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Heat Transfer and Cooling in Gas Turbines

(Le Transfert Thermique et le Refroidissement dans les Turbines à Gaz)

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Papers presented at the Propulsion and Energetics Panel 80th Symposium held in Antalya, Turkey, 12th—16th October 1992.

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REPORTS (R)

Application of Modified Loss and Deviation Correlations to Transonic Axial Compressors
AGARD R 745, November 1987

Rotorcraft Drivetrain Life Safety and Reliability
AGARD R 775, June 1990

Theme

Heat transfer and cooling in gas turbine engines are still key factors to achieve high performance, increased life and improved reliability. Any progress in this field will lead to a reduction of maintenance cost and fuel consumption.

The purpose of this Symposium was to bring together experts from industry, research establishments and universities to discuss fundamental and applied heat transfer problems relevant to gas turbines, to exchange practical experience gained and to review the state of the art.

The Symposium focused on turbine blade cooling (both external and internal heat transfer); heat transfer in combustors, to disks, in labyrinth seals, and in shafts; measurement techniques and prediction methods; as well as interactions.

Thème

Le transfert thermique et le refroidissement continuent à jouer un rôle clé dans l'obtention de meilleures performances, l'augmentation de la durée de vie et l'amélioration de la fiabilité des turbines à gaz. Tout progrès réalisé dans ce domaine permettra de réduire les coûts de maintenance et de diminuer la consommation de carburant.

L'objet du Symposium était de rassembler des spécialistes de l'industrie, des établissements de recherche et des universités pour discuter des problèmes fondamentaux et d'application en transfert thermique dans les turbines à gaz. La réunion a fourni l'occasion pour un échange d'expérience pratique et l'examen de l'état de l'art dans ce domaine.

Le Symposium a traité du refroidissement des aubes de turbine (le transfert thermique interne et externe), du transfert thermique dans les chambres de combustion, les disques, les presse-garnitures à labyrinthe et les arbres, ainsi que des méthodes de prévision et des techniques de mesure et les interactions qui en résultent.

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Contents

SESSION I – TURBINE BLADES: EXTERNAL HEAT TRANSFER

SESSION II – TURBINE BLADES: INTERNAL HEAT TRANSFER

SESSION III – MEASUREMENT TECHNIQUES

SESSION IV – ROTATING DISKS, LABYRINTH SEALS AND SHAFTS

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* Paper available and distributed but not presented at the Symposium.

COOLING PREDICTIONS IN TURBOFAN ENGINE COMPONENTS

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SUMMARY

The aim of this work is to show how the metal temperature measured in a convergent-divergent nozzle and in a turbine exhaust diffuser of a turbofan engine, can be predicted with reasonable approximation using the data available in the open literature. It is shown how the simplified fluid dynamic equations with the appropriate experimental correlation allow the prediction of these results in other flight conditions than those tested.

LIST OF SYMBOLS

$C_{p\infty}$= specific heat at main stream temperature.
C_{p2}= specific heat at cooling flow temperature.
D_h= Hydraulic diameter
d = local curvature diameter
h = convective heat flux coefficient
K = thermal conductivity coefficient
L = inner blade passage length
M =blowing factor $\frac{\rho_2 V_2}{\rho_\infty V_\infty}$
N_u= Nusselt number
P_r= Prandtl number
q = heat flux
R_{e2}= Reynolds number for the cooling parameters $\frac{\rho_2 V_2 . S}{\mu_2}$
$R_{e_{x,s,d}}$ = Reynolds number based on x,s,d.
s = cooling slot height.
T_g= absolute gas temperature.
T_r = recovery temperature = $T_\infty + 0,9\left[T_{\infty 0} - T_\infty\right]$
T_w= actual wall temperature
T_{aw} = adiabatic wall temperature
$T_\infty, T_{\infty 0}$ = static and stagnation temperature of the main stream.
$T_* = T_\infty + 0,72\left[T_{aw} - T_\infty\right]$
T_2= cooling injection temperature.
x = distance from the point of injection.
$\Delta\varepsilon$= correcting factor due to spectral overlapping.
ε_w= inner face wall emissivity.
ε_g= gas emissivity = $\varepsilon_{CO2}\rho_{CO2} + \varepsilon_{H2O}\rho_{H2o} - \Delta\varepsilon$
η = film cooling effectiveness
μ = gas dynamic viscosity
θ = angle from stagnation point in profile leading edge.
ρ = density
ρ_*, μ_*= density and viscosity at a temperature T_*
σ = Stefan-Boltzman constant.
ξ^* = non dimensional distance

$$= \frac{x}{M.S}\left[\frac{R_{e2}\,\mu_2}{\mu_*}\right]^{-0,25} \frac{\rho_*}{\rho_\infty}$$

