A STUDY OF TIP CLEARANCE INFLUENCE ON AXIAL FLOW COMPRESSOR PERFORMANCE

Jesuino Takachi Tomita, jtakachi@ita.br
João Roberto Barbosa, barbosa@ita.br
Instituto Tecnológico de Aeronáutica - ITA

Abstract. *This paper deals with the study of a single-stage, high performance axial flow compressor aiming at the selection of the appropriate tip clearance dimensions for both rotor and stator. High performance is associated with high pressure rise per stage, what imposes high over tip leakage and high pressure loss. Tip clearance must therefore be carefully examined and its influence predicted over the speed and mass flow ratio at which the compressor would work at design and off-design. Experimental data show that a region with high flow losses is located at the clearance over the rotor tip, due to the flow leakage from the pressure side to suction side of the blade. The compressor under analysis was designed using the mean line technique. The 3-D flow calculation is carried out by a commercial CFD code (NREC) that solves the 3D viscous and turbulent conservation equations at the blade passages, using structured grid and finite volume discretization. Turbulence is accounted for using the one-equation Spalart-Allmaras model. The pressure ratio and stage efficiency are evaluated for several tip clearances.*

Keywords: *axial compressor, tip clearance, gas turbines, CFD.*

1. INTRODUCTION

Computational fluid dynamics has been used in many engineering areas, such as aeronautical, automotive, chemical, petroleum and energy for both internal and external flows. In the last decades the study of flowfields in axial compressors has been greatly increased. The flow behavior in compressor is very complex, mainly the high speed ones, where shock waves, secondary flows, leakages and separation are present. High performance compressors need good inter-rows matching for stable operation. Experimental and numerical tools are necessary to predict and to correctly understand the flow along the compressor. Experimentally, data acquisition is difficult and expensive, due to test rig installation limitations. Numerically, the calculation need a robust numerical methods to face the complex fluid motion. A complete description of internal flows in axial compressors, including the study and the analysis of the loss mechanisms is found in the literature (Wu, 1951 and Tomita, 2003). Tip and hub clearances have great influence on compressor efficiency and stability, so that it is important its evaluation. Today's computational capability allows the computation and analysis of the three-dimensional flow in axial compressors for that purpose. In this work, the three-dimensional flow calculation is used for the study of the tip clearance influence on the efficiency of an axial compressor.

2. SOME COMMENTS ON GAS TURBINE ENGINE

Gas turbines efficiency depends heavily on its main components efficiency: compressor, combustion chamber and turbine. Other components like oil pumps, valves, ducts have less influence on the overall efficiency. Therefore, the aerothermodynamics involved in compressors and turbines design is a key issue to guarantee good gas turbine performance (Tomita, 2003 and Barbosa, 1987 and Tomita and Barbosa, 2004). Blade profile selection, flow matching between the rotor and stator rows, high heat exchange rates in turbine blades, material limitation are among the many tasks the designer must face and they all are flow-related. More recently, cost of fuel is driving the compressor design towards very high efficiency in a wide range of operation conditions, what makes the detailed study of losses and their mechanisms a must. CFD has proved to be a key tool for those studies, in particular for the hub and tip clearances loss. This paper shows one of the possible means.

3. TIP CLEARANCE INFLUENCE ON AXIAL COMPRESSOR PERFORMANCE

The flowfield in axial compressor is unsteady and highly three-dimensional. The gaps formed by the rotor blades and the shroud, as well as between stator blades and the rotating hub, cause the flow to leak from one side to the other side of the blades, driven by local pressure difference, with significant velocities that will cause disturbance to the main flow. Detailed description of the physics involved can be found in reference (Tan, 2006). It will include the boundary layer, secondary flow and scraping vortex and their interaction. In the gap the flow does not follow the directions given by the blades channel, so that it does not participate in the process of energy conversion, consequently causing performance degradation. Increasing the gap increases the flow leakage and the performance degradation: pressure ratio, mass flow rate and efficiency fall; surge line moves downwards, causing decrease of surge margin. In this context, flow calculation is important for the performance prediction and map generation.

