer Table 3), with exit temperature T_{i2} = 60 °C and h_{i2} = h_{i1}, the ambient air at the entrance section with temperature T_{o1} = 22 °C and convective heat coefficient h_{o1} = 8 Wm^{-2} K^{-1} (refer Table 3), the ambient air at the exit section with temperature T_{o2} = 24 °C and convective heat coefficient h_{o2} = h_{o1}, and the heater is located in a big room with wall or surrounding temperature T_{sur} = 20 °C. The results are shown in Table 4 and Fig. 2.

4.2. Example 2

A chiller constructed by a circular duct with conductivity K_A = 77 Wm^{-1} K^{-1} (refer Table 1) and various surface emissivities ε = 0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9 and 1 (refer Table 2), length L = 10 m, inner radius r_1 = 198 mm, outer radius r_2 = 200 mm, the cold water is flowing inside the heater with entrance temperature T_{i1} = 7 °C and convective heat coefficient h_{i1} = 30 and 5000 Wm^{-2} K^{-1} (refer Table 3), with exit temperature

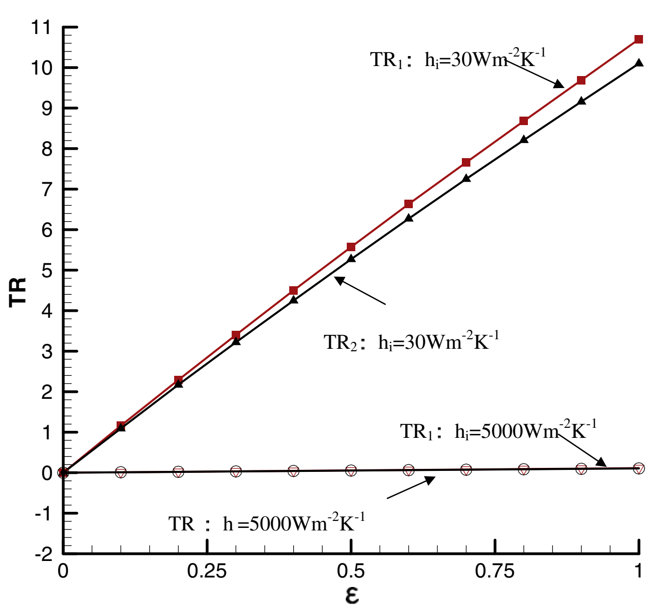

Fig. 3. The relations among surface temperature errors, TR_1 as well as TR_2, and emissivity, ε, for a heater.

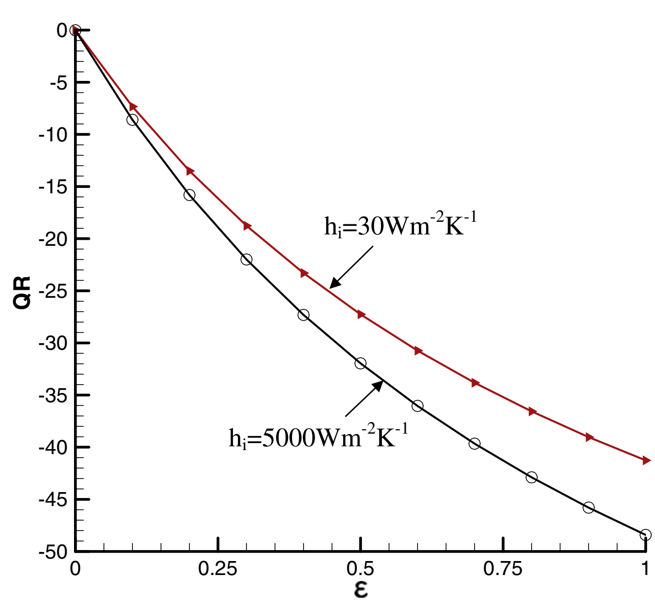

Fig. 4. The relations between heat transfer rate error, QR, and emissivity, ε, for a chiller.

