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By

D. R. BALLAL and A. H. LEFEBVRE

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A PROPOSED METHOD FOR CALCULATING FILM-COOLED WALL TEMPERATURES IN GAS TURBINE COMBUSTION CHAMBERS

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ABSTRACT

The growing awareness of pollutant emissions from gas turbine engines is highlighting the need to minimise the amount of air employed in liner film cooling and thereby reduce the output of carbon monoxide and unburned hydrocarbons. To this end an analysis has been made of experimental data obtained on a specially designed film-cooling rig which has led to the development of a calculation procedure for liner wall temperatures in the presence of film cooling.

The purpose of this paper is to describe the procedure and show the level of agreement between predicted values of wall temperature and the results of thermal paint tests carried out an an aircraft combustor at pressures up to 30 atmospheres. The implication of the results to the design of cooling slots for optimum performance is also discussed.

CONTENTS

FIGURES

NOMENCLATURE

A	=	cross sectional area, (m^2)
C	=	convective heat transfer, (W/m^2)
C_p	=	specific heat at constant pressure, (J/kg deg C)
D	=	diameter, (m)
e	=	gas emissivity
h	=	heat transfer coefficient, $(W/m^2$ deg C)
L	=	flame luminosity factor
ℓ	=	radiation beam length, (m)
M	=	mass flow, (kg/sec)
m	=	mass velocity ratio, $(\rho_c U_c/\rho_m U_m)$
Nu	=	Nusselt number based on x, (hx/λ)
n	=	number of slots
P	=	total pressure (atm)
Pr	=	Prandtl number $(C_p\mu/\lambda)$
R	=	radiation heat transfer, (W/m^2)
Re_s	=	Reynolds number based on slot height $(\rho_c U_c s/\mu_c)$
Re_x	=	Reynolds number based on x, $(\rho_c U_c x/\rho_c)$
r	=	fuel/air ratio by weight
St	=	Stanton number $(h/C_p\rho_c U_c)$
s	=	slot height (m)
T	=	temperature (^oK)
ΔT	=	combustion temperature rise (^oK)
t	=	lip thickness (m)
u	=	velocity (m/sec)
x	=	distance downstream of the slot (m)
Θ	=	ratio of diffusivity of heat to diffusivity of momentum $(\varepsilon_H/\varepsilon_M)$
η	=	effectiveness of film cooling = $(T_m - T_{w_{ad}})/(T_m-T_c)$
λ	=	gas conductivity (W/m deg K)
μ	=	gas viscosity (kg/m sec)
ρ	=	gas density (kg/m^3)

Subscripts

ad = adiabatic condition

an = annulus

c = pertaining to coolant (slot) flow

ℓ = pertaining to liner

m = pertaining to mainstream flow

w = pertaining to wall

1 = internal to the liner

2 = external to the liner

INTRODUCTION

Film cooling is widely used in gas turbine combustion chambers as a means of protecting the liner walls from the heating effects of the adjacent hot gas stream. On modern aircraft gas turbines up to one-third of the total combustor air flow is used in film cooling the liner. Unfortunately, the injection of such large amounts of cool air at the liner walls has an adverse effect on the radial temperature distribution in the outlet gas, and also tends to reduce combustion efficiency, especially at altitude cruise and idling conditions. As the emissions of carbon monoxide and unburned hydrocarbons are directly related to combustion inefficiency, there is an obvious need to reduce the amount of film-cooling air to an absolute minimum. This implies a better understanding of the film-cooling process in gas turbine combustors, and so provides the incentive for the present investigation.

In a previous paper (1) equations were developed for predicting film-cooling effectiveness downstream of two-dimensional, thin lipped, 'clean' slots featuring tangential injection of the coolant along an adiabatic wall. Attention was focussed on the important 'near-slot' region because most other workers had concentrated their studies on regions far downstream of the injection slot. Analysis of flow conditions in this region yielded simple expressions for cooling effectiveness in terms of the velocity ratio of the coolant and main-stream gases, and the non-dimensional distance downstream of the slot.

The purpose of the present work is to show how the effectiveness expressions, derived for clean slot geometries, may be modified to account for the effects of slot lip thickness. A further objective is to examine the equally important problem of estimating the local heat transfer coefficients to be used in conjunction with the effectiveness equations for calculating wall temperatures in film-cooled combustors.

HEAT TRANSFER COEFFICIENTS

In the calculation of heat fluxes at the liner wall, the normal practice is to use the standard impermeable wall turbulent heat transfer relationship.

$$h = \frac{C}{T_{w_{ad}} - T_w} \quad .. \quad (1)$$

This supposition has found experimental support in the work of Scesa (2), Hartnett, Birkebak and Eckert (3) and Seban (4). The work of Seban and Back (5, 6) and Pai and Whitelaw (7) has also shown that for $x/s > 50$ the Colburn equation for flat plate heat transfer is obeyed.

i.e. $$Nu = 0.0256 \left[Re_s\ (x/ms)\right]^{0.8} \quad \ldots (2)$$

assuming $Pr_{air} = 0.69$

Nearer the slot the geometry and flow field at the point of coolant injection become significant variables. In this near-slot region it is convenient to consider separately a boundary layer model for $m < 1.3$, and a wall-jet model which provides a more accurate description of the flow field when $m > 1.3$ (1).

