4. **Han L. S. and Cox R. W.** A visual study of turbine blade pressure-side boundary layers. *A.S.M.E. Paper No. 82-GT-47*

5. **Sharma O. P. and Graziani R. A.** Influence of endwall flow on airfoil suction surface mid-height boundary layer development in a turbine cascade. *A.S.M.E. Paper No. 82-GT-127*

6. **Walker G. J.** The turbulent boundary layer on an axial compressor blade. *A.S.M.E. Paper No. 82-GT-52*

7. **Hobbs D. E., Wagner J. H., Dannenhoffer J. F. and Dring R. P.** Experimental investigation of compressor cascade wakes. *A.S.M.E. Paper No. 82-GT-299*

8. **Kool P. and Hirsch Ch.** A prediction scheme for the decay of a turbomachine blade wake. *A.S.M.E. Paper No. 82-GT-273*

9. **Bryanston-Cross P. J. and Camus J. J.** Auto and cross correlation measurements in a turbine cascade using a digital correlator. *A.S.M.E. Paper No. 82-GT-132*

10. **Sieverding C. H.** The influence of trailing edge ejection on the base pressure in transonic turbine cascades. *A.S.M.E. Paper No. 82-GT-50*

11. **Motallebi F., Edwards S. J. and Norbury J. F.** Base pressure on a blunt base in transonic flow—some effects of base geometry and bleed air. *A.S.M.E. Paper No. 82-GT-317*

12. **Joslyn H. D., Dring R. P. and Sharma O. P.** Unsteady three-dimensional turbine aerodynamics. *A.S.M.E. Paper No. 82-GT-161*

13. **Dring R. P., Joslyn H. D., Hardin L. W. and Wagner J. H.** Turbine rotor-stator interaction. *A.S.M.E. Paper No. 82-GT-3*

14. **Leboeuf F., Bario F., Boris G. and Papailiou K. D.** Experimental study and theoretical prediction of secondary flows in a transonic axial flow compressor. *A.S.M.E. Paper No. 82–GT-14*

15. **Hunter I. H. and Cumpsty N. A.** Casing wall boundary layer development through an isolated compressor rotor. *A.S.M.E. Paper No. 82-GT-18*

16. **Pandya A. and Lakshminarayana B.** Investigation of the tip clearance flow inside and at exit of a compressor rotor passage. Part I: Mean velocity field. *A.S.M.E. Paper No. 82-GT-12*

17. **Lakshminarayana B., Pouagare M. and Davino R.** Three-dimensional flow field in the tip region of a compressor rotor passage, Part I: mean velocity profiles and annulus wall boundary layers. *A.S.M.E. Paper No. 82-GT-11*

18. **Lakshminarayana B., Pouagare M. and Davino R.** Three-dimensional flow field in the tip region of a compressor rotor passage, Part II: Turbulence properties. *A.S.M.E. Paper No. 82-GT-234*

19. **Bettner J. L. and Elrod C.** The influence of tip clearance, stage loading and wall roughness on compressor casing boundary layer development. *A.S.M.E. Paper No. 82-GT-153*

20. **Binder A. and Romey R.** Secondary flow effects and mixing of the wake behind a turbine stator. *A.S.M.E. Paper No. 82-GT-46*

21. **Gregory-Smith D. G.** Secondary flows and losses in axial flow turbines. *A.S.M.E. Paper No. 82-GT-19*

22. **Lacor C. and Hirsch Ch.** Rotational flow calculation in three-dimensional blade passages. *A.S.M.E. Paper No. 82-GT-316*

23. **Abdallah S. and Hamed A.** The elliptic solution of the secondary flow problem. *A.S.M.E. Paper No. 82-GT-242*

24. **Sheoran Y. and Tabakoff W.** A study of viscous flow in stator and rotor passages. *A.S.M.E. Paper No. 82-GT-248*

25. **Wang L. C., Hetherington R. and Goulas A.** The calculation of deviation angle in axial flow compressor cascades. *A.S.M.E. Paper No. 82-GT-230*

Heat transfer sessions

A. Brown*

In reviewing the four heat transfer sessions it is convenient to take the first two together and then treat the last two separately. Thus this report has three subdivisions: external blade heat transfer; film and transpiration cooling; and blade internal cooling.

Regardless of the continuing improvements made in the structural capabilities of high temperature alloys used in gas turbine components, many gas turbines operate at turbine entry temperatures where component cooling is necessary. Cooling is usually achieved by systems using relatively cold bled compressor air. The aim is to design the cooling system for minimum coolant flows compatible with the maximum stresses experienced during cyclic operation and an acceptable component stress rupture and creep operating life. Also excessive cooling is detrimental to engine efficiency as the use of compressor bled air for cooling partially offsets performance improvements obtained by higher turbine entry temperature. Thus turbine component cooling design engineers have to meet the dual challenge of minimising cooling air consumption and assuring that component temperatures are low enough to meet life requirements. Knowledge of the fluid mechanics of flows within and over turbine components and the consequent heat transfer between gases and components is essential. This knowledge is continually being improved upon by basic research and is being adapted in the design of modern high temperature gas turbines.

External blade heat transfer

Measurements of heat transfer on both suction and pressure surfaces of rotor and stator blades in cascades for a range of flow conditions were reported in six papers[1,3,4,6,9,10]. Of these six papers, five used transient techniques to measure heat transfer, the exception being the measurements of Krishnamoorthy[4]. Krishnamoorthy measured the heat transfer coefficient distribution over a blade profile at constant heat flux boundary conditions. The heat transfer test blade was made of low thermal conductivity

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material with a 0.25 mm thick stainless steel sheet covering its surface in which the constant heat flux boundary condition was generated by heating it with a high current, low voltage, ac electric power source. The computed heat transfer coefficients were corrected for conduction through the sheet thickness.