Subscripts

g = gas
w = wall
1,2 = cold gas injection

1 INTRODUCTION

The modern engine solid surfaces exposed to internal flow need improved thermal protection because high temperature cycles are used to increase performance. During the past decades, different cooling methods have been used to reduce the metal temperature, therefore minimizing the required amount of cooling air.

The simplest way to cool surfaces is by convective cooling. In this process, heat flows by conduction from exposed metal surfaces to un-exposed ones, which are cooled by air flowing usually parallel to it. Convective cooling is used whenever low levels of cooling effectiveness are required. This limitation exists due to the fact that the air supply is somehow limited. On the other hand, high effectiveness levels tend to increase thermal stress problems [1-3].
A special type of convective cooling is by means of impingement. It is used whenever large heat transfer coefficients are needed on the un-exposed surfaces [4-8]. Another

method to obtain high heat transfer coefficients is to place fins or ribs normal to the coolant flow path [9].

When higher levels of cooling effectiveness are required it is necessary to use more sophisticated alternatives. The most common method is to insert a secondary fluid into the boundary layer on the surface which is to be protected. There are different means of injecting this fluid such as ablation, transpiration and film cooling. In ablation cooling, a heat shield ablates and secondary fluids enter the boundary layer [10] In transpiration cooling the coolant enters the boundary layer through a porous material [11-13]. Both methods are used to protect the region where coolants are added. Unfortunately the application of these systems is difficult because ablation has a limited time span and porous materials are not strong enough to be used in engines. The basic mechanism of film cooling is the introduction of a secondary fluid at several locations along a surface to protect that surface not only in the injection area but also in the downstream region [12-14].

Thermal studies on aircraft engines, particularly modern ones with large flight envelopes, require a great amount of testing and instrumentation. Designers have different tools for obtaining engine component temperature predictions. These tools can be classified as: 1) theoretical, 2) experimental and 3) numerical methods. Theoretical analysis is usually aimed at finding parameters to describe the temperature of an adiabatic wall downstream of the coolant injection [15]. Real and three-dimensional effects are studied on test rigs with controlled conditions [16-18]. Predictive numerical calculations are being widely used nowadays for different cooling applications and configurations, because they help to speed up the design and to reduce the number of experiments [19-20].
In this paper a method to predict full scale engine temperatures is described. This method combines the fluid dynamic equations with semi-empirical data obtained from the open literature.

The result is an easy to use procedure.

2. NOZZLE COOLING PREDICTION

The prediction of the petal temperature of a convergent-divergent nozzle has been performed through a computer code that uses correlation models available in the open literature with actual data obtained from a test program that includes a reduced scale hot test, a full scale rig test and a DVE (Design Verification Engine) program.

The film cooling produced when the relative cold layer is injected at the beginning of the convergent petal of the nozzle, produces a reduction in the metal temperature along the petal.

The heat transfer rate is modelled by:

$$q = A.h \left[T_{aw} - T_{w}\right]$$

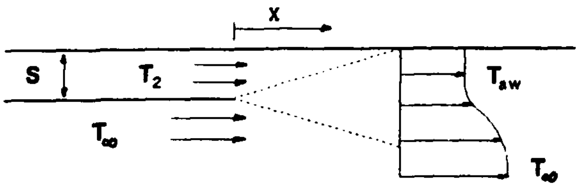

Fig.1. Single slot film cooling configuration.

The convective coefficient h is calculated independently from the adiabatic wall temperature T_{aw}.

This magnitude is related to the concept of effectiveness of the film cooling, figure 1:

$$\eta = \frac{T_r - T_{aw}}{T_r - T_2}$$

The correlation found more suitable for the flow conditions and geometry is the model of Kutatelache & Leanter, in reference [14] because the flow can be considered twodimensional and compressible and the difference of temperature between the wall and the main stream is important.

