

On heat load calculations in gas turbine combustors

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Abstract

Combined convective and radiative heat transfer governs the heat load on the combustor wall. This paper reviews engineering simplified models and more advanced ones. The latter ones are based on CFD approaches and a variety of methods to solve the radiative transfer equation. The principles of heat transfer in the combustor including the cooling aspects are briefly described. Details of the advanced modelling procedures are outlined and some results are discussed.

Keywords: heat load calculation, gas turbine combustor, radiation, convection, turbulence, combustion, soot.

1 Introduction

Several important parameters such as pattern-factor, emission of pollutants, combustion efficiency, and liner strength and stress, are coupled with the heat transfer in the combustion chambers. Heat from hot gases is transferred by radiation and convection to the liner. In order to predict these two components there are empirical and numerical methods with various level of complexity and precision. Empirical methods are simple and useful for preliminary design, but they predict the average radiative and convective heat flux to the liner and the average temperature in the liner. In cases where a solution with higher precision, or temperature distribution along the walls is needed, numerical methods are necessary. Numerical methods often model the convective heat transfer and in some cases also the radiative part. The complex nature of the heat transfer in combustion chambers have increased the need and development of sophisticated models. Thus, for these methods it is important to determine what level of sophistication being appropriate. The flow field in the combustion chambers is complex, and this complexity will increase with one of the following cooling



techniques: film cooling, effusion cooling or impingement cooling. Further, combustion and related parameters to this phenomenon will increase the complexity. In Figs. 1 and 3, some cooling methods are shown.

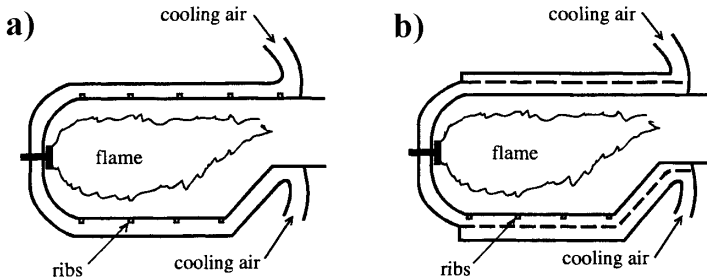


Figure 1: Forced convection a), and impingement b), cooled combustor with ribbed walls, Abdon [1].

2 Empirical methods

In these methods, the liner may be regarded as a container of hot flowing gases surrounded by a casing, with air flowing between the container and the casing. Depending on the geometry and operating conditions, radiation and convection from the hot gases heat up the liner to various extents. The liner is cooled by convection, but also by radiation. At steady state conditions, the rate of heat transfer into a wall must be balanced by the rate of heat transfer out. By assuming equal area on inside and outside of the liner, the basic relation for the heat transfer in the liner can be expressed as in eqn (1). Radiation, convection and conduction in the empirical analysis are expressed in summary as, Lefebvre [2]:

$$R_1 + C_1 = R_2 + C_2 = K_{1-2} \quad (1)$$

$$K_{1-2} = \frac{\lambda_w}{t_w} (T_{w1} - T_{w2}) \quad (2)$$

$$R_1 = 0.5\sigma (1 + \epsilon_w) \epsilon_g T_g^{1.5} (T_g^{2.5} - T_{w1}^{2.5}) \quad (3)$$

$$T_g = T_3 + \Delta T_{comb} \quad (4)$$

$$\epsilon_g = 1 - e^{-290PL(q_b)^{0.5} T_g^{-1.5}} \quad (5)$$

$$L = 0.0691 (C/H - 1.82)^{2.71} \quad (6)$$

$$R_2 = Z\sigma (T_{w2}^4 - T_3^4) \quad (7)$$



$$C_1 = 0.020 \frac{\lambda_g}{D_L^{0.2}} \left(\frac{\dot{m}_g}{A_L \mu_g} \right)^{0.8} (T_g - T_{w1}) \quad (8)$$

$$C_2 = 0.020 \frac{\lambda_a}{D_{an}^{0.2}} \left(\frac{\dot{m}_{an}}{A_{an} \mu_a} \right)^{0.8} (T_{w2} - T_3) \quad (9)$$

