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Assessment of a Radiative Heat Transfer Model for Gas Turbine Combustor Preliminary Design

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The early stages of gas turbine combustor design represent a challenging combination of analytical analysis, numerical simulation, and rig testing. The objective of this work is the development of a versatile radiation submodel within the constraints of a preliminary gas turbine combustor simulation tool. A network approach forms the basis of the design solution algorithm, dividing the domain into a number of independent semiempirical interconnected subsections. A novel application of the pressure correction methodology more commonly employed by computational fluid dynamics codes is utilized to solve the combined continuity equation and pressure-drop/flow-rate relationships. A coupled conjugate heat transfer analysis is employed to determine heat transfer to the combustor liner. Radiation represents the most difficult mode of heat transfer to simulate in the combustor environment. A novel variation of the discrete transfer radiation model is presented and validated for use within the network solver. The effect of the radiation model on the prediction of liner wall temperature is evaluated in an annular gas turbine combustor at a typical high-power operating condition. The importance of radial distributions of temperature and soot are evaluated by examining the flametube wall heat transfer mechanism.

Nomenclature

Cp = specific heat at constant pressure
C_0 = soot index of refraction constant
C_2 = Planck's second constant
d = external mass flow into node
$\mathrm{d}A$ = surface area associated with ray origin
E = energy source term
f_v = soot volume fraction
h = convective heat transfer coefficient
I = radiative intensity
I_w = incident radiative intensity
K = extinction coefficient
k_a = absorption coefficient
k_s = scattering coefficient
$\dot{m}$ = mass flow rate
P = pressure
Q = volumetric flow rate
q_+ = outgoing radiative flux
q_- = incident radiative flux
R = conductive heat transfer coefficient
s = distance
T = temperature
Δs = ray volume path length
ε = emissivity
θ = polar angle
ρ = density
σ = Stefan–Boltzmann constant
ϕ = azimuthal angle
Ω = ray solid angle

Subscripts
f = film
g = gas
n = entering volume element
$n + 1$ = leaving volume element
s = soot
T = total
w = wall

Introduction

GAS turbine combustor development involves a lengthy process of analysis, beginning with the conception of overall performance requirements. Given an idea of the sizing requirements of a combustor, a basic geometrical layout may be developed, providing the first step to detailed empirically-based design validation, followed by more complex computational fluid dynamics (CFD) simulations and rig testing. Despite the successes of detailed CFD simulations, empirically based models are still routinely employed as part of the gas turbine combustor design process. Many of today's new combustors are extensions of well-proven designs, and modified scale versions of these designs provide the first pass for the design validation procedures.

Semiempirical models have the advantage of rapid execution times, on the order of a few seconds or minutes. This is a significant advantage for the design engineer as it allows optimization with relatively little time expenditure. The more accurate this initial design process, the more rapid, and hence less costly, the following phases of design. The limitations of such tools include their restriction to simple geometries, being cumbersome to set up, and having difficulties with convergence when applied to more irregular flowfields. Network methods have the ability to model complicated and unusual geometries effectively and with little numerical difficulty while retaining the advantage of rapid execution.[1]

Despierre et al.[2] demonstrated the effectiveness of the network scheme described here to optimize a real gas turbine combustor using a genetic optimization algorithm to control the network solver. The speed of the network code allowed the complete flow and conjugate heat transfer simulation of 1000 combustor alternatives within an 8-h period, running on an entry-level Silicon Graphics Indy Workstation. Despierre et al.[2] were able to illustrate the success of this technique in optimizing the geometry for liner wall air cooling requirements and overall combustor pressure-drop within the framework of a series of realistic flowfield constraints.

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A network consists of a number of independent subflows linked together to model a physical process. The method has been used with success in modeling large pipe networks.[3] Because the orientation of the subflows are independent, multidimensional features such as total-static fed cooling rings may be modeled with ease. Each subflow is defined by a semiempirical pressure-drop/flow rate relationship and a heat transfer relationship.

Radiation represents the most difficult mode of heat transfer to simulate in the combustor environment. Previous comparisons, using a simplified heat transfer approach, have shown qualitative agreement, but highlighted the need for a more broadly applicable radiation procedure.[1] An effort has been made to extend the predictive radiative capability of the code beyond simple conventional empirical radiation correlations, improving the flexibility and accuracy of the radiation calculation to that of the underlying network solver. The approach includes a constrained equilibrium model, mixing/recirculation models, and discharge and film-cooling models. The flexibility of this model framework allows the relatively simple modeling of complicated geometries.

Radiation

Radiation is governed by a complex integro-differential equation, which must be simplified to be solved economically. Radiative heat transfer from the flame and combustion products to the surrounding walls may be computed, given knowledge of the radiative properties and temperature distributions within the medium. However, these requirements are generally unknown, and therefore, the total energy and radiative energy conservation equations must be coupled.

