Proceedings of the 10th Brazilian Congress of Thermal Sciences and Engineering -- ENCIT 2004
Braz. Soc. of Mechanical Sciences and Engineering -- ABCM, Rio de Janeiro, Brazil, Nov. 29 -- Dec. 03, 2004

Paper CIT04-0062

Design and Analysis of an Axial Flow Compressor for a 1 MW Gas Turbine

Jesuino Takachi Tomita
Instituto Tecnológico de Aeronáutica – ITA
Gas Turbine Group
jtakachi@mec.ita.cta.br

João Roberto Barbosa
Instituto Tecnológico de Aeronáutica – ITA
Gas Turbine Group
barbosa@mec.ita.cta.br

Abstract. This work deals with the conception of an axial flow compressor that would be part of a gas turbine to be designed, aiming at 1 MW power output at design conditions. Number of stages, overall dimensions and blading is determined using mean line and streamline curvature techniques. Off-design operation conditions are used to both determine the bleed and stators repositioning needs aiming at good part load performance. Compressor maps are generated and the curve shapes analyzed. The compressor is also analyzed as part of gas turbine. The operating line is calculated and the final bleeds and variable geometry of the stators are set.

Keywords*. Axial Compressor, Variable Geometry, Gas Turbine, Propulsion*

1. Introduction

The axial compressor is the most used compressor in aero and industrial gas turbines, due to its high efficiency, high mass flow per unit frontal area, high pressure ratio, simple mechanical design, so as to reduce manufacturing time and cost and high reliability. Nevertheless, the aerodynamic and thermodynamic design process of axial compressor is not simple, mainly at off-design conditions (Jansen and Moffatt, 1966) where the operation range is small.

Gas turbine engine requires a high efficiency turbomachinery, compressor and turbine. The function of a compressor is to increase the stagnation pressure of the gas stream to that required by the cycle while absorbing the minimum shaft power from turbine, see Fig. (1) from Saravanamuttoo et al. (2001).

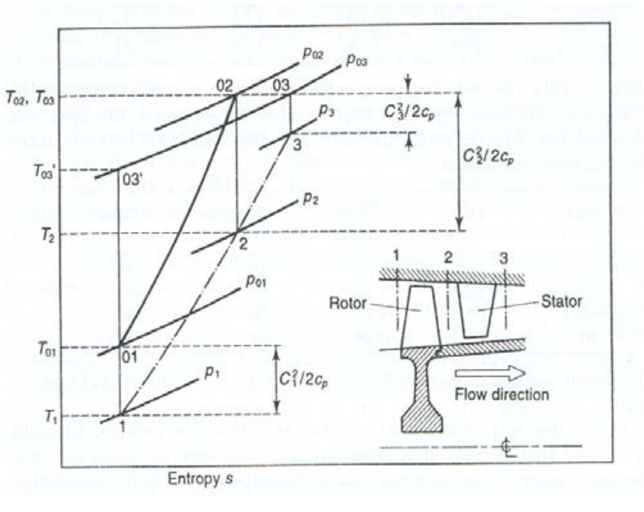

Figure 1. A single compressor stage represented by Temperature vs Entropy diagram.

The compressor power input is given by Eq. (1):

$$\dot{W} = \dot{m} c_p (T_{02} - T_{01}) \tag{1}$$

where, $\dot{W}$ is the compressor power; $\dot{m}$ is the mass flow rate; c_p is the specific heat at constant pressure and $\left(T_{02} - T_{01}\right)$ is the difference of outlet and inlet stagnation temperatures.

Another important point in the axial compressor design is the thermodynamics involved in a single stage. The flow in blade passages, rotor and stator, undergoes the following thermodynamic process: rotor increases the static pressure, stagnation pressure, static temperature, total temperature, absolute velocity, enthalpy and density and decreases the relative velocity along the blade chord; stator increase the static pressure, static temperature and density and decrease the stagnation pressure and absolute velocity, maintaining constant the stagnation temperature.

In this work, the methodology applied considers the streamline at blade mid height. The design and performance analyses of axial compressor are calculated with a computer code named AFCC (Axial Flow Compressor Code), (Tomita, 2003). The compressor map may be generated to create the compressor operation map, using variable geometry in the stators, and bleed. The case study in this work refers to an axial compressor used in a 5 kN turbojet and in a gas generator of the 1MW turboshaft. Detailed design uses, the streamline curvature method (Barbosa, 1987).