Boundary Layer Model

In (1) equations for predicting film-cooling effectiveness were developed from considerations of boundary layer thickness and skin friction coefficient. However, when skin friction factors are used with the Colburn analogy to determine heat transfer coefficients they tend to underestimate heat flux rates in the near-slot region (4,5). This discrepancy stems from the fact that h is dependent on temperature profile, and therefore on heated length rather than hydrodynamic length downstream of the slot. Analysis of available data (3-5, 7) reveals that heat transfer coefficients in the near slot region are adequately described for $m < 1.3$ by the following expression due to Seban (4)

$$Nu = 0.090 \left[Re_s\ (x/s)\right]^{0.7} \quad \ldots \quad (3)$$

The correlation obtained is illustrated in Fig.1

Wall Jet Model

A useful amount of experimental data exists in the literature that is directly relevant to the wall jet case. Research at Cranfield has also concentrated on this area since most modern combustors feature cooling slots in which the coolant mass velocity exceeds that of the main stream. The working section of the test rig comprises a circular duct, 14cm. in diameter, through which hot air flows, with provision to introduce an annular film of cold air at the wall. A tubular, quartz, infra-red heater is located along its central axis in order to provide a radiation component at the wall and thus allow heat transfer coefficients to be determined in addition to effectiveness. The results obtained on heat transfer coefficients are shown plotted in Fig.2, along with those of Pai and Whitelaw (7).

Several attempts at analytical prediction of heat transfer coefficient have been made (8). Myers et al (9) have derived an equation for local Stanton number having the form

$$St.\ Re_s^{0.2}\ 10^2 \left[1 - (\ell/x)^{0.45}\right]^{0.0625} = 7.9\ \theta^{0.89} (x/s)^{-0.56} \quad \ldots (4)$$

where ℓ is the unheated starting length.

By putting $\ell = 0$, i.e., by assuming the velocity and thermal boundary layers to grow simultaneously, and for Pr = 0.69, equation (4) simplifies to

$$Nu = 0.0545\ \theta^{0.89}\ Re_s^{0.8}\ (x/s)^{0.44} \quad \ldots \quad (5)$$

The value of θ depends on the flow field and the turbulence level. It varies from 1.0 for a flat plate to 2.4 for a free jet. Since a wall jet is a combination of a flat plate boundary layer and a free jet, especially in the near-slot region, it would appear best to use a value of 2.0 for θ, since this corresponds to the maximum influence of free jet hydrodynamics on the wall jet and is thus most appropriate for the near-slot region, say, $x/s < 50$. For $\theta = 2.0$, equation (5) becomes

$$Nu = 0.10\ Re_s^{0.8}\ (x/s)^{0.44} \quad \ldots \quad (6)$$

Equation (6) is shown plotted in Fig. 2 alongside corresponding equations obtained from (4) for different values of θ.

FILM COOLING EFFECTIVENESS

In recent years a growing awareness of the adverse effects of film-cooling air on exhaust emissions and temperature traverse quality has led to the development of new designs of cooling slot in which the air flow quantity is minimized by ensuring almost complete uniformity of velocity at the slot outlet. For this type of slot the equations previously derived (1) for predicting effectiveness in the near-slot region are directly applicable provided suitable correction is made for the effects of finite slot-lip thickness. The presence of the lip, whose thickness may, for reasons of mechanical integrity, represent an appreciable proportion of the slot height, introduces into the mixing zone a wake drag which tends to shorten the length of the potential core and extend the transition region. From analysis of the experimental data obtained by Kacker and Whitelaw (10, 11) and Sivasegaram and Whitelaw (12) on the effects of slot-lip thickness on effectiveness, the following empirical correction factor has been derived

$$C.F. = 1.83\ (x/s)^{0.1}\ (t/ms)^{-0.2}\ Re_s^{-0.15} \quad \ldots \quad (7)$$

Introduction of this correction factor gives:-

for $0.5 < m < 1.3$

$$\eta = 1.10 m^{0.65}\ (\mu_c/\mu_m)^{0.15}\ (x/s)^{-0.2}\ (t/s)^{-0.2} \quad \ldots \quad (8)$$

and for $1.3 < m < 4.0$

$$\eta = 1.28\ (\mu_c/\mu_m)^{0.15}\ (x/s)^{-0.2}\ (t/s)^{-0.2} \quad \ldots \quad (9)$$

The validity of equations (8) and (9) is demonstrated in Figs. 3 and 4 in which experimental data from (10, 11, 12) are compared with predicted values based on equations (8) and (9) for boundary layer and wall jet models respectively. Although these equations take no account of adverse or favourable pressure gradients, acceleration of the mainstream flow and initial thickness of the boundary layer, prediction capability is nevertheless high in the range of interest to combustor designers, because the works of Hartnett et al (3), Seban (4), Seban and Back (6) and Pai and Whitelaw (13) have shown that, for $x/s < 50$, these effects are insignificant and may be ignored. It has also been shown by Carlson and Talmor (14) that the mainstream turbulence has only a slight effect on effectiveness in the near-slot region.

CALCULATION OF FILM-COOLED WALL TEMPERATURES

For an uncooled liner wall the relevant heat transfer components are, from (15).

$$R_1 = 48.7\ 10^{-9}\ e_m T_m^{1.5}\ (T_m^{2.5} - T_w^{2.5}) \qquad \ldots \quad (10)$$

$$C_1 = 0.017\ (\lambda_m/\mu_m^{0.8})\ (M_m^{0.8}/A_\ell^{0.8}\ D_\ell^{0.2})\ (T_m - T_w) \qquad \ldots \quad (11)$$

$$R_2 = 22.9\ 10^{-9}\ (T_w^4 - T_2^4) \qquad \ldots \quad (12)$$

$$C_2 = 0.02\ (\lambda_m/\mu_m^{0.8})\ (M_{an}^{0.8}/A_{an}^{0.8}\ D_{an}^{0.2})\ (T_w - T2) \qquad \ldots \quad (13)$$

For equilibrium $R_1 + C_1 = R_2 + C_2$

For calculation of film-cooled wall temperatures the above expressions for R_1, R_2 and C_2 remain the same, but the internal convection component, C_1, is altered because the coolant flow changes both the velocity and the temperature of the gas adjacent to the wall. Dealing with velocity first, we have, from equations (3) and (6)

for $0.5 < m < 1.3$

$$C_1 = 0.090\ (\lambda/x)\ Re_x^{0.7}\ (T_{w_{ad}} - T_w) \qquad \ldots \quad (14)$$

while for $m > 1.3$

$$C_1 = 0.10\ (\lambda/x)\ Re_x^{0.8}\ (x/s)^{-0.36}\ (T_{w_{ad}} - T_w) \qquad \ldots \quad (15)$$

The gas temperature at the wall, $T_{w_{ad}}$ is obtained from the definition of η, i.e.

$$\eta = (T_m - T_{w_{ad}}) \big/ (T_m - T_c)$$

where η is the effectiveness value calculated from equations (8) or (9).