Three different transient techniques were used by the other five groups of authors. Litchfield and Norton[1] and Nicholson *et al*[3] used the Oxford University Light Piston Tunnel, the design and operation of which is well documented[22,23]. Kercher *et al*[6] used a shock tube to generate high temperature and pressure air flow and measurements of heat transfer were made using thin film heat transfer gauges on a flat plate in the shock tube and in a shock tunnel behind the shock tube and also on blades in cascade in the shock tunnel. The flat plate was used for datum measurement in the interpretation of blade measurements.

Barry *et al*[9] and Beacock *et al*[10] respectively have developed and used, with some degree of success, an extremely useful new transient technique for determining local heat transfer coefficients which could be used by further development under engine conditions. The method involves the measurement of the variation of wall temperature of a cooled, thin, high temperature alloy, hollow shell blade to controlled perturbations of coolant temperature. The unmodulated wall temperature, the amplitude ratio between wall and coolant temperature modulation and the phase lag between wall and coolant temperature modulation are measured. From either the measurements of amplitude ratio or phase lag response and the unmodulated wall temperature of both the gas and coolant side, heat transfer coefficients can be determined from simple linear relationships. Typical plots of amplitude and phase lag are shown in Fig 1. This method was checked via measurements on a hollow cylinder from which it was found that the greatest source of error was circumferential conduction; the overall estimated error in heat transfer coefficient was ±7%.

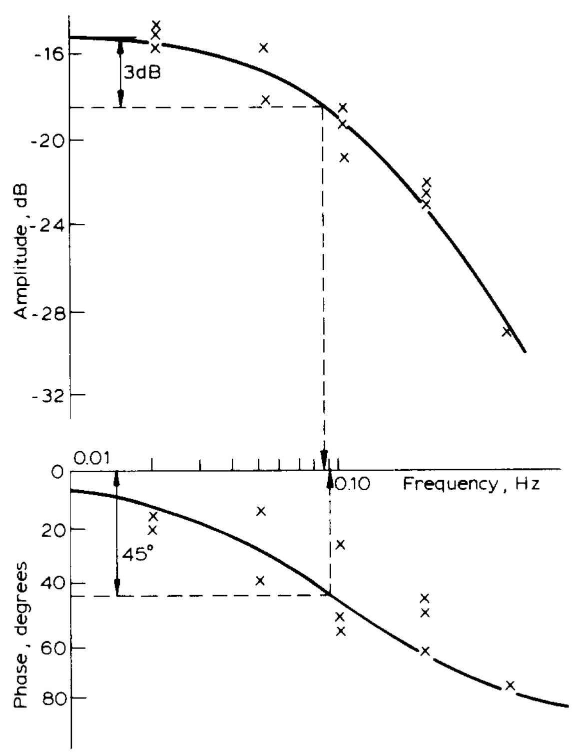

Fig 1 Amplitude ratios and phase lags obtained with the single harmonic technique (from Beacock et al[10]*)*

Krishnamoorthy investigated the effect of free stream turbulence intensity on the local heat transfer coefficient. He considered turbulence intensities up to 0.36 with exit Mach number and Reynolds number based on chord and exit velocity of up to 0.4 and 4.5×10^5 respectively. The velocity distribution on the suction surface was favourable for the first 40% of surface with the remaining 60% adverse, whereas the pressure surface velocity distribution was adverse for the first 20% with the remaining 80% favourable and the complete pressure surface velocity distribution was concave upwards. The findings of this work were:

1. the local heat transfer coefficient increases linearly with local freestream turbulence intensity for turbulent boundary layers for all turbulence intensities and for laminar boundary layers for turbulence intensities up to 0.02. In laminar boundary layer regions, the heat transfer coefficient is proportional to the square root of local turbulence intensity for turbulence intensities between 0.02 and 0.12;
2. turbulence intensity has only marginal effect on the start of boundary-layer transition and virtually no effect on the length of boundary-layer transition on suction surfaces;
3. in turbulent boundary-layer regions on suction surfaces, turbulence intensity has little effect on the heat transfer coefficient;
4. the heat transfer coefficient variation over pressure and suction surfaces is of the same order.

Most of these findings are in agreement with the published literature for low speed flows except conclusion 3 (see Seyb[24], Dhawan and Narasimha[25] and Debruge[26]).

Litchfield and Norton[1], Nicholson *et al*[3], Kercher *et al*[6] and Beacock *et* al[10] all made measurements at more nearly engine gas velocities than Krishnamoorthy. Throat Mach numbers of up to 1.75 on suction surfaces and exit Mach and Reynolds numbers up to about 1.0 and 2×10^6 respectively were used. Beacock *et al* demonstrated the use of the Barry *et al*[9] technique to measure heat transfer coefficients on rotor blades and nozzle guide vanes and compared their measurements with the Oxford University technique at lower pressures and temperatures; the agreement was generally good giving confidence in the modulated coolant temperature technique. Kercher *et al* demonstrated the use of their transient technique and found the Eckert reference temperature for evaluating fluid properties to be valuable in correlating their measurements. Both Beacock *et al* and Kercher *et al* observed that the heat transfer on their pressure surfaces was higher than that on the suction surfaces and higher than that predicted by flat plate theory. The latter suggested this may be due to the existence of a separation bubble on the pressure surface at about 10% surface

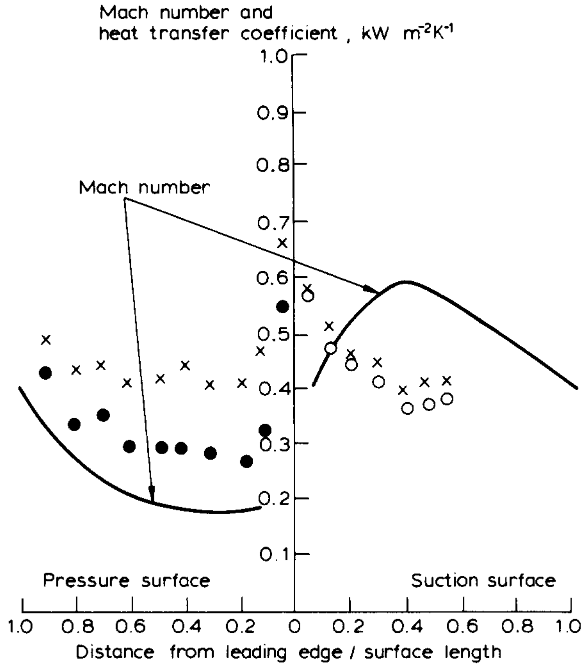

Fig 2 Example of the variation of heat transfer coefficient with surface position for a trailing edge Reynolds number = 4.5×10^5; ×—turbulence intensity = 0.178; ●—turbulence intensity = 0.128; ○—turbulence intensity = 0.018. (from Krishnamoorthy[4])

