Estimation and Optimization of the Film Cooling Requirements in a Gas Turbine Combustion Chamber

By

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A study of film cooling requirements of a modern aero gas turbine combustion chamber over a typical aircraft flight path is undertaken. A computational procedure is used to estimate and optimize the coolant flow for a given value of the metal temperature along the whole length of the liner wall. These results are presented for a variety of mainstream and slot flow conditions. The effects of different film cooling geometries, mainstream turbulence, hot gas acceleration etc. are also investigated.

It is found that for an actual liner wall, optimum values of mass flow ratio deviate significantly from unity. Increasing inlet pressure leads to increasing metal temperatures and the optimum value of mass flow ratio depends mainly upon internal radiation in the primary zone, convective components in the intermediate zone and the mainstream temperatures in the dilution zone respectively. Optimization of the combustion chamber studied here leads to between 8-15% (of the total mass flow) saving in the cooling air requirements.
# CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0 INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>2.0 EQUIPMENT AND TEST CONDITIONS</td>
<td>1</td>
</tr>
<tr>
<td>3.0 PREDICTION ANALYSIS</td>
<td>3</td>
</tr>
<tr>
<td>3.1 Effects of individual variables on $T_w$</td>
<td>3</td>
</tr>
<tr>
<td>3.1.1 Mainstream variables</td>
<td></td>
</tr>
<tr>
<td>3.1.2 Slot parameters</td>
<td></td>
</tr>
<tr>
<td>3.2 Combined effects of the inlet parameters on $T_w$</td>
<td>4</td>
</tr>
<tr>
<td>3.3 Significance of the calculated flame tube temperatures</td>
<td></td>
</tr>
<tr>
<td>4.0 COOLING AIR REQUIREMENTS</td>
<td>4</td>
</tr>
<tr>
<td>4.1 Effects of the individual mainstream variables on $m_{opt}$</td>
<td>4</td>
</tr>
<tr>
<td>4.2 Effects of altitude and mach no. on $m_{opt}$</td>
<td>5</td>
</tr>
<tr>
<td>4.3 Effects of the individual slot parameters on $m_{opt}$</td>
<td>5</td>
</tr>
<tr>
<td>5.0 STUDY OF VARIOUS SLOT CONFIGURATIONS</td>
<td>5</td>
</tr>
<tr>
<td>5.1 Slots with tangential injection of the coolant</td>
<td>5</td>
</tr>
<tr>
<td>5.1.1 Clean slots</td>
<td></td>
</tr>
<tr>
<td>5.1.2 Clean slots with a finite lip</td>
<td></td>
</tr>
<tr>
<td>5.1.3 Dirty slots (stacked and machined rings)</td>
<td></td>
</tr>
<tr>
<td>5.2 Inclined slots and holes</td>
<td>6</td>
</tr>
<tr>
<td>5.2.1 $30^o$ slot</td>
<td></td>
</tr>
<tr>
<td>5.2.2 $90^o$ slot</td>
<td></td>
</tr>
<tr>
<td>5.2.3 $35^o$ hole</td>
<td></td>
</tr>
<tr>
<td>5.3 Inclined holes with the cover plate protection</td>
<td>7</td>
</tr>
<tr>
<td>5.3.1 Splash cooling geometries</td>
<td></td>
</tr>
<tr>
<td>6.0 EFFECTS OF THE MAINSTREAM TURBULENCE</td>
<td>9</td>
</tr>
<tr>
<td>7.0 EFFECTS OF THE HOT GAS ACCELERATION</td>
<td>10</td>
</tr>
<tr>
<td>8.0 OPTIMIZATION OF THE COOLING AIR REQUIREMENTS</td>
<td>11</td>
</tr>
<tr>
<td>8.1 Optimization for the maximum permissible metal temperature</td>
<td></td>
</tr>
<tr>
<td>8.2 Optimization for the minimum metal temperature</td>
<td>11</td>
</tr>
</tbody>
</table>
8.3 Optimization for minimum mixing

9.0 CONCLUSIONS

10.0 REFERENCES

TABLES

FIGURES

NOTATIONS

\(A_0/A_e\) Outlet area/slot open area (3dim. slot geometry)

\(a\) Constant equation (6)

\(C\) Convective heat transfer (W/m²)

\(C_p\) Specific heat at constant pressure (J/kg deg.C)

\(D\) Diameter (m)

\(e\) Emissivity

\(H\) Enthalpy (W/m²)

\(h\) Heat transfer coefficient (W/m² deg.C)

\(L\) Luminosity factor

\(\ell\) Radiation beam length (m)

\(M\) Mass flow (Kg/sec)

\(m\) Mass velocity ratio \((\ell_c U_C/\ell_m U_m)\)

\(P\) Pressure (atm)

\(R\) Radiation heat transfer (W/m²)

\(Re\) Reynolds number

\(r\) Fuel-air ratio by weight

\(s\) Slot height (m)

\(T\) Temperature (°K)

\(t\) Lip thickness (m)

\(U\) Mean velocity (m/sec)

\(u'\) Turbulence intensity (m/sec)

\(X_l\) Generalized film cooling parameter, equation (6)

\(x\) Distance downstream of the slot (m)

\(z\) Transverse distance (m)
NOTATIONS (Contd.)