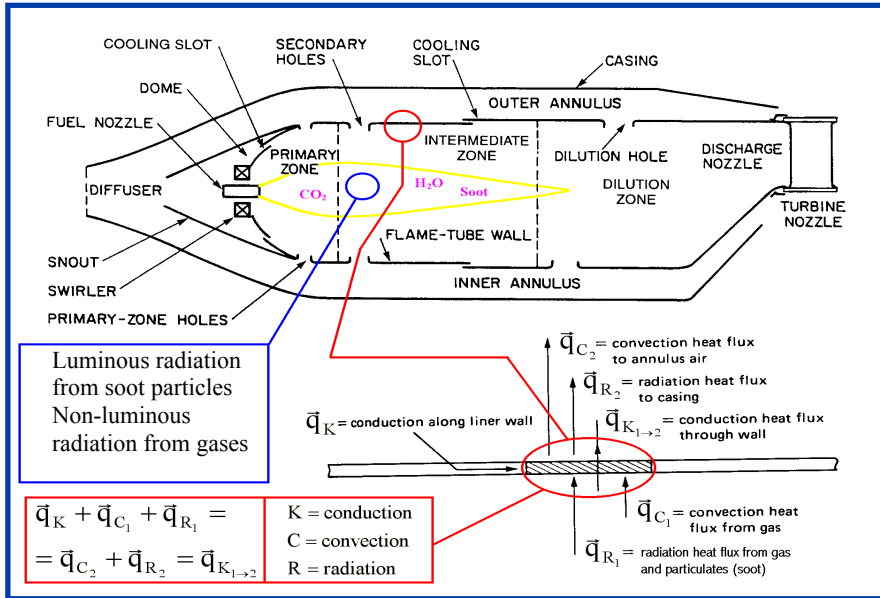


Figure 2: A principle sketch of the heat transfer mechanism for a combustor and liner wall.

Internal radiation R_1 consists of gas and soot radiation, see Fig. 2. The gas temperature T_g can be calculated according to eqn (4). In this equation ΔT_{comb} is the increase of temperature due to combustion and it can be obtained from standard charts for this purpose. R_2 in eqn (7) is the external radiation between the liner wall and the casing. With eqn (7) a reasonable value of the radiation can be obtained if Z is selected properly. Z depends on the casing emissivity. For aluminium casing this coefficient should be around 0.4 and for a steel casing around 0.6. Internal convection C_1 in the primary zone can be calculated if one changes the constant 0.02 to 0.017 in eqn (8), and if one uses the mixture flow rate in this zone. These relations are for a liner without film cooling. However in some cases some kind of film cooling is used. In the analysis of these cases, R_1 , R_2 and C_2 remain the same, but the internal convection C_1 is altered, because the coolant flow changes both the velocity and temperature of the hot gases near the wall. It can then be calculated as, Lefebvre [2]:

$$m = \frac{(\rho U)_a}{(\rho U)_g}, \quad Re_x = U_a \rho_a \frac{x}{\mu_a}$$

$$\text{if } 0.5 < m < 1.3: \quad C_1 = 0.069 \frac{\lambda_a}{x} Re_x^{0.7} (T_{w,ad} - T_{w1}) \quad (10)$$

$$m > 1.3: \quad C_1 = 0.10 \frac{\lambda_a}{x} Re_x^{0.8} \left(\frac{x}{s}\right)^{-0.36} (T_{w,ad} - T_{w1}) \quad (11)$$

Here the adiabatic wall temperature $T_{w,ad}$ is obtained from the cooling effectiveness η :

$$\eta = \frac{T_g - T_{w,ad}}{T_g - T_a} \quad (12)$$

Two approaches are established for obtaining η . One is based on a turbulent boundary layer model and the other on a wall jet model. Both models are useful, but when the velocity of the cooling air is significantly higher than that of the main stream, the wall jet model gives better predictions. The relations for calculation of η have been proposed in Lefebvre [2]. Gosselin *et al* [3] introduced another similar approach for prediction of the heat load. In their approach some new empirical relations have been used. In summary, they basically used the same equations for calculation of radiative, convective and conductive heat transfer. The difference is that they divided the combustor into four zones: recirculation zone (RZ), primary zone (PZ), secondary zone (SZ) and dilution zone (DZ), and introduced empirical equations for calculation of the temperature in each zone. For better predictions, the luminosity factor L is only considered in RZ. In this method film-cooling was considered as conduction to the wall, and the following empirical relations were introduced:

$$\left(\frac{\lambda}{\mu^{0.8}}\right) = 74.811 + 1.674T^{0.75} - 0.0081599T^{1.5} + 2.2539 \cdot 10^{-5}T^{2.25} - 2.5287 \cdot 10^{-8}T^3 \quad (13)$$