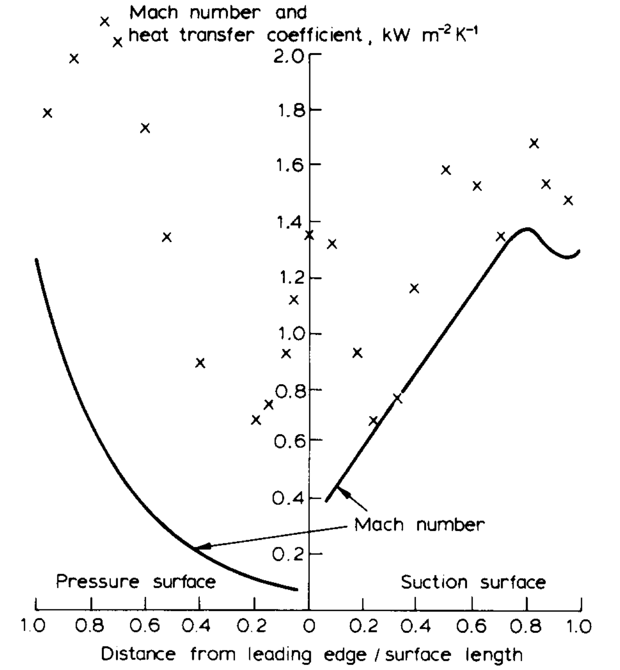

Fig 3 Example of the variation of heat transfer coefficient with surface position for a trailing edge Reynolds number = 3.3×10^6; turbulence intensity = 0.044 (from Litchfield and Norton[1])

position from the leading edge. I would agree that a separation bubble probably exists on the Kercher *et al* pressure surface but that alone is not the explanation of high heat transfer on blade pressure surfaces. The same high heat transfer on pressure surfaces was noted by Litchfield and Norton in their transonic measurements and has been widely discussed and recorded in much published literature. Litchfield and Norton reduced pressure surface heat transfer by film cooling with injection at about 70% surface position and injection rates up to 2.2% of free stream flows. Beyond this injection rate the heat transfer was greater than that without injection. They also reduced suction surface heat transfer near the trailing edge by having a local surface concavity but the penalty at design conditions was higher velocities on pressure than suction surfaces in this region and a resultant local negative lift. Nicholson *et al*, having recognised the problem of high heat transfer on pressure surfaces and reduced this by boundary-layer optimisation, made measurements on high and low stagger blades; on the former the boundary-layer on the pressure surface relaminarised at about 30% surface position and remained laminar to the trailing edge, and on the latter the boundary layer on the pressure surface was turbulent from about 10% surface position and remained so, to the trailing edge. The measurements of Nicholson *et al* show that a significant reduction in heat transfer to pressure surfaces can be achieved by boundary-layer optimisation during blade profile design without loss of aerodynamic performance. This conclusion is in line with the predictions of Brown and Martin[27]. Typical results from the papers discussed above are shown in Figs 2–4.

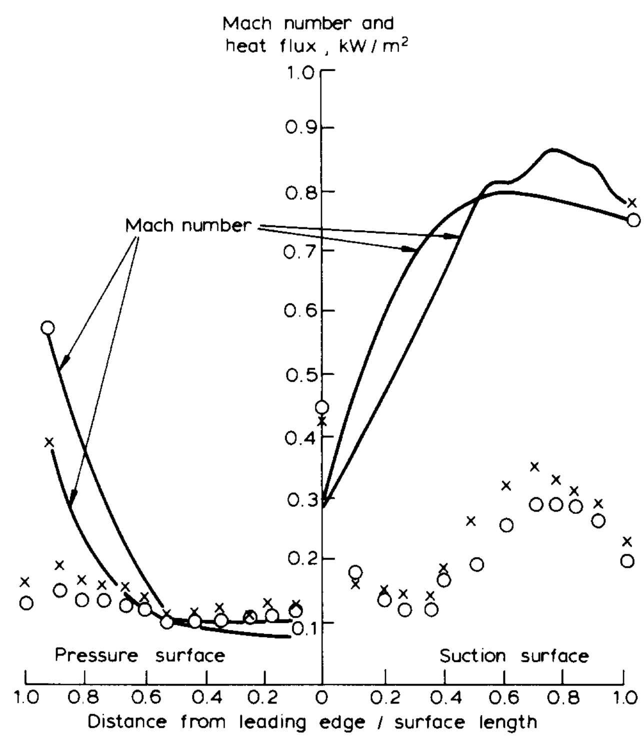

Fig 4 Example of the variation of heat flux coefficient with surface position for a trailing edge Reynolds number = 1.113×10^6 and turbulence intensity = 0.04. ×—low stagger profile; ○—high stagger profile (from Nicholson et al[3]*)*

Baines *et al*[5] describe a short-duration (three to five seconds running time) blowdown tunnel for aerodynamic studies on gas turbine blading which one may describe as an adjunct to the Oxford University Isentropic Light Piston Tunnel used for heat transfer and aerodynamic measurements. The blowdown tunnel can handle blades of either 100 mm or 300 mm chord covering a range of trailing edge Reynolds numbers from 3×10^5 to 6×10^7 m^{-1} for an exit Mach number of unity and throat Mach numbers up to 1.45. The downstream pressure in the tunnel is controlled by tandem ejectors which themselves, in conjunction with the blade cascade tank, are supplied with constant pressure air via a pressure regulator. The first ejector is an adjustable peripheral nozzle, the throat setting of which, in conjunction with the supply pressure, controls the cascade exit pressure. The second ejector consists of four discrete nozzles each separately controlled, feeding into the exhaust at a small angle. Aerodynamically the blow-down tunnel compares favourably with the well proven isentropic light piston tunnel and should prove extremely useful in the future for collecting accurate airfoil data.