\( \eta_c \)
Combustion efficiency (\%)

\( \eta \)
Adiabatic effectiveness \( (T_m - T_{w_{ad}}/T_m - T_c) \)

\( \rho \)
Density

\( \mu \)
Dynamic viscosity

Subscripts

\( \text{ad} \)
Adiabatic condition

\( \text{an} \)
Annulus

\( \text{c} \)
Coolant (slot) flow

\( \text{f} \)
Flame

\( \text{h} \)
Hole

\( \text{m} \)
Mainstream flow

\( \text{w} \)
Wall condition

\( \text{opt} \)
Optimum condition

\( \text{l} \)
Internal to the liner

\( \text{2} \)
External to the liner

Abbreviations

P.Z., I.Z., D.Z. Primary, Intermediate, Dilution Zones respectively.

\( \Delta \)
Difference

A.F.R. Overall air-fuel ratio
1.0 INTRODUCTION

It has been known for sometime that in a modern aero gas turbine combustion chamber, film cooling is an effective means of maintaining acceptable liner wall temperatures and thus prolonging its operational life. The last decade has seen a vast accumulation of useful experimental data on various film cooling systems and recently it has even become possible to predict to design accuracy, the performance of the commonly used film cooling geometries by semi-analytical techniques (1). Two interesting possibilities arise as a result of these developments and they are:

i) In order to achieve maximum cycle benefits, is it possible to optimize the film cooling air requirements, slot design and other parameters. The combustion chamber designer recognizes the importance of such an exercise.

ii) The actual design of an optimized cooling system is mainly governed by the engine specifications and the engine-aircraft combination. Therefore, a wholly general approach to this problem cannot be pursued. Is it however possible to set up certain important guidelines which significantly reduce the rather ad-hoc approach to the combustor cooling design.

In the present work, attention has been focussed on these two problems. Based on the available data, study of the optimum cooling air requirements and the resulting wall temperatures is carried out. This complete procedure is demonstrated for the case of a modern gas turbine combustion chamber over its operational flight path. The cooling potential of various slot geometries, and the effect of individual and combined inlet flow variables on the wall temperature and the cooling air requirement are studied. Wherever possible, comparison with the experimental data obtained on a full scale test rig is also given.

2.0 EQUIPMENT AND TEST CONDITIONS

The main parameters involved in the prediction procedure of Ref. (1) are:

a) Slot parameters: \( \eta, h_1 \)

b) Mainstream parameters: Inlet flow factors \( M_T, P_C, T_C, \) AFR etc. Mainstream parameters \( T_m, h_2, \) etc.

c) Mainstream radiation: \( \varepsilon_m, L, R_1 \)
Knowledge of the factors under the groups (a), (b) and the assumptions underlying their use in the film cooling calculations have been discussed in the past (1). The mainstream radiation at high pressures is a partly unknown, but probably the single most important factor influencing the metal temperatures. The purpose of this investigation therefore was to measure the unknown parameters related to the mainstream radiation, which otherwise preclude the possibilities of calculating the absolute magnitude of the wall temperatures.

A tubular type combustion chamber of Fig.1 featuring an air blast atomizer was used in the study. A detailed description of the test rig, instrumentation and measurements is given by Norster and Lefebvre (2). Experimental tests were carried out to embrace the whole aircraft flight envelope. Of importance from the point of view of the liner cooling are the tests which simulate

i) maximum heat loads on the combustor walls which occur during the take off and the rapid acceleration.

ii) continuous heat loads during cruise.

The results of these tests are compared with their predicted values in Figs. 2 and 3. The increase of radiation and emissivity with pressure can be calculated using the method and assumptions outlined in (1) i.e.

\[ C_m = 1 - \exp\left\{ -8.85 \times 10^3 \cdot L \cdot P \cdot (h) \cdot (T_m^\delta - T_w^\delta) \right\} \quad (1) \]

\[ R_1 = 48.7 \times 10^{-9} \cdot C_m \cdot T_m^{1.5} \left\{ T_m^{2.5} - T_w^{2.5} \right\} \quad (2) \]

Comparison of equations (1) and (2) with the experimental values of the Ref. (2,3) show that over most of the pressure range, values of \( C_m \) and \( R_1 \) can be predicted fairly accurately. An approximate value of \( L = 4.0 \) in the primary zone, linearly decreasing along the length of the combustor to about 2.0 in the dilution zone seems appropriate for this high pressure combustor. However as the primary zone combustion efficiency continues to increase, emissivity values are bound to decrease rapidly until eventually they correspond with the non-luminous emissivity due to carbon dioxide and water vapour alone.

Fig.3 also shows the comparison of the flame tube metal temperatures with pressure in the primary zone. Again the agreement between the experimental and the predicted values is very good. Both the Figs. 2 and 3 therefore establish reasonably well the validity of the prediction procedure of Ref. (1). This procedure is used in the calculations to follow.
3.0 PREDICTION ANALYSIS

The test conditions examined in the prediction procedure are outlined in Table I. Fig. 4 shows a general computational scheme for the calculation of the wall temperatures. The enthalpy balance equation $R_1 + C_1 = R_2 + C_2$ used in this prediction exhibits minimum values of $T_W$ at values of $m > 1.0$ for all three zones, Fig. 5. A rapid reduction in $T_W$ occurs as $m$ exceeds unity, but further reduction is difficult to achieve viz. in the primary zone, $m = 2.0$ is a fairly good approximation to the minimum $T_W$. Further increase in $m$ upto 4.0 only yields 20 K reduction in the wall temperature. After which a small increase in $T_W$ in the vicinity of $m = 5.0$ is followed by a continual decrease in $T_W$, because of the presence of a large quantity of cold air near the wall. The slots located in the intermediate and the dilution zones behave in a similar manner, but at the lower values of $m$. Wherever optimization is carried out, usually the lowest value of $m$ beyond which appreciable reduction in wall temperature is not forthcoming, $(dT_W/dm = 0)$ is taken as $m_{opt}$.

