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# EFFECT OF THERMAL RADIATION IN THE HEAT TRANSFER IN SMOKETUBE STEAM GENERATORS

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***Abstract.*** *This work presents a numerical analysis of the heat transfer combining convection and thermal radiation in the turbulent flow of participating gases inside circular tubes of smoketube steam generators. Two gaseous mixtures are considered, being a typical products of the combustion of methane and fuel oil. The gases temperature profiles and the heat transfer are solved by the numerical solution of the energy conservation equation. The diffusive and the convective terms are treated by the Flux-Spline control volume method, while the thermal radiation is dealt with the zonal formulation. The gases radiative properties are modeled by the weighted-sum-of-gray-gases. The results are presented for typical conditions found in smoketube steam generators, and include the effect of thermal radiation on the convective, radiative and total Nusselt numbers, on the gases bulk temperature and on the heat transfer. The analysis shows that thermal radiation can have an importance in the thermal process, especially for tubes with larger diameters.*

*Keywords: smoketube steam generators, participating gases, combined heat transfer mode, zonal method, control volumes*

## 1. Introduction

The solution of the heat transfer combining convection and thermal radiation is of great importance in many engineering applications. Examples include steam generators, furnaces for materials processing, combustors and boilers of power plants. In such processes, heat transfer from the high temperature combustion gaseous products to the system surface results from both the convective and radiative mechanisms.

Viskanta (1998) discusses a number of fundamental difficulties that are related to the inclusion of thermal radiation in the analysis. The first one is related to the treatment of the complex dependence of the gases radiative properties, more specifically the absorption coefficient, on the wavelength. The second difficulty concerns its nature of propagation, which is not only limited to the vicinity of a point but involves the entire space around the point, and so it requires the integration of the radiation intensity arriving from all directions in the space. A third difficulty is predicting the effect of fluctuations in turbulent flows temperatures. These difficulties have given rise to a number of specific methods for the solution of the radiative transfer equation. Selection of the appropriate methods depends on the problem considered.

Several works dealing with the combined heat transfer by convection and thermal radiation are available in the literature. Einstein (1963), Echigo et al. (1975), Smith et al. (1985), França and Goldstein (1995) considered the propagation of thermal radiation in both the radial and axial direction inside the tube. With the exception of the second work, which employed finite difference approximation, the other three works used a zonal-type formulation. Smith et al. (1986) and Alturki and Smith (1987) employed the zonal method with weighted-sum-of-gray-gases to account for the presence of particles in the participating medium. Campo and Schuler (1988), Schuler and Campo (1988), and Seo et al. (1994) used the *p*-1 approximation to simplify the integral formulation of the radiation exchange to a first-order differential equation, and considered only the heat transfer in the radial direction. More recently, Sediki et al. (2002) solved the combined radiation-convection heat transfer in a circular tube employing mode advanced gas models, the CK-Correlated-k and the ADF-Absorption Distribution Function.

In this work, it is considered the combined heat transfer by convection and thermal radiation in the flow of combustion gases in the interior of circular tubes of smoketube steam generators. The results include the gases bulk temperature distributions and the heat transfer by radiation and convection for typical engineering applications. While it is common practice to consider only convection heat transfer in such systems, this work shows that thermal radiation can be of significance, especially for tubes with larger diameters. The analysis is based on the numerical solution of the energy conservation, where the diffusive-advective terms are treated by the Flux-Spline control volume, and the radiation terms are solved with the zonal method, considering radiation transfer in both the radial and axial directions.

The dependence of the gas absorption coefficient on the wavelength is taken into account with the weighted-sum-of-gray-gases model. Two gaseous mixtures at a total pressure of 1.0 atm are considered. The first one, named gas mixture 1, is composed of water vapor (0.2 atm), carbon dioxide (0.1 atm), and nitrogen (0.7 atm), and is a typical product of stoichiometric combustion of methane. The second one, gas mixture 2, is composed of water vapor (0.1 atm), carbon dioxide (0.1 atm), and nitrogen (0.8 atm, and is a typical product of stoichiometric combustion of fuel oil. The tube walls are diffusive, gray emitters and absorbers. The gases physical properties are computed at the average bulk temperature, that is, the arithmetic mean between the gas inlet and outlet temperatures. The fluid flow is turbulent and developed in the entrance, while the temperature inlet is assumed uniform, so that the thermal entrance region is considered. The inlet and outlet reservoirs are treated as black surfaces at the temperature of the reservoirs.

## 2. Problem analysis

Figure 1 presents a schematic view of the system. The tube inner diameter and length are indicated by $D$ and $L$, respectively. The internal surface is indicated by ε. The gas inlet and outlet temperatures are indicated by $T_{g1}$ and $T_{g2}$, and the surface temperature is uniform and indicated by $T_s$. The condition of uniform temperature in the tube is a satisfactory approximation for tubes where vapor is formed in its outside surface, due to the very high heat transfer coefficients that typically occur in nucleate boiling. Thus, the surface temperature can be approximated by the water liquid-vapor saturation temperature at the operation pressure of the vapor line. The internal flow is turbulent and developed at the entrance, but the gas inlet temperature is assumed uniform. As it occurs in steam generators, the gas temperature is higher than the tube temperature, so that heat is transferred from the gas to the tube surface. Both the radiative and the convective mechanisms are included in the analysis.

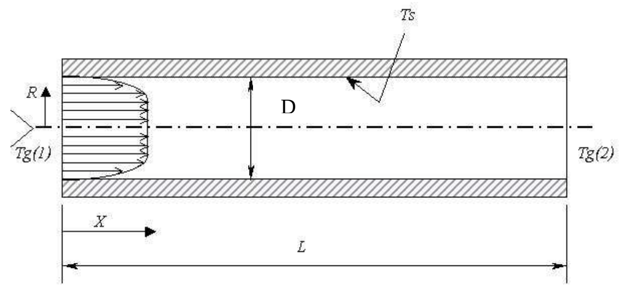


Figure 1. Problem geometry.