The leakage jet and the main flow are in different angles, inducing a vortex in the main flow. This vortex depends on the turbulence level and Reynolds number, so that, in certain conditions, it may not exist due to the dissipation and diffusion processes. Its existence can be proved through two-dimensional cascade tests. The viscous effects in the tip region may result in flow separation at the blade suction side, but the leakage may be beneficial when blade tip corner separation is present because the leakage flow "wash out" the the separated region. The general effect is very complex to understand mainly because the leakage flow occurs at the endwall region. In the endwall region the lower axial velocity results in a higher flow incidence at blade leading edge, causing higher blade loading. Rotating stall can appear in some cases at the tip region. Some data acquisition made at compressor facilities (Camp and Day, 1997 and Day, 1993) have demonstrated that there are two ways to cause rotating stall. The first is characterized by the growth of disturbances of small amplitude, essentially two-dimensional long wave amplitude that extend axially through the compressor in the direction of the main flow. These disturbances are referred as modal stall waves (Moore and Greitzer, 1986). The second way is characterized by the development of three-dimensional "spike" disturbances localized at rotor tip region and have a length scale on the order of blade pitch.

There are some empirical models to account for the tip clearance effects (Cumpsty, 1989 and Denton, 1993 and Vo, 2001 and Koch and Smith, 1976). The detailed explanation of those models and a complete tip clearance discussion is not the scope of this work. To fully understand the tip clearance and tip leakage it is required to know details of the following important topics: secondary flow, tip clearance vortical flow, tip clearance vortex core trajectory, vortex core trajectory in passage and downstream, tip clearance flow blockage assesment, endwall blockage quantification, vortex core trajectory in passage and downstream, effects of asymmetric clearance on compressor performance and stability, tip clearance flow on compressor stall onset, relation between the tip clearance flow features and spike stall inception, interaction of blade rotor tip leakage vortex and streamwise vortex with stator blades, effect of upstream unsteady flow conditions on rotor tip leakage flow, interface angle between main flow and tip flow, interaction of rotor tip vortex with steady upstream wakes.

4. COMPUTED CHARACTERISTICS OF AXIAL COMPRESSOR

The compressor chosen for this work is a single-stage axial-flow compressor that, at design point runs at the speed $N = 37500rpm$, delivers the pressure ratio $PR = 1.57$ with the efficiency $\eta = 80\%$. Ambient conditions are: pressure $P_0 = 101325Pa$ and temperature $T_0 = 288.15K$. Losses were taken into account using the loss correlation of Koch and Smith. The rotor and stator blade heights at leading edges are $0.040m$ and $0.030m$ respectively. The airfoil profiles are multiple circular arc (MCA). The number of blades are 16 for the rotor and 40 for the stator. The rotor solidity ($\sigma = chord/bladespacing$)varies linearly from 1.25 at the tip, to 1.48 at the mid blade hight and up to 1.92 at the hub and for the stator from 1.38 at the tip, to 1.61 at the blade mid height and up to 2.02 at the hub. Figure 1 shows views of the compressor.