3.1 Effects of Individual Variables on Wall Temperature

3.1.1 Mainstream variables:

Fig. 6a shows this effect in the primary zone for $m = 1.3$. A similar trend is observed in intermediate and dilution zones. Most of these results are to be expected. The predictions show a gradually decreasing influence of pressure on wall temperatures, an observation very similar to the experimental evidence of Ref. (3). Since $e_m$ increases exponentially with pressure, (equation 1) at higher pressures, its rate of increase is curtailed. On the other hand, dissociation is suppressed and $T_m$ increases. A closer observation of the predicted values however indicate an increase in convective cooling components and hence the tendency of the wall temperature to approach a constant value at higher pressures.

3.1.2 Slot parameters:

Fig. 6b shows their effect on $T_W$. The film cooling effectiveness $n$, the external convective heat transfer coefficient and the internal radiation represented here by $e_m$ and $n_C$ are the three most important factors contributing to the changes in the wall temperatures.

3.2 Combined Effects of the Inlet Parameters on the Wall Temperature

Fig. 7 shows the combined effects of $M_t, P_C, T_C (M_t\sqrt{T_C}/P_C = constant)$ with the altitude and the mach number. Also presented are the curves for the uncooled combustion chamber wall temperatures calculated by the Lefebvre and Herbert (4) method, for the same input conditions. At the lower altitudes and mach numbers, considerable benefits are to be gained from the film cooling of the liner wall in all the
three zones. The explanation for this can be sought in Figs. 8 and 9, where the individual components of the enthalpy balance equation are plotted. For the cooled and the uncooled wall, the radiation components $R_1$ and $R_2$ are virtually the same. However at the lower altitudes

$$\Delta C_1 >> \Delta C_2$$

where $\Delta C_2$ Convective heat loss without the film cooling - convective heat loss with the film cooling present.

This results in a substantial reduction in the wall temperatures with the film cooling at the lower altitudes and mach nos. since in this range, the value of $C_1$ is reduced more than that of the $C_2$.

3.3 **Significance of the Calculated Flame Tube Temperatures**

The liner wall temperatures calculated above are fundamental to the geometric design and the heat loading of the combustor. However various assumptions have been made in their calculations (1,4). No allowance is made for the fact that severe hot spots can occur along the liner due to the combined effects of the local annulus velocity being too low or the cooling air film near the intermediate and the dilution holes has broken down. The calculations however show

i) the extent of the influence of the internal radiation $R_1$, Fig. 9a

ii) the importance of the external cooling i.e. $C_2$, Fig. 9b

4.0 **COOLING AIR REQUIREMENTS**

It has been known that the film cooling process tends to overcool the wall from the point of injection, up to the point where the maximum permitted wall temperature is reached. Therefore, there is a considerable incentive to minimize the amount of cooling air. The film cooling effectiveness $\eta = f(x/s, m)$. Having chosen the $x/s$ value for each station in the primary, intermediate and the dilution zones, the optimized value of $m$ is calculated for the combustor of Fig.1.

4.1 **Effects of Individual Mainstream Variables on $m_{opt}$**

Computations are performed for the take off condition No.1 for different inlet temperatures $T_C$ and the results plotted in Fig.10. As the cooling air temperature rise, almost a linearly rising quantity of air is required to offset the effects of lower internal and external convective components in increasing the wall temperature $T_w$. In the primary zone, the major factor governing $T_w$ and $m_{opt}$ is $R_1$, in intermediate zone it is the convective component, and in the dilution zone the $T_m$. Table II gives some idea of the magnitude of this effect for the present study.
The effects of $P_C$, $M_t$ and A.F.R. can be explained based on the above observation.

4.2 Effects of Altitude and Mach No. on $\dot{m}_{opt}$

Fig. 11 shows this effect for $M_t \sqrt{T_c/P_c} = 1.15$. Of immediate significance is the primary zone curve which rapidly approaches a value of $\dot{m}_{opt} = 1.3$ as the altitude goes up. Under the cruise conditions therefore, it seems adequate to have $\dot{m}_{opt} = 1.3$ for all the three zones.

Also plotted here is the increase in cooling requirement with the mach no. As the aircraft is accelerated, there is a noticeable increase in the heat loading of the primary zone. One would therefore expect the higher cooling requirements especially in this zone. At mach no. = 2.0 and beyond, the cooling air required may approach the take off condition.

4.3 Effects of Individual Slot Parameters on $\dot{m}_{opt}$

This is shown in Fig. 12. As the adiabatic effectiveness $\eta$ decreases, in order to get minimum wall temperatures, $\dot{m}_{opt}$ also has to decrease towards unity, (since minimum mixing then occurs). On the other hand the wall temperatures are rising. As shown, about 25% reduction in effectiveness brings $\dot{m}_{opt}$ to unity.

