[논문] 태양에너지
Solar Energy
Vol. 15. No. 3, 1995

Non−gray Radiation in the Entrance Region of a Smooth Tube

Taebeom Seo*,

* Samsung Corning

평편한 튜브의 입구 영역에서의 비회복사

서태범

* 삼성코닝 생산기술과

요 약

튜브 내의 입구영역에서 대류와 비회복사(mon-gray radiation)가 동시에 일어날 때의 열전달 특성을 수치해석적으로 연구하였다. 작동유체는 이산화탄소, 수증기, 질소의 혼합가스라 하였고, 유동은 속도장과 온도장이 동시에 발달하는 층류 유동으로 가정하였다. 복사전달방정식을 풀기 편하게 하기 위해 P−1 근사법이 사용되었고 혼합가스의 비회흡수계수(non-gray sbsorption coefficient)는 지수광폭밴드모형(exponential wide band model)을 이용해서 구하였다. 열전달 특성에 대한 온도조건의 영향을 조사하기 위해 튜브의 축방향에 대한 평균온도와 뉴셀트수(Nusselt number)의 변화를 몇 가지 다른 온도조건에 대해 나타내었다. 속도장과 온도장이 동시에 발달하는 경우의 뉴셀트수를 속도장은 완전히 발달되어 있고 온도장만 발달하는 경우의 뉴셀트 수와 비교하였다. 또한, 가스의 성분조성이 대류와 비회복사 뉴셀트수에 미치는 영향을 조사하였다.

Abstract

Non-gray radiation with convection in the entrance region of a smooth tube is numerically investigated. The fluid is a mixture of carbon dioxide, water vapor, and nitrogen to simulate combustion products of propane. The flow is assumed to be laminar and hydrodynamically and thermally developing. The P-1 approximation is used to simplify the radiative transfer equation and the exponential wide band model is adapted to model the spectral absorption coefficients of non-gray gas mixture. The bulk mean temperature and Nusselt number variation along the tube axis are shown for several inlet and wall temperature pairs to show the effect of temperature on the heat transfer characteristics. Nusselt numbers for simultaneously developing flow are compared to those for thermally developing flow. In addition, the effect of the mole fraction of the non-gray gases on convective and radiative Nusselt numbers is investigated

NOMENCLATURE

a : absorption coefficient(m^{-1})
c_p : specific heat at constant pressure (J/kgK)
D : diameter(m)
D_h : hydraulic diameter(m)
e_b : emissive power(W/m^2)
G : intergrated radiation intensity(W/m^2)
h : heat transfer coefficient(W/m^2K)
i : radiation intensity(W/m^2 μm sr)
k : thermal conductivity(W/mK)
N_R : the ratio of conduction to radiation ($=\frac{k \cdot a}{4\sigma T_w^3}$)
Nu : Nusselt number
P : pressure(Pa)
Pr : Prandtl number
q : heat flux(W/m^2)
q_r : radiative heat flux vector(W/m^2)
R : radius(m)
Re : Reynolds number($=\frac{VD_h}{\nu}$)
r : coordinate(m)
s : unit vector along the radiation path
T : temperature(K)
T_m : bulk mean temperature(K)
u : local axial velocity(m/s)
V : mean velocity(m/s)
v : local radial velocity(m/s)
x^* : dimensionless length($=\frac{x}{D_h \cdot Re \cdot Pr}$)
x : coordinate(m)
Y : mole fraction
α : integrated band intensity
λ : wavelength(μm)
ν : kinematic viscosity(m^2/s)
θ : dimensionless temperautre($=\frac{T-T_w}{T_{in}-T_w}$)
ρ : density(kg/m^3)
τ_o : optical radius($=a \cdot R$)
ω : solid angle exponential decay width

Subscripts

c : convective
i : i^{th} band
in : inlet
m : mean
r : radiative
t : total
w : wall
λ : spectral

Introduction

In order to design a high temperature heat exchanger and to evaluate its performance, a thorough analysis of heat transfer characteristics in a heat exchanger element is required. The analysis is complicated because thermal radiation becomes important. In these applications, the working fluids are usually products of combustion consisting mainly of carbon dioxide, water vapor, and nitrogen. Carbon dioxide and water vapor can emit and absorb thermal radiation, and the absorptivity is a strong function of wave number and temperature. Therefore, not only convective heat transfer but also non-gray radiative heat transfer in the participating media must be taken into account in any analysis of a high temperature heat exchanger tube element. In addition, the length to diameter ratios of typical tubes are short enough so that entrance effects become important. Hence, simultaneously developing flow and heat transfer with convective and radiative heat transfer should be used to model heat transfer in a high temperature heat exchanger.