The effectiveness is given from this reference by the expression:

$$\eta = \frac{1}{1 + \frac{C_{p\infty}}{C_{p2}} \left[0,33\ (4,0 + \xi^{*})^{0,8} - 1\right]}$$

When the cooling injection is made by two consecutive slots, the total effectiveness can be treated as the combination of two cooling layers where the external temperature for the inner flow is the "adiabatic wall temperature" for the external one [21], figure 2.

$$\eta_1 = \frac{T_\infty - T_{aw1}}{T_\infty - T_1}$$

$$\eta_2 = \frac{T_{aw1} - T_{aw2}}{T_{aw1} - T_2}$$

and the total effectiveness is:

$$\eta = \frac{T_\infty - T_{aw2}}{T_\infty - T_2} = \eta_1 + \eta_2\ (1 - \eta_1)$$

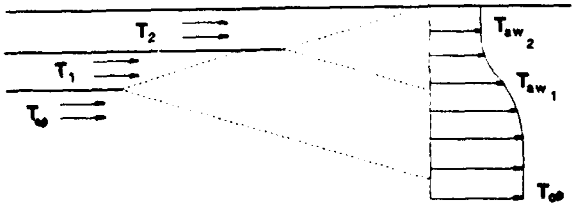

Fig.2. Double slot film cooling configuration.

The convective coefficient h is obtained from semiempirical correlation for film cooling with a high velocity compressible boundary layer.

As the blowing factor $M < 2$ the correlation used is the one proposed by Lefebre [22].

$$x \leq 0{,}01\ m \qquad h_x = N_{ux} \cdot \frac{K_2}{x}$$

$$N_{ux} = 0{,}057\ R_{ex}^{0,7}$$

$$R_{ex} = \frac{\rho_2\ V_2\ . X}{\mu_2}$$

$$x > 0.01\ m \qquad h_x = \frac{N_{ux}\ K_2}{x}$$

$$N_{ux} = 0{,}0256 \left[R_{es}\ \frac{x}{M.S}\right]^{0,8}$$

$$R_{es} = \frac{\rho_2\ V_2\ .S}{\mu_2}$$

The wall temperature of the petal is obtained for an equilibrium model where the convective and radiative heat transfer with the cavity between the nozzle petals and the fairing flap, and the conduction along the petal are modeled in the classical way.

It may be of interest to note that the radiative heat transfer with the hot gas of the main flow is modeled as [15 & 23].

$$\frac{q}{A} = \frac{1}{2}\ \sigma \left(1 + \varepsilon_v\right)\ \varepsilon_g\ T_g^{1,5} \left(T_g^{2,5} - T_v^{2,5}\right)$$

3. NOZZLE THERMAL TESTS

The model resulting from this information gave good results for wall temperature estimation always with a conservative margin that the test in the scale model helped to reduce (figure 3). The full scale results with fixed convergent-divergent nozzle tested in a hot rig showed very good agreement between the predictions and the actual values from the thermocouples and thermal paints. (see figures 4 and 5).

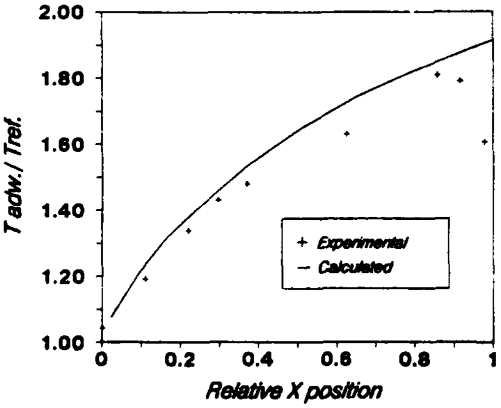

Fig.3. Initial prediction and scaled hot test results along nozzle petal.