In the same figure, the $h_1$ vs. $\dot{m}_{opt}$ curve shows the advantages to be gained by having higher heat transfer coefficients of the film and therefore preferably a wall jet type of flow. Again since internal convection features predominantly in the intermediate zone, this curve also shows a rapid reduction in $\dot{m}_{opt}$ with $h_1$. Similar conclusions apply to the curves of $e_m$ and $\eta_c$ in the primary zone.

5.0 STUDY OF VARIOUS SLOT CONFIGURATIONS

5.1 Slots with Tangential Injection of the Coolant

5.1.1 Clean Slots:

Long narrow clean slots have been known to be the most efficient cooling geometry available to the combustion chamber designer, since they project an undisturbed thin film of cold air on the walls. Their performance can be computed by using the appropriate effectiveness and heat transfer relationships of the near slot region as shown in Table III. Fig. 13a shows the results. As mentioned previously in Section 3.0 $\dot{m}_{opt} \neq 1.0$ in an actual combustion chamber. With a non-adiabatic wall, heat is able to dissipate through the external convection and radiation and the deviation of $\dot{m}_{opt}$ from unity is proportional to the enthalpy balance due to $R_1$, $C_2$, $R_2$ reached at the wall.
5.1.2 Clean Slots with a Finite Lip:

This type of slot is usually formed by introducing a spacer viz. a wigglestrip in the clean slot, for reasons of mechanical strength. Fig.13a shows the results of the computation which are:

i) As $m$ increases beyond unity ($m>2.0$), enhancement of the mixing due to the lip turbulence, increases $T_W$ substantially above the clean slot valves.

ii) In the near slot region, continual increases in $m$ leads to an increase in $T_W$ for a finite lip slot.

iii) For larger values of $x/s$ (>20), $m_{opt}$ is the same, both for the clean and the finite lip slots because of the decay of the lip turbulence downstream.

5.1.3 Dirty slots (stacked and machined rings):

Sturgess (5) has recommended a correlation for the stacked and the machined rings. His effectiveness formula is

$$\eta = A \left[ 1 + B S_N^{0.65} \right]$$

where $A$ and $B$ are constants and $S_N$ is the potential core length in

$$S_N = \frac{x-x_p}{u_{ms}} \left( \frac{u_c}{u_m} \right)^{-0.15} \left( \frac{A_v}{A_c} \right)$$

In view of the difficulty of predicting $x_p$ with any degree of certainty, we neglect this term while calculating the wall temperatures in the far downstream region. The plotted curves of Fig.13a are for $A_v/A_c = 1.5$. Since there is no optimum value of $m$ as such, one can only observe that in order to maintain the wall temperatures at a value of $T_{Wmin}$ for the finite lip slots, the additional cooling air requirement is as under.

<table>
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<th>P.Z.</th>
<th>I.Z.</th>
<th>D.Z.</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta M_c/M_t$</td>
<td>1.5%</td>
<td>3%</td>
<td>5.5%</td>
</tr>
</tbody>
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5.2 Inclined Slots and Holes

5.2.1 30° Slot:

Haatnett, Birkebak, Eckert (6), Sivesegaram and Whitelaw (7) and Metzger, Carper, Swank (8) provide the required experimental data on this type of discrete slot. The effectiveness data of Ref. (6,7) is used in the computation. As regards the local heat
transfer rates, adequate tests are lacking in the region of $x/a < 30$. A limited data for $m < 1.23$ is provided by Ref. (6) and this along with their extrapolated values are used here. Fig.13a shows the results. In the primary zone, the advantages of using an inclined slot (especially when $m < 1.0$) are clearly obvious. This is because of the considerable increase in the heat transfer coefficients which outweighs the decrease in effectiveness and reduces $T^c$. As the slot inclination is raised and $M_c$ goes up, ($m > 2.0$), the effectiveness takes over and $T^c$ is raised. Thus this type of slot seems ideal for use in the primary zone cooling rows. In the dilution zone again, increased $m$ leads to lower wall temperatures because after traversing a certain distance downstream, the coolant flow is forced against the wall by the mainstream thus resulting in an improved heat transfer.

5.2.2 90° Slot:

Here the experimental values of Ref. (7,8) are used and the results plotted in Fig.13b. A small coolant flow through a normal slot is deflected by the hydrodynamic force of the mainstream towards the wall and therefore the $T^c$ drops. Further increase in coolant however causes the slot jet to increasingly penetrate the mainstream and expose the wall downstream to the influence of hot mainstream. Thus $m_{opt} = 0.75$ seems a best compromise for all the three zones, although $T^c$ is greater than for a 30° slot throughout the liner.

5.2.3 35° Hole:

Mechanical considerations may prevent the use of porous walls or continuous clean slots in the high pressure combustors. Inclined holes are therefore considered for cooling. Use is now made of the effectiveness data of Goldstein, Eckert and Ramaey (10) together with an observation of Metzger (11) that the heat transfer coefficient for a 20° injection angle circular hole is increased only slightly above the flat plate value. The results of this prediction are plotted in Fig.13b. Lowest wall temperatures are obtained for $m_{opt} = 0.5$ for all the three zones, corresponding to the maximum values of effectiveness Ref. (10). The lateral wall temperatures at $F/D = 0.5$ also follow the same pattern. For $m_{opt} = 0.5$, inclined circular holes give almost the same wall temperatures as the normal (90°) slot.