Echigo et al.[1] analyzed two-dimensional combined convective and radiative heat transfer with fully developed laminar flow of a gray gas in a pipe. DeSoto [2]. considered the interaction of radiation with conduction and convection in a non-isothermal, non-gray gas(CO_2) in a circular tube with isothermal black walls; the flow was hydrodynamically fully developed and thermally developing. The exponential wide band model [3] was used to represent the temperature and frequency dependence of the spectral absorption coefficients. Im and Ahluwalia [4-6] studied combined convective and radiative heat transfer in a rectangular and a circular duct with a non-gray gas. They considered simultaneously developing turbulent flow and heat transfer of a mixture of carbon dioxide, water vapor, and particles. Using the moment method [7], they simplified the radiative transfer equation. The exponential wide band model and the Mie theory were used to calculate the spectral properties of a non-gray gas and scattering particles. Soufiani and Taine [8] considered non-gray radiation using a statistical narrow band model [9]. They applied this model to coupled radiation and convection in a laminar flow of a water vapor and air mixture between two parallel, isothermal walls but only showed the water vapor results. The Curtis-Godson approximation [10] was used to compensate for the effect of the non-isothermal temperature field. Yang and Ebadian [11] investigated combined convection and radiation in the thermal entrance region of a parallel plate duct with axial heat conduction and radiation of very low Peclet number flow (Pe = 10~100). The weighted sum of gray gas model [12] was used to approximate the non-gray carbon dioxide gas. The moment method was applied to solve the radiative transfer equation. Hirano et al. [13] numerically analyzed heat transfer for fully developed laminar flow of a gray gas and a non-gray gas (CO_2) between infinite parallel plates using a finite difference techniqeu. The exponential wide band model was used for non-gray band radiation. In another paper [14], Hirano et al. studied enhancement of radiative heat transfer by the use of a radiating plate between infinite parallel plates. A mixture of carbon dioxide, water vapor and nitrogen was used to simulate products of combustion. It was shown that by installing a solid plate in the gas flow, the heat transfer enhancement was 80% for carbon dioxide and 36% for combustion gas. Schuler and Campo [15] numerically analyzed combined convective and radiative heat transfer of a turbulent gray flow for the thermally developing region of a circular pipe using the P-1 approximation and the method of lines [18]. Yang and Ebadian [16] and macGrath et al. [17] calculated combined heat transfer of a gray gas in several different geometries. The method of lines, which is an explicit finite difference scheme, was used to solve the governing equations in [15-17].

From this literature review, it was found that useful information for high temperature heat transfer is sparse, and that simultaneously developing non-gray gas laminar flows in a cylindrical geometry have not been investigated. Hence, in the present study, non gray radiation and convection in simultaneously develping flow and heat transfer in a smooth tube is considered. The continuity, momentum, energy, and radiative transfer equations are solved numerically using a finite difference technique. The exponential wide band model is used to obtain the spectral absorption coefficient of the gas mixture and the P-1 approximation is used to simplify the radiative transfer equation.

Numerical Modeling

Combined convection and non-gray radiation in the entrance region of laminar flow in a smooth circular tube is numerically modeled. It is assumed that the gas is incompressible and its physical properties are constant and the axial diffusion and radiation of heat are negligible compared to the radial diffusion and radiation. Viscous energy dissipation is neglected. The gas is assumed to be non gray medium which absorbs and emits radiation, but does not scatter.

Governing equations

Based on these assumptions, the equations governing this flow are as follows:

Continuity :

$$\frac{1}{r}\frac{\partial}{\partial r}(rv) + \frac{\partial u}{\partial x} = 0 \quad (1)$$

Momentum :

$$v\frac{\partial u}{\partial r} + u\frac{\partial u}{\partial x} = -\frac{1}{\rho}\frac{dP}{dx}$$

$$+v\{\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial u}{\partial r}) + \frac{\partial^2 u}{\partial x^2} \quad (2)\}$$

Energy :

$$\rho c_p u\frac{\partial T}{\partial x} = k\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T}{\partial r}) - \nabla \cdot q_r \quad (3)$$

The appropriate boundary conditions are :

$$\text{at } r = 0 : \frac{\partial u}{\partial r} = 0, \ \frac{\partial T}{\partial r} = 0 \quad (4)$$

$$\text{at } r = R : u = 0, \ T = T_w \quad (5)$$

$$\text{at } x = 0 : u = u_{in}, \ T = T_{in} \quad (6)$$

The quantity $\overline{q_r}$ in Eq. (3) is the radiation flux vector which is defined as follows :

$$\overline{q_r} = \int_{4\pi}\int_0^\infty i_\lambda \bar{s} d\lambda d\omega \quad (7)$$

where i_λ is the radiation intensity and s is the unit vector. The radiation intensity can be obtained from the radiative transfer equation [10] :

$$\frac{di_\lambda}{ds} + a_\lambda i_\lambda = a_\lambda i_{\lambda b} \quad (8)$$

Integration Eq.(8) over all directions,

$$-\nabla \cdot q_{r\lambda} = a_\lambda(G_\lambda - 4e_{b\lambda}) \quad (9)$$

where

$$G_\lambda = \int_{4\pi} i_\lambda d\omega \quad (10)$$

is the radiation intensity integrated over all directions, and

$$e_{b\lambda} = \pi \cdot i_{b\lambda} \quad (11)$$

is the emissive power

Radiation model

In order to apply the P-1 approximation to non-gray radiation, the participating bands were divided into a number of sub-bands, and the absorption coefficient of each sub-band was assumed constant. The radiation intensity for each sub-band is calculated using the following equation[7] :

$$\frac{1}{r}\frac{\partial}{\partial r}(\frac{r}{a_i}\frac{\partial G_i}{\partial r}) = 3a_i(G_i - 4e_{bi}) \quad (12)$$

where

$$G_i = \int_{\lambda i}^{\lambda i+1} G_\lambda d\lambda \quad (13)$$

is the direction and band-integrated intensity.