Of these papers presented in the external blade heat transfer sessions two are mainly theoretical, that is, Amano[8] and So *et al*[7]. Amano undertook a numerical study of heat transfer to liquid cooled blades; in computing turbulent flows, he used a hybrid of central and upwind finite differences with the Boussinesq turbulent viscosity concept and the standard κ-ε turbulence model. I suggest that the major contribution of Amano is in his choice of wall boundary conditions. He assumed that beyond the viscous sublayer the turbulent length scale is universal, increasing linearly with distance from the wall. Amano compared his predictions with the measurements of May[28]. The comparison with experiment was good in some respects, increasing free stream turbulence intensity increased heat transfer, but poor in others. It is surprising that Amano used the measurements of May for comparison purposes as May's measurements were restricted to a Reynolds

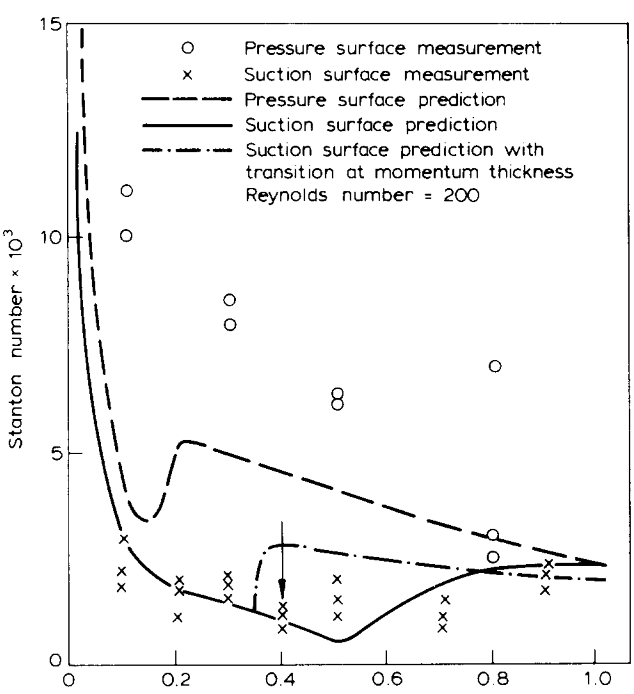

Fig 5 Comparison of calculated and measured Stanton number on blade pressure and suction surfaces (from So et al[7]*)*

number of 4.5×10^4 based on trailing edge fluid conditions and blade chord. It would be of interest for Amano to use experimental measurements nearer engine conditions for comparison purposes.

So *et al* modified STAN 5, a two-dimensional boundary-layer programme[29], so that it can be used to calculate heat transfer coefficients around turbine blades, especially for turbines with high inlet temperatures. The programme accounts for compressibility, pressure gradient, boundary-layer transition, streamline curvature, free stream turbulence and fluid property variations through the Eckert reference temperature. (In passing one might comment that the Eckert reference temperature is a useful parameter for accounting for property variations; the ratio of wall to freestream temperature[30] is equally useful). So *et al* make two important contributions in modifying STAN 5. They introduce a stagnation point flow calculation that allows flow to start from a forward stagnation point and eliminates the necessity of having known initial conditions and secondly they introduce start and length of boundary-layer transition criteria. Comparison is made of prediction with the measurements of Kercher *et al*[6] obtaining fair agreement for the suction surfaces but poor agreement for pressure surfaces (Fig 5).

The final paper to be discussed in this section is due to Hannis and Smith[2] who made measurements on a 4.5 MW industrial gas turbine with turbine entry temperature 1000 °C, pressure ratio 7.8, mass flow rate 22 kg/s at a rotational speed of 11 400 r/min. The cooled rotor blades had three single-pass cooling channels, the leading and trailing edge channels were basically triangular in section without surface ribbing, whereas the central channel was rectangular in section with ribbed surfaces. The cooled nozzle guide vane was fabricated, having a central coolant supply duct giving leading edge impingement cooling followed by convection cooling of the suction and pressure surfaces leading to trailing edge cooling via a pillared channel and venting on the pressure surface (Fig 6). Temperatures in the nozzle guide vane were measured with thermocóuples; the more difficult task of monitoring rotor blade conditions was achieved with a marked degree of success by an optical pyrometer located between combustion chambers. The use of optical pyrometers proved so successful that Hannis and Smith hope to use them in production industrial gas turbines as continuous monitors. The nozzle guide vane measurements were nearly as predicted but the measurments on the rotor blade were above prediction (Fig. 7). Their predictions drew on Seyb[31], Brown and Burton[32], Hall[33], Chupp[34], McAdams[35], Stachiewicz[36], Theoclitis[37] and Faulkner[38]. Hannis and Smith and their employers, Ruston Gas Turbines Ltd, should be complimented on their work. They had the courage to make engine tests instead of model, cascade tests and have shown that certainly as far as industrial gas turbines are concerned, continuous on-line rotor blade monitoring is possible with optical pyrometers.

In conclusion to this section, it is clear that knowledge of and, therefore, the ability to predict the effect of fluid properties and blade geometry on heat transfer to suction surfaces have improved dramatically, but the same is not true for pressure surfaces. Also, in general, heat transfer to pressure surfaces is greater than that to suction surfaces and some redesign, maybe along the lines of Nicholson *et al*, could be one way forward.