A number of solution strategies exist for radiative heat transfer. The simplest make use of semiempirical correlations to simulate the radiation in a one-dimensional domain.[4] The application of these models require sweeping assumptions concerning system geometry, gas temperature, and radiative properties. The accuracy of the correlations is limited within complex geometries and at elevated pressures, regimes beyond those used to originally formulate the relationships.

More complex models take direct account of the geometry. A balance must be made between solution accuracy and solution economy.[5,6]

Statistical methods such as the Monte Carlo scheme provide solutions approaching those of the exact solution.[7] The method has been successfully employed in complex geometries and it accounts for spectral effects. A factor limiting the use of this method is the excessive computational time required for practical engineering calculations.

The zonal method has been widely used for solutions to practical problems.[7] Direct exchange area factors between the surface and volume elements must be computed, and the total exchange area calculated. This becomes a time-consuming task as the geometry increases in complexity. Further difficulty arises for an absorbing and emitting medium as the attenuation of radiation along a path connecting area elements must be accounted for in the calculation of exchange areas.

Radiation intensity is dependent upon location, the direction of radiation propagation, and wavelength. The problem is complicated by the angular dependence of the intensity. If the assumption is made that the intensity is uniform over specified solid angles, the problem may be simplified, and the integro-differential radiative transfer equation is reduced to a series of coupled linear differential equations. This procedure forms the basis of the flux methods.[7] They include multiflux models, moment methods, spherical harmonics P_N approximations, and discrete-ordinate models. Essentially, they only differ in the derivation of weighting coefficients for the intensities in each direction. Although these methods provide accuracy with computational economy, they involve mathematical complexity.

Hybrid solution strategies combine the best of existing models. The discrete transfer method is such a model, combining elements of the zonal, Monte Carlo, and discrete-ordinate methods.[8] A variation on the original formulation of the discrete transfer method is presented in this paper. Axisymmetric discrete transfer is a formulation specifically for three-dimensional ray tracing calculations achieved using an underlying axisymmetric description of the geometry and combustion products' thermodynamic properties.

Radiation in Gas Turbine Combustors

A large proportion of the heat transferred to the liner wall from the hot combusting gases and particles within the combustor flametube is by radiation. It follows therefore that an accurate liner heat balance may only be performed with the assistance of a reliable radiation model.

Predictions of radiative flux are dependent upon the distribution of gas temperature and radiative properties. A number of examples exist of the implementation of radiation models to compute radiative flux in gas turbine combustors. Rizk and Mongia[9] were able to achieve satisfactory wall temperature results using simple semiempirical correlations for radiative transfer in a gas turbine combustor. The correlations were implemented as part of a three-dimensional analysis. Menguc et al.[10] successfully employed a spherical harmonics (P_1 and P_3) model on a cylindrical enclosure at gas turbine combustor conditions. It was necessary to make some assumptions in obtaining a soot concentration profile, allowing the effects of differing profiles to be evaluated.

Carvalho and Coelho[11] presented a three-dimensional model of a can combustion chamber. The discrete transfer method was employed to model the radiative transfer within the can. A simple conjugate heat transfer scheme was coupled to the CFD solution to directly calculate wall temperatures from the computed internal heat fluxes.

Bai and Fuchs[12] performed a numerical analysis of the radiative heat transfer in the reacting flow of a gas turbine combustor. The discrete transfer method was used to compute the radiative heat source terms. The mean temperature profile exhibited errors of up to 10% when the effects of radiation were neglected. The effect of turbulent temperature fluctuations in the heat radiation were also evaluated. As expected, the turbulent fluctuations were significant where there were very large amplitude fluctuations. The mean values of temperature presented a reasonable approximation when the levels of temperature fluctuation were globally low. It is inappropriate to attempt to solve transport equations for turbulent kinetic energy with the network approach. Therefore, the effects of turbulent fluctuations on temperature and species concentrations may only be indirectly incorporated via empirical correlations.

Network Algorithm

The domain of interest is modeled by overlaying a network on the system geometry. The network consists of a number of elements and nodes. The elements represent actual physical features in the domain, e.g., duct sections, holes, etc. The nodes join the elements to one another, thus combining independent features into a meaningful overall structure. The overall governing equations are solved within the nodes, while semiempirical relationships may be employed to describe the flow through an element.

The procedure used for obtaining a solution to the flow equations is based upon a pressure-correction methodology. The one-dimensional flow may be compressible or incompressible. The overall governing flow equations are the continuity equation and a pressure-drop/flow rate relationship.[3] The continuity equation may be specified as

$$\sum_{j=1}^{J} \rho_{i,j} Q_{i,j} = -d_i \qquad i = 1, 2, \ldots, J \tag{1}$$

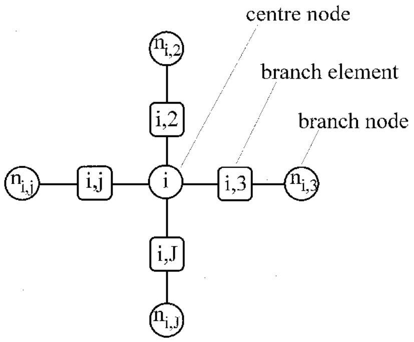

Fig. 1 Network nomenclature.