The manner in which the equations may be used to calculate liner wall temperatures in the presence of film-cooling air is illustrated in the sample calculation presented in the Appendix.

With the aid of a computer a large number of calculations has been carried out on various representative types of gas turbine combustor.

The results obtained on a single, aircraft type, tubular combustor are shown in Figs. 5 and 6, corresponding to pressure levels of 22 and 30 atmospheres respectively. Also plotted on these figures for comparison are hatched areas representing the metal temperature levels indicated by thermal paint tests carried out on the combustor at the stipulated operating conditions. It is clear from inspection of the figures that the proposed method provides a very satisfactory prediction of the rate of increase of wall temperature downstream of a film-cooling slot. In regard to absolute values of wall temperature, the figures show that the best prediction is obtained using values of combustion efficiency of 85% in the primary zone, 90% rising to 95% along the intermediate zone, and 99% in the dilution zone.

It should be emphasized that precise calculations of wall temperature in the dilution zone is impossible owing to the wide difference which exists between the mean gas temperature in this region and the gas temperature adjacent to the wall. The line shown in the figures was calculated on the assumption that the hot gas temperature adjacent to the wall was reduced by the gradual mixing in of 50% of the dilution air along the length of the dilution zone. This somewhat arbitrary approach gives reasonable answers, as shown in the figures, but should be used with caution if the method is applied to annular chambers, where the dilution-zone airflow pattern is basically different.

DESIGN CONSIDERATIONS

It is of interest to examine the derived equations for film-cooling effectiveness with a view to obtaining guidance on important design factors, such as the optimum height and spacing of cooling slots. This process is fairly simple because for any given combustor, at any fixed operating condition, one can assume that μ_c, μ_m, ρ_c, ρ_m and u_m are constant and $m \propto M_c/s$.

(a) $0.5 < m < 1.0$

From equation (8) for a constant lip thickness, t, we have

$$\eta \propto \frac{M_c^{0.65}}{x^{0.2}\, s^{0.25}} \qquad \ldots \quad (16)$$

This equation shows that film-cooling effectiveness is improved by increasing the amount of cooling air and by reducing the height of the slot.

Consider now the effect of increasing the number of slots while keeping both the slot height and the total amount of cooling air constant. The cooled distance is now x/n, where n is the number of slots.

From equation (16)

$$\eta \propto \frac{M_c^{0.65}}{x^{0.2}\ s^{0.25}} \quad (\eta^{\ 0.45}) \qquad \ldots \quad (17)$$

Comparing equations (16) and (17) it is seen that effectiveness is reduced by the factor $n^{-0.45}$.

It is also of interest to consider the effect of increasing the number of slots and reducing the slot height proportionately so that the cooland velocity remains constant. The cooled distance is again x/n. From equation (16)

$$\eta \propto \quad (M_c/\eta)^{0.65} \quad (x/n)^{\ 0.2} \quad (n/s)^{0.25}$$

$$\text{or} \qquad \eta \propto \frac{M_c^{0.65}}{x^{0.2}\ s^{0.25}} \quad (n^{\ 0.2}) \qquad \ldots \quad (18)$$

Comparison of equations (16), (17) and (18) shows that effectiveness is still reduced but not as much as before. Thus, under conditions where the coolant velocity is less than the mainstream velocity, for any given total amount of cooling air, it always pays to have the minimum number of cooling slots and each slot should be of minimum height.

(b) $m > 1.0$

Under these conditions calculations carried out by the same procedure described above show that effectiveness is now improved by an increase in number of slots, corresponding to a reduction in u_c towards the optimum value of $m = 1.0$. Thus, the main conclusion to be drawn from this simple analysis is that any modification to the geometry of the liner or cooling slot that tends to move m towards an optimum value of unity will lead to either a reduction in cooling air requirements or to a more efficient utilization of the available air.

This generalization is, of course, based solely on considerations of effectiveness. In practice the task of minimizing the convective heat flux to the wall depends not only on effectiveness but also on the heat transfer coefficient and the influence of gas radiation. In most gas turbine combustors the effect of taking gas radiation into account is to raise the optimum value of m above unity by an amount which depends on the ratio of R_1/C_1. This is illustrated in Fig.7 which shows the results of calculations carried out on the tubular combustor described earlier, when operating at its normal take-off condition. Calculations on other combustor types show that a good general design target for m is around 2.0 for the primary zone and about 1.4 for the dilution zone. Further increase in m at first impairs film cooling by increasing the rate of mixing between the coolant and mainstream gases, but continued increase in m eventually leads to reduction in wall temperatures due to the presence of large quantities of coolant adjacent to the wall. It is of interest to note in Fig.7 that in this particular combustor the actual values of m are far from ideal. A reduction in cooling air to almost half the present values would effect appreciable performance gains and also reduce the general level of metal temperature.

Fig.8 shows for the same combustor the effect of varying the slot height in all three zones while maintaining the same amount of cooling air through each slot. As would be expected from the discussion of Fig.7, with this particular liner an increase in slot height would be generally beneficial, but especially so in the dilution zone. It should, however, be emphasized that the main purpose of Figs. 7 and 8 is not to provide design rules but merely to illustrate the type of calculation that ought to be carried out on every cooling slot in the combustor design stage in order to optimize its cooling potential and thereby accomplish economies in cooling air, with attendant benefits in terms of good temperature traverse quality and low pollutant emissions.