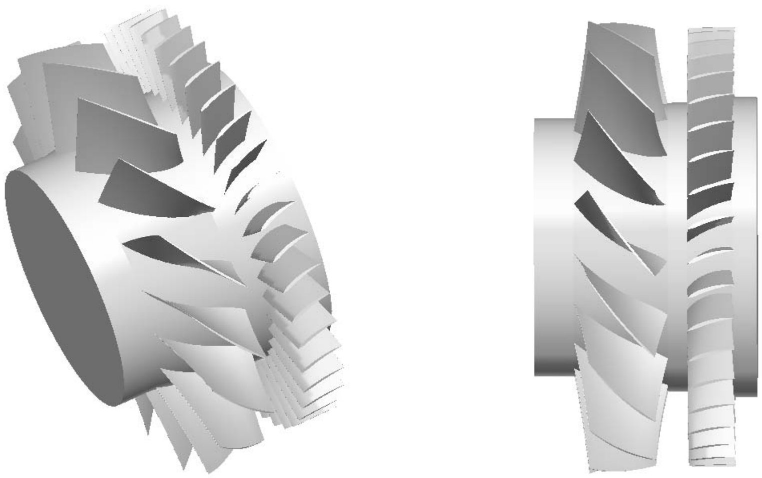

Figure 1. Views of the designed single-stage axial flow compressor

5. NUMERICAL SCHEME FOR CFD CALCULATION

The numerical scheme used for the calculations is based on finite-volume formulation. The time integration is performed using a four-step Runge-Kutta scheme and the spatial integration is performed with second-order centered-scheme. Artificial dissipation is used, following the recommendations of the literature (Jameson and Schmidt and Turkel, 1981), to avoid numerical instabilities caused by the convective terms of the Navier-Stokes equations. Turbulence is modeled using the one-equation Spalart-Allmaras turbulence model (Spalart and Allmaras, 1992. At the solid boundaries the log-law is used.

5.1 GRID GENERATION

A $37x71x137$ hexahedral-elements structured H-grid was used in the computational domain, with five layers of hexahedral elements placed radially in the tip gap. The stretching factor in the 3 directions are 1.17, 1.15 and 1. Figure 2 shows details of the grid structure. The grid independence was achieved.

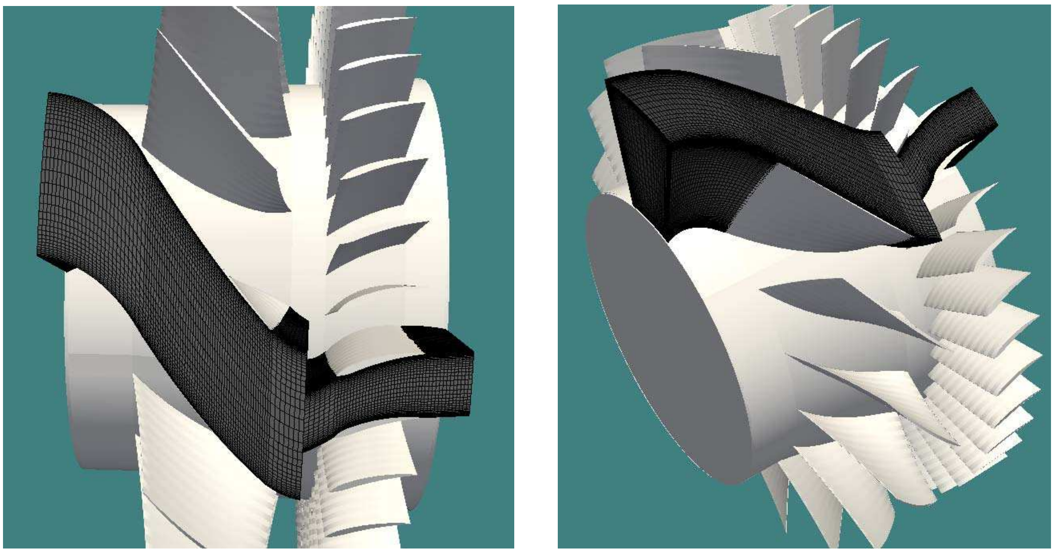

Figure 2. Details of the H-type grid and of the compressor rows

5.2 BOUNDARY CONDITIONS

Figure 2 shows the grid structure along one of the compressor stage passages, where the flow is calculated, to reduce the computational requirements. Therefore, flow is considered periodic in the sense that the blade passages are identical. Conditions at the boundary were calculated using the compressor design computer program already developed [Tomita, 2003]. Transition from the compressor rotating grid to the stationary stator one is made through a mixing plane (Belardini, 2003), at which circumferential averages of the rotor-leaving flow properties in the radial direction are calculated and transferred to the following stator-entering static grid.

6. TIP AND HUB CLEARANCES

The influence of tip clearance on the compressor performance was carried out in this work through the analysis of five different clearances 0.5%, 1.0%, 1.5%, 2.0% and 2.5% of the blade height, for both rotor and stator. Usually, as in this work, the design clearance is set to 1%. Table 1 shows details of the used gaps. It is worth noting that the gaps are small, requiring fine grids in that region.