The radiative boundary condition at the wall is given by [4] :

$$\frac{\partial G_{wi}}{\partial r} = -\frac{3}{2}a_i(\frac{\varepsilon_w}{2-\varepsilon_w})(G_{wi} - 4e_{bwi}) \quad (14)$$

From the radiation intensity calculated by Eq. (12), the divergence of the radiative heat flux can be written as follows :

$$-\nabla \cdot \overline{q}_r = \sum_{i=1}^{N} a_i(G_i - 4e_{bi}) \tag{15}$$

The spectral absorption coefficients of carbon dioxide and water vapor are ditermined from the exponential wide band parameters(the integrated band intensity, the exponential decay width, and the mean line-width-to-spacing parameter) [3]. In the present study, the 2.7, 4.3, and 15μm bands for carbon dioxide and the 2.7, 6.3μm, and rotational bands from water vapor are considered because these bands are important from a heat transfer point of view compared to othere small bands(2.0, 9.4, and 10.4μm bands for carbon dioxide and 1.38 and 1.87μm for water vapor). The exponential wide band model was developed from experimental data to obtain band absorption. When approximate spectral calculations are made(approximate because the model shape is an idealization of the true band shape), the spectral-integrated band absorption (A_Q) is little different from the four-region expression for band absorption (A). According to the chart of the ratio of A/A_Q in Edwards and Balakrishnan [20], the maximum discrepancy is about 40%. Fortunately, the chart also shows that the spectral calculations for the bands considered in the present study had only 1−3% discrepancies except the rotational band of water vapor. The discrepancy of the rotational band of water vapor was about 10%. If the adjusted parameters recommended by Edwards and Balakrishnan [20] are used to compensate for the discrepancy, the exponential wide band model can be successfully applied to engineering problems without significant error.

Since the bands of carbon dioxide and water vapor overlap, the influence of band overlapping must be taken into account. The absorption coefficients of overlapped bands are simply calculated by adding the absorption coefficients of each band [3].

Method of solution

In order to solve the governing equations, a finite difference technique, which employs an implicit scheme based on the SIMPLEST algorithm [21], was used.

Total, convective, and radiative heat fluxes transferred from the wall are given as follows :

$$q_t = \frac{1}{R \cdot \Delta x}\left[\int_0^R \rho u c_p T r dr\,|_{x+\Delta x} - \int_0^R \rho u c_p T r dr\,|_x\right] \tag{16}$$

$$q_c = -k\left(\frac{\partial T}{\partial r}\right)_{r=R} \tag{17}$$

$$q_r = q_t - q_c \tag{18}$$

From these heat fluxes, total, convective and radiative Nusselt numbers can be difined as :

$$Nu_t = \frac{q_t \cdot D}{k(T_w - T_m)} = Nu_c + Nu_r \tag{19}$$

$$Nu_c = \frac{q_c \cdot D}{k(T_w - T_m)} \tag{20}$$

Table 1. Comparison of the results for different grid sizes

No. of grids ($r \times x$)	$x = 0.1m$		$x = 0.5m$	
	Nu_c	Nu_r	Nu_c	Nu_r
20×100	15.84	22.02	8.518	18.89
40×100	16.08	21.85	8.655	18.75
60×100	16.12	21.82	8.688	18.72
40×50	15.86	21.83	8.645	18.74
40×150	16.07	21.86	8.651	18.75

$$Nu_r = \frac{q_r \cdot D}{k(T_w - T_m)} \qquad (21)$$

Results have been obtained for the flow of the products of combustion of natural gas and propane in a 0.1 m diameter smooth tube. Hence, numerical analysis of combined convective and radiative heat transfer has been performed for carbon dioxide, water vapor, and nitrogen mixtures. In order to simulate stoichiometric combustion of propance, mole fractions of each gas component were selected as follows : $Y_{CO_2} = 0.10$, $Y_{H_2O} = 0.15$, and $Y_{N_2} = 0.75$.(Combustion of four representative natural gases will be discussed below.) An inlet temperature of 1,200K, wall temperatures of 900, 600, and 300K, and Re = 2,000 were used. The fluid properties for each run were evaluated at 1,050K, 900K, and 750K, respectively; the Prandtl number ranged from 0.78 to 0. 80.

The domain was divided into 40 radial and 100 axial control volumes spaced non-uniformly. A fine grid was used near the wall and close to the inlet of the tube. Grid sensitivity was examined by comparing the results with $T_w = 900K$ at progressively refined grids as summarized in Table 1. The Nusselt numbers, Nu_c and Nu_r from a 20 (the radial direction)$\times$100(the axial direction) grid are about 2% and 1% different from those from a 60×100 grid, respectively. Nu_c and Nu_r using a 40×100 grid are less than a 0.5% different from those with a 60×100 grid. The 40×50 grid was not used because accurate information close to the inlet of the tube could not be obtained. Based on these results, all subsequent calculations were performed with the $40\times$ 100 grid.

Pure convection heat transfer in simultaneously developing flow and combined heat transfer with gray radiation in a thermally developing flow were calculated, and these results were compared to the results in the pien literature as a validation of the code. All sub-band absorption coefficients (a_i) were set to zero for pure convection and constant for gray radiation, respectively. The Nusselt number for pure convection was compared to the results in Shah and London [22]. The two results are almost identical. The bulk mean temperature, the heat fluxes, and the temperature profiles for combined heat transfer with gray radiation were compared with the results($\tau_0 = 1.0$ and $N_R = 0.01$) of Echigo et al. [1]. The differences between these two results were indistinguishable.

Results

For flow in a circular tube, the bulk mean temperature variation and Nusselt number variation for three different wall temperatures are shown in Figs. 1 and 2, respectively. In order to visualize the effect of radiation heat transfer, the bulk mean temperature and the temperature profile for pure convection are also presented in Fig. 1. Because of radiation, the bulk mean temperature with combined heat transfer grows much faster than that with pure convection. In addition, the bulk mean temperature for $T_w = 900K$ decreases more rapidly than the others because radiation at high temperature is more dominant than in other cases.