CONCLUSIONS

In spite of doubts and uncertainties in estimating the flow of heat to the liner wall, especially in the primary combustion zone where the luminous radiation component is a significant but partly unknown factor, it is possible to calculate liner wall temperatures in the presence of film cooling to a level of accuracy that is satisfactory for most design purposes. Predictions of wall temperature from the derived equations show good agreement with thermal paint tests carried out on the actual liner, and therefore support their application in the design stage to determining the flow requirements and also the optimum number and height of cooling slots for any given combustor.

REFERENCES

1. BALLAL, D.R. and LEFEBVRE, A.H. — The prediction of film cooling effectiveness in the near-slot region. - To be published.

2. SCESA, S. — Effect of local normal injection on flat plate heat transfer. Ph.D. Thesis. University of California - Berkeley, U.S.A. 1954.

3. HARTNETT, J.P., BIRKEBAK, R.C. and ECKERT, E.R.G. — Velocity distributions, temperature distributions, effectiveness and heat transfer for air injection through a tangential slot into a turbulent boundary layer. Journal of Heat Transfer, Trans ASME, Series C. Vol. 83, No.3 August 1961, pp.293-306.

4. SEBAN, R.A. — Heat transfer and effectiveness for turbulent boundary layers with tangential fluid injection. Journal of Heat Transfer, Trans ASME, Series C, Vol.82, No.4, November 1960, pp. 303-313

5. SEBAN, R.A. and BACK, L.H. — Velocity and temperature profiles in turbulent boundary layers with tangential injection. Journal of Heat Transfer, Trans ASME, Series C, Vol. 84, February, 1962, pp.45-54.

6. SEBAN, R.A., and BACK, L.H. — Effectiveness and heat transfer for a turbulent boundary layer with tangential injection and variable free-stream velocity. Journal of Heat Transfer. Trans ASME, Series C, Vol. 84, August 1962, pp.235-242.

7. PAI, B.R. and WHITELAW, J.H. — The Prediction of wall temperature in the presence of film cooling. International Journal of Heat and Mass Transfer, Vol.14, March 1971, pp.409-425.

8. LEONTE'V, A.I. — Heat and mass transfer in turbulent boundary layers. Advances in Heat Transfer, Vol.3, Academic Press, New York, 1966, pp.80-100

9. MYERS, G.E., SCHAUER, J.J. AND EUSTIS, R.H. — Heat transfer to plane turbulent wall jets, Journal of Heat Transfer, Trans ASME, Series C, Vol. 85, August 1963, pp.209-214.

10. KACKER, S.C. & WHITELAW, J.H. An experimental investigation of the influence of slot lip thickness on the impervious wall effectiveness of the uniform density, two-dimensional wall jet. International Journal of Heat and Mass Transfer, Vol. 12, 1969, pp.1196-1201.

11. KACKER, S.C., & WHITELAW, J.H. The effect of slot height and slot turbulence intensity on the effectiveness of the uniform density two-dimensional wall jet. Journal of Heat Transfer, Trans ASME, Series C, Vol. 90, 1968, pp.469-475.

12. SIVASEGARAM, S. and WHITELAW, J.H. Film cooling slots: the importance of lip thickness and injection angle". Journal of Mechanical Engineering Science, Institution of Mechanical Engineers, London, Vol. II, 1969, pp.22-27.

13. PAI, B.R., and WHITELAW, J.H. The influence of strong pressure gradients on film cooling, EHT/TN/A/15, April 1969, Department of Mechanical Engineering, Imperial College, London.

14. CARLSON, L.W., and TALMOR, E. Gaseous film cooling at various degrees of hot-gas acceleration and turbulence levels. International Journal of Heat and Mass Transfer, Vol. II, 1968, pp. 1695-1713.

15. LEFEBVRE, A.H., and HERBERT,M.V. Heat transfer processes in gas turbine combustion chambers". Proceedings of the Institution of Mechanical Engineers, London, Vol. 174, No.12, 1960, pp.463-473.

APPENDIX

It is proposed to calculate wall temperature at a point $x/s = 18$ downstream of the first row of cooling slots in the primary zone of a combustor operating at an inlet pressure and temperature of 30 atm. and 880^oK respectively. The procedure is as follows:-

Flow Parameters:

From the liner geometry it is known that:

$x/s = 18.0$; $t/s = 0.4$; $A_c = 5.95\ cm^2\ (0.92\ in^2)$

$M_c = \rho_c U_c A_c = 0.289$ Kgs/sec. (0.636 lb./sec.)

Hence $Re_s = \rho_c U_c s/\mu_c = 1.74 \times 10^4$; $Re_x = \rho_c U_c x/\mu_c = 3.17 \times 10^5$

Also $A_m = 137.0\ cm^2\ (21.2\ in^2)$; Local fuel/air ratio (LFAR) = 0.0588

$M_m = \rho_m U_m A_m = 2.62$ Kgs/sec (5.78 lb/sec)

Assume a local combustion efficiency = 85%; then $T_m = 2{,}280^oK$.

Film Cooling Effectiveness:

Specific flow ratio $m = \rho_c U_c/\rho_m U_m = 2.54$

Since $m > 1.3$, using equation (9) of the text gives $\eta = 0.785$

Hence $T_{w_{ad}} = 1180^oK$ . . . (19)

Heat Transfer Calculation:

Since $m > 1.3$ we use equation (6)

$\therefore\ Nu = h.x/\lambda = 0.1\ Re_x^{0.8}\ (x/s)^{-0.36}$

$\therefore\ h = 1.96\ kw/m^2\ ^oC\ (0.0955\ CHU/ft^2\ sec\ ^oC)$ . . . (20)

From (15) $e_m = 1 - \exp\left[-8.85 \times 10^3\ L.P\ (r l)^{0.5}\ T_m^{-1.5}\right] = 0.60$

For enthalpy balance at the wall -

$$R_1 + C_1 = R_2 + C_2 \qquad \ldots (21)$$

Where R_1, R_2, C_2 are given by equations (10), (12) and (13) respectively, while C_1 is given by equation (15). Since $T_{w_{ad}}$ and h are known from equations (19) and (20), the only unknown in equation (21) is the wall temperature T_w. Equation (21) can be solved by iteration to yield

$$T_w \sim 1270^{o}K$$

The thermal paint charts for this combustor at the above operating conditions indicate a liner temperature within the range 1230-1290^{o}K.