Table 1. Values of the gaps at the rotor and stator tips and hubs

	ROTOR	STATOR
	Blade height=0.040m	Blade height=0.030m
%	Tip clearance (m)	Hub clearance (m)
0.5%	0.000230	0.000640
1.0%	0.000450	0.000330
1.5%	0.000670	0.000500
2.0%	0.000900	0.000658
2.5%	0.001120	0.000822

7. INFLUENCE OF TIP AND HUB CLEARANCES ON COMPRESSOR MASS FLOW, EFFICIENCY AND PRESSURE RATIO

Figure 3 shows the variation of compressor inlet mass flow, efficiency and pressure ratio for 100% speed as function of tip and hub clearances. The points shown are mass-averages obtained from the CFD calculations, in agreement with the literatures aforementioned.

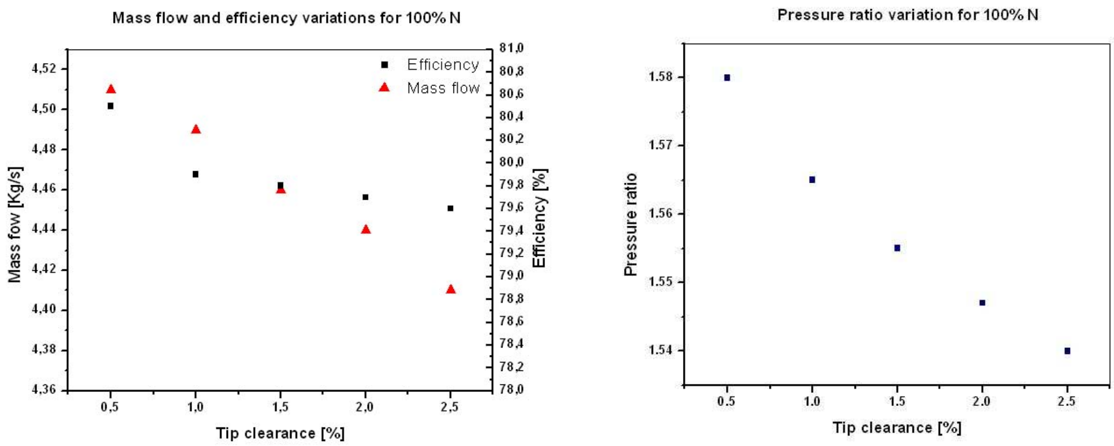

Figure 3. Influende of tip clearance on mass flow, efficiency and pressure ratio, for 100%N

The criterion to stop the numerical simulation can be based on the variables as mass flow error, efficiency and pressure ratio or number of iterations. In this work, the CFD calculations were considered converged after 5,500 iterations, as can be seen from Fig. 4.