and the pressure-drop/flow-rate relationship as

$$\Delta P_{i,j} = P_{n_{i,j}} - P_i = \frac{Q_{i,j}}{|Q_{i,j}|} g_{i,j}(\rho_{i,j}) f_{i,j}(|Q_{i,j}|) \tag{2}$$

The functional relationships are derived from semiempirical formulations for combustor features and internal flows. The nomenclature is illustrated in Fig. 1. The relationships are written as coefficients in the overall solution matrix, and simultaneously solved using a direct method.[3]

The energy equation is satisfied by ensuring an enthalpy balance at each node within the network. This may be specified at nodes with branch elements containing mass transfer as

$$T_i = \sum_{j=1}^{J} (E_{i,j} + \dot{m}_{i,j} Cp T_{n_{i,j}})_{\text{inflow bits}} \Big/ \sum_{j=1}^{J} (\dot{m}_{i,j} Cp)_{\text{inflow bits}} \tag{3}$$

A semi-implicit formulation is used to compute node temperatures on boundaries or within walls, i.e., at nodes where the branch elements contain no mass flow. On the flow boundaries, where conduction, convection, and radiation are present, this may be expressed as

$$T_i = \frac{\left(\sum_{j=1}^{J} R_{i,j} T_{n_{i,j}}\right)_{\text{conduction}} + \left(\sum_{j=1}^{J} h_{i,j} T_{n_{i,j}}\right)_{\text{convection}} + [(\varepsilon q_-)_i]_{\text{radiation}}}{\left(\sum_{j=1}^{J} R_{i,j}\right)_{\text{conduction}} + \left(\sum_{j=1}^{J} h_{i,j}\right)_{\text{convection}} + [(\varepsilon \sigma T^3)_i]_{\text{radiation}}} \tag{4}$$

and within the solid where conduction is the only mode of heat transfer

$$T_i = \left(\sum_{j=1}^{J} R_{i,j} T_{n_{i,j}}\right)_{\text{conduction}} \Big/ \left(\sum_{j=1}^{J} R_{i,j}\right)_{\text{conduction}} \tag{5}$$

The heat transfer coefficients in Eq. (4) are evaluated using semiempirical correlations and data for various cooling types found in gas turbine combustors. The effect of film cooling has a significant effect on the wall temperatures, and must be modeled accurately. The calculations take the form of Nusselt number correlations, employing numerous experimental data. A wide range of cooling effects are modeled, including Z-rings, lipped rings, slots, effusion patches, and transply patches. Heat pickup by the fluid moving through the flametube wall is computed. Multiple films at the same location, originating from different features, are accounted for when computing the effective heat transfer coefficient. A constrained equilibrium calculation is used to obtain a mean flametube combustion gas temperature. Full details of the network methodology may be found in Stuttaford and Rubini.[1]

Discrete Transfer Radiation Model

The discrete transfer (DT) method combines the advantages of other radiation procedures to provide a numerically exact and flexible solution algorithm. It allows for simple coupling to the existing overall flow and heat transfer solver. A further important advantage is the ease with which radiation properties models can be incorporated into the DT calculation.

The transfer equation for thermal radiation along a ray in a direction s may be written as[8]

$$\frac{dI}{ds} = -(k_a + k_s)I + k_a \frac{E_g}{\pi} + \frac{k_s}{4\pi} \int_{4\pi} P(\Omega, \Omega') I(\Omega') \, d\Omega' \tag{6}$$

where $E_g = \sigma T_g^4$ represents the emission of radiant energy, and $P(\Omega, \Omega')$ represents the probability that the incident radiation in the direction Ω' will be scattered into the increment of solid angle $d\Omega$ about Ω. The assumption is made that the soot is made up of small particles in this study, and so, scattering is neglected. The formulation then reduces to

$$\frac{dI}{ds} = -k_a I + \frac{k_a \sigma T_g^4}{\pi} \tag{7}$$

Given a representative ray, the intensity distribution can be calculated along it. This equation can be integrated to give the recurrence relationship

$$I_{n+1} = (\sigma T_g^4/\pi)(1 - e^{-k_a \Delta s}) + I_n e^{-k_a \Delta s} \tag{8}$$

Thus, the intensity may be calculated stepwise as the ray passes through successive control volumes within the domain. The initial intensity is calculated, assuming that the surface is a gray, Lambert one

$$I_0 = \frac{q_+}{\pi} = (1 - \varepsilon_w)\frac{q_-}{\pi} + \varepsilon_w \frac{\sigma T_w^4}{\pi} \tag{9}$$

Then, at any given point on the boundary, the incoming heat flux caused by radiation is

$$q_- = \int_{2\pi} I_w(\Omega) \cos\theta \, d\Omega \tag{10}$$

Clearly, q_- may only be calculated given knowledge of the initial intensity of the ray impinging on the surface of interest. Thus, the process is iterative with updated initial ray intensities used at the beginning of each new iteration loop. The net wall radiative heat flux is simply the difference between the energy flux away from the surface and toward the surface.