5.3 Inclined Holes with the Cover Plate Protection

It has been demonstrated above that the film cooling scheme using the discrete hole configuration is inferior to that of the conventional two dimensional slot injection. Recently however, the designers have been using the inclined holes with a cover plate protection, where the coolant is admitted through spaced holes, impinges against a cover plate, turns and mixes to form a uniform two dimensional flow before its exposure to the hot mainstream gas.
Metzger et. al (12) have recently studied this geometry and provided effectiveness and heat transfer values. Their results are used in the computation to yield the curves of Figs. 13b. They lie between the two dimensional and the three dimensional geometries and thus represent a compromise between the mechanical and the aerodynamic requirements of the cooling system.

5.3.1 Splash cooling geometries:

Interest in the three dimensional film cooling geometries is recent and has arisen because of the rapid developments in the alloys that can withstand the high temperatures. It is now possible to operate the combustors at much higher wall temperatures than ever before and hence the attention to the splash cooling techniques. The effectiveness values of Nina and Whitelaw (13) are utilized here. Since the coolant flow after the slot exit is almost two dimensional, heat transfer coefficients recommended in Ref.(1) were used for the calculations here. Fig. 13b shows all the results. The following conclusions may be drawn from these predictions.

i) Comparison of the geometry number 6 and 13, in the primary zone shows that the splash cooling slot with the holes off line is generally to be preferred to a finite lip slot but with low open area ratio (i.e. contraction). Geometry number 13 offers substantial reduction in $T_w$ and $m_{opt}$ as compared with the thick lip geometry no.6 and finally a cover plate protection results in a much lower wall temperature than a splash cooling slot.

ii) In the intermediate zone, since convective component $C_i$ is important, a large open area ratio geometry 1 is compared with the splash cooling geometry number 12 (holes in line). The very important influence of the open area ratio is thus well confirmed.

iii) Fig.13b also shows a comparison of the geometry numbers 3 and 9 in the dilution zone, the former having twice the open area ratio, lip length and hole diameter than the latter.

A comparative study of the various geometries leads to the following choice for the three zones:

Primary Zone: Splash cooling slot (holes off line No.13).
Intermediate Zone: Three dimensional geometry (No.1)
Dilution Zone: Three dimensional geometry (No.3)

Table III summarizes the details of all the film cooling schemes discussed above, for the combustor of Fig.1.
6.0 EFFECTS OF THE MAINSTREAM TURBULENCE

The effect of the slot turbulence in reducing the film cooling effectiveness, has been found to be negligible, up to a turbulence intensity of 10%, Ref. (14). Therefore the attention has been focused on the hot mainstream turbulence. This has been investigated by Carlson and Talmor (15) and their correlation is

\[
\frac{1-n}{n} = 0.329 a \cdot X_1
\]

where the constant \( a \) has values 1, 2, 3 for the mainstream turbulence of 3%, 12% and 22% respectively.

Equation (6) however is only accurate for \( X_1 > 4 \), Ref. (15) i.e. in the far-slot region. Therefore, values of \( X_1 \) were computed from the slot and the mainstream parameters of the combustor (Fig. 1) and the corresponding experimental values of effectiveness were obtained for 3%, 12% and 22% turbulence intensities respectively from Ref. (15). Calculations of the heat transfer coefficients are made using either

i) Colburn flat plate equation for 12% and 22% intensities

ii) Heat transfer coefficients of Ref. (6) for the turbulence intensities of up to 3% since the slot configurations of Ref. (6) and Ref. (15) is almost identical.

Fig. 14 shows the results in the three zones. In the near slot region, entrainment of the hot mainstream in the film coolant has just begun while the turbulence intensity of the mainstream is maximum (viz. in the primary zone). Far downstream of the slot however the mixing layer has spread but the mainstream turbulence has decayed to a lower value. The metal temperature now depends upon the combined effect of these two factors. The distribution of the turbulence intensity along the length of the combustion chamber is so chosen that the effects of additional turbulence and mixing due to the jet penetration in the intermediate and the dilution zones, is partly accounted for in these results. Based on this distribution some conclusions may be drawn.

i) The effects of turbulence are greater in the primary and dilution zone than in the intermediate zone.

ii) The preservation of the coolant film is very important e.g. by increasing \( m \) slightly, the film can be restored and a good reduction in \( T_w \) achieved because of the immediate increase in the heat transfer rates above their flat plate values.
iii) The formation of the mixing layer between the hot mainstream and the coolant tends to isolate the direct effects of the mainstream turbulence. Due to recirculation in the primary zone however, $T_w$ falls less rapidly with $m$ here, than in the other two zones.

It must be noted that these effects of turbulence are confined only to the clean slots with no lip thickness. When a thick lip is introduced ($t/s = 0.5$ say), and if it is assumed that this generates a turbulence intensity of 22% at about 5 slot heights downstream - then we have

$$\left(\frac{u'}{U}\right)^2 = \frac{1}{(x/t)}$$

which gives $x/s = 47.0$ for $u'/U = 5\%$. i.e. as high as 5% turbulence intensity exists at $x/s = 47.0$. This turbulence generated by the wake of the lip, must protect the slot flow from the effects of the mainstream turbulence, since Fig.14 shows a much lower wall temperature prediction for the finite lip slot.