To determine the heat transfer, it is necessary to determine the gas temperature distribution, which in turn requires the solution of the energy conservation equation. In dimensionless form, this equation is given by:

$$\frac{\partial}{\partial x}\left(ut - \frac{1}{\mathrm{Re}\,\mathrm{Pr}}\frac{\partial t}{\partial x}\right) - \frac{1}{r}\frac{\partial}{\partial r}\left[r\left(\frac{1}{\mathrm{Re}\,\mathrm{Pr}} + \frac{1}{\mathrm{Re}}\frac{\varepsilon_H}{\nu}\right)\frac{\partial t}{\partial r}\right] = -\frac{1}{\mathrm{Re}\,\mathrm{Pr}\,N_{CR}}q_R^* \quad (1)$$

where $t$ is the gas dimensionless temperature, $\overline{T}/T_g$ ; $x$ and $r$ are the dimensionless coordinates, $X/D$ and $R/D$, respectively; $N_{CR}$ is the conduction-radiation parameter, given by $k/\left(D\sigma T_g^3\right)$, in which $T_g$ is the gas inlet temperature; Re and Pr are the Reynolds and Prandtl number, and $\varepsilon_H$ is the thermal eddy diffusivity to account for the thermal turbulent transport. The thermophysical properties of the gas, such as the density ρ, the kinematic viscosity ν and the thermal conductivity $k$ are assumed constant and evaluated at the gas average bulk temperature. In this solution, the velocity profile is assumed as developed, and is given by the equation proposed by Reichardt (1951), as presented in Kays e Crawford (1980), while the thermal eddy diffusivity $\varepsilon_H$ is computed from the model proposed by Kays and Crawford (1980) for ducts with circular cross section.

The term $q_R^*$ is the dimensionless net volumetric radiative heat rate, given by $q_R D/\left(\sigma T_g^4\right)$, where σ is the Stefan-Boltzman constant, equal to $5.67\times10^{-8}$ W/m$^2$·K4. In the energy equation, $q_R^*$ corresponds to the radiative energy emitted minus the energy absorbed per unit of volume in the gas element, and is evaluated by the application of the zonal method combined to the weighted-sum-of-gray-gases (WSGG) model. The zonal method involves the division of the enclosure into surface and volume zones, in which all radiative quantities (emissive power, radiosity and irradiation) are assumed uniform. Figure 2 shows the volume and surface zones in the tube, which are coincident to the control volumes used for the discretization of the diffusive-advective terms. According to the zonal method and the WSGG model, the net volumetric radiative heat rate for volume zone $V_\gamma$ is given by:

$$q_{R_\gamma} = \frac{1}{V_\gamma}\left[4V_\gamma \sum_{i=0}^{I}\left(C_{e,i}\left(T_\gamma\right)a_i\right)\sigma T_\gamma^4 - \sum_{\gamma^*=1}^{\Gamma}\overrightarrow{g_{\gamma^*} g_\gamma}\sigma T_{\gamma^*}^4 - \sum_{j=1}^{J}\overrightarrow{s_j g_\gamma} q_{o,j}\right] \tag{2}$$

On the right-hand side of the above equation, in the bracket, the first term accounts for the radiation emitted by the volume zone $V_\gamma$, while the second and the third accounts for the absorption of radiation from the other volume zones $V_\gamma$ and surface zones $A_j$, respectively. The gas-to-gas and the surface-to-gas directed-flux areas, $\overrightarrow{g_{\gamma^*} g_\gamma}$ e $\overrightarrow{s_j g_\gamma}$, are obtained from:

$$\overrightarrow{g_{\gamma^*} g_\gamma} = \sum_{i=0}^{I}\left[C_{e,i}\left(T_{\gamma^*}\right)\left(\overline{g_{\gamma^*} g_\gamma}\right)_i\right] \tag{3}$$

$$\overrightarrow{s_j g_\gamma} = \sum_{i=0}^{I}\left[C_{e,i}\left(T_j\right)\left(\overline{s_j g_\gamma}\right)_i\right] \tag{4}$$

The coefficients $C_{e,i}$ are polynomial functions that are solely dependent on the radiation source temperatures, $a_i$ is the absorption coefficient of each gray gas of the WSGG model. For the two gas mixtures considered in this work, the values of $C_{e,i}$ and $a_i$ are obtained from Smith *et al.*(1982).

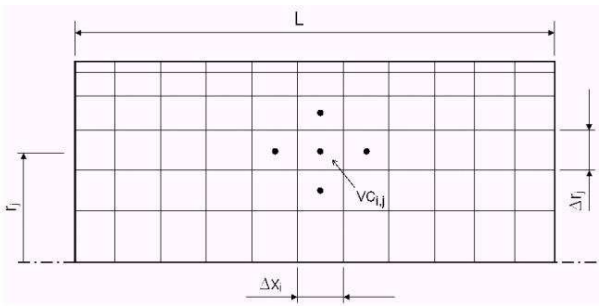


Figure 2. Surface and volume zones in the interior of a circular tube.

The gas-to-gas and the surface-to-gas direct-exchange areas, $\overline{g_{\gamma^*} g_\gamma}$ and $\overline{s_j g_\gamma}$, respectively, for each absorption coefficient $a_i$ are given by Sika (1991). The radiosity, $q_{o,j}$, corresponds to the sum of the emission and the reflection from surface $A_j$, and is given by:

$$q_{o,j} = \varepsilon_j \sigma T_j^4 + \left(1-\varepsilon_j\right)q_{i,j} \tag{5}$$

where $q_{i,j}$ is the irradiation on surface zone $A_j$ and $T_j$ is its temperature.