The internal and external convective equations are:

$$C_1 = 0.017 \left(\frac{\lambda_c}{\mu_c^{0.8}}\right) \left(\frac{\dot{m}_c}{A_c}\right)^{0.8} d_{h,c}^{-0.2} (T_c - T_{w1}) \quad (14)$$

$$C_2 = 0.02 \left(\frac{\lambda_a}{\mu_a^{0.8}}\right) \left(\frac{\dot{m}_a}{A_a}\right)^{0.8} d_{h,a}^{-0.2} (T_{w2} - T_a) \quad (15)$$



where $T = T_c$ in eqn (14) and $T = T_a$ in eqn (15). The value of T_c is found by employing the correlation from Odgers and Winter [4], where the film-cooling effectiveness is defined as:

$$\eta_c = \frac{T_g - T_c}{T_g - T_3} \quad (16)$$

The X parameter in fig. 4 is defined as, Odgers and Winter [4]:

$$X = K \left(\frac{\dot{m}_g A_c}{\dot{m}_c A_g} \right)^{0.8} \left(\frac{\bar{T}_g}{T_3} \right)^{0.6} \frac{x^{0.8}}{w} \quad (17)$$

where \bar{T}_g is average temperature along the cooling section, and:

$$K = \begin{cases} 1 & \text{for } \frac{u_g}{u_c} = \frac{\dot{m}_g \bar{T}_g A_c}{\dot{m}_c T_c A_g} \geq 0.8 \\ \left(\frac{u_g}{u_c} + 0.2 \right)^{-1.25} & \text{for } 0 < \frac{u_g}{u_c} < 0.8 \end{cases} \quad (18)$$

The link between η_c and X depends upon the type of cooling device used. Figure 3 shows different types, while Fig. 4 gives their performances. It is possible to predict the wall temperatures with the use of eqn (2). If the mean thickness of the wall is small, and its thermal resistance can be neglected, then $T_{w1} = T_{w2} = T_w$.

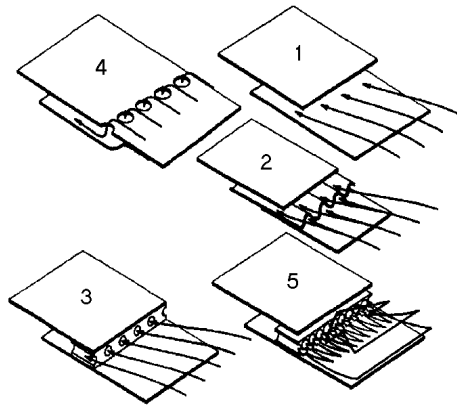


Figure 3: Typical film cooling methods.

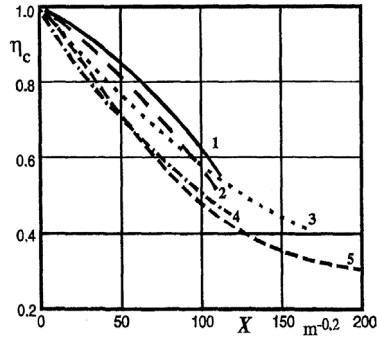


Figure 4: Film cooling performance with different methods, 1: Unobstructed Gap, 2: Wiggle Strip, 3: Plain Hole Machined Ring, 4: Splash Cooling, 5: Modified Machined Ring.

3 Numerical methods

Numerical methods are powerful tools in analysis of flow field and heat transfer in combustion chambers and with a goal of conformity to reality and experimental data they have progressed very well. For complete simulation of heat transfer in combustion chambers, a coupled system of the equations from Table 1, together with the modelling of combustion is needed. Further the energy balance in the liner, eqn (1), must be accounted for in order to update the boundary conditions in the energy equation.

3.1 Convection, turbulence and governing equations

Convective heat transfer, on both the hot and cold side of the liner, is basically described by eqn (19). When numerical calculations are performed on the cold side of the liner one usually uses the Reynolds averaged equations. A very important issue is to predict the near-wall turbulence in the best way possible, see the section on turbulence.

