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AXISYMMETRIC COMPUTATIONS OF FLOW AND HEAT TRANSFER IN A PRE−SWIRL ROTOR−STATOR SYSTEM

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ABSTRACT An axisymmetric, turbulent, multigrid computational fluid dynamics (CFD) code is used, on a parallel computing facility, to predict the mixing between superposed flows in a pre−swirl rotor−stator system, being a simplified representation of the internal air cooling system in gas turbine engines. Strong recirculations formed by the pre−swirl cross−flow lead to entrainment of disc−cooling air from the heated rotor into the underside of the cold pre−swirl jet, raising the temperature of the air leaving the system to cool the turbine blades. Disc speed, flow rates and seal geometry all affect the detailed structure of the mixing process, however many common features allow insight into the thermal contamination of the cross−flow.

1. Introduction

Effective cooling of the rotor discs and turbine blades of a gas turbine engine can increase its efficiency and operating life. Much useful information on the behaviour of flows in the internal air cooling system can be obtained from the study of purpose−built experimental rigs, allowing detailed measurements to be taken under controlled flow and heating conditions. Such studies promote basic understanding, provide useful information for designers, and supply data for the validation of numerical simulations of the idealised systems.

Fig. 1a,b shows the axisymmetric computational geometry representing a pre−swirl rotor−stator rig, designed and built at the University of Bath, for the study of heat transfer. Cold air is introduced (into the pre−swirl chamber formed by inner and outer shrouds) through angled nozzles on the stator and leaves through holes on the rotor (supplying cooling air for the turbine blades in a real engine). In addition, disc−cooling air is introduced at the axis to remove heat from the rotor and ’windage’ caused by frictional heating at the disc surface [1].

2. Computational Model and Numerical Techniques

To minimise computing times an axisymmetric system is considered, in which the stator nozzles and rotor holes are merged to form equivalent area slots on the stator and rotor discs respectively, as illustrated in Fig. 1a (the slot widths are small compared with the hole diameters but the injection velocity is the same). Fig. 1b illustrates the 113x343 (axial x radial) cylindrical−polar grid used for the computations. The inner shroud is modelled by two block obstructions within the grid, which is concentrated at solid surfaces using geometric expansion factors of around 1.2. The outer shroud is simplified as a stator−side flow exit. The rotor−disc temperature distribution was fitted to experimental data and stationary surfaces were assumed adiabatic.

The steady, turbulent axisymmetric computations were carried out using an elliptic multigrid solver incorporating the SIMPLEC iterative method [2] for solution of the steady−state Reynolds averaged Navier−Stokes and energy equations on a staggered grid [3,4] The low Reynolds−number k−ε (LR k−ε) turbulence model (modified from [5]) is capable of predicting transition to turbulent flow in the radial boundary layer on the rotor−disc, and gives improved agreement with heat transfer data for rotating−disc systems (compared with wall function treatments) [6], however a very fine grid is required in order to resolve the viscous sub−layer near the wall.