The radiative heat flux on surface zone $A_j$ is:

$$q_{R,j} = \left(q_{i,j} - q_{o,j}\right) \tag{6}$$

where, according to the zonal method, the irradiation $q_{i,j}$ is computed by:

$$q_{i,j} = \frac{1}{A_j}\left(\sum_{\gamma=1}^{\Gamma}\overrightarrow{g_\gamma s_j}\sigma T_\gamma^4 + \sum_{k=1}^{K}\overrightarrow{s_k s_j} q_{o,j}\right) \tag{7}$$

The gas-to-surface and the surface-to-surface, $\left(\overrightarrow{g_\gamma s_j}\right)$ e $\left(\overrightarrow{s_k s_j}\right)$ are given by:

$$\overrightarrow{g_\gamma s_j} = \sum_{i=0}^{I} C_{e,i}\left(T_\gamma\right)\left(\overline{g_\gamma s_j}\right)_i \tag{8}$$

$$\overrightarrow{s_k s_j} = \sum_{i=0}^{I} C_{e,i}(T_k)\left(\overline{s_k s_j}\right)_i \tag{9}$$

where $\left(\overline{g_\gamma s_j}\right)$ and $\left(\overline{s_k s_j}\right)$ are the gas-to-surface and the surface-to-surface direct-exchange areas for a gray gas with absorption coefficient of $a_i$.

An important relation for the direct-exchange areas for each gray gas $a_i$ is given by:

$$4V_\gamma a_i = \sum_{\gamma^*=1}^{\Gamma}\left(\overline{g_\gamma g_{\gamma^*}}\right)_i + \sum_{j=1}^{J}\left(\overline{g_\gamma s_j}\right)_i \tag{10}$$

$$A_j = \sum_{\gamma^*=1}^{\Gamma}\left(\overline{s_j g_{\gamma^*}}\right)_i + \sum_{k=1}^{K}\left(\overline{s_j s_k}\right)_i \tag{11}$$

The above relation guarantees global conservation of the radiative energy in the enclosure. Since the computation of the direct-exchange areas involves the numerical calculation of integral terms, the above relations will be only approximately obeyed. For most of the pairs of zones, the error was as low as 0.1 %. Even so, for convergence of the numerical solution, it was necessary to multiply all the direct-exchange areas by a factor that forced the equality in Eqs. (10) and (11).

For the solution of the gas temperature from the energy conservation in Eq. (1) the following boundary conditions are adopted: at the tube entrance, $x = 0$, $r \geq 0$, the gas dimensionless temperature is uniform and equal to $t = 1.0$; at the tube wall, $x \geq 0$, $r = 1.0$, the gas dimensionless temperature corresponds to the tube wall dimensionless temperature, $t_s = T_s/T_g$. Taking advantage of the axis-symmetry, for $x \geq 0$, $r = 1.0$, the gas temperature gradient at is null, $\partial t/\partial r = 0$.

The global energy balance must be verified for each solution, as given by:

$$\dot{m}c_p\left(T_S - T_e\right) = \int_A q_T \, dA \tag{12}$$

where $q_T = q_R + q_C$, is the total heat flux on the tube wall, obtained by the sum for the radiative and convective heat fluxes, $q_R$ and $q_C$, respectively. Since the gas enters with a temperature above the tube wall temperature, its temperature will decrease as it flows in the tube. The global energy conservation states that the decrease in the gas enthalpy is given by integration of the heat flux on the tube wall.

## 3. Smoketube steam generators

This type of steam generator can be divided into two types: vertical and horizontal. This work is focused on small and medium size, horizontal steam generators having a bank of tubes that are mounted on the headers. Gaseous combustion products flow in the interior of the tubes and transfer heat to the tube, where water in the outside is vaporized to form vapor. The tubes of medium and medium size steam generators have diameters ranging from 2 to 3 inches, while large size units can have diameter of 4 inches. The gases flow velocities range from 10 to 40 m/s to allow operation in the turbulent regime, where the heat transfer coefficient is more elevated (Shields, 1961). There is nowadays a large variety of sizes and configurations of steam generators to cover the different applications. Table 1 presents some general specifications of typical steam generators.

Table 1. Typical specification of industrial steam generators
(Shields, 1961; Babcock and Wilcox, 1972).

| **Steam Generators** | | | |
|---|---|---|---|
| Size | Small | Medium | Large |
| Load (kg of vapor per hour) | 200 - 3500 | 3500 - 20000 | 20000 – 75000 |
| Heating surface (m²) | 6 - 85 | 85 - 500 | 500 – 2000 |
| Work pressure (atm) | 1 - 10 | | 10 – 25 |

In this work, tubes having diameters ranging from 2 to 3½ inches were considered, according to ASTM Standards (ref. A178, A179, A192, A210 and A214). The length of the tubes depends on the specified vapor generation capacity. The length of the tubes was kept at the value of 5.0 m for all cases presented in this work.

## 4. Numerical solution

The determination of the gas temperature distribution depends on the solution of Eq. (1). The coupling between the convection and radiation mechanisms adds considerable complexity to the problem. An iterative numerical procedure is proposed in this work. The diffusive-advective terms of Eq. (1) are tackled with the Flux-Spline control volume (Varejão, 1979). The method is a development of the conventional control volume method, in the sense that the diffusive and advective fluxes are assumed to vary linearly between two grid points instead of being constant (Patankar, 1980). The choice of the method aimed at allowing a less refined grid resolution for the diffusive-advective terms, still keeping an adequate accuracy, so that each control volume and radiative zone could be coincident. If a much refined grid for the diffusive-advective terms were used, it could make the computation time of the zonal method excessively expensive. In addition, the grid mesh in the $x$ direction was set as uniform, that is $\Delta x$ constant, for it allows the reduction of the time to compute all pairs of direct-exchange areas by symmetry. In the radial direction, the grid was non-uniform to observe the steep variations on the gas temperature and velocity close to the tube wall.

In all cases, the domain in the radial direction, within the interval $0 \le r \le 0.5$, was divided into $M$ = 10 elements, being more refined in the region close to the surface. The closest element to the tube wall was set to have the same dimension of the laminar sublayer, $\Delta r = 5y^+$, and the size of the other elements followed an exponential increase. Since different dimensional lengths were considered, different numbers of elements in the axial direction were employed, so that $\Delta x$ = 1.0. That is, for $L/D$ = 60, $N$ = 60; for $L/D$ = 100, $N$ = 100; and so on. With this grid choice, the solution proved to be grid independent.