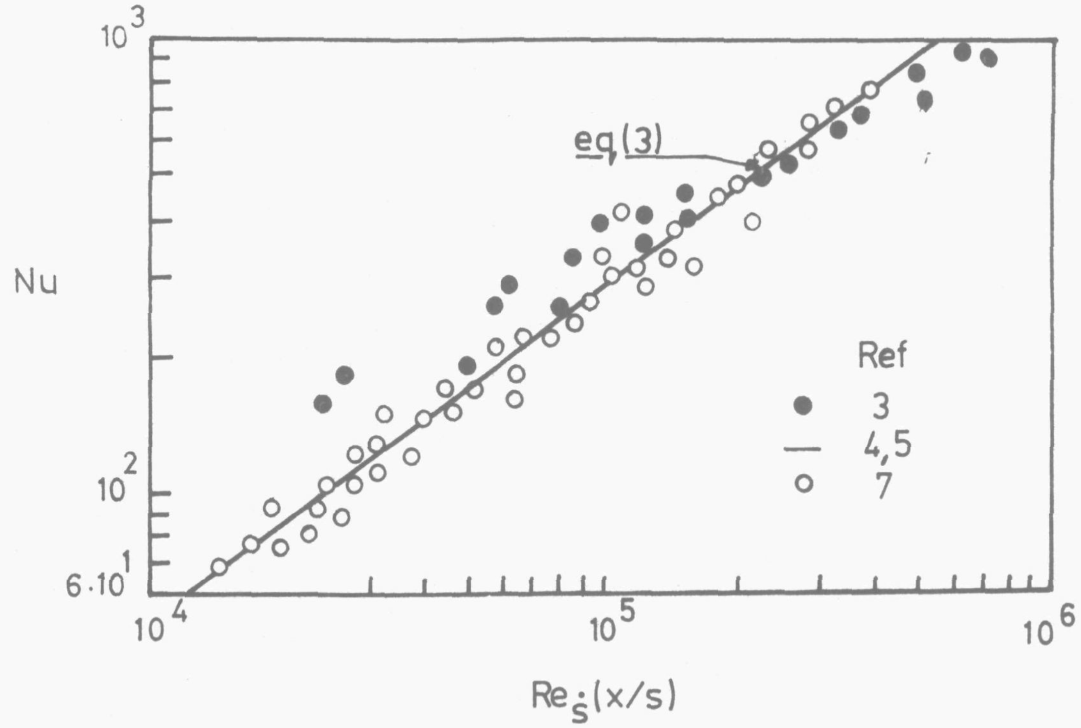

FIG. 1: HEAT TRANSFER RELATIONSHIP FOR THE NEAR-SLOT REGION, M < 1.3.