Conventionally, the discretization of the solid angle in Eq. (10) is achieved by equal division of polar and azimuthal angles[13] (as illustrated in Fig. 2). This choice leads to weightings proportional to sin 2θ, which produces a bias to rays closest to $\theta = \pi/4$. Cumber[14] obtained improvements in efficiency for

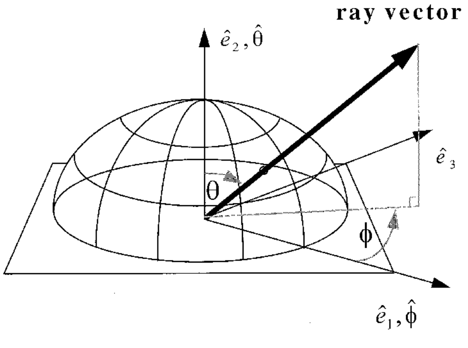

Fig. 2 Discretization of a solid angle hemisphere.

a limited number of rays. A more detailed analysis has been performed by Bressloff et al.,[15] who compared results over a range of ray numbers. Significant improvements in computational efficiency were demonstrated over a range of cases.

Axisymmetric DT

The ray-tracing procedure represents the most complex component in the implementation of a DT model. The problem is unique in this case because a three-dimensional radiation model is required for a pseudo-one-dimensional solver. For the purposes of simplicity, the combustor geometries are assumed to contain an axis of symmetry, for aerospace applications this is a reasonably valid assumption. The combustor geometry is described in axisymmetric terms, while allowing a fully three-dimensional radiation calculation, this is referred to as ADT.

The lengths of every segment of each ray through the control volumes making up the domain must be computed before any intensity calculation can be performed. Although the calculation is complex and time consuming, it needs to be performed only once prior to the main solution procedure. Each surface is described by an analytical expression.

Rays are launched from the axial midpoint of every volume boundary surface. The equations describing the rays are generated in spherical coordinates about the midpoint, which is specified as the origin of the local frame of reference (Fig. 3).

The equations may be transformed from spherical to Cartesian coordinates as follows:[16]

$$\begin{aligned} x' &= R \sin\theta \sin\phi \\ y' &= R \cos\theta \\ z' &= R \sin\theta \cos\phi \end{aligned} \tag{11}$$

where the ray is represented by the length R from the origin to point P.

The local coordinate system is then translated and rotated into the global coordinate system (Fig. 4). In this way, all of the rays and surfaces are described in the exact same global Cartesian coordinate system.

The translation and rotation are represented as

$$\begin{aligned} x &= x' \\ y &= yy + yy' \cos\zeta + zz' \sin\zeta \\ z &= zz + zz' \cos\zeta - yy' \sin\zeta \end{aligned} \tag{12}$$

The equations describing the ray are now solved by simple substitution, and the result, written in parametric form in the global coordinate frame as

$$\begin{aligned} x &= (\sin\theta \sin\phi)t \\ y &= yy + (\sin\theta \cos\phi \sin\zeta + \cos\theta \cos\zeta)t \\ z &= zz + (\sin\theta \cos\phi \cos\zeta - \cos\theta \sin\zeta)t \end{aligned} \tag{13}$$

Thus, a simple description of the ray is obtained dependent only upon its origin (the boundary surface midpoints) and the

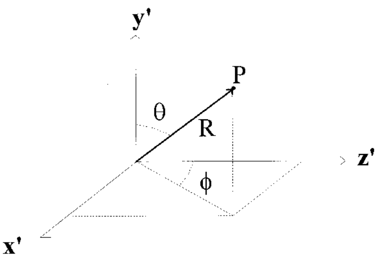

Fig. 3 Transformation from spherical to rectangular coordinate system.

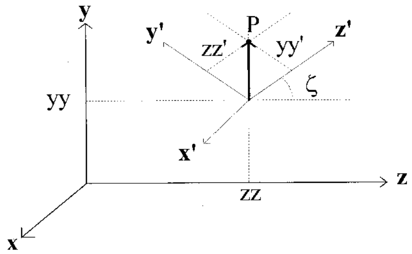

Fig. 4 Rotation and translation from local to global coordinate system.

user-specified polar and azimuthal angles. The boundary surfaces (planes, cylinders, and cones) must also be described by equations in the global Cartesian coordinate frame of reference.[16]

Combustion Products Radiative Properties

Radiative properties of combustion gases present a difficult obstacle in performing an accurate, reliable radiation calculation. The radiative properties are dependent upon temperature, pressure, composition, wavelength, and path length. The concentrations of the constituent species are required to be known, including soot volume fraction. This initial step typically introduces the largest unknowns into the calculation.