Film and transpiration cooling

Measurements of flow rates and distributions, film cooling effectiveness and heat transfer and comparisons with prediction procedures have been made for single surfaces and blades in cascade for cold and hot situations and for single and double rows of holes and for full-coverage film cooled blades. This work is reported in five papers[11-15]. The work described by Han and Jenkins[14] on the film cooling effectiveness of steam and that of El-Masri[13] on two-phase transpiration cooling are important contributions to the general fields of film and transpiration cooling. Although both these contributions are important, the work reported by Afejuku *et al*[11], Jones and Loftus[12] and Yoshida *et al*[15] have more relevance to current design and practice.

Jones and Loftus investigated the effect of the ratio of wall to free stream temperature on heat transfer in the presence of film cooling from one row of

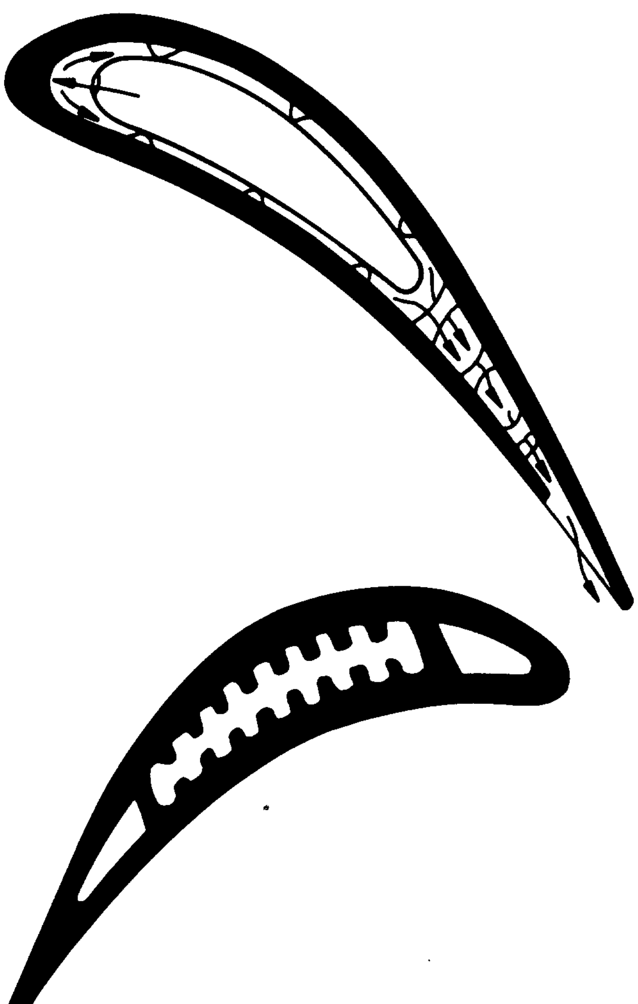

Fig 6 Stator and rotor blade sections (from Hannis and Smith[2])

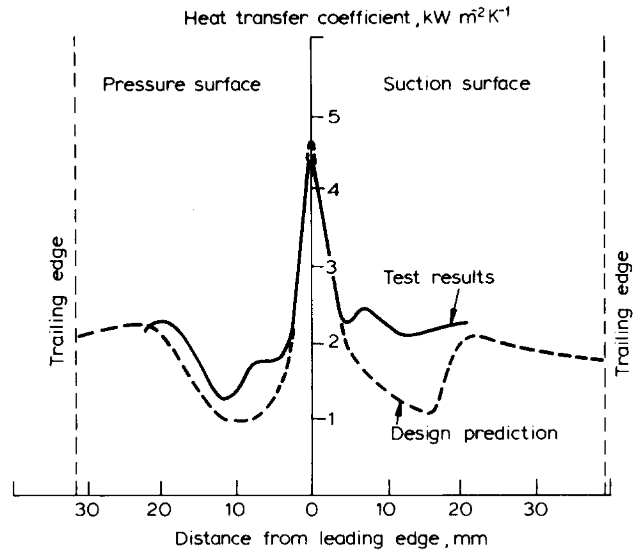

Fig 7 Rotor blade external heat transfer (from Hannis and Smith[2])

holes under conditions which realistically simulate a gas turbine environment. First, using the Oxford University Isentropic Light Piston Tunnel they checked that temperature superposition as proposed by Kays and Crawford[39] for turbulent boundary layers with wall temperature variation was valid. This they did and showed that when the ratio of wall to free stream temperature was less than unity the temperature ratio has no effect on the heat transfer as predicted by Kays and Crawford. Having established superposition and the accuracy of the experimental apparatus, Jones and Loftus extended the work to film cooling situations. They concluded that temperature superposition can be applied in the presence of film cooling under gas turbine conditions where the wall to free-stream temperature ratio is less than unity. The usefulness of this piece of work tends to be detracted from by a number of typographical errors in the paper, the most important and obvious of which is in the misplacement of the graphs 8 and 9. Afejuku *et al* carried out film cooling experiments using a mass-transfer technique to investigate the distribution of coolant from two rows of holes with blowing rates controlled independently. Staggered and in line configurations were used with pitch to diameter ratios of three and row spacings of ten, twenty, thirty and forty hole diameters, the injection angles were 35° or 90°. The blowing rates considered approximated to 0.5, 1.0, 2.0 and 3.0, the free-stream had a zero pressure gradient and the boundary layer at the point of injection of the first row of holes was turbulent with the ratio of displacement thickness to hole diameter equal to 0.16. The ratio of injected to free-stream density was 2.0 and this was achieved by using a mixture of Freon-12 and air as injectant. Foot prints of Freon-12 concentrations and, therefore, film cooling effectiveness were measured on the surface for a range of film cooling conditions; also, concentrations above the surface were measured showing the jet formation, (Figs 8 and 9). As expected, staggered rows of holes are more effective than in-line and from their measurements Afejuku *et al* suggest that the momentum flux of the upstream film between holes of the downstream row is crucial in control of the jets from the downstream row. This momentum flux suggestion helps to explain the findings of Jabbari and Goldstein[40] and Bergeles *et al*[41]. The effect of the first row film on the ejection from the second row is to reduce the tendency for jet lift off and if lift off still occurs the film from upstream tends to fill in under the second row jets.