Nusselt number variation along the tube length is shown in Fig.2 compared to the pure convection case. Because the convective Nusselt numbers for the three different wall temperatures are not much different from each other, the convective result for $T_w = 600K$ is shown in the graph. The convective Nusselt numbers with combined heat transfer are greater than those with pure convection, because of radiation coming from the hot gas near the center line. In other words, radiation from the center region makes the gas temperature near the wall higher than that of the pure convection case so that the temperature gradient at the

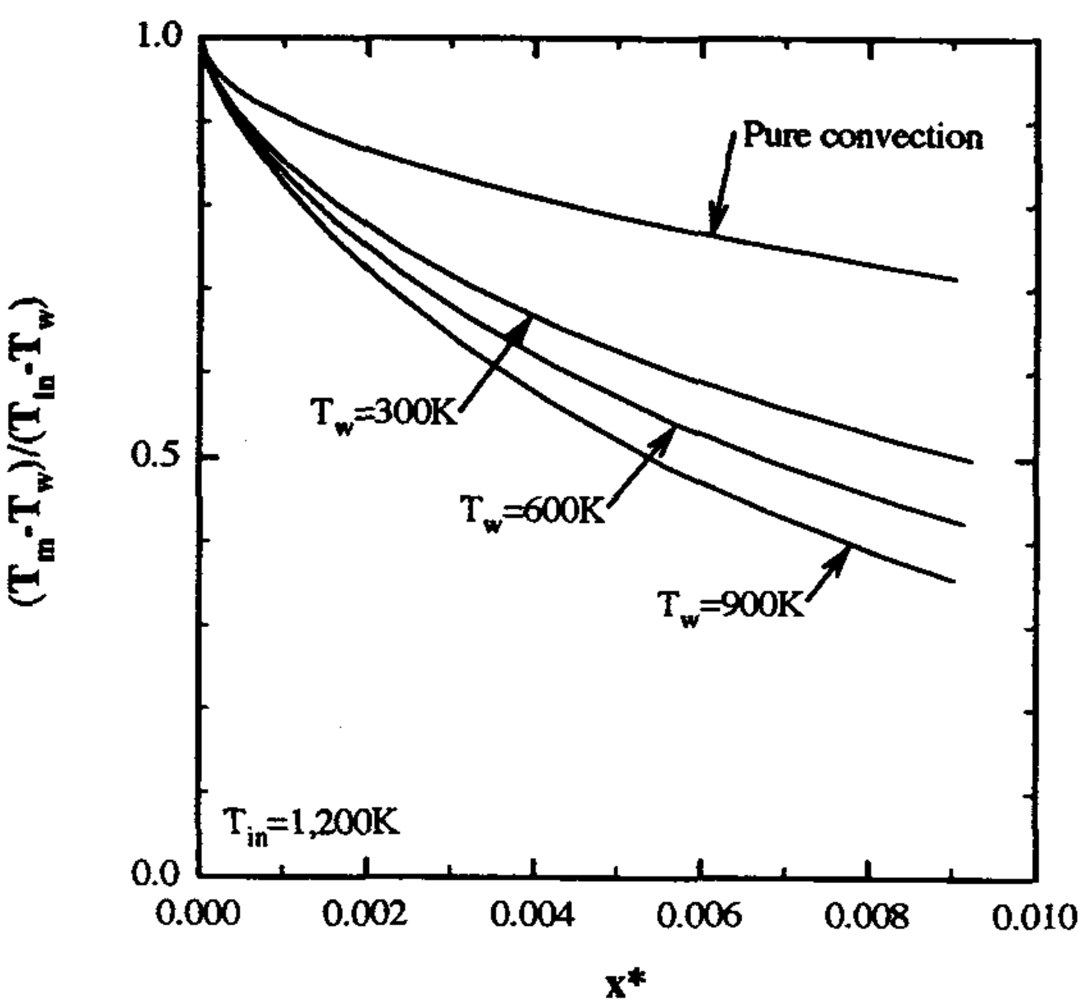

Fig 1. Bulk mean tempertature variation(Re=2, 000, D=0.1m, Y_{CO_2}=0.10, Y_{H_2O}=0.15, Y_{N_2}=0.75)