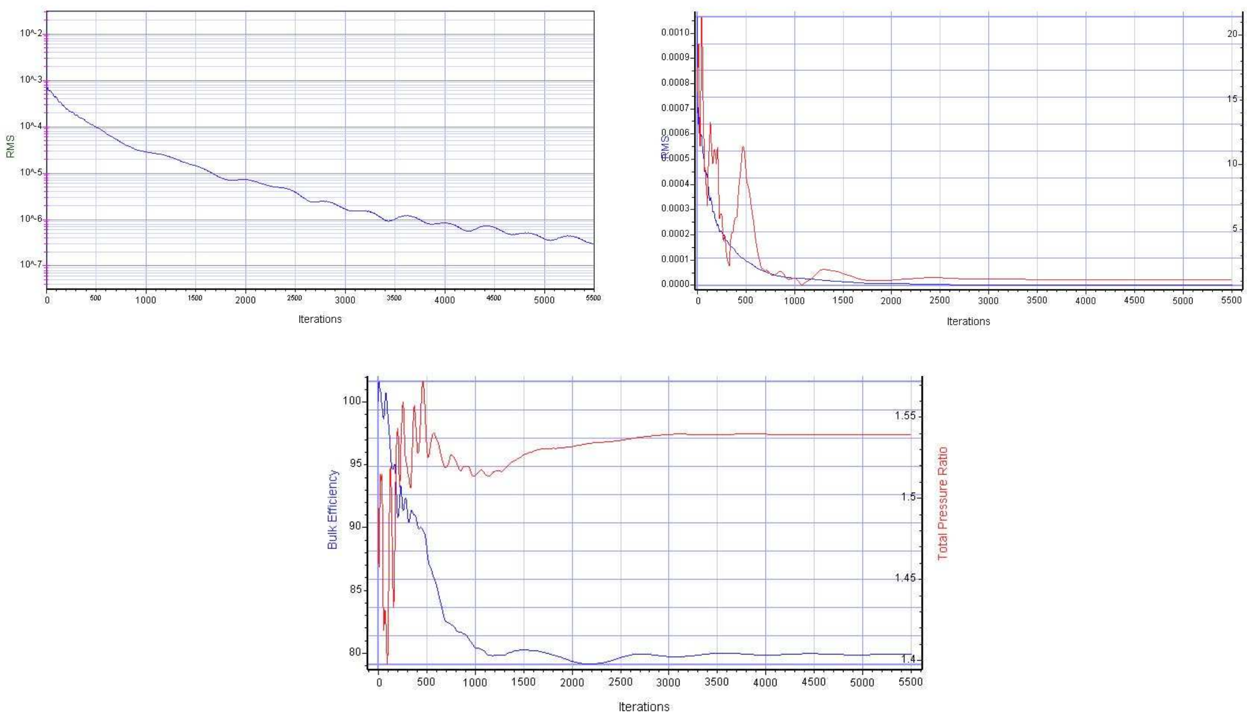

Figure 4. Convergence parameters

Figure 5 shows the the velocity vectors in the gap at the rotor tip, giving indication of the leakage flow.

Figure 6 shows high values of entropy at the walls and at the gaps for both rotor and stator, giving indication that the phenomena that occur in those regions must be carefully treated for the sake of performance improvement. Also, in the regions of boundary layers the entropy also rises. There is no indication of reverse flows, what indicates that stage blading is correct.

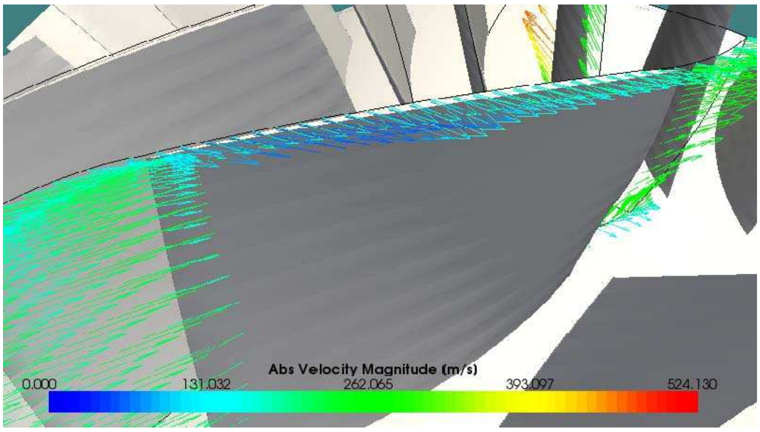

Figure 5. Velocity vectors at rotor tip (1% tip clearance)