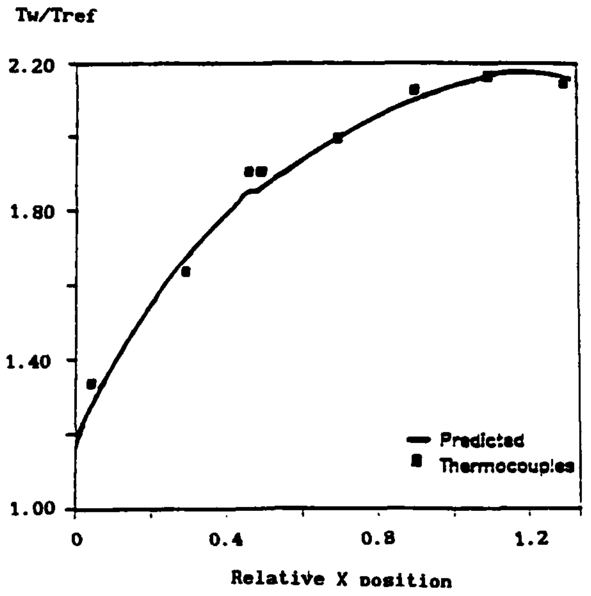

Fig.4. Comparison of predicted temperatures and fixed condi nozzle test results.

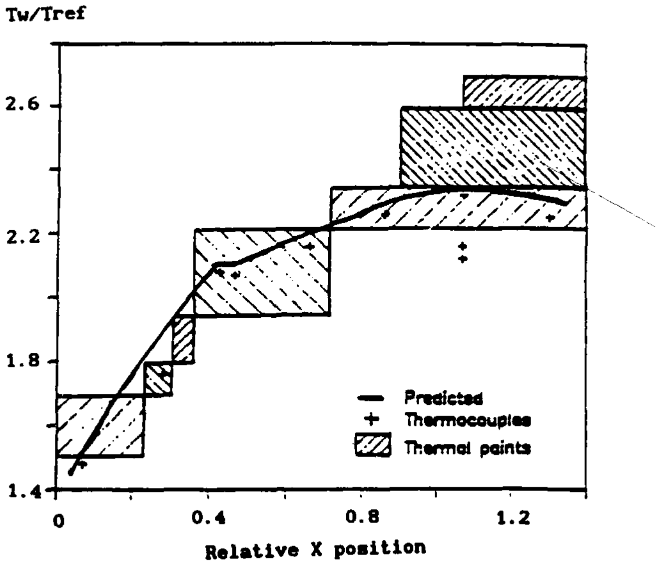

Fig.5. Thermal points, predicted temperature and thermocouples values in fixed condi nozzle.

When the true variable convergent-divergent nozzle was tested in the Design Verification Engine (DVE) the differences between the prediction and the measurement in the beginning of the divergent petal were important. But if a cooling ingestion through the gap in the throat is considered, this disagreement disappears and the model appears to be consistent and to give accurate temperature predictions. (Figure 6).

4. EXHAUST DIFFUSER COOLING CIRCUIT

The cooling circuit of the exhaust diffuser has been designed with a method that modelise the more relevant fluid factors

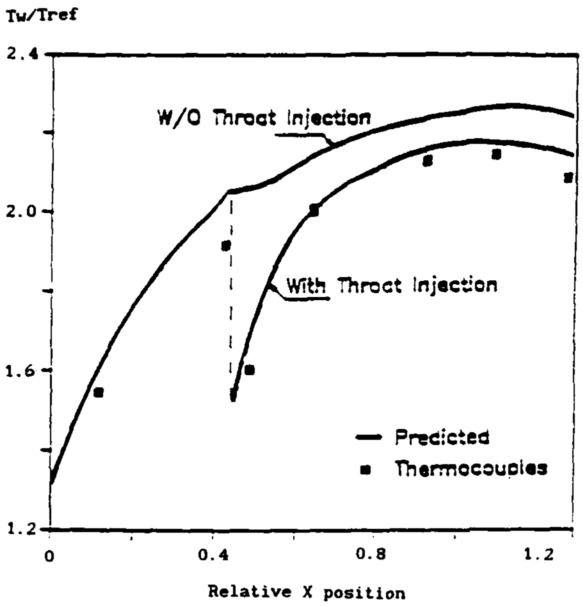

Fig.6. Predicted and measured temperatures with and without throat cold air injection.

around the circuit and use the available data in the open literature. This prediction method includes theoretical analysis and semienpirical correlations coupled to numerical simulation wherever further refinement is required. A more detailed description of this design method has been given elsewhere [24]. Therefore here we will only comment at the aspects related to the general objectives of this paper.