## 5. Results and dicussion

Results for the heat transfer are presented in terms of the convective, radiative and total Nusselt number, indicated by $Nu_C$, $Nu_R$ and $Nu_T$, respectively. The convective Nusselt number is given by:

$$Nu_C\left(x\right) = -\frac{1}{t_m\left(x\right)-t_s}\left.\frac{\partial t\left(x,r\right)}{\partial r}\right)_{r=0.5} \tag{13}$$

where $t_m(x)$ is the gas dimensionless bulk temperature at position $x$; $t_s$ is the tube wall dimensionless temperature, and is uniform in this problem. The radiative Nusselt number is given by:

$$Nu_R\left(x\right) = -\frac{1}{N_{CR}}\frac{1}{t_m\left(x\right)-t_s}\frac{q_o\left(x\right)-q_i\left(x\right)}{\sigma T_o^4} \tag{14}$$

The radiative Nusselt number $Nu_R\left(x\right)$ takes into account the radiation exchanges between the surface element with all the gas elements, with the other surface elements and with the inlet and outlet reservoirs. The total Nusselt number, $Nu_T$, considers both mechanisms, radiation and convection. Thus,

$$Nu_T\left(x\right) = Nu_R\left(x\right) + Nu_C\left(x\right) \tag{15}$$

For pure convective heat transfer, the total and the convective Nusselt numbers are the same.

### 5.1 Thermal radiation effect

For a study case, it is considered that the gas inlet and the surface temperatures are $T_g$ = 1500 K and $T_s$ = 500 K, the tube length and diameter are $L$ = 5.0 m and $D$ = 8.336 cm, so that the dimensionless length is $L/D$ = 60, and the Reynolds number is Re = 10000. These conditions are typical of smoketube steam generators. Two gaseous mixtures at a total pressure of 1.0 atm are considered: gas mixture 1 ($p_{H2O}$ = 0.2 atm, $p_{CO2}$ = 0.1 atm, and $p_{N2}$ = 0.7 atm) and gas mixture 2 ($p_{H2O}$ = 0.2 atm, $p_{CO2}$ = 0.1 atm, and $p_{N2}$ = 0.8 atm).

Figures 3 and 4 presents the total Nusselt number for the pure convective and the combined mode heat transfer for gas mixtures 1 and 2, respectively. The figures show that for both gas mixtures, the inclusion of the radiation mechanism causes a considerable elevation on the total Nusselt number. Figures 3 and 4 also present the radiative Nusselt number for both cases. In opposition to the convective Nusselt number, the radiative Nusselt number $Nu_R$ does not reach a constant value characterizing a radiative thermal development. In fact, it decays along the tube length until the proximity of the tube end, where the outlet reservoir, modeled as a blackbody surface at the gas outlet temperature, emits considerable amount of energy to the surface. The same elevation on the radiative Nusselt number is observed in the proximity of the inlet reservoir. As a consequence, the total Nusselt number also did not reach a constant, and so one no longer observes a thermal development when thermal radiation is present.

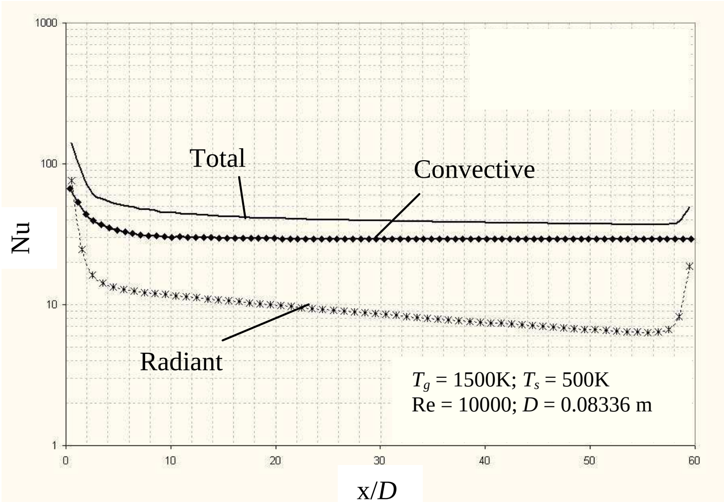


Figure 3. Total, convective and radiative Nusselt numbers. Gas mixture 1.

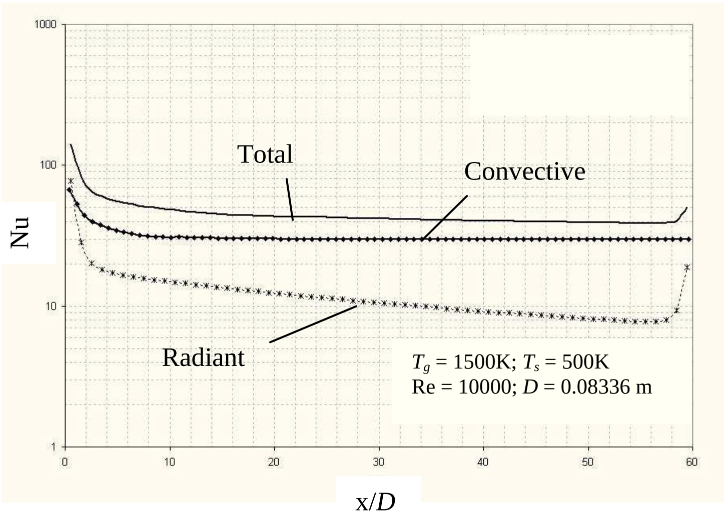


Figure 4. Total, convective and radiative Nusselt numbers. Gas mixture 2.

Figures 5 and 6 present the gas bulk temperature for the pure convection and for the combined mode heat transfer for gas mixtures 1 and 2, respectively. As it can be observed, the gas bulk temperature decreases with the inclusion of the thermal radiation into the solution, since it adds another mechanism to transfer energy to the tube wall surface. When thermal radiation is taken into account, the computed heat transferred increased in about % 15 and % 20 in comparison to the pure convective heat transfer case for gas mixtures 1 and 2, respectively. As it will be seen, the increase can be much more relevant for tubes with larger diameters, when thermal radiation can be the dominant mechanism in the heat transfer process.