If the Reynolds number is high enough turbulence will be present. In direct numerical simulations (DNS) all scales are resolved and no turbulence model is needed. In large eddy simulation (LES) only the large eddies are fully resolved whereas the smaller eddies need to be modelled, Nilsson [5]. Either Reynolds or Favre averaging and decomposition should account for turbulence in the governing equations. In Table 1 the governing equations are presented in a general form.

$$\dot{q}_w'' = \dot{q}_{w,jam}'' + \dot{q}_{w,t}'' = -\lambda \left(\frac{\partial \bar{T}}{\partial y} \right)_w + \rho c_p \overline{u'T'} \quad (19)$$



Table 1: Governing equations in general form, Nilsson [5].

$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial(\rho U_j \phi)}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\Gamma_\phi \frac{\partial \phi}{\partial x_j} \right) + S_\phi$			
ϕ	Γ_ϕ	S_ϕ	eqn
1	0	0	(20)
U_i	$\mu + \mu_t$	$-\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left((\mu + \mu_t) \left(\frac{\partial U_j}{\partial x_i} - \frac{2}{3} \frac{\partial U_k}{\partial x_k} \delta_{ij} \right) - \frac{2}{3} \rho k \delta_{ij} \right)$	(21)
h	$\mu / Pr + \mu_t / Pr_t$	$\frac{\partial P}{\partial t} - \frac{\partial q_{R,j}}{\partial x_j}$	(22)
Y_α	$\mu / Sc + \mu_t / Sc_t$	$\dot{\omega}_\alpha$	(23)
k	$\mu + \mu_t / \sigma_k$	$P_k - \rho \varepsilon$	(24)
ε	$\mu + \mu_t / \sigma_\varepsilon$	$f_1 C_{1\varepsilon} \frac{\varepsilon}{k} P_k - \rho \varepsilon f_2 C_{2\varepsilon} \frac{\varepsilon}{k}$	(25)
$\mu_t = \rho f_\mu C_\mu k^2 / \varepsilon$			

Reynolds averaged Navier–Stokes (RANS) models are based on the Reynolds decomposition of the variable, and they are usually used for incompressible flows. For a compressible or a reactive flow field the Favre averaging is rather applied on the governing equations. The fluctuating properties in the equations need to be modelled, i.e., one has a closure problem, and thus extra relations are needed. Reynold stress models (RSM) generate too many new differential equations and unknowns, and instead the approach by Boussinesq is used where a turbulent eddy–viscosity (μ_t) is predicted. To model the turbulent fluxes, the gradient diffusion model based on the eddy–viscosity (μ_t) is commonly used. Many different approaches exist for modelling of μ_t . Zero equation (algebraic) and one equation models are outdated to predict μ_t . By far the two equation models are the most common ones, and for engineering applications the most common model is the k – ε –model, where ε is the dissipation rate. The k – ε –model is applied in the core region. Near–wall turbulence is very important when the heat fluxes are predicted, and thus a low Reynolds number model may be needed, see eqns (24) and (25) in Table 1. In the two–equation models one multiplies the eddy–viscosity as well as the different terms in ε –equation by damping functions (f). Further, a much finer grid is required in the “inner” near–wall regions, than that in the “outer” core region. Another approach, which requires less number of grid points, is the wall functions, but these usually give less accurate results.

The energy equation, eqn (22) in Table 1, is related to the equations of the flow field and to the RTE. The source term (divergence of radiative heat flux vector) in the energy equation is a result of the radiation field. The incident radiation is needed in the source term of eqn (22). The incident radiation results



from the solution of the radiation model. The influence of combustion in the energy equation is taken care of by updating the total enthalpy, when the species equations, eqn (23) in Table 1, are solved.

3.2 Radiative heat transfer

A considerable proportion of the heat transfer to the liner wall may be from radiation of the hot combusting gases and particles. Therefore, an accurate liner heat balance may only be performed with the assistance of a reliable radiation model, see Table 2. Prediction of radiative heat flux depends on the radiative model, the selection of models for predicting radiative properties, gas temperature, pressure, and the distribution of the species. It is favourable to choose a model that will be compatible with the solution techniques for the other governing equations. It is also useful to know how simple/complex the medium geometry is, if there are steep temperature and species concentration distribution in the medium, what kind of model that is suitable for predicting radiative properties of gases and in some cases soot. If scattering is important, a proper approximation of the scattering phase function must be selected.