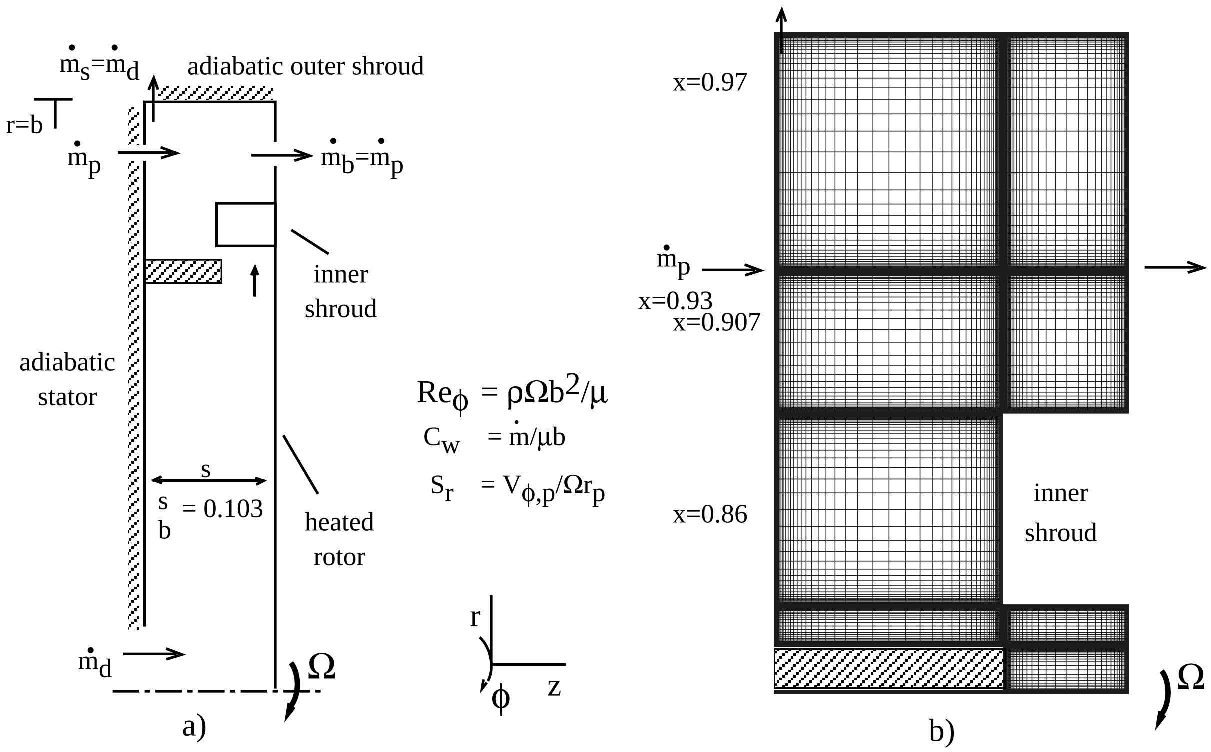

Fig. 1 a) Schematic diagram of rotor−stator rig and b) pre−swirl chamber computational geometry

Multigrid convergence acceleration was used in the form of a fixed V−cycle, Full Approximation Storage scheme. For simplicity, turbulence model equations were solved only on the finest grid, and restricted values for turbulence quantities were used unchanged on the coarser−grid levels. Test cases suggest a speed−up of around six for 4−level multigrid compared with corresponding single grid computations, although the latter were not attempted in the present work. Further time savings were achieved by carrying out the computations in parallel on 8 nodes of a Meiko M10/MK086 machine, having 16 Intel 1860 processors linked by T800 transputers. The complete grid was partitioned across the 8 nodes as horizontal slabs with equal numbers of grid points per slab. This gave a parallel speed of around six while preserving the efficiency and stability of the multigrid algorithm [4].

3. Computational Results

Several cases were computed from an extensive experimental programme. Parallel computation time for each case was approximately 9 hours. Results are presented for a representative high speed (600 rev/min) case for which $Re_\phi=1.71\times10^6$, $C_{w,d}=2981$, $C_{w,p}=3.081\times C_{w,d}$ and Sr=0.68. Fig. 2 illustrates the development of the flow in terms of profiles of radial velocity, tangential velocity and temperature at four radial locations within the pre−swirl chamber (Fig. 1b). At x=0.86 and x=0.97 the flow exhibits classical rotor−stator behaviour with a uniform temperature, constant−velocity rotating core between thin boundary layers on the discs. The tangential velocity profile at x=0.93 shows penetration of the core of the annular pre−swirl jet into the wheelspace. At x=0.907 the impinging pre−swirl flow gives rise to a powerful flow radially inward on the rotor. Good agreement has been observed in comparisons between predictions and measurements of tangential velocity and Nusselt number in several cases, for locations in the rotor−stator system beneath the inner shroud. These comparisons will be reported elsewhere.