Figures 7 and 8 present the local convective Nusselt numbers for gas mixture 1 and gas mixture 2. The figures present the convective Nusselt number when thermal radiation is included and when it is neglected. It can be observed that for both mixtures the effect of thermal radiation on $Nu_C$ was small. As seen, the convective Nusselt number suffered only a small increase. Another interesting result shown in the figures is that the local convective Nusselt in the presence of thermal radiation also reached a constant value, so the thermal development of the convective heat transfer in internal flow continues valid. One difference, though, is that the thermal development was somewhat retarded when thermal radiation was included in the analysis. For pure convection heat transfer, the thermal development was reached in about ten diameters, $x/D \cong 10$, a result commonly presented in the literature (Kays and Crawford, 1980). In the presence of thermal radiation, the convective thermal development is delayed to about eighteen diameters, $x/D \cong 18$. Although this solution considered the flow to be developed in the entrance of the tube, it can be inferred from the results in Figs. 7 and 8 that the convective Nusselt numbers including or not the thermal radiation would also be close when considering the development of the fluid flow in the tube.

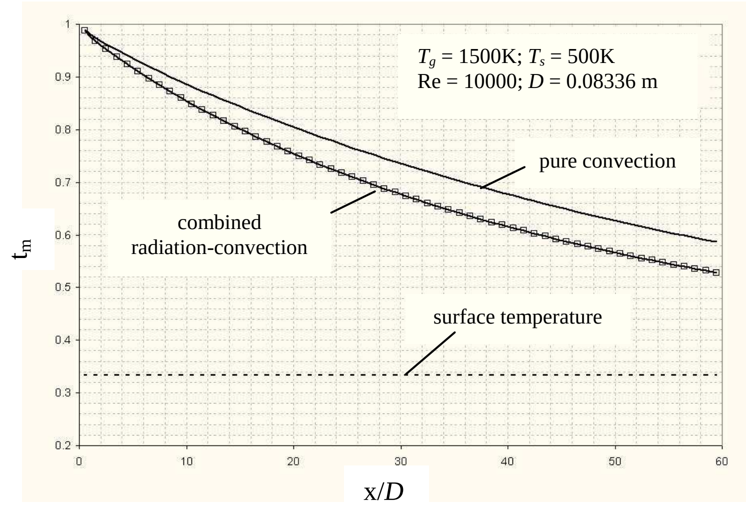


Figure 5. Gas bulk temperature for pure convection and combined radiation-convection heat transfer.
Gas mixture 1.

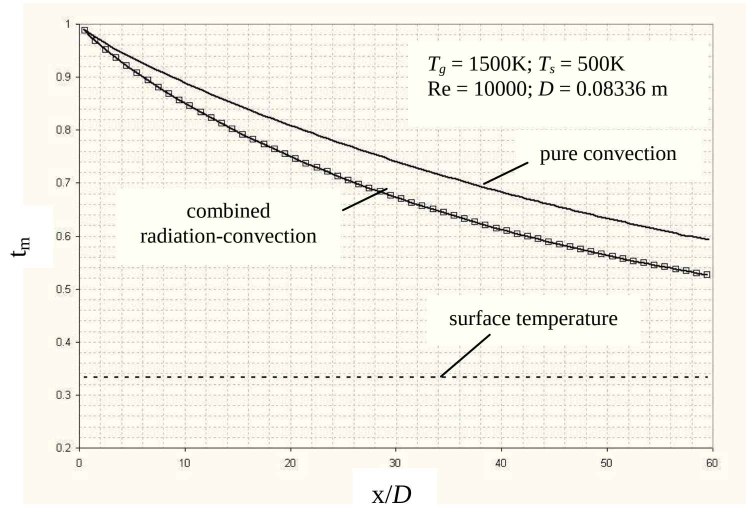


Figure 6. Gas bulk temperature for pure convection and combined radiation-convection heat transfer.
Gas mixture 2.

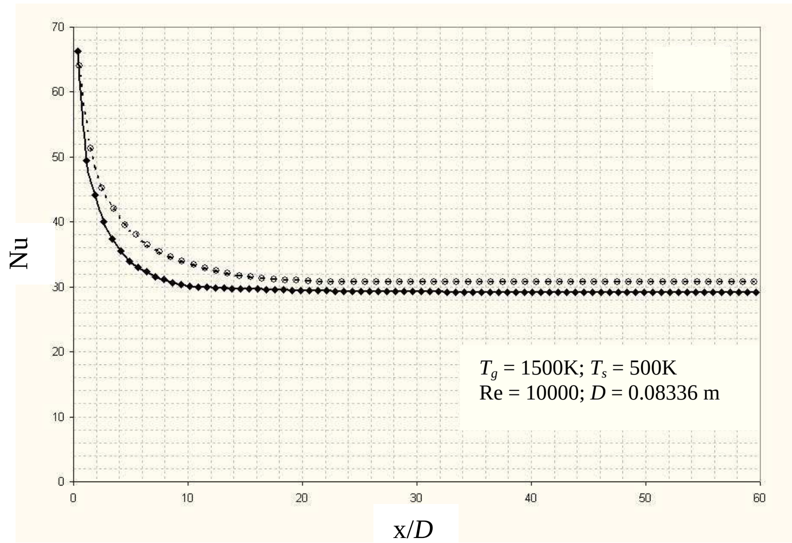


Figure 7. Local convective Nusselt numbers for pure convection and combined radiation-convection heat transfer.
Gas mixture 1.