The geometry of the cooling circuit can be seen in figure 7, the fixed blade diffuser cascade has an inner passage with cooling capillaries in the trailing edge. The thermal analysis of the blades requires a detailed numerical process with NASTRAN code for the solid pieces and CFD code for the convective heat transfer in the blade boundary layer. But for the purpose of the circuit flow prediction a simpler model is used estimating the internal friction coefficient using [25] and the inner heat flux with the expression from [2].

$$N_u = \sqrt{\frac{T}{T_w}} \cdot 0{,}036\ R_e^{0{,}8} P_r^{1/3} \left(\frac{D_h}{L} \right)^{0{,}035}$$

The mean blade temperature is fixed by a heat equilibrium analysis where the external heat transfer is modelled in the leading edge with the correlation

$$N_u = 1{,}14\ R_{ed}^{1/2}\ P_r^{0{,}4} \left| 1 - \frac{2\theta}{\pi} \right|$$

and in the rest of the blade surface with the standard Nusselt value for turbulent boundary layer.

The remaining heated flow which reaches the plenum is forced to cross the perforated wall to impinge onto the rear wall. This is needed to enhance the cooling of this wall which receives strong radiative heat flux from the afterburner. The dominant feature in this part of the cooling circuit is the combination of cross-flow and impingement cooling. The radial geometry simplifies the mathematical model which follows the scheme of [26].

The calculation of the flow through the holes in each station is also modified to allow for the cross flow. Correlation from [27] is used to take this effect into account. The convective heat coefficient produced by the impingement is modelled following [4 & 28].

To reproduce this correlation here is beyond our limits, so our advice is to consult the original papers.

Wall temperature is obtained by a thermal balance between external and internal radiative heat flux and the impingement as already commented. Upon the reheat luminous flame emissivity and radiative terms of this equilibrium are modelled after references [3 & 15].

The flow behavior inside the lateral double wall is calculated in a similar way with the simplification that no impingement effect is present.

The mass flow and thermal predictions are performed jointly in an iterative mode. It starts with a hypothetical mass flow at the beginning of the circuit. Its` purpose being to match the pressure, at the exit located downstream of the cascade, obtained by the program with the actual value.

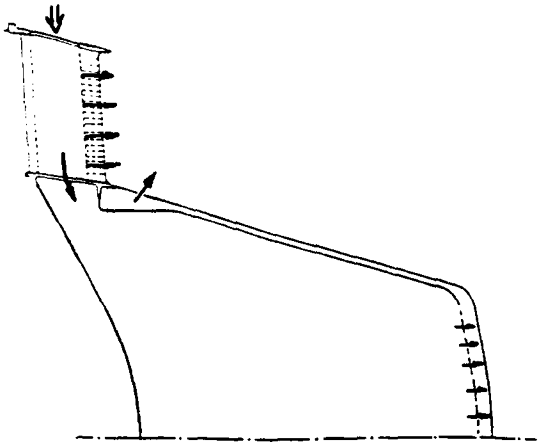

Fig.7. Exhaust diffuser cooling circuit.

5. EXHAUST DIFFUSER TEMPERATURE ASSESSMENT

Calibration tests were carried out to check the mass flow predicted through the cooling circuit. It was performed in a rig that reproduces the geometry and flow conditions, and is divided into two phases to obtain separately the mass flow through out both capillary trailing edge tubes and discharge exit downstream blade trailing edge.

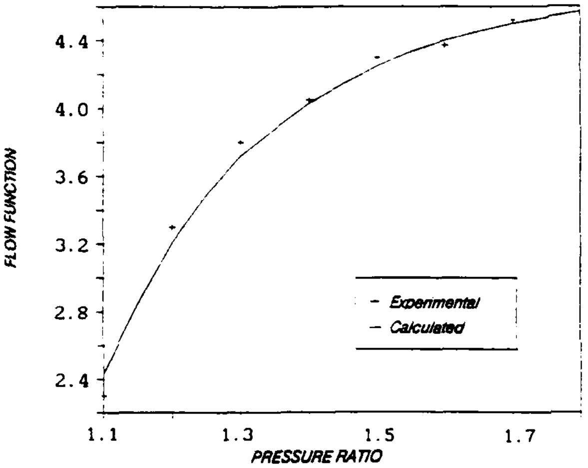

Fig.8. Results of calibration test mass flow through capillary cooling tubes in the blade trailing edge.