7.0 EFFECTS OF THE HOT GAS ACCELERATION

As shown in Fig.1, downstream of the dilution zone, the discharge nozzle is formed by a rapidly converging wall section which accelerates the hot gases before leaving the combustor. This converging wall has to be cooled since it bears the direct impingement of the accelerating hot gas flow. The effectiveness results of Carlson and Talmor (15) have been utilized together with the flat plate heat transfer equation corrected for the wall convergence, Ref.(15). The analytical and the experimental results are shown in Fig.15. As the wall inclination increases, $T_w$ rises because of the

i) increase in impinging action of the hot mainstream gases;

ii) boundary layer film of the cold air protecting the wall becomes increasingly thin due to the rapid increase in the hot outer scrubbing velocity.

Fig.15 also shows that the effects of mainstream turbulence at these higher rates of hot gas acceleration are insignificant because the acceleration through this converging discharge nozzle reduces the turbulence.

8.0 OPTIMIZATION OF THE COOLING AIR REQUIREMENTS

By referring to Table III, various schemes for the optimization of the cooling air can be suggested. The ideal optimization for the minimum cooling flow requirement must bring the values of $\omega_{opt}$ and $T_w$ to a minimum simultaneously. However, in practice this would limit the choice of the slot geometries and excessively constrain the combustor cooling design. Three different schemes of optimization which do not have this limitation, can be visualized.
8.1 Optimization for the Maximum Permissible Metal Temperature

Here the value of $m_{opt}$ is chosen so that $T_w = T_{max}$ permitted. This would operate the cooling system at its peak efficiency. Depending upon the limits of the maximum permitted wall temperatures, a proper choice of the slot configuration to suit the mechanical requirements can be made. Thus, here the operational life of the combustor is sacrificed at the cost of an increase in its efficiency and hence compactness.

8.2 Optimization for the Minimum Metal Temperature

Here, the cooling air requirement is fixed by a minimum mixing (of hot mainstream and cold film) criterion i.e. $m_{opt} = 1.0$. This does not always lead to the minimum metal temperatures, except in the case of an adiabatic wall.

8.3 Optimization for Minimum Mixing

Here, the cooling air requirement is fixed by a minimum mixing (of hot mainstream and cold film) criterion i.e. $m_{opt} = 1.0$. This does not always lead to the minimum metal temperatures, except in the case of an adiabatic wall.

The results of optimization based on the above three methods are shown in Table IV and Fig.16, both of which are self explanatory.

9.0 CONCLUSIONS

An analytical study, (supported by the experiments wherever possible) is made of the estimation and optimization of the cooling air requirements for a modern aero gas turbine combustion chamber over its operational flight path. The conclusions of this study are:

a) The optimum value of $m$ for minimum wall temperatures deviates significantly from unity and this depends upon what fraction of the heat gained due to the internal radiation is lost in the internal and the external convection by the wall.

b) Increasing pressure leads to increasing wall temperatures, but this effect is gradually reduced at higher pressures. The conclusions of Marsland, Odger and Winter (3) in this respect are confirmed.

c) External heat convection removes anywhere between 60-80% of the total heat flux from the wall, over the whole of the altitude and mach number range of operation of the combustor.

d) Film cooling is generally most beneficial at lower altitudes and mach numbers.
e) The optimum cooling air requirement is most affected by $R_1$ in the primary zone, $C_1$ and $C_2$ in the intermediate zone and $T_m$ in the dilution zone.

f) Long narrow clean slots with tangential injection or inclined slots give low $m_{opt}$ and $T_w$ values, only under no mainstream turbulence conditions. Clean finite lip slots or geometries like inclined holes with cover plate, protect the ensuing coolant flow from the deteriorating effects of mainstream turbulence and in general yield better results in the practical situations.

g) Splash cooling geometries are to be preferred in the primary zone. In the intermediate and dilution zones large open area ratios are required to give reduced metal temperatures.

h) The effects of mainstream turbulence are greater in the primary and the dilution zone than in the intermediate zone. Any small increase in $m$ which restores the coolant film immediately reduces the wall temperatures.

i) The predicted effects of hot gas acceleration show good agreement with the actual wall temperatures obtained by using thermal paint tests.

j) The optimization study reveals that for a combustor of Fig.1, only by readjusting the coolant flow proportions, anything between 8-15% (of the total combustor mass flow) saving in the cooling air requirement can be obtained.
10.0 REFERENCES


REFERENCES (Contd.)

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    "Discussion on above Paper - (Ref.10)," pp.393, October (1968)

    "Heat transfer to film cooled combustion chamber liners"
    ASME paper No. 72-WA/HT-32, Nov.(1972)

    "The effectiveness of film cooling with three-dimensional slot geometries"

    "The effects of slot height and slot turbulence intensity on the effectiveness of the uniform density, two-dimensional wall jet"

15. Carlson, L.W. and Talmor, E.
    "Gaseous film cooling at various degrees of hot-gas acceleration and turbulence levels"