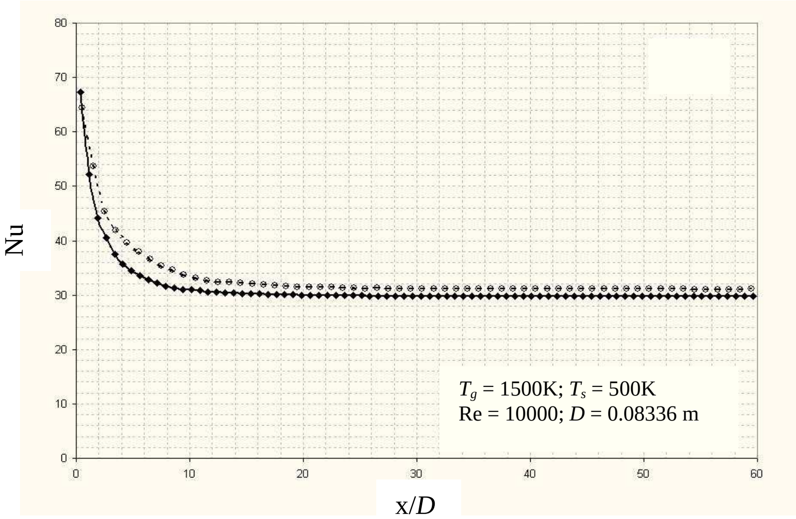


Figure 8. Local convective Nusselt numbers for pure convection and combined radiation-convection heat transfer. Gas mixture 2.

In many engineering applications, such as the sizing of the steam generator, it is of more interest for the engineer to determine the average Nusselt number, $\overline{Nu_C}$, which is given by the integration of the local Nusselt number $Nu_C$ along the tube length. Table 2 presents the average convective Nusselt number $\overline{Nu_C}$ for pure convection and combined radiation-convection heat transfer. Values of the Reynolds numbers and of the dimensionless lengths $L/D$ are typical of smoketube steam generators. As seen in the table, the change in the value of $\overline{Nu_C}$ is very small when thermal radiation is included and not included in the analysis.

Table 2. Comparison of the average convective Nusselt numbers for pure convection and combined radiation-convection heat transfer. Gas mixtures 1 and 2.

| | $T_g$ (K) | $T_s$ (K) | $L/D$ | $Re_D$ | Pure convection heat transfer $\overline{Nu_C}$ | Combined heat transfer $\overline{Nu_C}$ | Deviation (%) |
|---|---|---|---|---|---|---|---|
| Gas mixture 1 | 1000 | 500 | 60 | 10000 | 31.35 | 31.28 | 0.22 |
| | 1200 | 600 | 110.5 | 25000 | 62.49 | 62.34 | 0.24 |
| | 1500 | 500 | 77.6 | 30000 | 74.31 | 73.71 | 0.81 |
| | 2000 | 500 | 77.6 | 20000 | 54.66 | 53.96 | 1.28 |
| Gas mixture 2 | 1000 | 500 | 60 | 10000 | 32.15 | 31.56 | 1.87 |
| | 1200 | 600 | 110.5 | 25000 | 63.84 | 63.42 | 0.66 |
| | 1500 | 500 | 77.6 | 30000 | 76.29 | 75.31 | 1.28 |
| | 2000 | 500 | 77.6 | 20000 | 57.05 | 55.94 | 1.94 |

**5.2 Effect of the diameter**

Table 3 presents the average convective and the radiative Nusselt numbers for six different values of diameters for gas mixtures 1 and 2. In all cases that are shown, the tube length was kept the same, $L$ = 10 m. To keep a basis of comparison for the heat transfer in a bundle of tubes having different diameters, it is considered the same total area for all the chosen diameters. So, for the tube having a diameter $D$ = 2.0 m, one single tube was considered, $N$ = 1; for $D$ = 1.0 m, $N$ = 2; and so on up to the tube having diameter $D$ = 0.0625 m, $N$ = 32 tubes were considered. In all cases, the total gas mass flow rate is 2 kg/s. Considering that the flow rate is equally distributed in the tubes of the bundle, the same Reynolds number will be obtained in each tube.

As seen in Table 3, for the largest diameters, radiation is the dominant heat transfer mechanism, while the trend is the opposite for the smallest diameters: heat transfer is mainly governed by convection. This results from the strong dependence of the gas radiation with the system optical thickness, which is given by the product of the gas absorption coefficient $a$ and the system characteristic length, which in this case is the diameter $D$. Since the gas is non-gray, the absorption coefficient can be taken as an average of the gray gases from the WSGG model. Since for the different tubes, the Reynolds number was kept approximately the same, the convective Nusselt number should be kept approximately constant, except for the effect of thermal radiation. Table 3 shows that, while the effect of thermal radiation on the convective Nusselt number for smaller tubes is of minor importance, as discussed in the previous section, it becomes more important for larger diameters. Since the total Nusselt number increases with the diameter, it is interesting to verify whether using tubes with larger diameter increases the heat transfer. Figures 9 and 10 present the heart transfer in the bundle of tubes as a function of the number of tubes (and indirectly on the tubes diameters), considering gas mixtures 1 and 2, respectively. As seen, the heat transfer is larger for bundles with larger number of tubes, or tubes with smaller diameters. This is explained by the fact that the average heat transfer coefficient, computed by $\overline{h_T} = \overline{Nu_T} k / D$, decreases with the diameter. That is, the increase of $\overline{Nu_T}$ with the diameter $D$ is not enough to compensate $D$ in the denominator. The results shown in Figs. 9 and 10 indicate that the choice of small diameter tubes is more interesting from the sole point of view of heat transfer. The optimum choice needs to account for other costs involved with the choice of a smoketube generator having a large number of tubes.