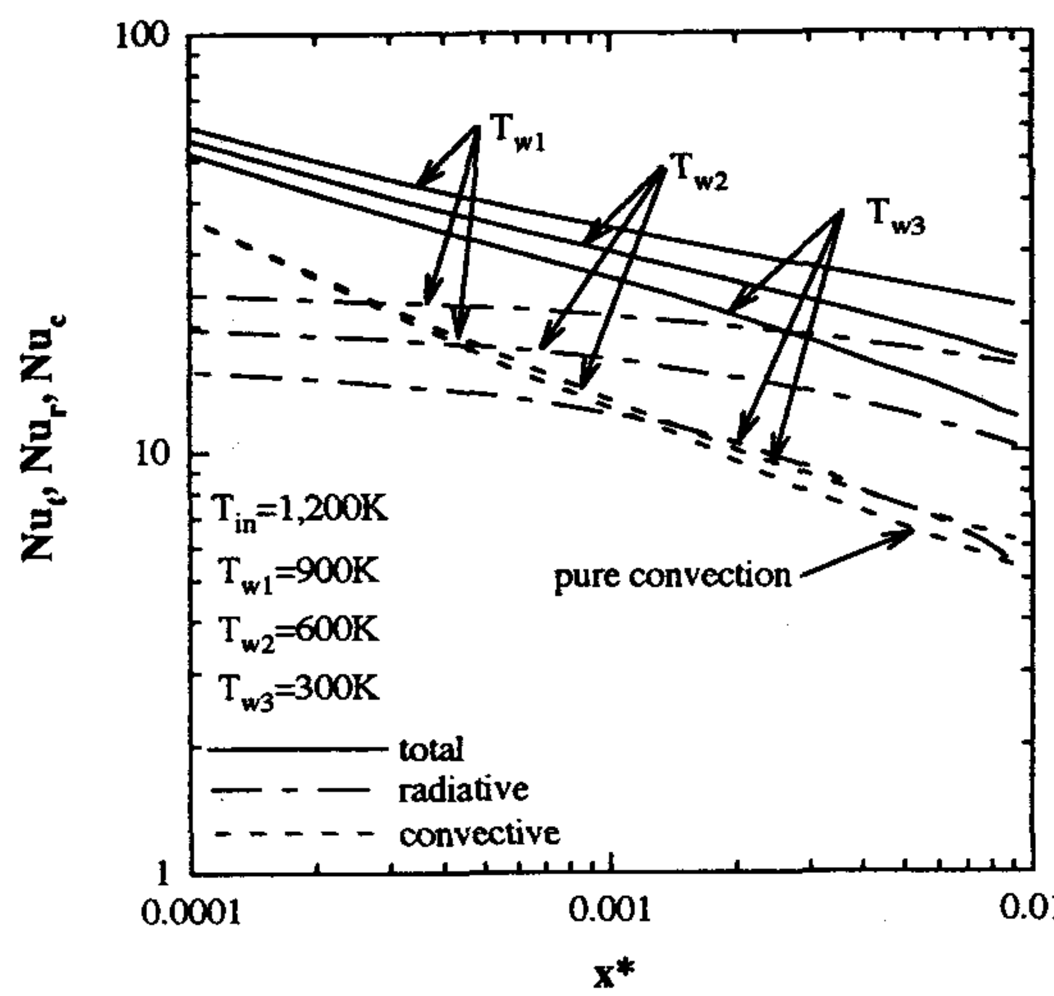

Fig 2. Local Nusselt number variation(Re=2, 000, D=0.1m, Y_{CO_2}=0.10, Y_{H_2O}=0.15, Y_{N_2}=0.75)

wall is steeper than it would be in pure convection. It was found that the difference in convective nusselt Numbers between the combined heat transfer and pure convection case at $x^* = 0.009$ is about 13%. However, the difference between the convective Nusselt numbers for the three different wall temperatures is only about 1%. The radiative Nusselt numbers decrease as the difference between the inlet and the wall temperature increases ; the radiative heat transfer is dominant when the temperature difference is small and the temperature level is high. Therefore, the total Nusselt number at $T_w = 900K$ is the largest among the three cases, although the absolute quantity of heat transfer is the smallest.

Nusselt numbers for thermall and simultanesusly developing flow are compared in Fig.3. Both convective and radiative Nusselt numbers for simultaneously developing flow are greater than those for thermally developing flow. At $x^* = 0.005$, convective, radiative, and total Nusselt numbers for simultaneously developing flow are about 19%, 9%, and 11% greater than those for thermally developing flow, respectively. At $x^* = 0.009$, the difference decreases to 15%, 7%, and 9%, respectively. In order to simplify the flow calculation, thermally developing flow is often assumed. However, as seen from these results this assumption can introduce significant errors.

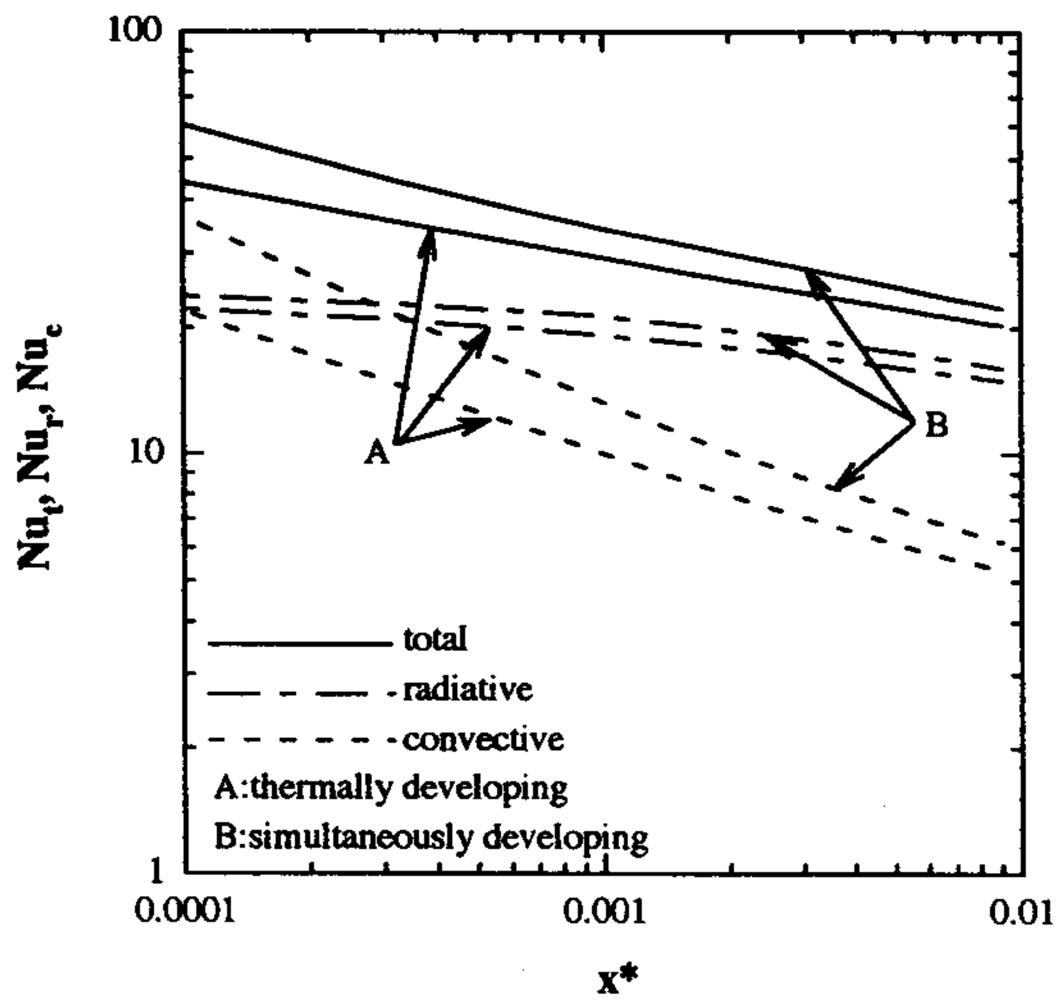

Fig 3. Nusselt number comparison of thermally and simultaneously developing flows(Re=2,000, D=0.1m, T_{in}=1,200K, T_w=900K, Y_{CO_2}=0.10, Y_{H_2O}=0.15, 75)