The radiative transfer from an opaque wall can often be represented by a simple model of gray emission, absorption, and reflection. Unfortunately, the same assumption cannot be made about a molecular gas, because the radiative properties vary significantly across the gas spectrum, and are strongly dependent upon wavelength. A detailed discussion and comparison of suitable methods for use within DT has been performed by Bressloff et al.[17]

The operating environment of a gas turbine combustor at elevated pressure severely restricts the selection of a gas properties model. Much of the work in this area has been performed at atmospheric pressure, or at least an order of magnitude lower pressure than that found in a typical gas turbine combustor. Unfortunately, the effects of pressure broadening cannot be ignored, and models that account for them must rely on extrapolation. In the present work, the correlation from Leckner[18] was employed for combustion products containing mixtures of H_2O and CO_2. A concise description of the model is provided by Modest.[6]

The luminous component of radiation may far outweigh the nonluminous radiation in some areas of the combustor flametube, particularly in the fuel-rich pockets within the primary zone. At the high pressures present within the gas turbine combustor, the soot particles attain sufficient size to radiate as blackbodies.[4] When a parallel beam of radiation passes through a domain of particles, the strength of the beam decreases exponentially, as follows[4]:

$$I_{n+1}/I_n = \exp(-K\Delta s) \tag{14}$$

Assuming that the scattering is negligible, K is equivalent to the absorption coefficient. If the emissivity is equal to the absorptivity, the emissivity may be defined as[5]

$$\varepsilon_s = (I_n - I_{n+1})/I_n \tag{15}$$

Substituting from Eq. (21), the formulation may be rewritten as

$$\varepsilon_s = 1 - \exp(-k_a\Delta s) \tag{16}$$

where

$$k_a = 3.72 f_v C_0 T / C_2 \tag{17}$$

The preceding formulation is only applicable to situations where very small soot particles exist. The extinction coefficient will increase with increasing particle sizes. The value of C_0 is taken to be 7.0 (Ref. 10). Values range from 3 to 10, depending on the fuel type and test conditions.[5]

The process of soot formation and oxidation is extremely complex. Quantitatively accurate methods for calculating soot production within the operating regimes of a gas turbine combustor have yet to be developed. Soot is not an equilibrium product of combustion.[19] It depends upon physical processes such as atomization, evaporation, and turbulent mixing as much as the chemical kinetics.

Quasiglobal models have been developed to describe the process of soot formation. Najjar and Goodger[20] developed a successful model after studying soot formation in a gas turbine combustor burning kerosene and gas oil. Soot oxidation tends to dominate in the regions downstream of the primary zone, at lower fuel/air ratios, and correspondingly higher temperatures. Mongia[21] supplemented empirical data into multidimensional calculations to predict the sooting process of formation and oxidation within a gas turbine combustor. Sudarev and Antonovsky[22] developed a formulation for soot concentration based on a detailed experimental study of a class of combustors. The accuracy of these models becomes limiting as the operating conditions of the combustor differ significantly from the regime of the formulation of the correlation.

Once the soot and gas emissivities have been computed, the effective total emissivity may be obtained from the following:

$$\varepsilon_T = \varepsilon_g + \varepsilon_s - \varepsilon_g \varepsilon_s \tag{18}$$

The total emissivity calculated using Eq. (18) has been found to give good agreement with values computed using a more detailed nongray analysis.[5]

Validation of the ADT Radiation Model

The validation of the ADT ray-tracing procedure was accomplished by comparison to proven benchmark radiation test cases. Malalasekera and James[23] performed radiation calculations in three-dimensional complex geometries using the DT method. The accuracy of their model was successfully evaluated using exact solutions and the results of Chui et al.[24] The geometry used is shown in Fig. 5. The cylindrical enclosure is 6 m long and 2 m in diameter. The walls are black at a constant uniform temperature of 500 K. The gas temperature varied axially along the cylinder. The results shown in Fig. 5 were obtained for three different absorption coefficients (units of absorption coefficient in the graph are 1/m), using an exact calculation, and the finite volume method of Chui et al.[24]

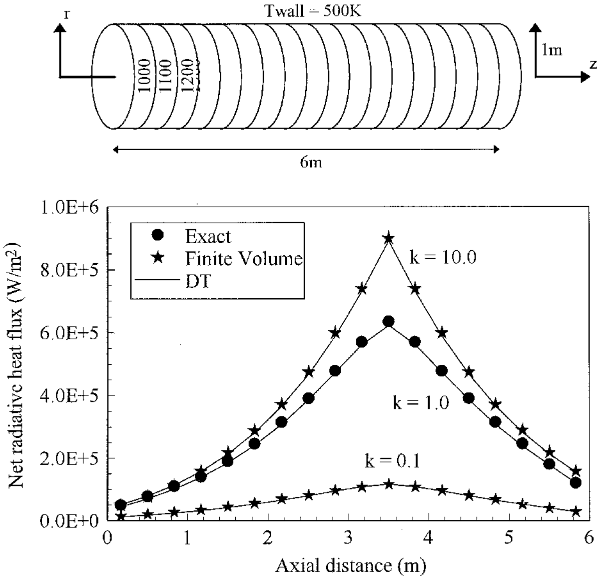

Fig. 5 Radiative flux comparison on cylinder.