Table 3. Average convective and radiative heat transfer for different numbers of tubes.

| | | Gas Mixture 1 | | Gas Mixture 2 | |
|---|---|---|---|---|---|
| *Number of ducts* | $D_i$(m) | $\overline{Nu_C}$ | $\overline{Nu_R}$ | $\overline{Nu_C}$ | $\overline{Nu_R}$ |
| 1 | 2.0000 | 113.61 | 750.17 | 122.01 | 831.20 |
| 2 | 1.0000 | 94.70 | 293.21 | 98.17 | 347.77 |
| 4 | 0.5000 | 87.69 | 105.64 | 89.52 | 132.26 |
| 8 | 0.2500 | 84.04 | 37.02 | 85.86 | 47.95 |
| 16 | 0.1250 | 84.61 | 13.51 | 82.79 | 17.11 |
| 32 | 0.0625 | 87.52 | 5.09 | 91.08 | 6.37 |

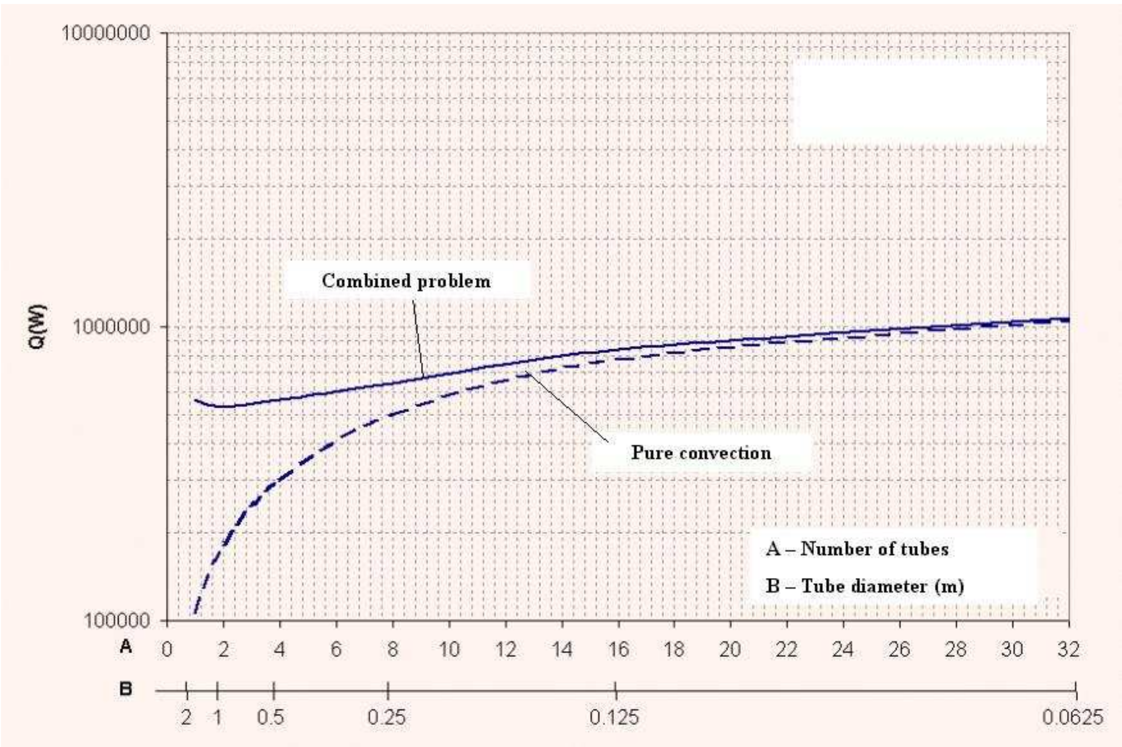


Figure 9. Heat transfer rate for different number of tubes in the bundle. Gas mixture 1. $T_g/T_s = 2$.

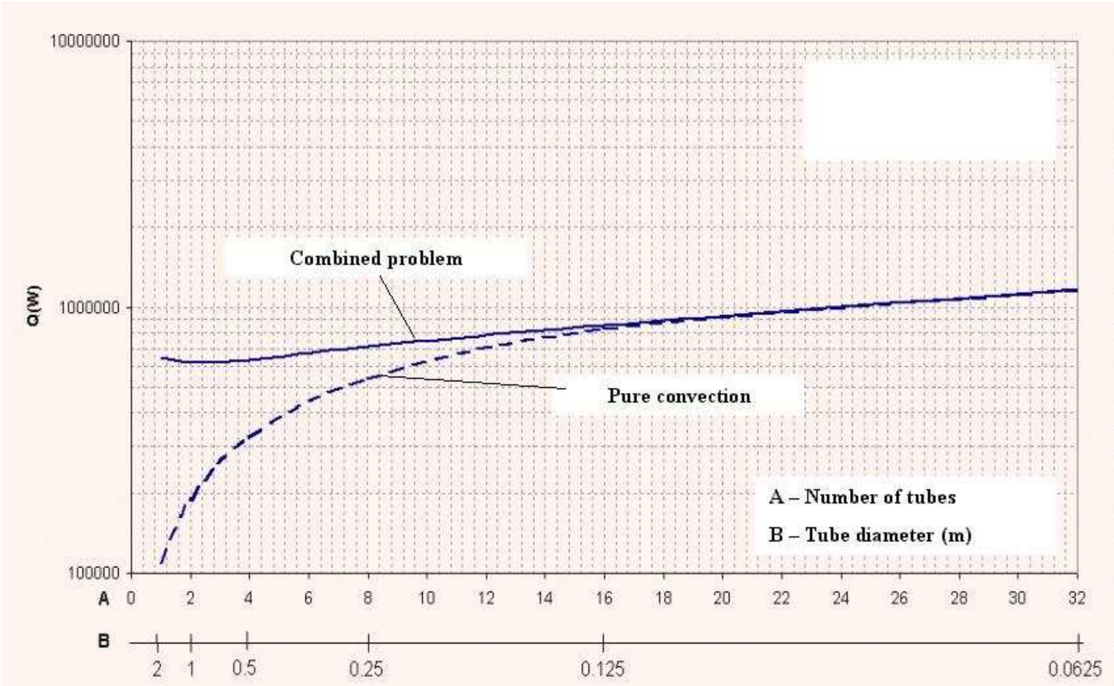


Figure 10. Heat transfer rate for different number of tubes in the bundle. Gas mixture 2. $T_g/T_s = 2$.

## 6. Conclusions

This paper presented a numerical analysis of the thermal radiation on the heat transfer of turbulent flow of participating gases in the interior of circular tubes. The analysis was mostly focused on the conditions found in smoketube steam generators. In this case, the tube wall temperature can be assumed uniform and equal to the vapor saturation temperature at a given operational pressure. The diffusive-advective and the radiative terms of the energy equation were treated by the Flux-Spline control volume and by the zonal methods, respectively. Two gaseous mixtures were considered, being typical products of the combustion of methane and fuel oil. Their radiative properties were modeled by the weighted-sum-of-gray-gases.