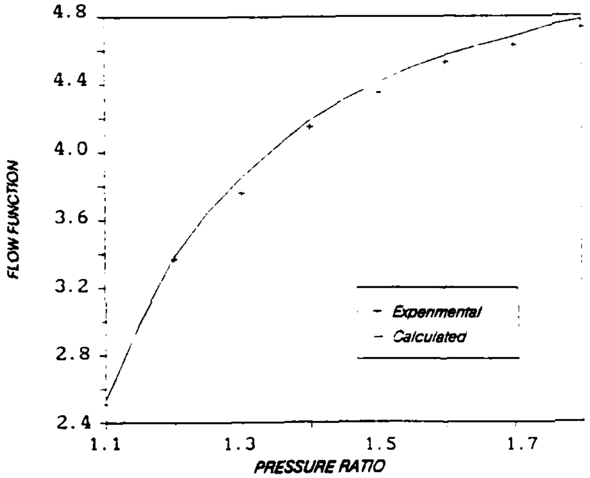

Fig.9. Results of calibration test. Total cooling mass flow in exhaust diffuser cooling circuit.

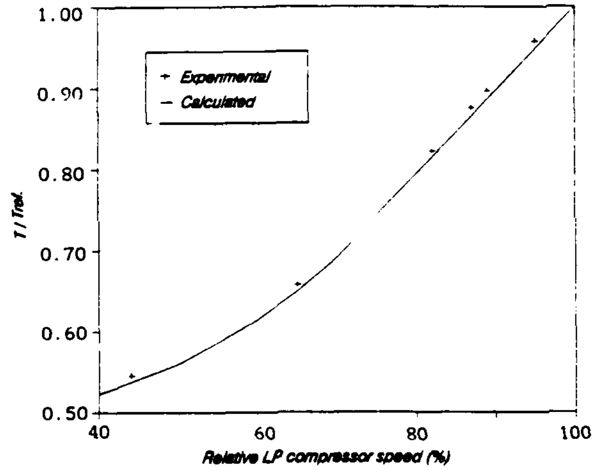

Fig.10. Comparison between predicted and experimental values of temperature at the center of the diffuser rear wall.

Figures [8 and 9] show the comparison between experiments and theoretical predictions for the capillary tubes and the total cooling mass flow.

The engine validation tests were performed to assess the pressure and temperature predictions. We will focus on the cone, lateral and rear walls where no further calculation was performed. The figures [10,11 and 12] show the comparison of predicted and measured temperature in the range of compressor speed, while figure [13] shows the relative error for both temperature and pressure in several circuit locations.

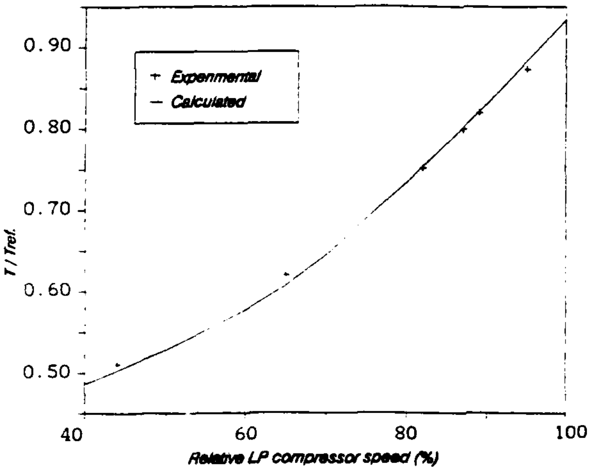

Fig.11. Comparison between predicted and experimental values of temperature at the periphery of the diffuser rear wall.