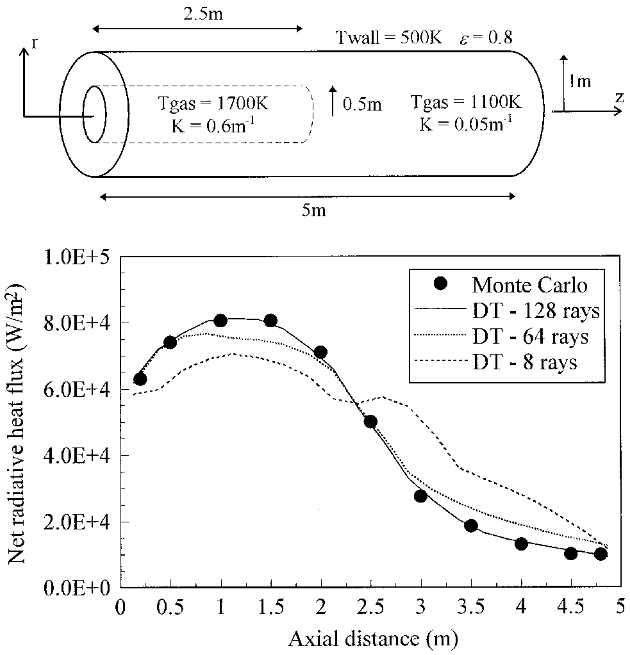

Fig. 6 Comparison of radiative flux on cylinder.

The elements within the network calculation model only one-dimensional flow. This is a severe limitation when computing radiative transfer, as radial variations in temperature and properties will significantly affect the radiative heat flux to the combustor walls. Therefore, to approximately represent such distributions, radial profiles allow the ADT calculation to be performed, taking into account variations in gas temperature and species concentrations. The profiles may be either absolute or nondimensional values about the computed mean obtained from the network solver. The profiles must be specified a priori, either based on practical experience, or from experimental and CFD analysis of a particular combustor. The overall profiles within the combustor should ideally remain relatively similar over a broad range of operating conditions.

The effect of the highly nonuniform distribution of temperature and soot typical of a gas turbine combustor was required to be evaluated by the radiation model. A second test case evaluated the ability of the new model to utilize radial profiles of temperature and radiative properties. The geometry that was considered consisted of a finite cylinder 5 m long and 2 m in diameter. The walls are at 773 K and an emissivity of 0.8. The gas contains a hot region at 1700 K and an absorption coefficient of 0.6 m^{-1}. The remainder of the cylinder is at 1100 K and an absorption coefficient of 0.05 m^{-1} (Fig. 6). Comparisons were made between a detailed accurate Monte Carlo calculation and the DT calculation using 8, 64, and 128 rays (Fig. 6).

The importance of using a sufficient number of rays is clearly illustrated. If too few rays are used, the calculation suffers from the ray effect. The number of representative rays used must capture variations in the gas temperature and properties. This may lead to problems in gases with steep gradients as the rays may completely miss a peak value.

Gas Turbine Combustor Evaluation

The network shown in Fig. 7, representative of a modern high-pressure ratio gas turbine combustor, was selected to evaluate the effects of the new radiation modeling approach on the predicted liner wall temperatures.