16. Ballal, D.R. and Lefebvre, A.H.
    "Film cooling effectiveness in the near-slot region"
Table I

Combustor inlet conditions over the engine flight path

<table>
<thead>
<tr>
<th></th>
<th>$T_2$ (°K)</th>
<th>$P_2$ (atm)</th>
<th>$M_e$ (Kgs/sec)</th>
<th>AFR</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1) Take-Off</td>
<td>800</td>
<td>25.2</td>
<td>6.8</td>
<td>50</td>
<td>Sea Level</td>
</tr>
<tr>
<td>2) Cruise</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>770</td>
<td>17.7</td>
<td>4.9</td>
<td>65</td>
<td>Altitude $6.1 \times 10^3$ m</td>
</tr>
<tr>
<td></td>
<td>670</td>
<td>6.8</td>
<td>2.04</td>
<td>70</td>
<td>Mach No. 12.2</td>
</tr>
<tr>
<td></td>
<td>670</td>
<td>2.72</td>
<td>0.82</td>
<td>75</td>
<td>0.8</td>
</tr>
<tr>
<td></td>
<td>670</td>
<td>1.02</td>
<td>0.32</td>
<td>80</td>
<td>18.3</td>
</tr>
<tr>
<td></td>
<td>670</td>
<td>1.02</td>
<td>0.32</td>
<td>80</td>
<td>24.4</td>
</tr>
<tr>
<td>3) Acceleration</td>
<td>600</td>
<td>2.04</td>
<td>0.635</td>
<td>65</td>
<td>Mach No. 0.4</td>
</tr>
<tr>
<td></td>
<td>620</td>
<td>2.60</td>
<td>0.82</td>
<td>70</td>
<td>Level flight at</td>
</tr>
<tr>
<td></td>
<td>660</td>
<td>3.74</td>
<td>1.13</td>
<td>75</td>
<td>18.3 x 10^3 m</td>
</tr>
<tr>
<td></td>
<td>720</td>
<td>4.76</td>
<td>1.35</td>
<td>75</td>
<td>r = 1.25</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>6.80</td>
<td>1.82</td>
<td>80</td>
<td>2.0</td>
</tr>
</tbody>
</table>

$m_{opt} = 1.15$  
$m$ varied from 0.25 to 5.0

Table II: Change in cooling air requirement $m_{opt}$ due to 70% increase in heat transfer components.
### Table III
Study of various slot configurations for combustor of Fig.1 (T/Off Cond.)

<table>
<thead>
<tr>
<th>Slot Geometry</th>
<th>$n$</th>
<th>$h_1$</th>
<th>$\tau_{\text{opt}}$</th>
<th>$T_W$ at $\tau_{\text{opt}}$ ($^\circ$K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1) Clean slots with tangential injection</td>
<td>Prediction eq. of Ref.(16)</td>
<td>Prediction eq. of Ref.(1)</td>
<td>2.5</td>
<td>1.5</td>
</tr>
<tr>
<td>2) Clean Slots with finite lip t/s=0.5</td>
<td>Prediction eq. of Ref.(1)</td>
<td>-ditto-</td>
<td>2.5</td>
<td>1.5</td>
</tr>
<tr>
<td>3) Stacked or machined rings (Dirty Slots)</td>
<td>Correlations due to Sturgess (5) $\tau_p = 0$, $A_o/A_e = 1.50$</td>
<td>-ditto-</td>
<td>No optimum &amp; hence values at $T_W=1100^\circ$K</td>
<td>2.5</td>
</tr>
<tr>
<td>4) Inclined Slots a) 30$^\circ$ b) 90$^\circ$</td>
<td>Expt. data of Ref. (6,7)</td>
<td>Expt. data of Ref. (6)</td>
<td>2.0</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>Ref. (7,9)</td>
<td>Ref. (9)</td>
<td>0.5</td>
<td>0.7</td>
</tr>
<tr>
<td>5) Inclined Holes. a) Z/D = 0.0 35$^\circ$ b) Z/D = 0.5 35$^\circ$</td>
<td>Expt. data of Ref.(10)</td>
<td>Flat Plate (Colburn) eq. Metzger's (11) comments</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Metzger (12) et. al. Modified Expt. Data</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>6) Inclined Holes with cover plate protection</td>
<td>*M.etzer (12) et. al. Modified Expt. Data</td>
<td>Metzger (12) et. al. Modified Expt. Data</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>1160</td>
<td>1100</td>
</tr>
<tr>
<td>7) 3D and Splash Cooling geometries</td>
<td>Nina &amp; White-law (13) expt. data</td>
<td>Prediction eq. of Ref.(1)</td>
<td>1.3</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(6)</td>
<td>(1)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(13)</td>
<td>(12)</td>
</tr>
<tr>
<td>8) Effects of mainstream Turbulence 32Z 122 222</td>
<td>Expt. data of Carlson &amp; Talmor (15)</td>
<td>Expt. data of Ref.(6)</td>
<td>1.8</td>
<td>2.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flat Plate Eq.</td>
<td>4.5</td>
<td>3.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flat Plate Eq.</td>
<td>5.5</td>
<td>4.0</td>
</tr>
<tr>
<td>9) Effects of Hot Gas acceleration 7.52 122</td>
<td>Expt. data of Carlson &amp; Talmor (16)</td>
<td>Expt. data of Carlson &amp; Talmor (15)</td>
<td>1.5</td>
<td>1.75</td>
</tr>
</tbody>
</table>