Yoshida *et al* describe a laminated, diffusion bonded technique for manufacturing a full-coverage film cooled turbine nozzle guide vane. Having made a blade by this technique they then tested it in a cascade by measuring flows and pressure drops through coolant injection holes in the absence of free-stream flow, and then film cooling effectiveness with free-stream flow at various positions on the blade surfaces. The measurements of flow rates and pressure drop in the absence of free-stream flow allowed Yoshida *et al* to correct their numerical analysis for determining coolant flow rates through the blade internal passages and then apply it to the blade in the presence of a free-stream flow. They found that the average film cooling effectiveness was of the order of 0.7 when the ratio of coolant flow rate per blade to free-stream flow rate per cascade channel was about 0.06.

In conclusion to the section on film and transpiration cooling, it is clear that the basic research work of the 1960's and early 1970's is now being refined to take account of more second order variables. Also, some of the earlier predictions on flow distributions are now being validated. There is a need to carry out film and transpiration cooling measurements in engines or at least in blade cascade rigs under as near as possible engine conditions both adiabatically and with heat transfer. As an example of this last comment, Litchfield and Norton[1] found under near engine conditions in a cascade rig that pressure surface heat transfer was reduced by ejection of cooling air on the surface.

Blade internal cooling

Understanding the fluid flow characteristics inside turbine blades and the resultant heat transfer in what today are extremely complicated coolant passage geometries is necessary for design purposes. The complicated geometries arise firstly from the method of introducing the coolant into the root of the blade particularly of rotor blades which involves a catchment, plenum and manifold. Secondly over the last twenty years, as knowledge of the critical cooling regions of blades has increased, the coolant passage geometries have been refined. Initially single-pass convection cooling with venting at the blade tip was used. This has progressed to multiple-pass geometries followed by the inclusion of leading edge impingement, leading edge venting and film cooling, film and transpiration cooling of both suction and pressure surfaces, extended surfaces via arrays of pillars in the trailing edges and trailing edge venting and, of course, surface ribbing of channels. In addition to the blade internals, efforts are made, usually through labyrinth seals, to prevent hot gases reaching

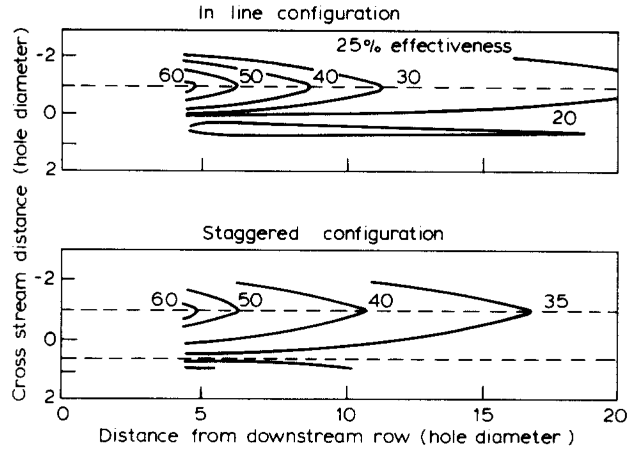

Fig 8 Film cooling foot prints (from Afejuku et al[11]*)*

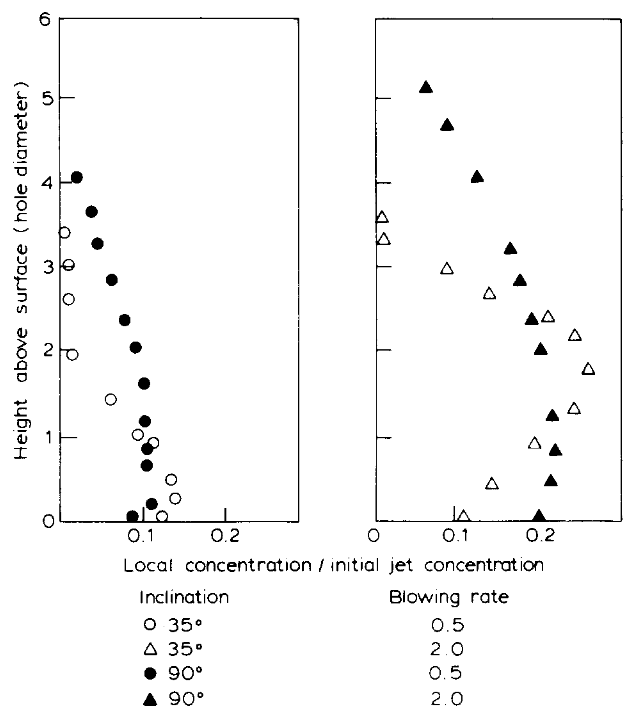

Fig 9 Jet development (from Afejuku et al[11]*)*

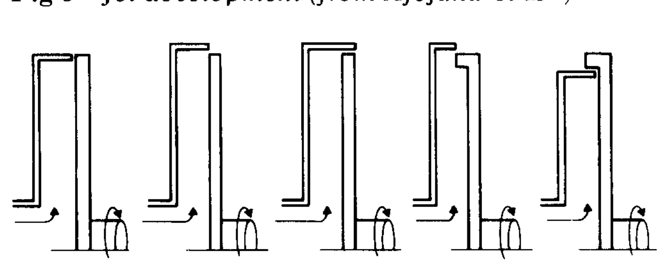

Fig 10 Rotor-stator seal configurations (from Phadke and Owen[17])

turbine disks. Six papers were presented on blade internal cooling and seals[16–21].