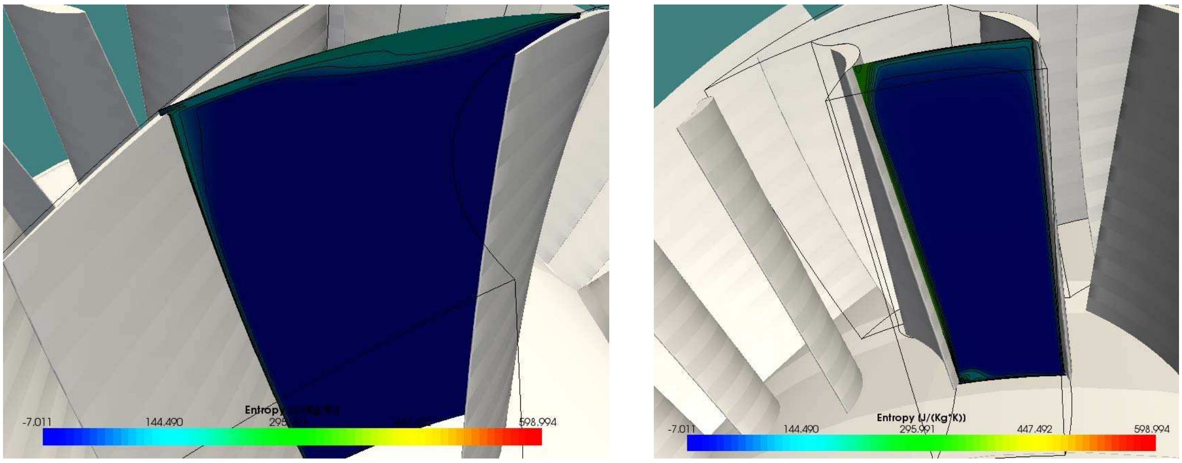

Figure 6. Contours of entropy in the compressor streamwise (1% tip clearance)

Figure 7 shows region of high relative velocity at mid blade height, indication of high loss, but unavoidable due to the high pressure ratio developed by the stage, what requires high speed flows.

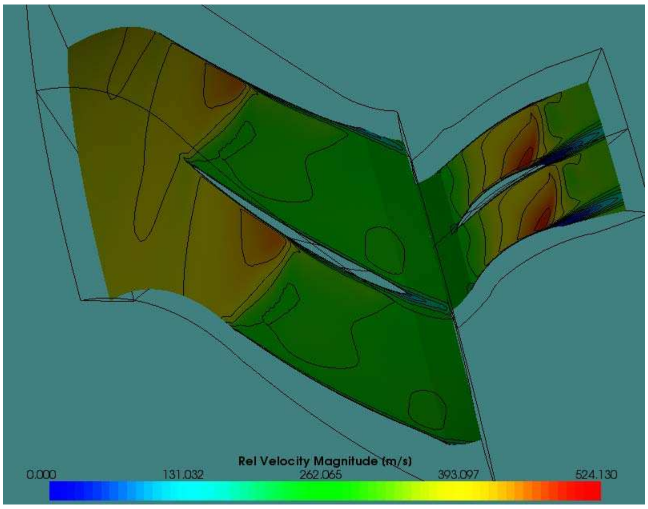

Figure 7. Contours of relative velocity in the compressor stage (1% tip clearance)

The study of the influence of the tip and hub clearances on the compressor performance was made with the aid of compressor performance maps, whose points were obtained from the CFD calculations using two different gaps, namely, 1.0% and 1.5% of the blade height.

It is expected that the operational range, at each rotational speed, be small, due to the high performance characteristic of the compressor, what means nearly vertical speed lines. Actual compressor, if build, certainly would require bleed valves and/or variable geometry to accommodate larger mass-flow range and avoid surge. Accordingly, there were many diverged calculations, so that a reduced number of converged solutions were taken into account.

Figure 8 present a comparison between tip/hub clearances for 100% and 90% of rotational speeds. The 1.0% tip/hub clearances are represented by blue triangles and the 1.5% by red circles. These figures show efficiency drop and the pressure ratio decreasing when the tip/hub clearances increased due to the high leakage flow on the blade tip region.

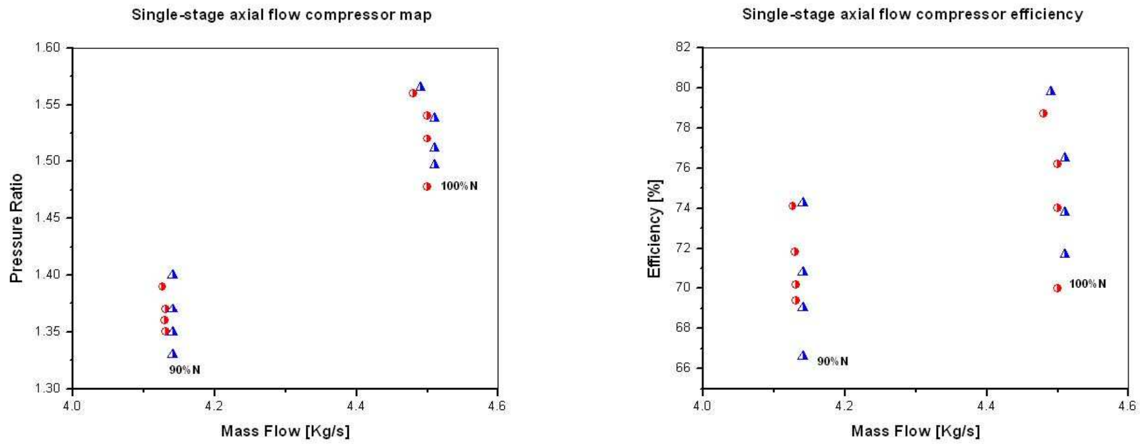

Figure 8. Variation on compressor pressure ratio and efficiency for different rotational speed and tip/hub clearances