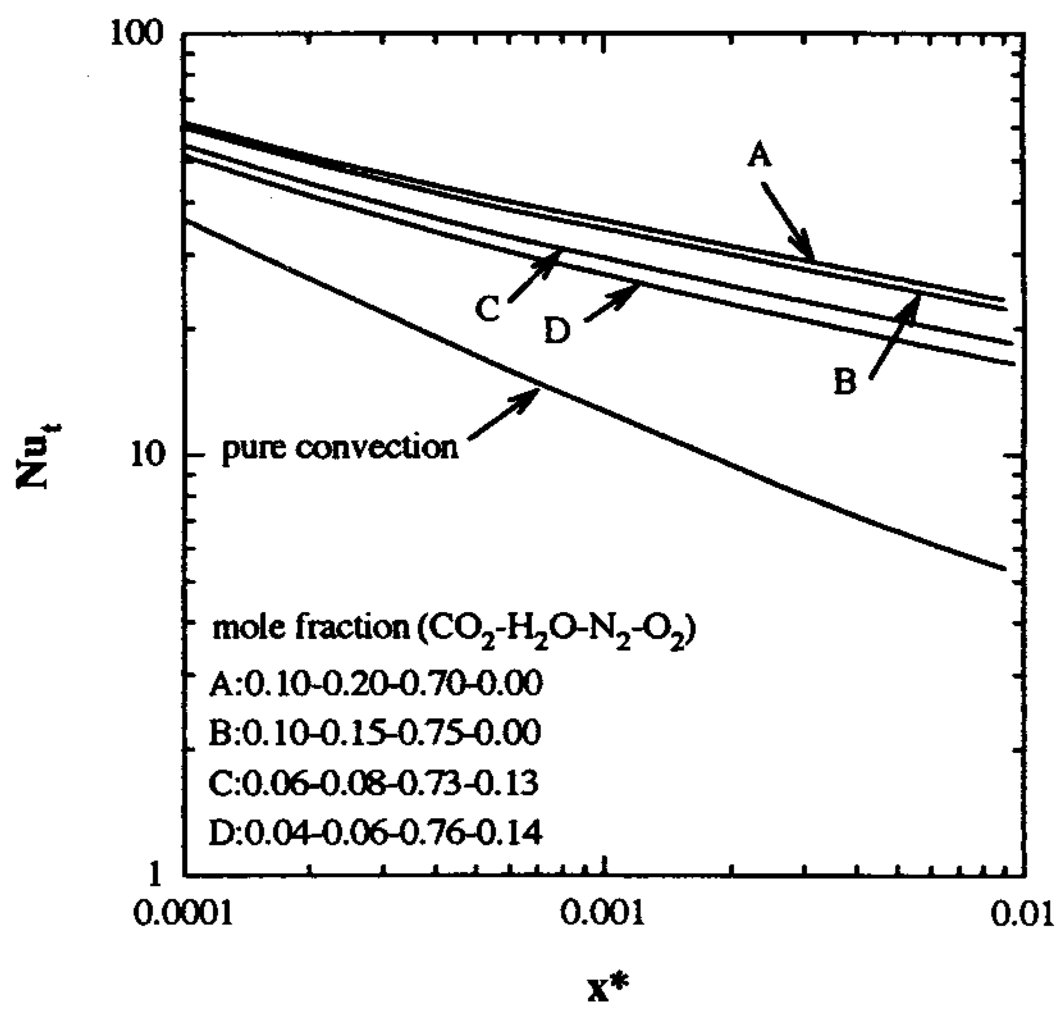

Fig 4. Total Nusselt number comparison for several gas mixture(Re=2,000, D=0m, T_{in}=900K)

Mixture A represents stoichiometric combustion products of natural gas 1 and 2. Mixture B represents stoichiometric combustion products of natural gas 3, 4, and propane. Mixtures C and D represent combustion products of propane with 100% excess air and natural gases 2-4 with 200% excess air respectively.

Table 2. Mole fractions of typical natural gases

constituent	natural gas 1	natural gas 2	natural gas 3	natural gas 4
methane	0.939	0.601	0.674	0.543
ethane	0.036	0.148	0.168	0.163
propane	0.012	0.134	0.158	0.162
butane	0.013	0.042	0.000	0.074
nitrogen	0.000	0.075	0.000	0.058

Total Nusselt numbers for several different gas mixtures are compared in Fig. 4. Since propane and natural gases are used as fuel by a number of industries, their combustion products were used as the non-gray working fluids in the study. Four kinds of natural gases are considered [24], and their combinations appear in Table 2. Mixture A represents the stoichiometric combustion products of natural gas 1. Mixture B is the stoichiometric products of an approximate average of natural gases 2, 3, and 4, and mixture C represents stoichiometric combustion of propane. The mixtures D and E represent the combustion products of propane with 100% excess air and the natural gases 2-4 with 200% excess air. Again, the Prandtl number ranged from 0.77 to 0.82. The total Nusselt numbers for the non-gray gas mixtures are much larger than those for the pure convection case. As the total mole fraction of the radiating gases (CO_2, H_2O) decreases, the total Nusselt numbers decrease. Although the total mole fraction of the radiating gases is only 0.10 in mixture E, the total Nusselt number is more than twice the pure convection Nusselt number. Thus, even a small amount of radiating gases can change the overall heat transfer characteristics dramatically.