Phadke and Owen[17] carried out experiments to investigate the performance of radial-clearance seals. They found to their advantage that radial-clearance seals can exhibit a pressure inversion effect with increasing disk speed which is not the case with axial-clearance seals[42,43]. From flow visualisation and pressure measurements, the performance of radial-clearance seals was measured, correlated and compared with axial-clearance seals. A schematic diagram of the seals is shown in Fig 10 and the inversion effect occurs at high flow rates and increases in strength with increasing stationary shroud overlap on the rotor. This finding is important for labyrinth seal design.

It is interesting to see the recent increase in work aimed at understanding which fluid flow properties affect discharge coefficients of impingement and film cooling holes. This is a good example of changing design for one reason, in this case heat transfer, showing up the weaknesses of accepted practice and knowledge in an associated flow process. Hay *et al*[18] and Florschuetz and Isoda[20] have measured discharge coefficients in geometries common to current blade design modelling practical flow situations. Hay *et al*[18] investigated the effect of crossflows on the discharge coefficients of film cooling holes and Florschuetz and Isoda[20] have also investigated the effect of crossflows on discharge coefficients and flow distributions but in their case for jet array impingement. Generally Hay *et al* use the ratio of coolant stagnation pressure to free-stream static pressure as their correlating parameter. At low values of the pressure ratio the film cooling hole discharge coefficient is a strong function of this ratio, increasing with increasing pressure ratio. As the pressure ratio increases the discharge coefficient increases to a maximum value beyond which a plateau exists. Both coolant flow and free-stream Mach numbers influence the value of the pressure ratio at which the discharge coefficient plateau occurs. In nearly all cases the plateau will have been reached at a pressure ratio of 1.6 and the discharge coefficient at the plateau is between 0.7 and 0.8. The exceptions to the general rule are 30° angle of injection and at high coolant flow Mach numbers when the discharge coefficient can be as high as 0.97. Florschuetz and Isoda determined flow distributions for jet arrays with ten spanwise rows of normal holes and found that the flow distributions ranged from uniform to highly non-uniform depending on the geometric configuration and the ratio of initial crossflow to jet velocities. They found that for some practical situations the jet orifice discharge coefficient is a function of the crossflow to jet velocity ratio. The function reduces to a constant for initial crossflow rates equal to or greater than the total jet flow rate provided the ratio of spanwise jet hole spacing to channel height (jet plate to impingement surface spacing) is greater than 12. Quite good agreement between numerical predictions and measurement was achieved for most situations. The primary reason for deviations between prediction and measurement is presumed by Florschuetz and Isoda to be due to the lack of an adequately precise friction factor model for these complex flows. One telling statement on flows in blade internal passages is the final conclusion of Hays *et al*: 'no analytical model exists for the prediction of discharge coefficients in the film cooling situation'.

Behbahani and Goldstein[19] measured the two-dimensional variation of local heat transfer on a flat surface due to impinging jets from a staggered array

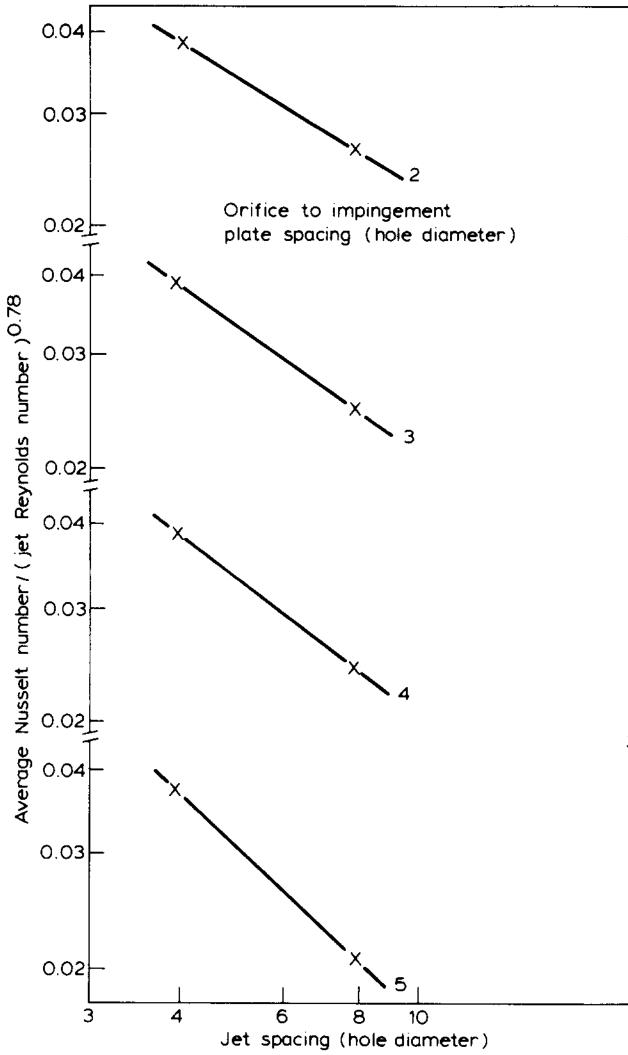

Fig 11 Correlation for impingement cooling (from Behbahani and Goldstein[19])

of circular jets, the plane of the jet plane being parallel to the plane of the impinged heat transfer plate. The coolant flow from the impinging jets was constrained to flow over the heat transfer surface in one direction only. Two different pitches of impinging jets were considered and the spacing between the jet and impinged heat transfer surfaces as well as the jet Reynolds numbers were varied. As may be expected for this coolant geometry the local heat transfer coefficients varied both in the flow direction and across the span with the highest heat transfer occurring at stagnation regions. The stagnation regions of individual jets move further in the downstream direction as the amount of cross flow due to the upstream jets increases. Behbahani and Goldstein were unable to correlate local heat transfer but had a high degree of success for averaged heat transfer. They found that area-averaged Nusselt number has a weak dependence on jet to impingement plate spacing of four and then decreases at higher spacing (Fig 11). There is a strong dependence between the averaged Nusselt number and the jet Reynolds number and jet pitching. It is found for the flow conditions examined, that the averaged Nusselt number is proportional to the jet Reynolds number raised to 0.78 and the jet pitch to diameter ratio raised to between −0.626 and −0.7 depending on the jet to impingement plate spacing. In view of this correlation it follows that the area-averaged heat transfer coefficient is approximately inversely proportional to jet diameter.