The advantage, disadvantages, range of applicability, and versatility of some RTE models can be found in Table 2. A number of investigations exist where radiation models have been implemented to compute the radiative heat flux in gas turbine combustors. For example, Mengüç *et al* [6] successfully employed spherical harmonics (P_1 and P_3), Carvalho and Farias [7] employed the discrete ordinates method (DOM), and Krebs *et al* [8] employed a spectral approach.

Table 2: Comparison of radiative models, Howell and Mengüç [9].

Model	Angular resolution	Spatial resolution	Spectral resolution	Scattering medium,2D	Scattering medium,3D
DTM	2	4	4	2	2
DOM	3	4	4	3	3
Spherical harmonics	2	3	4	3	3
Zonal method	2	3	2	2	2
Monte Carlo	4	3	3	4	4
Finite element techniques	2	4	3	2	2

4 = very good, 3 = good, 2 = acceptable, 1 = not good

3.3 Combustion

There are two major types of gas turbines, i.e., aircraft and industrial (stationary). In aircraft gas turbines the fuel is usually kerosene (a liquid), which needs to be atomised to form a proper spray. If the fuel and oxidiser are not mixed, one has a non-premixed (diffusion) combustion. In LPP combustors the liquid fuel is pre-vaporised and pre-mixed. For stationary gas turbines the fuel is often some kind



of natural gas (methane), and for this type of gas turbines one may have a premixed combustion. The chemical reactions describe how the transformation of energy stored in the chemical bonds to heat occurs, what kind of species being formed, and how much soot being produced. Thus, great care must be given combustion in order to predict the correct soot and gaseous species concentrations. The different species are treated by the species equation, see eqn (23) in Table 1. A separate model is needed for soot. The rate of formation of a species, the source term in the species equation, is determined by chemical kinetics. The use of elementary reactions is not practically applicable, and especially not if turbulence is present. As a consequence a reduced global reaction scheme has to be used where only the major species are treated. Experience shows that reduced schemes are acceptable in terms of major species concentrations and temperature fields and will suffice for engineering applications. Turbulence changes the mixing and heat transfer process, and influences the chemical reactions in a highly non-linear manner. A complete mathematical solution of the physical process is not possible and several models may be used. If a statistical distribution is known, then the joint PDF or the presumed PDF approaches could be used to estimate the reaction rate. The eddy break up (EBU) model is based on the assumption that chemical reactions are fast in comparison with the heat transfer and mixing of fuel and oxidant. This model is simple to implement, but a constant has to be tuned for different problems. The eddy dissipation concept (EDC) model is similar to EBU model, i.e., the chemical rates play no role in determining the burning rates. The flamelet concept has been widely used in the modelling of premixed and non-premixed turbulent combustion Nilsson [5]. In the flamelet library approach (FLA), detailed chemical kinetics is employed to calculate the flamelet library. By using the FLA soot can be simulated in turbulent combustion with relatively low computer capacity.

4 Results and discussion

CFD for practical combustion systems based on the finite difference (or finite volume) method of the 3D forms of the conservation equations have become viable in the past two decades. Jones *et al* [10], Jones and Priddin [11], Jones and McGuirk [12], and Coupland and Priddin [13], have all used a CFD approach to predict the flow and the combustion in gas turbine combustors. Because of the multiplicity of the phenomena involved, these works did not consider radiation heat transfer. However, in gas turbine combustion chambers, a significant part of the total heat flux to the walls is by radiation of hot gases and soot. Especially with the presence of soot in the combustion products, the radiative part imposes serious attention regarding liner durability.

Boysan *et al* [14] used a flux model to account for the effects of radiation in a gas turbine combustor solving three second order linear differential equations for the fluxes in the axial, radial and tangential directions. The contribution of the radiation was considered in the enthalpy equation, through a source term calculation from those fluxes. The study was carried out at atmospheric pressure.



The authors did not specify the type of boundary conditions used for calculation of the radiation heat transfer, nor did they discuss its influence on the results.

Sampath and Ganesan [15] used the same flux model to compute radiation heat transfer in a gas turbine combustor. However, heat transfer was not the aim of the investigation and therefore the heat flux was not presented. Lockwood *et al* [16] used DTM for prediction of radiative heat flux and assumed an adiabatic flame, but they did not solve the enthalpy equation. The distribution of temperature was determined as function of the instantaneous mixture fraction using the equilibrium method of Gordon and McBride [17]. Therefore the effects of radiative heat transfer on the flow, temperature field, and chemical reaction were not accounted for. The radiative heat fluxes on the wall were calculated a posteriori assuming a uniform wall temperature equal to the inlet air temperature. They concluded that the total radiative heat flux to the walls was less than 25% of the energy supplied to the combustor, justifying the adiabatic assumption on which the flow and reaction calculations were based.