Results were presented for the total Nusselt number, which is given by the sum of the convective and the radiative Nusselt numbers, and for the gas bulk temperature. Including thermal radiation into the analysis led to an increase in the total Nusselt number and, therefore, a decrease in the gas outlet bulk temperature in comparison to the case of purely convective heat transfer. The convective Nusselt number was less affected with the inclusion of the thermal radiation for typical diameters that are used in smoketube steam generators. In fact, the convective Nusselt number obtained can be well approximated by the usual correlations available for pure convection heat transfer in tubes. For larger diameters, the analysis showed that thermal radiation can have a considerable effect on the convective heat transfer coefficient and be the dominant heat transfer mechanism. This is explained by the fact that the mechanism is strongly dependent on the optical thickness of the participating medium, which in this case is given by the product of the tube diameter and an average value of the gas absorption coefficient. The numerical analysis was also employed to compare the heat transfer from a bank of tubes with different diameters, but having the same total area and mass flow rate. Although the total Nusselt increased with the diameter, due to the effect of the thermal radiation, the results showed that the heat transfer is more effective for the banks having small tube diameters.

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## 8. References

Alturki, A. M., Smith, T. F., 1987, "Radiative and Convective Transfer in a Cylindrical Enclosure for a Gas/Soot Misture, J. Heat Transfer, Vol. 109, pp. 259-262.
ASTM A178 / ASME SA178, Electric Resistance Welded Carbon Steel Boiler Tubes.
ASTM A179, Seamless Cold Drawn Low Carbon Steel Heat Exchanger and Condenser Tubes.
ASTM A192 / ASME SA192, Seamless Carbon Steel Boiler Tubes for High Pressure Service.
ASTM A210 / ASME SA210, Seamless Medium Carbon Steel Boiler and Superheater Tubes.
ASTM A214 / ASME SA214, Electric Resistance Welded Carbon Steel Heat Exchanger and Condenser Tubes.

Campo, A., e Schuler, C., 1988, "Thermal Radiation and Laminar Forced Convection in a Gas Pipe Flow", Wärme-und Stoffübertragung 22, pp. 251-257.

Echigo, R., Hasegawa, S., e Kamiuto, K., 1975, "Composite Heat Transfer in a Pipe with Thermal Radiation of Two-Dimensional Propagation - in Connection with the Temperature Rise in Flowing Medium Upstream from Heating Section", Int. J. Heat Mass Transfer, Vol. 18, pp. 1149-1159.

Einstein, T.H., 1963, "Radiant Heat Transfer to Absorbing Gases Enclosed in a Circular Pipe with Conduction, Gas Flow, and Internal Heat Generation", NASA TR R-156, 28 pp.

França, F. H. R., e Goldstein Jr., L., 1998, "Effects of Temperature and Geometry on the Heat Transfer from Turbulent Flow of a Participating Gas Through a Duct", Heat Transfer Engineering, Vol. 19, pp. 25-33.

Khan, Y. U., Lawson, D. A., Tuckers, R. J., 1997, "Simple Models of Spectral Radiative Properties of Carbon Dioxide", Int. J. Heat Mass Transfer, Vol. 15, pp. 3581-3593.

Kays, W. M., e Crawford, M. E., 1980, "Convective Heat and Mass Transfer", McGraw Hill Book Company, 2nd edition.

Patankar, S. V., 1980, "Numerical Heat Transfer and Fluid Flow", Hemisphere Publishing Corporation.

Reichardt, H., 1951, Arch. Ges. Warmetechnik, Vol. 6/7, citado em Kays e Crawford (1980).

Schuler, C, e Campo, A., 1988b, "Numerical Prediction of Turbulent Heat Transfer in Gas Pipe Flows Subject to Combined Convection and Radiation", Int. J. Heat and Fluid Flow, Vol. 9, No. 3, pp. 308-315.

Sediki, E., Soufiani A., Sifaoui, M. S., 2002, "Spectrally Correlated Radiation and Laminar Forced Convection in the Entrance Region of a Circular Duct", Int. J. Heat and Mass Transfer, Vol. 45, pp. 5069-5081.

Seo, T., Kaminski, e Jensen, M. K., 1994, Combined Convection and Radiation in Simultaneously Developing Flow and Heat Transfer with Nongray Gas Mixtures", Numerical Heat Transfer, Part A, Vol. 26, pp. 49-66.

Shields, C. D., 1961, "Boiler: Types, Charateristics and Functions", F. W. Dodge Co. New York.

Sika, J., 1991, "Evaluation of Direct-Exchange Areas for a Cylindrical Enclosure", ASME Journal of Heat Transfer, Vol. 113, pp. 1040-1044.

Smith, T. F., Byun, K. H., Ford, M. J., 1986, "Heat Transfer for Flow of an Absorbing, Emitting, and Isotropically Scattering Medium Through a Tube with Reflecting Walls, $8^{th}$ Int. Heat Transfer Conf., Vol. 2, pp. 803-808.

Smith, T. F., Shen, Z. F., e Friedman, J. N., 1982, "Evaluation of Coefficients for the Weighted Sum of Gray Gases Model", ASME Journal of Heat Transfer, Vol. 104, pp. 602-608.

Smith, T. F., Shen, Z. F., e Alturki, A. M., 1985, "Radiative and Convective Transfer in a Cylindrical Enclosure for a Real Gas", ASME Journal of Heat Transfer, Vol. 107, pp. 482-485.

The Babcock & Wilcox Company, 1972, "Steam: Its Generation and Use". 38th edition. New York.

Varejão, L. M. C., 1979, "Flux-Spline Method for Heat, Mass, and Momentum Transfer". Minnesota: Faculty of the Graduate School of the University of Minnesota, 235 p. Tese (Doutorado).

Viskanta, R., 1998, "Overview of Convection and Radiation in High Temperature Gas Flows", Int. J. Engineering Science, Vol. 36, pp. 1677-1699.

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