The bulk mean temperature variations calculated with the gray Planck mean absorption coefficients are shown in Fig.5. Three different Planck mean absorption coefficients are used. The first Planck mean absorption coefficient is evaluated at the inlet temperature. The second Planck mean

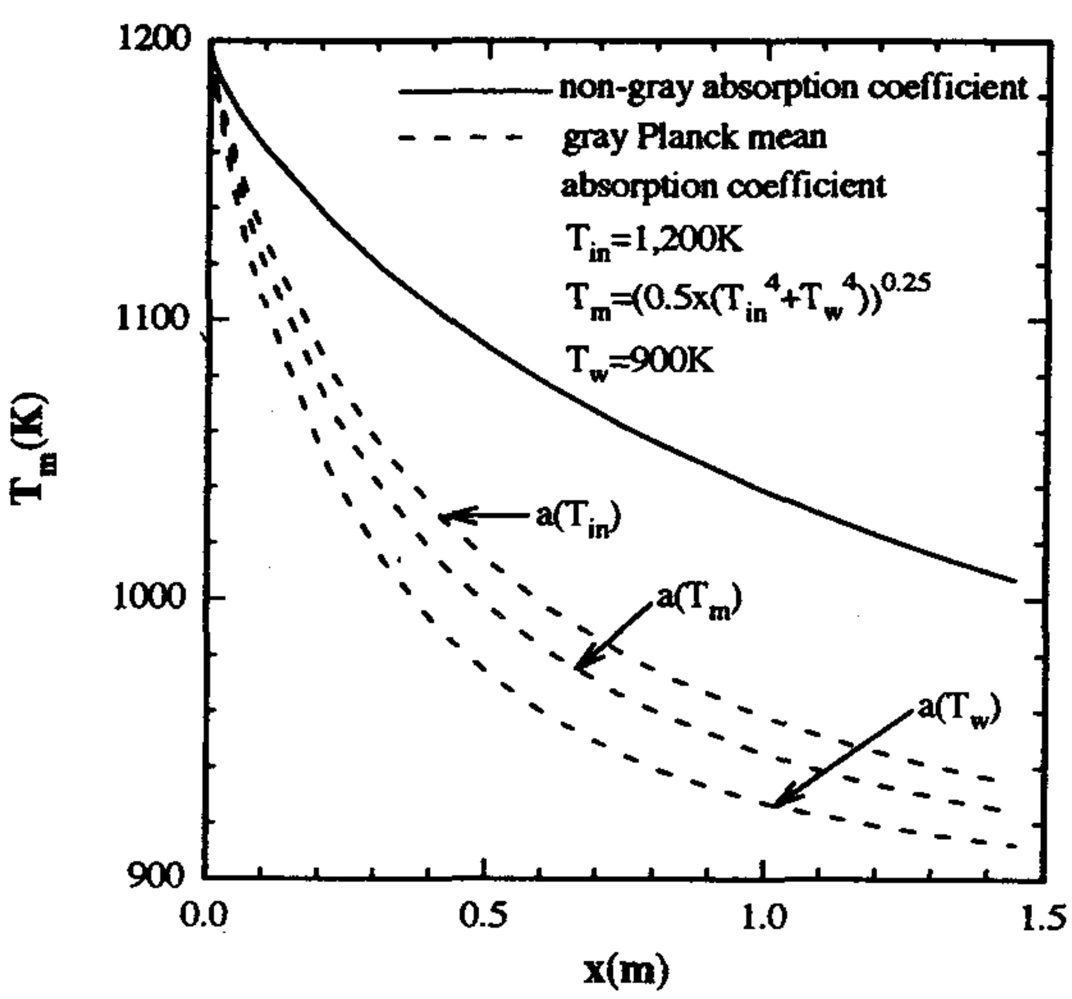

Fig 5. Bulk mean temperature comparison for non-gray absorption coefficients and gray Planck mean absorption coefficients(Re=2,000, D=0.1m, Y_{CO_2}=0.10, Y_{H_2O}=0.15, Y_{N_2}=0.75)

absorption coefficient is calculated at a weighted average temperature given by:

$$T_{ave} = \left[\frac{T_w^4 + T_{in}^4}{2} \right]^{\frac{1}{4}} \qquad (23)$$

The third Planck mean absorption coefficient is evaluated at the wall temperature. None of these three gray Planck mean absorption coefficients predicts well the combined convective and radiative heat transfer with non-gray gases. As can be seen, the gray Planck mean absorption coefficient overestimates the non-gray radiation significantly.

Conclusions

From this study, the following conclusions are drawn:

1) The bulk mean temperature variation and the Nusselt numbers for flowing non-gray gas mixtures are very different from those of pure convection. The convective Nusselt numbers in combined heat transfer are greater than those for pure convection. In addition, radiative heat transfer becomes dominant when the temperature difference between the wall and the inlet is small and the temperature level is high.
2) Both convective and radiative Nusselt numbers for simultaneously developing flow are greater than those for thermally developing flow.
3) The heat transfer characteristics of non-gray gas mixtures depends strongly upon the mole fraction or non-gray gases, and a small amount of radiating gases can change the heat transfer significantly.
4) A gray analysis using a gray Planck mean absorption coefficient overestimates radiative heat transfer significantly.