*Data used in Metzger's (12) correlation.*
## Table IV

### Cooling Air Optimization for Combustor of Fig.1 (T/off condts)

<table>
<thead>
<tr>
<th>Method of Optimization</th>
<th>SLOT Geometry used</th>
<th>% reduction in total (Mt) cooling air</th>
<th>$\Delta T_W=T_W-1100^\circ K$ Metal temp. variation along combustor length</th>
</tr>
</thead>
<tbody>
<tr>
<td>1) Use of $T_W=T_{\text{max}}$ permitted wall temperature criterion</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2) Use of $M_C=M_{\text{opt.}}$ Criterion</td>
<td>i) Clean slot</td>
<td>6%</td>
<td>-130 to -100°K</td>
</tr>
<tr>
<td></td>
<td>ii) Finite lip slot</td>
<td>4-5%</td>
<td>-40°K</td>
</tr>
<tr>
<td></td>
<td>iii) 30° inclined clean slot</td>
<td>7%</td>
<td>-100 to -150°K</td>
</tr>
<tr>
<td></td>
<td>iv) Inclined holes with cover plate protection</td>
<td>16%</td>
<td>+ 60 to + 20°K</td>
</tr>
<tr>
<td></td>
<td>v) 3D and splash cooling geometry</td>
<td>up to 1%</td>
<td>+ 50 to -50°K</td>
</tr>
<tr>
<td>3) Use of $m_{\text{opt}}=1.0$ Criterion</td>
<td>i) Clean slots</td>
<td></td>
<td>-50°K</td>
</tr>
<tr>
<td></td>
<td>ii) Finite lip slots</td>
<td></td>
<td>70°K to -50°K</td>
</tr>
<tr>
<td></td>
<td>iii) 30° inclined clean slots</td>
<td></td>
<td>-50°K</td>
</tr>
<tr>
<td></td>
<td>iv) Incl. holes with cover plate protection</td>
<td>16%</td>
<td>60 to 40°K</td>
</tr>
<tr>
<td></td>
<td>v) 3D &amp; splash cooling geometry</td>
<td></td>
<td>+ 30 to -50°K</td>
</tr>
</tbody>
</table>

No optimization has been carried out for 2 discharge nozzle splash strips and 3 flare splash strips since the level of turbulence and hot gas acceleration are unknown.
FIG 1: SCHEMATIC DIAGRAM OF THE GAS TURBINE COMBUSTION CHAMBER USED IN THE PRESENT STUDY.
FIG 2: COMPARISON OF PREDICTED AND EXPERIMENTAL VALUES OF EMISSIVITY.
FIG 3: COMPARISON OF PREDICTED AND EXPERIMENTAL VALUES OF $R_1$ AND $T_w$
FIG 4: COMPUTATION SCHEME FOR THE CALCULATION OF $T_w$ AND $M_{opt}$
FIG 5: INTERATIVE SOLUTION OF ENTHALPY BALANCE EQUATION 
($R_1 + C_1 = R_2 + C_2$) AT THE LINER WALL.

(TABLE I: INLET CONDITION 2a)
FIG 6a: EFFECTS OF THE INDIVIDUAL FLOW PARAMETERS ON WALL TEMPERATURE IN THE PRIMARY ZONE. (m = 1.3)

FIG 6b: EFFECTS OF THE SLOT FLOW PARAMETERS ON WALL TEMPERATURE IN THE PRIMARY ZONE. (m = 1.3)
FIG 7: COMBINED EFFECT OF INLET PARAMETERS IN THE PRIMARY ZONE.
(TABLE I: INLET CONDITIONS 2a TO 3e)
FIG 8: MAGNITUDE OF THE INDIVIDUAL COMPONENTS OF HEAT TRANSFER IN THE PRIMARY ZONE. (m = 1.3)
FIG 9a: MAGNITUDE OF THE INDIVIDUAL COMPONENTS OF HEAT TRANSFER IN THE PRIMARY ZONE. (m = 1.3)

FIG 9b: MAGNITUDE OF EXTERNAL CONVECTIVE COMPONENT C₂ OVER A TYPICAL AIRCRAFT FLIGHT PATH.
FIG 10: EFFECTS OF THE INDIVIDUAL MAIN STREAM PARAMETERS ON $M_{\text{opt}}$ ($T/\text{off}$).
FIG 11: EFFECTS OF ALTITUDE AND MACH NO. ON $m_{opt}$
FIG 12: EFFECTS OF INDIVIDUAL SLOT PARAMETERS AND HEAT TRANSFER COMPONENTS ON $M_{\text{opt}}$
FIG 13a: EFFECTS OF VARIOUS SLOT CONFIGURATIONS ON THE WALL TEMPERATURE (T/off).
FIG 13b: EFFECTS OF VARIOUS SLOT CONFIGURATIONS ON THE WALL TEMPERATURE ($T_{\text{off}}$).

SLOT GEOMETRY NO. (TABLE III)

- 5a
- 6a
- 7a
- 6b
- 7b
FIG 14: EFFECTS OF MAIN STREAM TURBULENCE ON THE WALL TEMPERATURE. (T / off).
FIG 15: EFFECTS OF HOT GAS ACCELERATION IN THE COOLING OF DISCHARGE NOZZLE WALL. (I: EXPERIMENTAL RESULTS FOR THE COMBUSTOR OF FIG 1. T/off).
FIG 16: COOLING AIR OPTIMIZATION FOR THE COMBUSTOR. (T/off CONDITION).