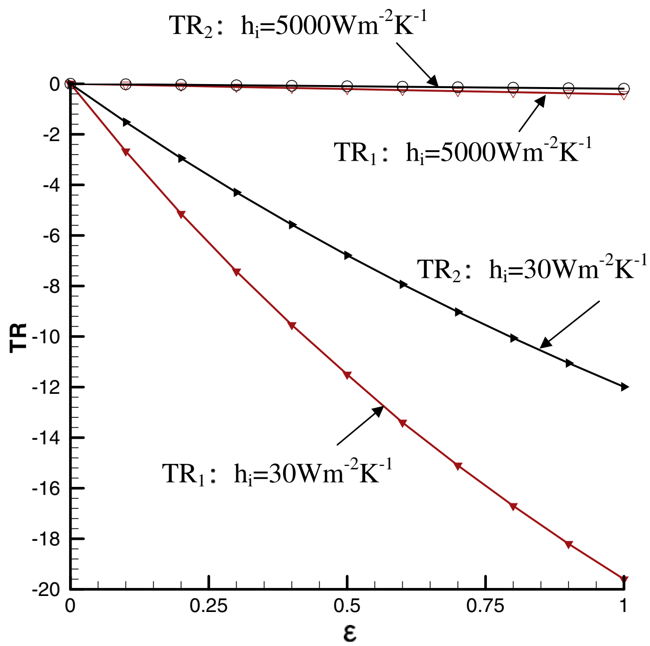

Fig. 5. The relations among surface temperature errors, TR_1 as well as TR_2, and emissivity, ε, for a chiller.

T_{i2} = 12 °C and $h_{i2} = h_{i1}$, the ambient air at the entrance section with temperature T_{o1} = 24 °C and convective heat coefficient h_{o1} = 8 Wm^{-2} K^{-1} (refer Table 3), the ambient air at the exit section with temperature T_{o2} = 26 °C and convective heat coefficient $h_{o1} = h_{o2}$, and the heater is located in a big room with wall or surrounding temperature T_{sur} = 30 °C. The results are shown in Table 5 and Fig. 3.

From Tables 4 and 5 and Figs. 4 and 5 show that in situations of $\varepsilon = 0$, $QR = 0$, $TR_1 = 0$ and $TR_2 = 0$; the greater the ε is, the greater the absolute values of QR, TR_1 and TR_2 will be; the greater the h_{o1} and h_{o2} are, the smaller the TR_1 and TR_2 will be; while $\varepsilon > 0$, all the QR are negative, it means in the situations of neglecting heat radiation, the smaller absolute values of heat transfer rate, $|Q|$, will be obtained (i.e., $|Q| < |Q_a|$), and most absolute values of QR are so big and out of the engineering acceptable range; and in situations of $\varepsilon > 0$, the greater the values of h_{i2} and h_{i1}, the greater the absolute values of QR will be, but the smaller the TR_1 and TR_2 will be; for the heater, TR_1 and TR_2 are positive (i.e., $T_{S1} > T_{21}$ and $T_{S2} > T_{22}$), it means in the situations of neglecting heat radiation, the greater surface temperatures (over-prediction of hot surface) will be obtained; for the chiller, TR_1 and TR_2 are negative (i.e., $T_{S1} < T_{21}$ and $T_{S2} < T_{22}$), it means in the situations of neglecting heat radiation, the smaller surface temperatures (over-prediction of cold surface) will be obtained, these predictions may lead to misjudge the condensation occurred on the surface.

5. Conclusions

From the practical numerical results of this study, it demonstrates that one will take a very big risk to neglect the influence of heat radiation especially in the situations of low outer ambient air convection coefficients and greater surface emissivity of a heat exchanger; because in the situations of higher emissivities, the very big errors of heat transfer characteristics may be obtained. Thus, in order to obtain accurate results of a heat exchanger, the log mean heat transfer rate (LMHTR) method should be applied in stead of the conventional log mean temperature difference (LMTD) method. The exact numerical heat transfer results of heat exchangers with circular duct can be obtained by any one-dimensional computer code (such as LabVIEW programming in this study) within one second computing time by a common PC.

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