For each rotational speed four points were calculated. The machine time for each operation point was about 9 hours in a P4 2.4GHz with 2GB RAM.

8. COMMENTS AND CONCLUSIONS

The tip and hub clearances influence compressor pressure ratio and efficiency, as shown in Figs. 8 and 9. Therefore, investigation of acceptable clearances is necessary during compressor design. The procedure described in this work can be used for that purpose.

Since the compressor used in this work is of high performance, the speed lines are nearly vertical, suggesting the need of variable geometry to handle large flow variation. The same procedure may be used for the variable geometry study.

9. ACKNOWLEDGEMENTS

The authors thank FINEP (Financiadora de Estudos e Projetos), CNPq (Conselho Nacional de Desenvolvimento Científico e Tecnológico) and Termomacaé (Petrobrás), for the support that made this reaserch possible through projects developed at the Center for Reference on Gas Turbines (ITA).

10. REFERENCES

Barbosa, J.R., 1987, "A Streamline Curvature Computer Programme for Performance Prediction of Axial Flow Compressors", PhD Thesis, Cranfield Institute of Technology, England.

Belardini, E., 2003, "State of the Art in Rotor-Stator Interaction for Turbines", QNET-CFD, Vol. 2, Nř 3, Dec 2003.

Camp, T.R., Day, I.J., 1997, "A study of Spike and Modal Stall Phenomena in a Low-Speed Axial Compressor", ASME Journal of Turbomachinery, Vol. 120, pp. 393-401.

Tan, C.S., 2006, "Three-Dimensional and Tip Clearance Flows in Compressors", von Karman LS 2006-06, Brussels.

Cumpsty, N.A., 1989, "Compressor Aerodynamics", Longman Group UK Limited, London.

Day, I.J., 1993, "Stall Inception in Axial Flow Compressors", ASME Journal of Turbomachinery, Vol. 115, pp. 1-9.

Denton, J.D., 1993, "Loss Mechanisms in Turbomachines", ASME Journal of Turbomachines, Vol. 115, pp. 621-650.

Jameson, A., Schmidt, W. and Turkel, E., 1981, "Numerical Solutions of the Euler Equations by Finite Volume Methods Using Runge-Kutta Time-Step Schemes", AIAA Fluid and Plasma Dynamics Conference, Pablo Alto, California.

Koch, C.C. and Smith, L.H., 1976, "Loss Sources and Magnetudes in Axial Compressors", ASME Journal of Engineering and Power, 98, pp. 411-424.

Moore, F.K. and Greitzer, E.M., 1986, "A Theory of Post-Stall Transients in Axial Compression Systems: Part I - Development of Equations", Transactions of the ASME, Vol. 108, January 1986, pp. 68-76.

Spalart, P. and Allmaras, S., 1992, "A One-Equation Turbulence Model for Aerodynamics Flows", AIAA-92-0439, Aerospace Sciences Meeting & Exihbit.

Tomita, J.T., 2003, "Numerical Simulation of Axial Flow Compressors", MSc Thesis, ITA, Brazil.

Tomita, J.T. and Barbosa, J.R., 2004, "Design and Analysis of an Axial Flow COmpressor for a 1MW Gas Turbine", 10th Brazilian Congress of Thermal Sciences and Engineering-ENCIT, CIT04–0062.

Vo, H.D., 2001, "Role of Tip Clearance Flow on Compressor Stability", PhD Thesis, MIT, USA.

Wu, C.H., 1951, "Survey of Available Information of Internal Flow Losses Through Axial Turbomachines", NACA RM E50J13, January 1951.