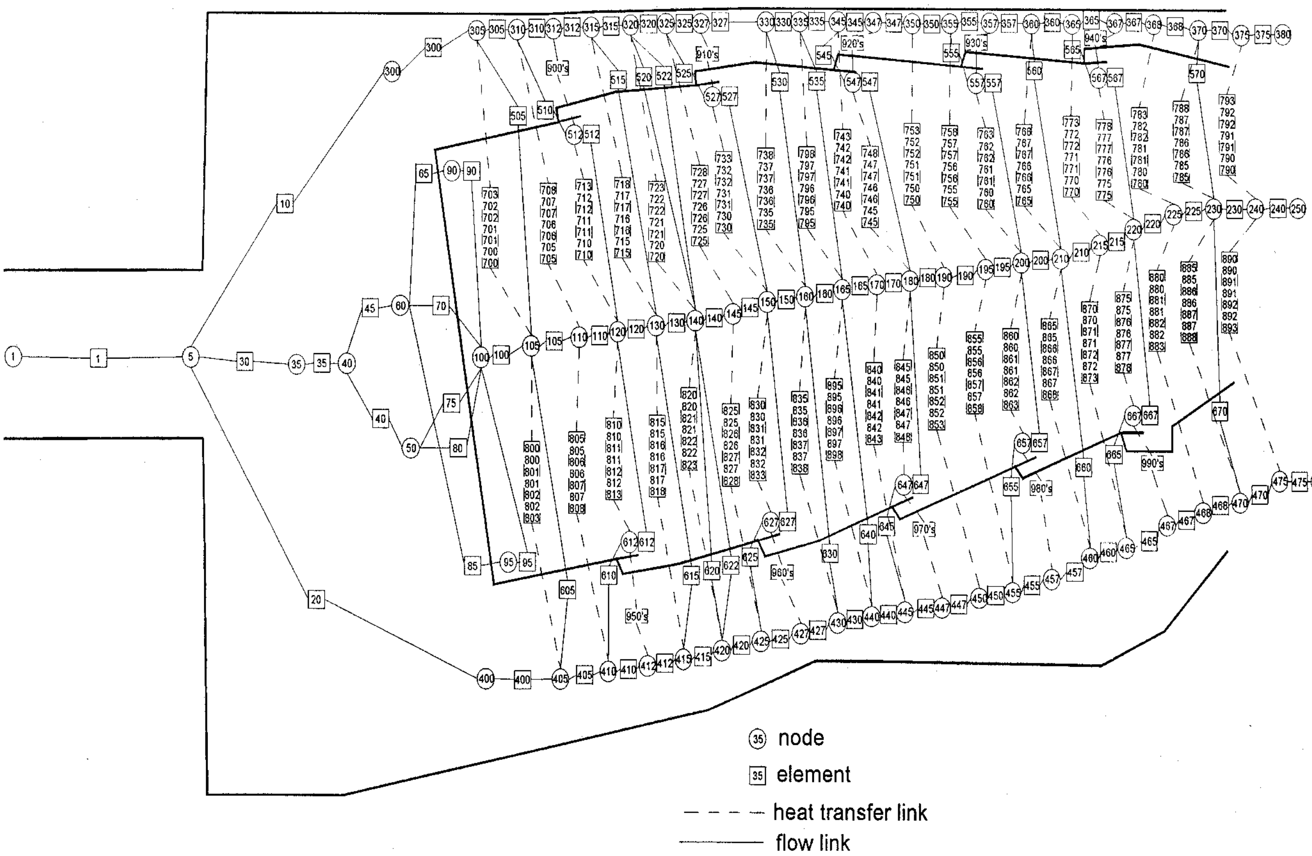

Fig. 7 Network diagram for annular combustor case study.

Combustor liner inner wall temperatures derived from thermal paint experimental data, taken from the hot side of the inner thermal barrier coating, were provided as a basis for the comparisons. The data were obtained at a typical full-power condition during an engine test. Although the hot-side thermal paint data have a scatter of ± 20 K at the maximum temperatures presented, here, it is one of the few methods available to the combustor engineer in obtaining detailed hot-side liner wall surface temperature distributions, at these operating conditions. The complete network solution including the new radiation procedure, with 25 axial cells and 40 radial stations within the flametube, was obtained within 3 min on the previously described Silicon Graphics Indy Workstation.

The dark outline is a "not to scale" representation of the combustor general features. The solid network of elements and nodes refers to the flow computational cells, in which the flow and energy equations are solved. The dashed network represents the heat transfer sequence of elements and nodes required to model the overall heat transfer from the flametube through the liner to the annuli.

Mass flow splits and pressure-drops were computed in the flow logic links, and included appropriate correlations for diffuser, liner hole features, and duct flows. The heat transfer network allows for the effects of film cooling, thermal barrier coating, and liner feature heat pickup. Heat transfer through the double-skin regions was also computed. Radial profiles of temperature and soot were included in the radiation calculation within the flametube. These profiles were obtained from predictions compared with measurements taken within a combustor flametube.[25]

Results and Discussion

The network solver was previously validated against a number of annular and reverse flow combustors.[1] The solver demonstrated accuracy and versatility in modeling complex geometries in terms of mass flow splits and pressure drops. This investigation concentrates solely on the heat transfer aspects of the simulation. Comparisons between experimental measurements and predicted wall temperatures are shown in Figs. 8 and 9.

The semi-empirical prediction uses a simple semi-empirical calculation based on the bulk gas properties at a particular axial location for computing the radiative flux; radial profiles are not included.[4]

The DT-soot1 prediction involves a full ADT simulation of the combustor flametube, utilizing circumferentially-averaged radial profiles of temperature and soot volume fraction. The soot volume fractions employed in this case were taken from CFD predictions, based upon a model correlated by measurements in a confined turbulent jet flame.[25] The model does not correctly compute the oxidation of soot, and hence, the soot concentrations are one or two orders of magnitude higher than measured values.

The DT-soot2 prediction uses the same temperature profile as the previous prediction, but the soot profile is derived from a model incorporating detailed reaction kinetics.[25] The soot volume fractions in this case compared more closely with the measured data.

The soot2 prediction provided a moderately better comparison with wall temperatures than the soot1 calculation. The maximum difference between the soot2 prediction and the thermal paint measurements was 80 K. However, the difference was much less than this over most of the inner and outer annulus liner walls. The trends of the prediction closely matched those of the measurements.