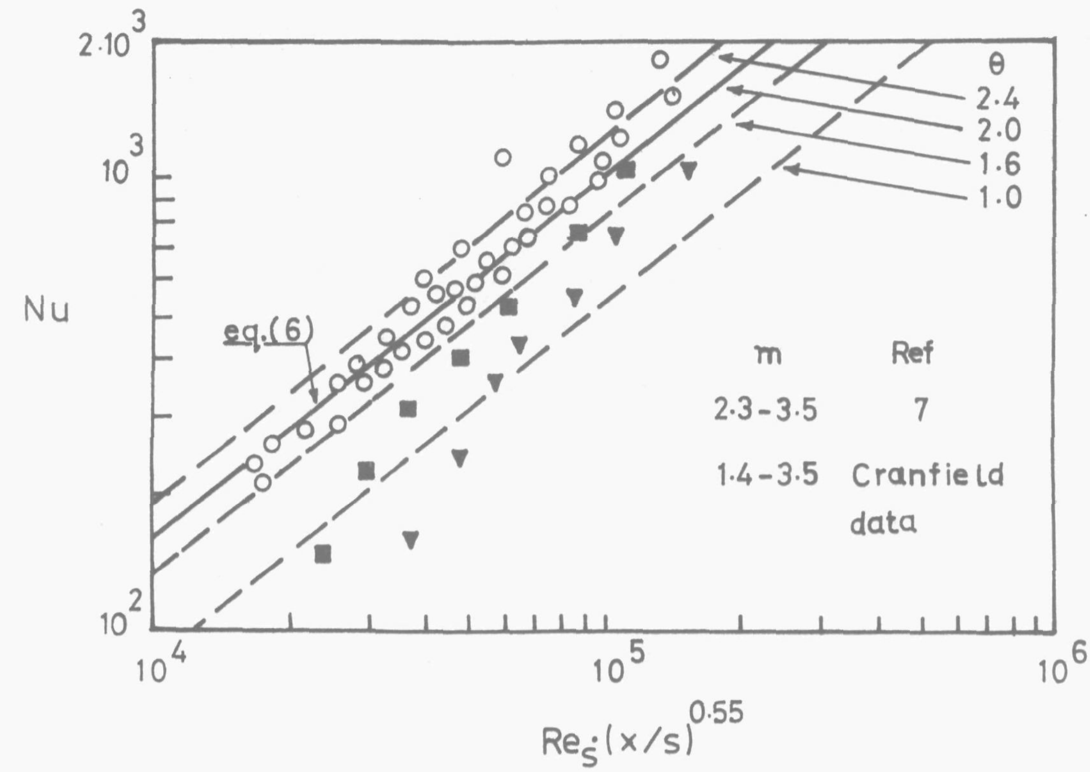

FIG. 2: HEAT TRANSFER RELATIONSHIP FOR THE NEAR-SLOT REGION, M > 1.3

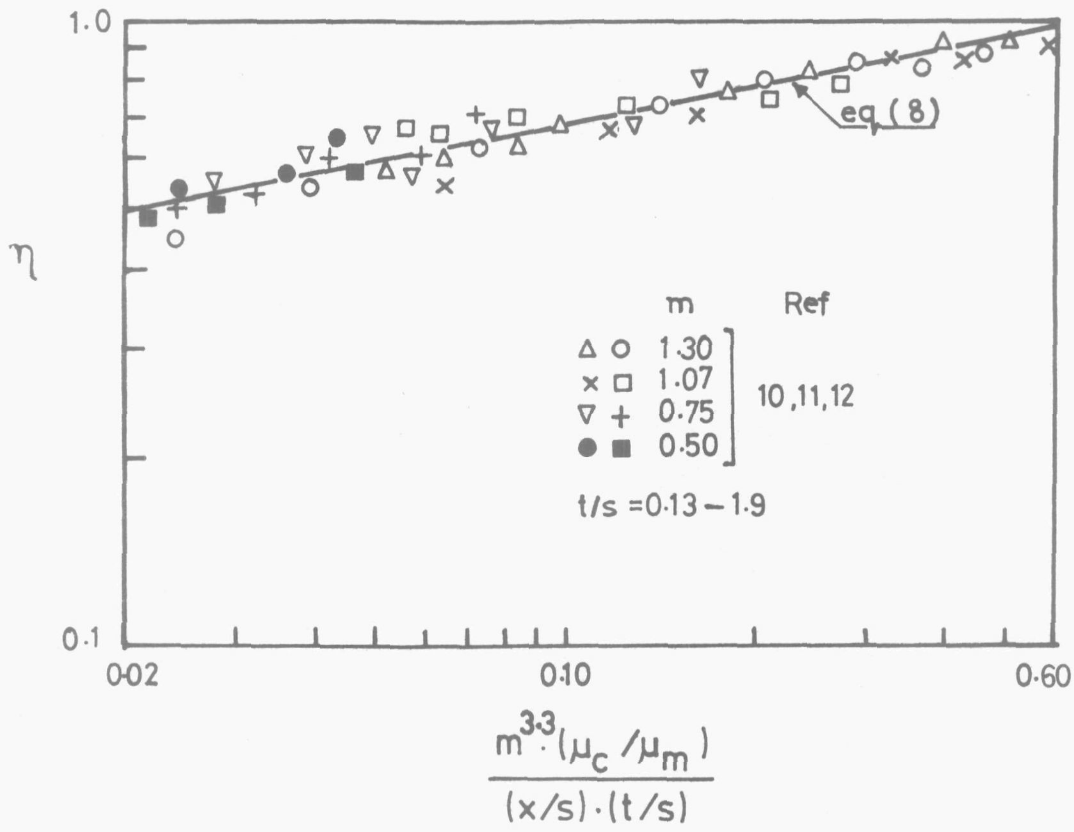

FIG. 3: CORRELATION OF EFFECTIVENESS FOR FINITE LIP THICKNESS $M < 1.3$

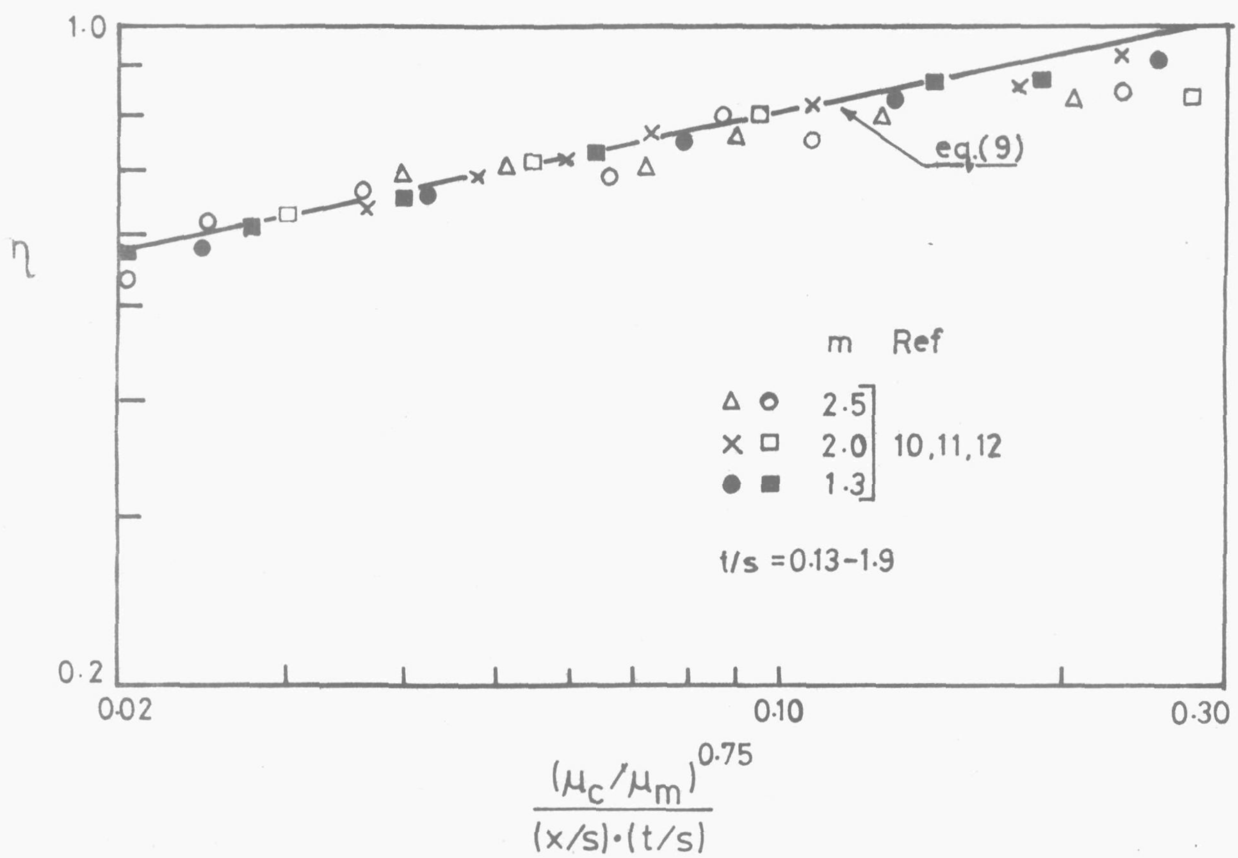