Carvalho *et al* [18] performed a study using the DTM for the prediction of radiative heat transfer in a can combustor. In that study, they solved the enthalpy equation with a source term accounting for the heat transferred by radiation. The wall temperature was assumed uniform and equal to the inlet air temperature. Two sets of predictions were made for pressures at 6 and 25 bar. Radiative heat flux to the wall increased markedly with the pressure. In another study by Carvalho and Coelho [19] an annular combustor was studied. They studied the magnitude of radiative and convective heat flux to the walls for different pressures. The turbulence was handled by the standard k - ϵ model. For prediction of radiation they used DTM and for properties of gases (water vapour and carbon dioxide) they utilised the “two grey plus a clear gas” fit, but they did not consider soot production and its effect on radiation. They solved the equations for continuity, momentum, energy and mixture fraction and assumed that reactions were fast compared to the transport processes. They also assumed that species and heat diffuse at the same rate, and that the heat loss to the surroundings can be neglected compared with the heat transfer released by chemical reaction. Thus instantaneously, the gas composition can be determined as a function of a strictly conserved scalar variable. The boundary condition was based on the energy balance of the liner eqn (1). They neglected conduction in the liner wall and for the internal convection Reynolds analogy was used. Eqns (7), (9) and (1) were used in order to generate boundary conditions for the numerical simulation. The calculation procedure may be summarised as follows:

1. guessing the liner wall temperature (flame side) is equal to the inlet air temperature.
2. the internal radiation and internal convection are calculated.
3. using eqn (1), the liner wall temperature (coolant side) is calculated. A non-linear equation has to be solved because R_2 is proportional to T^4 .
4. setting $R_1 + C_1 = K_{1-2}$ a new value for the liner wall temperature (flame side) is obtained and it is compared with the value used in step 2.
5. steps 2-4 may be repeated according to a certain convergence criterion.



Although they did not consider soot, they found that the convective and radiative heat fluxes were of the same order of magnitude, and that they increase with an increased pressure. The convective part was larger than the radiative one for all pressures.

Crocker *et al* [20] compared the empirical approach described above, with CFD calculations carried out on both the cold and hot side of the liner. They found good agreement for the predicted temperatures, between the two approaches, after they had modified the constants in eqns (8) and (9) to 0.046 and 0.040, respectively. In the numerical approach they used DOM with a modified k - ε model (RNG). The flame was non-luminous (no soot).

5 Conclusions

The empirical models are satisfactory as an overall estimation of the heat load is needed, and they should only be used in the preliminary design of the gas turbine combustor. Numerical (CFD) approaches are needed in the final stage of the design, where detailed data of the liner temperature and heat loads are needed. Some CFD investigations neglect radiation and almost all neglected the influence from soot. Thus, it is very important that turbulence and chemistry are included, and that these are modelled in the best way possible. The influence of pressure is also important, because it greatly influences the formation of soot.

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Nomenclature

A	Area	X	See eqn (17)
C/H	Carbon to hydrogen ratio	x	Distance downstream of slot
c_p	Specific heat	Y_α	Species mass fraction
D, d	Diameter	ε	Emissivity, Eddy dissipation rate
f_μ, f_1, f_2	Damping functions	λ	Conductivity
h	Total enthalpy	μ	Dynamic viscosity
K	See eqn (18)	ρ	Density
k	Turbulent kinetic energy	σ	Constant
L	Luminosity factor	$\dot{\omega}_\alpha$	Formation rate
l_b	Mean beam length	Index	
\dot{m}	Mass flow rate	1, 2	Hot and cold side



P	Pressure	3	From compressor
Pr	Prandtl number	a	Air
q	Fuel/air ratio by mass	ad	Adiabatic
\dot{q}''	Heat flow rate	an	Annulus
Sc	Schmidt number	c	Cooling-film
s	Depth of film cooling slot	g	Gas
T	Temperature	h	Hydraulic
t	Thickness, Time	L	Liner
U, u	Velocity	lam	Laminar
w	Width	w	Wall

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