The increased levels of soot in the soot1 prediction resulted in a higher liner wall temperature than the soot2 data, as expected. However, the lower gas temperatures adjacent to the liner wall may result in the cooler soot shielding the wall. If the soot profiles are flat as in soot1, this effect is strengthened.

The error in the experimental measurements ranges from 20 K in the hottest regions to 100 K in the cooler regions. The

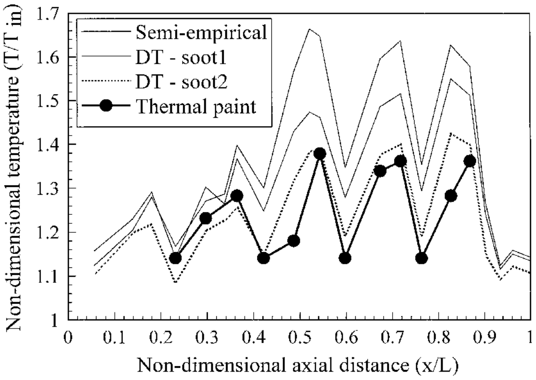

Fig. 8 Annular combustor outer annulus liner wall temperature comparison.

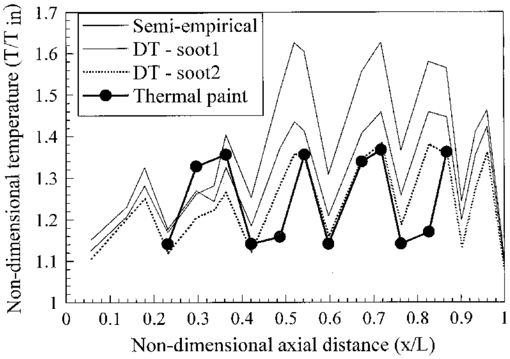

Fig. 9 Annular combustor inner annulus liner wall temperature comparison.

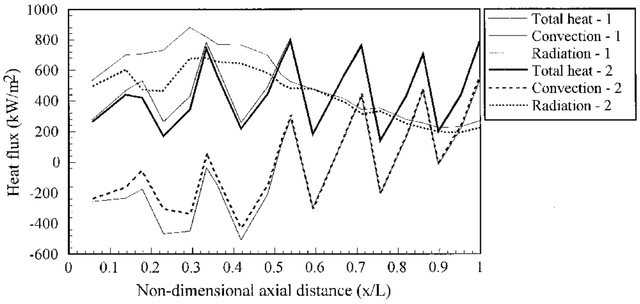

Fig. 10 Annular combustor outer annulus liner wall profile heat flux comparison.

thermal paint data consists of discreet point values that have been joined for clarity, as plotted in Figs. 8 and 9.

The axial location of the measurement points was not precise between bay endpoints. Thus, the discrepancies in these temperature points (at 0.49 on the outer wall, and 0.49 and 0.84 on the inner liner) were not of concern. Furthermore, the error bands in the measurements in this temperature range were 100 K. The maximum measured temperatures do fall within the narrow 20 K error band on both the inner and outer liner wall.

When designing a combustor, it is unlikely that sufficient information will be available to generate temperature and soot profiles within the flametube. Figure 10 examines the sensitivity of the conjugate heat transfer for a given profile of soot and temperature and corresponding bulk values, i.e., no profile. The curves designated 1 represent a single bulk soot and temperature, whereas the curves designated 2 make use of the profiles. The primary zone fluxes were most affected by this simplification. Because the walls are film cooled, the increasing negative convective flux somewhat compensates for the increasing radiative flux. The maximum difference in predicted wall temperatures between the two cases is 60 K in the primary zone.

The importance of nonluminous effects were also evaluated. In regions of lower soot concentration, downstream of the primary zone, the gas component of emission is not necessarily negligible. The effects are dependent upon the soot levels within a particular combustor. Further details of the combustor radiation interaction and the underlying network strategy may be found in Stuttaford.[26]

Conclusions

A three-dimensional radiation model was implemented with an underlying axisymmetric description of geometry and gas properties. The model was evaluated within a gas turbine combustor preliminary design code based upon a network approach. Significantly improved radiative predictions were obtained when using ADT in conjunction with radial profiles of gas temperature and soot concentrations, compared to simpler global radiation models. Absolute quantitative accuracy is reasonable considering the underlying assumptions upon which the network solver is based. For preliminary design purposes, the demonstrated accuracy is most likely sufficient. However, where absolute accuracy is still important, in, for example, line wall life calculations, more detailed simulations are still necessary.

Accurate simulations require a description of the combustor flowfield in the form of radial profiles, these must be obtained from experimental data or detailed CFD predictions. However, the results have demonstrated that if physically realistic assumptions are employed when such data are not available, the results still provide a model more accurate than a simple semi-empirical calculation. Such simulations were able to closely match the wall temperature trends in an annular combustor. The model also provided a reasonable match on the magnitudes of liner temperature.

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