FIG. 4: CORRELATION OF EFFECTIVENESS FOR FINITE LIP THICKNESS, $M > 1.3$

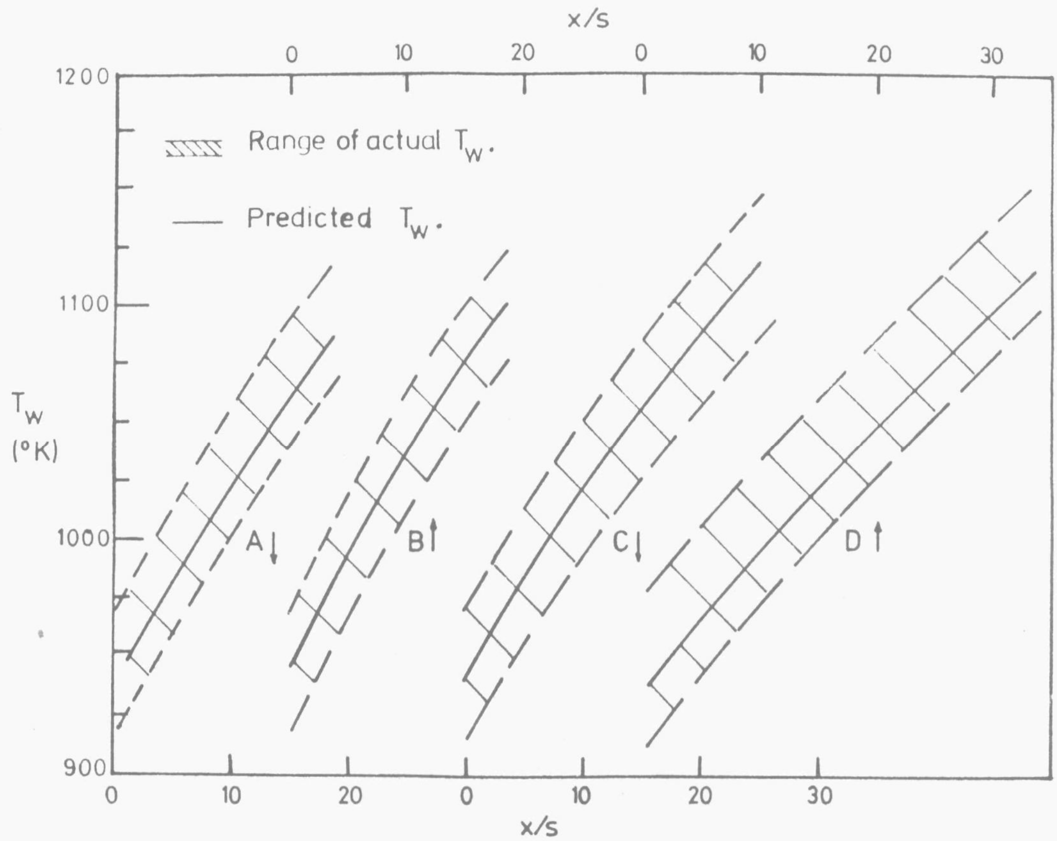

FIG. 5. COMPARISON OF ACTUAL AND PREDICTED LINER WALL TEMPERATURES AT P_c = 22 atm. (Combustion efficiency Zone A = 85%, B = 90%, C = 95%, D = 99%)