Fig. 3 shows profiles of axial velocity in the pre−swirl chamber at three axial locations. These profiles illustrate the recirculating flow above and below the pre−swirl flow centreline. Reverse flow on the surface of the inner shroud (Fig. 3c) forces the rotor boundary layer to separate from the disc. This is illustrated in Fig 4, using visualisation techniques to track the progress of the disc flow in the r−ϕ plane. The heated disc flow cools as it is carried by a clockwise recirculation and entrained into the underside of the pre−swirl cross−flow. The temperature distribution in the fluid (Fig. 4) illustrates the core of the annular pre−swirl jet and downward deflection of the cross−flow, due to anti−clockwise recirculation of heated air above the centreline. This explains the temperature rise along the centreline shown in Fig. 2c.

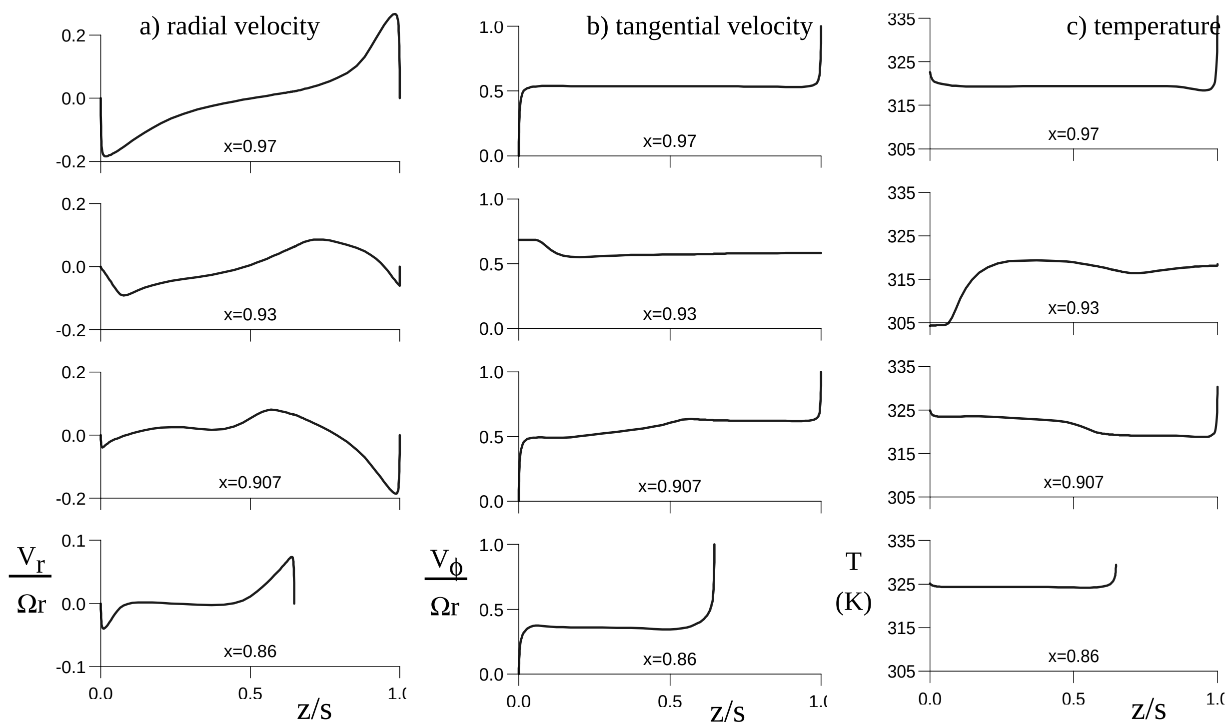

Fig. 2 Velocity and temperature profiles at different radial locations in the pre−swirl chamber

4. Conclusions

Axisymmetric steady−state computations of turbulent flow and heat transfer have been carried out in order to provide insight into the mixing between disc−cooling air and a pre−swirl cross−flow in a heated rotor−stator system. The powerful pre−swirl jet gives rise to strong recirculations in the pre−swirl chamber, leading to separation of the heated rotor boundary layer and thorough mixing with the cross−flow. The bulk temperature of the blade−cooling air at the rotor exit (Fig. 4) is lower than would be the case if the rotor boundary layer remained attached to the heated disc inside the chamber [7]. Axisymmetric and 3D turbulent computations deliver similar qualitative predictions of the recirculating motion and entrainment [8], and it is expected that the more economical axisymmetric computations will provide sufficiently accurate predictions of heat transfer for practical design purposes.