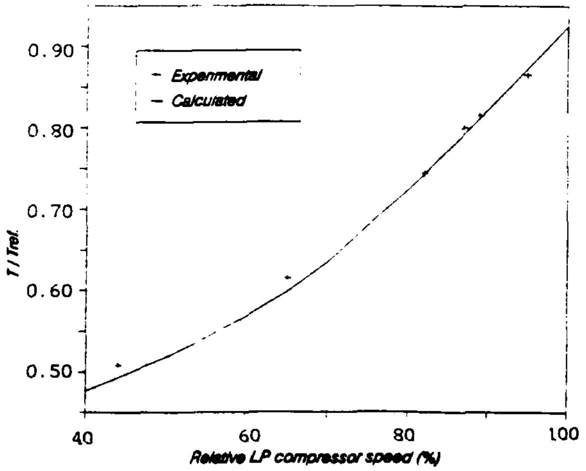

Fig.12. Comparison between predicted and experimental values of temperature at an intermediate section of the lateral wall.

6. CONCLUSIONS

The comparison between predicted and actual test results of calibration rigs and engine test beds shows that for design purposes the accuracy obtained is enough and no other complicated method is needed.

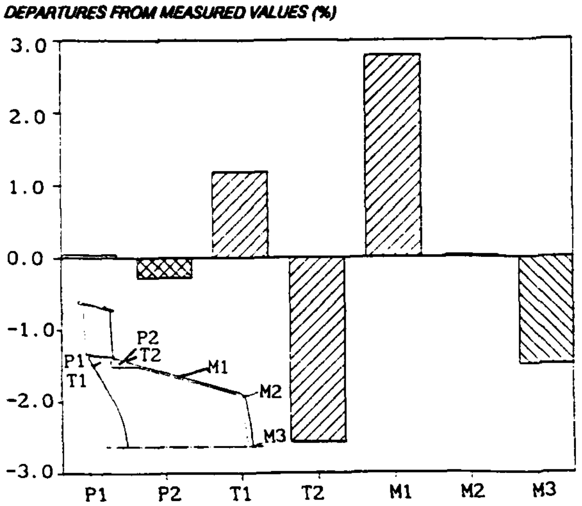

Fig.13. Differences between prediction and measured values. P, T and M are pressure, air temperature and metal temperature respectively.

Relevant engineering design parameters, such as metal temperatures, pressures and mass flow were predicted within 3% of accuracy. It is possible to explore a broad range of flight envelope points using this simple design tool. Further improvements on the modeling of reheat radiative properties would be needed if greater accuracy were required.

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Discussion

QUESTION 1:

DISCUSSOR: J. Salva Monfort, Escuela Tecnica Superior de Ingenieros Aeronauticos

Can you explain in more detail how you have calculated the cooling flux through the capillary tubes in the trailing edges of the blades?

AUTHOR'S REPLY:

The flow through the cooling capillary tubes in the trailing edge of the blades is calculated using the same turbulent one-dimensional equations as for the main passages of the blade. Some problems of choking due to the high heat transfer can arise. In this case, it is necessary to detect the Mach number increase and to reduce the mass flow for that capillary tube accordingly.

QUESTION 2:

DISCUSSOR: D.T. Vogel, DLR

You used many of empirical constants in your calculations. Are these constants related to your special problem, or is it possible to calculate other cooling configurations?

AUTHOR'S REPLY:

The correlations used are all chosen from open literature taking into account the geometry and dimensionless parameters of the problem. Please refer to the original paper to find out if it can be applied to other configurations.

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Rotating discs
Seals technology
Turbine blades
Turbulent flow
Wake flow

14. Abstract

The Conference Proceedings contains the 38 papers presented at the Propulsion and Energetics Panel 80th Symposium on "Heat Transfer and Cooling in Gas Turbines" which was held from 12th—16th October 1992 in Antalya, Turkey. The Technical Evaluation Report and the Keynote Address are included at the beginning, and discussions follow most papers.

The Symposium was arranged in the following Sessions: Turbine Blades: External Heat Transfer (9); Turbine Blades: Internal Heat Transfer (6); Measurement Techniques (4); Rotating Disks, Labyrinth Seals and Shafts (7); Combustors (4); Design, Interactions (3); and Prediction Methods (5).

Heat transfer and cooling in gas turbines are still key factors for achieving high performance, increased life and improved reliability. Any progress in this field will lead to a reduction of maintenance cost and fuel consumption. The purpose of the Symposium was to bring together experts from industry, research establishments and universities to discuss fundamental and applied heat transfer problems relevant to gas turbines, to exchange practical experience gained and to review the state of the art.

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