Arrays of short circular pillars inside the trailing edge region of turbine blades with trailing edge venting are becoming common practice with many engine manufacturers. Metzger and Haley[21] have carried out some flow visualization and heat transfer experiments for staggered arrays of constant length, short pin fins spanning the full height of the duct and with pitch to diameter ratios of 1.35 or 2.5. In both cases the streamwise development of heat transfer, averaged across the duct width (which corresponds to blade span in practice) was resolved to single pin row spacing. An interesting feature of Metzger and Haley's work is the comparison of measurements for arrays of conducting pins and of non-conducting pins. They concluded from these measurements that tests using non-conducting pin arrays are a useful and relatively inexpensive method for measuring the comparative performance of a wide variety of array geometries. There are a large number of parameters involved in the fluid flow and heat transfer through the pin arrays found inside blade trailing edges making it difficult to interpret the measurements. Invariably it is best to present the results in terms of either the averaged Nusselt number for the complete array or the averaged Nusselt number per pin row, (Figs 12 and 13). Finally, Metzger and Haley make an interesting comment on turbulence levels in that: 'the turbulence level is highest in the forward portion of the array and decreases to a lower level downstream'. Bearing in mind that their arrays were of constant length pins and the duct had constant width then this finding of Metzger and Haley is important.

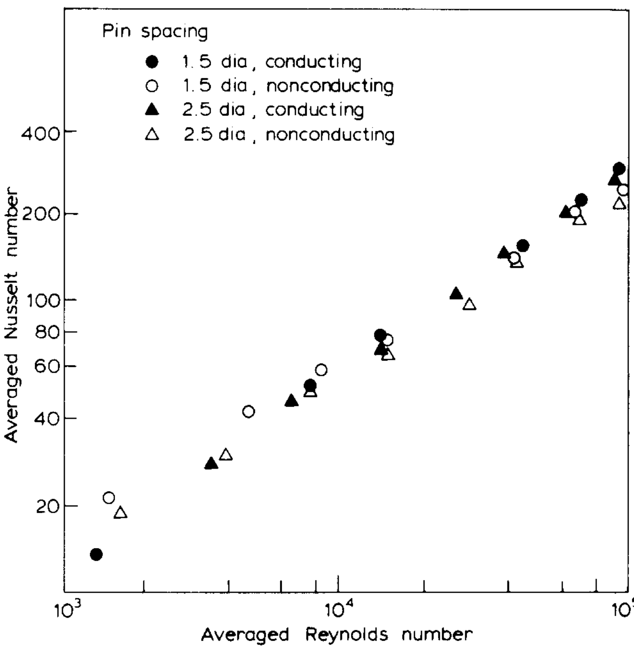

Fig 12 Variation of array averaged Nusselt number with averaged Reynolds number (from Metzger and Haley[21])

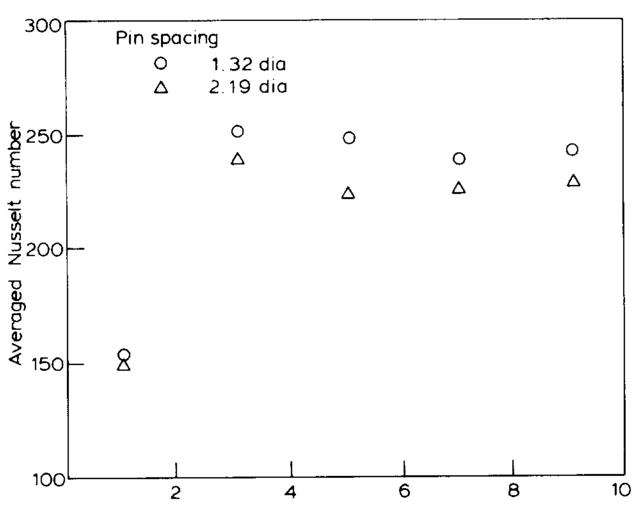

Fig 13 Variation of pin averaged Nusselt number per pin row (from Metzger and Haley[21])

The final paper to be considered is due to Owen and Onur[16] who carried out flow visualization, velocity and heat transfer measurements in a rotating cavity with either an axial throughflow or radial outflow of coolant. The purpose of this work was to try and obtain a better understanding of the conditions inside turbine rotors. In the axial throughflow tests, flow visualization revealed the presence of spiral vortex breakdown and that this breakdown depends on the Rossby number. For small Rossby numbers, where vortex breakdown is reduced, the mean Nusselt number of the heated downstream disk of the cavity can be correlated in terms of the gap ratio of the cavity, that is the ratio of axial width to outer radius of the cavity, the axial Reynolds number and the rotational Grashof number. From the radial outflow tests for a constant flow rate with the downstream disk heated, a critical rotational speed was observed. Above this speed the classic structure of Ekman layers on the disks and a central inviscid core broke down into a chaotic flow with oscillations of the inner layer. The occurrence of this behaviour depends on the flow rate and rotational Grashof number and Owen and Onur suggested this heralded the onset of free convection. For this 'chaotic' flow regime the measurements were well correlated in terms of the mean Nusselt number and the Grashof number raised to 0.286.

In conclusion to this section on internal flows the shear complexity of blade internal geometries with the additional complication of rotation about an axis for rotors makes fluid flow and heat transfer analysis impossible. Correlations of measurements coupled to numerical techniques and engine component performance and life data must be used to improve design.

Comments

As a general conclusion to this article ASME should be complimented on their ability year in and year out to organise among many other functions the annual gas turbine conference and exhibit. The gas turbine conference is always of high technical quality and well attended. As an author on a number of occasions, session organiser for the last two conferences and a frequent delegate for many years, I cannot speak too highly of the ASME Gas Turbine Division and the Heat Transfer Committee in particular.

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