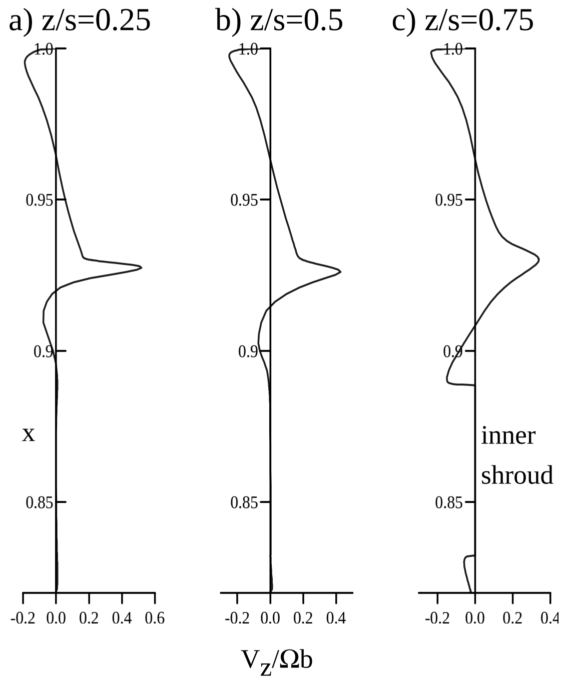

Fig. 3 Axial velocity profiles

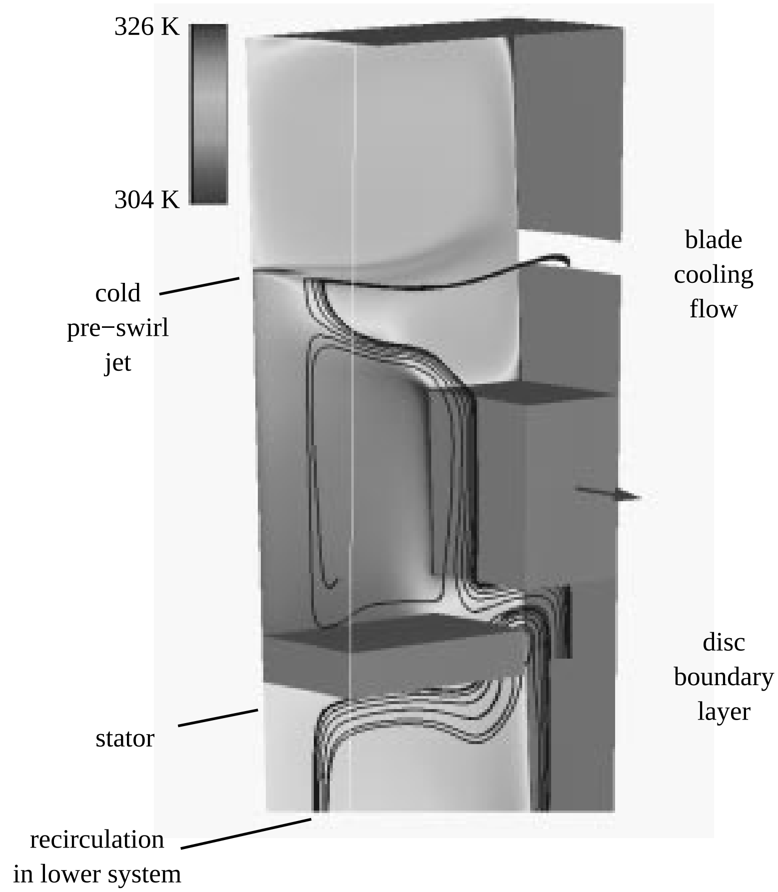

Fig. 4 Entrainment of the disc boundary layer

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