References

1. R. Echigo, S. Hasegawa and K. Kamiuto, Composite Heat Transfer in a Pipe with Thermal Radiation of Two Dimensional Progagation-in Connection with the Temperature Rise in Flowing Medium Upstream from Heating Section, Int, J. Heat Mass Transfer, vol.18, pp.1149−1159, 1975.
2. S. DeSoto, Coupled Radiation, Conduction and Convection in Entrance Region Flow, Int. J. Heat Mass Transfer, vol. 11, pp.30−53, 1968.
3. D.K. Edwards, Molecular Gas Band Radiation, Advances in Heat Transfer, vol.12, pp.115−193, Academic Press, New York, 1976.
4. K.H. Im and R.K. Ahluwalia, Combined Convective and Radiation in Rectangular Ducts, Int. J. Heat Mass Transfer, vol.27, no.2, pp.221−231, 1984.
5. K.H. Im and R.K. Ahluwalia, Radiative Transfer in Spectrally Dissimilar Absorbing-Emitting-Scattering Adjacent Mediums, AIAAJ, vol.20, no.1, pp.134−141, 1983.
6. R.K. Ahluwalia and K.H. Im, Radiative

Heat Transfer in Segregated media, AIAAJ, vol.22, no.2, pp.317−319, 1984.
7. M.N. Ozisik, Radiation Transfer and Interactions with Conduction and Convectin, pp.343−346, Wiley, New York, 1973.
8. A. Soufiani and J. Taine, Application of Statistica Narrow Band Model to Coupled Radiation and Convection at High Temperature, Int. J. Heat Mass Transfer, vol.30, no.3, pp.437−447, 1987.
9. A. Soufiani, J.M. Hartmann, and J. Taine, Validity of Band Model Calculations for CO_2 and H_2O Applied to Radiative Properties and Conductive-Radiative Transfer, J. Quat. Spectr. Rad. Trans., vol.33, pp.243−257, 1985.
10. R. Siegel and J.R. Howell, Thermal Radiation Heat Transfer, 3rd ed., chap. 17, hemisphere Publishing Co., 1992.
11. G. Yang and M.A. Ebadian, Radiative-Convective Heat Transfer in a Parallel Plate Duct with Non-gray Medium, Advances in Heat Exchanger Design, Radiation, and Combustion, HTD-vol.182, pp.51−58, 1991.
12. H.C. Hottel and A.F. Sarofim, Radiative Transfer, chap.6, McGraw Hill Book Co., New York, 1967.
13. M. Hirano, T. Miyauchi, and Y. Takahira, Heat Transfer Analysis of a Nongray Gas in a Flow System(Part 1. The Case of Small Temperature Differences between the Gas and a Heat Absorbing Surface), Heat Transfer-Japancese Research, vol.17, no.2, 1988.
14. M. Hirano, T. Miyauchi, and Y. Takahira, Enhancement of Radiative Heat Transfer in the Laminar Channel Flow of Non-gray Gases, Int. J. Heat mass Transfer, vol.31, no.2, pp. 367−374, 1988.
15. C. Schuler and A. Campo, Numerical Prediction of Turbulent Heat Transfer in Gas Pipe Flows Subject to Combined Convection and Radiation, Int. J. Heat Fluid Flow, vol.9, no.3, pp.308−315, 1988.
16. G. Yang and M.A. Ebadian, Combined Radiation and Convection Heat Transfer in Simultaneously Developing Flow in Ducts with Semi-circular and Right Triangular Cross Sections, Warme-Stoffubertragung, vol.27, pp. 141−148, 1992.
17. D. MacGrath, G. Yang, and M.A. Ebadian, Conjugated Heat Transfer in a Concentric Annular Pipe, Nuclear Engineering and Design, vol.132, pp. 393−402, 1992.
18. O.A. Liskovets, The Method of Lines (review), Differential Equation, vol.1, pp.1308−1323, 1965.
19. Y. Bayazitoglu and J. Higenyi, Higher-Order Differential Equation of Radiative Transfer : P-3 Approximation, AIAAJ., vol.17, no.4, pp.424−431, 1979.
20. D.K. Edwards and A. Balakrishnan, Thermal Radiation by Combustion Gases, Int, J. Heat mass Transfer, vol.

16, pp.25−40, 1973.

21. D.B. Spalding, mathematical Modeling of Fluid-Mechanics, Heat-Transfer and Chemical-Reaction Processes, A Lecture Course, CFDU Report, HTS/80/1.
22. R.K. Shah and A.L. London, Laminar Flow Forced Convection in Ducts, Advances in heat transfer, Supplement 1, pp.141, Academic Press, New York, 1978.
23. R.A. Strehlow, Combustion Fundamentals, McGraw-Hill, New York, 1984.
24. G.J. Van Wylen and R.E. Sonntag, Fundamentals of Classical Thermodynamics, 3rd, John Wiley and Sons, pp.471, 1986.

Non−gray Radiation in the Entrance Region of a Smooth Tube

Taebeom Seo*,

* Samsung Corning

Abstract

Non-gray radiation with convection in the entrance region of a smooth tube is numerically investigated. The fluid is a mixture of carbon dioxide, water vapor, and nitrogen to simulate combustion products of propane. The flow is assumed to be laminar and hydrodynamically and thermally developing. The P-1 approximation is used to simplify the radiative transfer equation and the exponential wide band model is adapted to model the spectral absorption coefficients of non-gray gas mixture. The bulk mean temperature and Nusselt number variation along the tube axis are shown for several inlet and wall temperature pairs to show the effect of temperature on the heat transfer characteristics. Nusselt numbers for simultaneously developing flow are compared to those for thermally developing flow. In addition, the effect of the mole fraction of the non-gray gases on convective and radiative Nusselt numbers is investigated