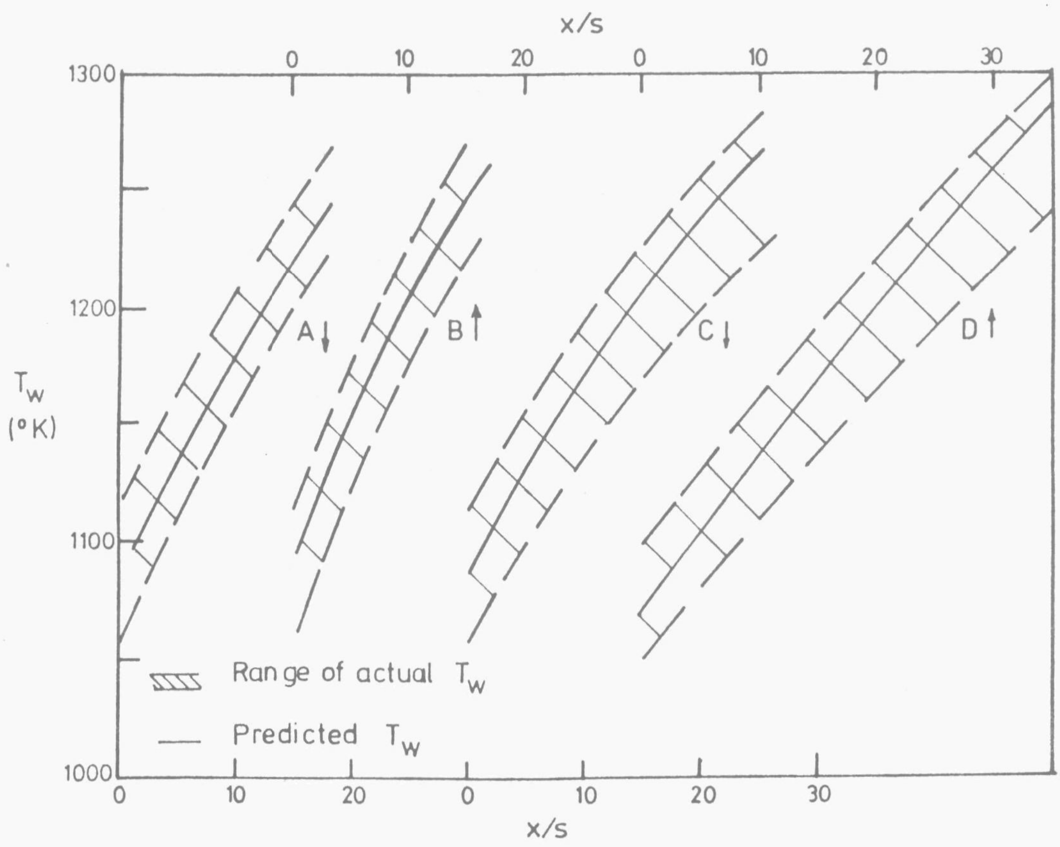

FIG. 6: COMPARISON OF ACTUAL AND PREDICTED LINER WALL TEMPERATURES AT P_c = 30 atm. (Combustion efficiency, Zone A = 85%, B = 90%, C = 95%, D = 99%).

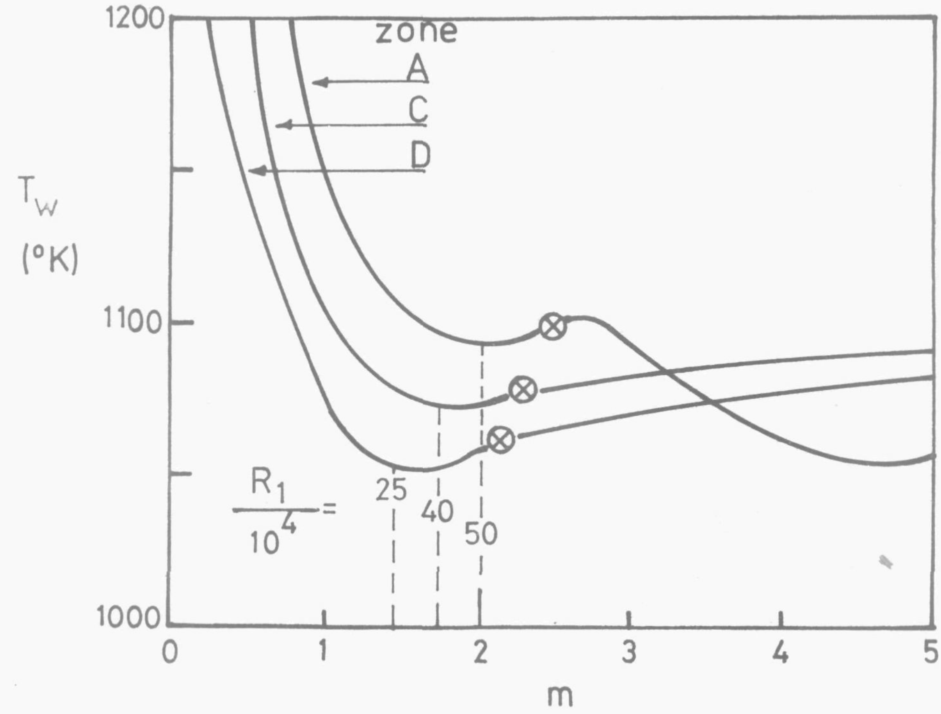

FIG. 7: INFLUENCE OF MASS VELOCITY RATIO ON CALCULATED WALL TEMPERATURES (Reference conditions: T_c = 800°K, P_c = 25 atm. M_{total} = 6.8 kg./sec., Air/Fuel ratio = 50, s = constant. ⊗ - actual combustor values.)

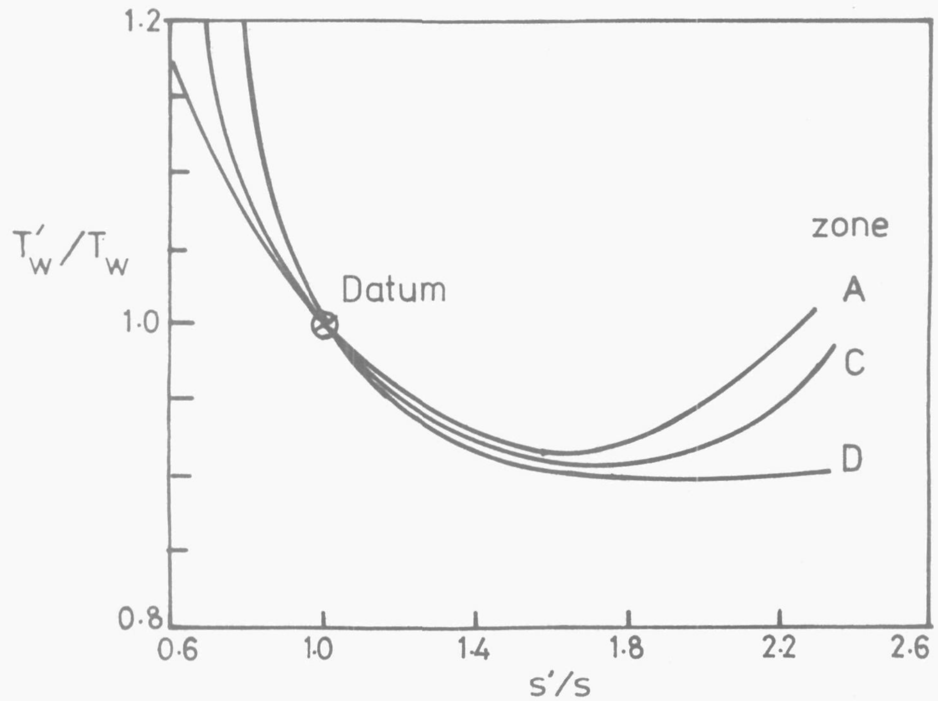

FIG. 8: INFLUENCE OF SLOT HEIGHT RATIO ON CALCULATED WALL TEMPERATURES (Reference Conditions: Same as Fig. 7.)