2. Design and Off-Design Performance

For the axial flow compressor design, the parameters used are: inlet stagnation temperature and pressure, pressure ratio, efficiency, inlet and outlet Mach number, number of stages, hub-tip ratio, shape of axial channel (constant outer diameter, constant inner diameter, variable mean diameter, constant mean diameter), mass flow, blade tip speed or rotational speed. High efficiency is necessary since the gas turbine thrust and its efficiency increase with the pressure ratio from high efficiency and performance compressor and turbine inlet temperature. High mass flow capacity is also important compressor attribute because the maximum frontal area of the engine is controlled by compressor diameter, thus high flow capacity is required to reduce drag force. The blades of a high performance axial compressor must tolerate high relative Mach number so that high flow capacity and high relative speeds can be realized. The losses (Denton, 1993; Miller and Wasdell, 1987; Wu, 1951) are the most important factor to be considered for compressor design. The design methodology and loss models are very important for the accuracy and reliability of the results.

The compressor must also be able to provide suitable performance at other operating conditions (off-design) imposed upon it. Therefore the compressor needs to provide satisfactory performance over a range of speeds, pressure ratios and mass flow during all the operating conditions.

The analysis of the off-design conditions requires investigation of the matching of the compressor to the other engine components. The losses at off-design operation (Carter, 1949; Carter and Hughes, 1948) are higher than at design point. Variable geometry in the stator blades and flow bleed are the used to improve efficiency at off-design conditions (Tomita, 2003; Tomita and Barbosa; Bobula and Soeder, 1983; Serovy, 1968).

To produce a compressor map prior to its fabrication is a must during the gas turbine design. It will provide insight into the compressor and into the engine behavior, thus allowing unwanted behavior during engine operation.

3. The Axial Compressor Design and Performance

In this work, an axial compressor is designed and its performance calculated using the AFCC program.

The compressor would equip the gas generator of a turbojet and of a turboshaft. The design parameters are:

- Inlet stagnation pressure: 101.325 kPa;
- Inlet stagnation temperature: 288.15 K;
- Inlet Mach number: 0.50;
- Outlet Mach number: 0.26;
- Pressure ratio: 5.0;
- Polytropic efficiency: 89 %;
- Number of stages: 5;
- Tip speed: 355 m/s;
- Inlet hub-tip-ratio: 0.40;
- Axial channel: Constant Outer Diameter (COD).

Other assumptions were made based on literature data (Saravanamuttoo et al., 2001; Walsh and Fletcher, 1998; Mattingly, 1996). The loss model is based on the model used by (Barbosa, 1987). The program has 2 modules: design and performance analysis. The performance analysis module may accept the geometry calculated by another design program. Initially a simple design, without variable geometry and bleed was carried out, aiming at low cost and weight.

The calculated channel is sketched in Fig. (2).

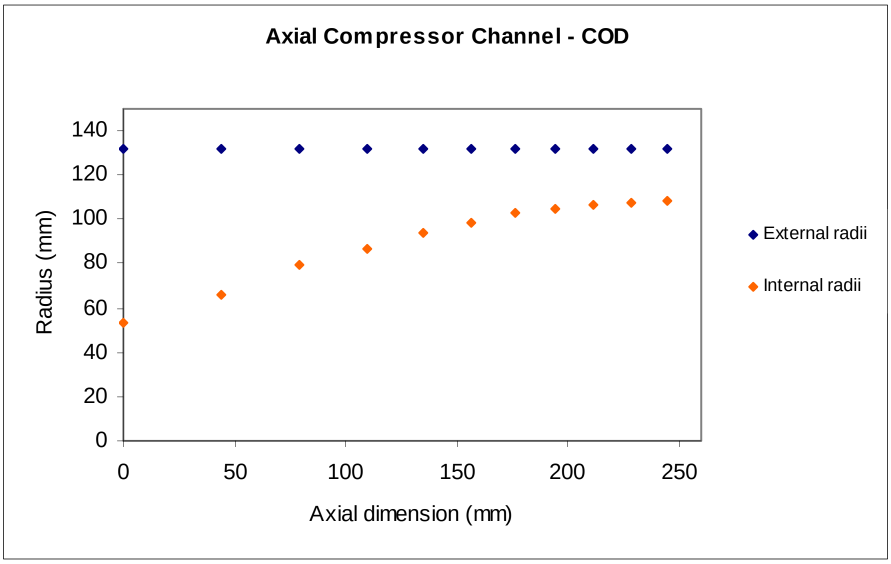

Figure 2. Axial compressor channel.

The stagnation temperature distribution for each stage and the relative Mach number are presented in Figs. (3) and (4).

Figure (3) presents the stagnation temperature increase for each stage, around 45 K.

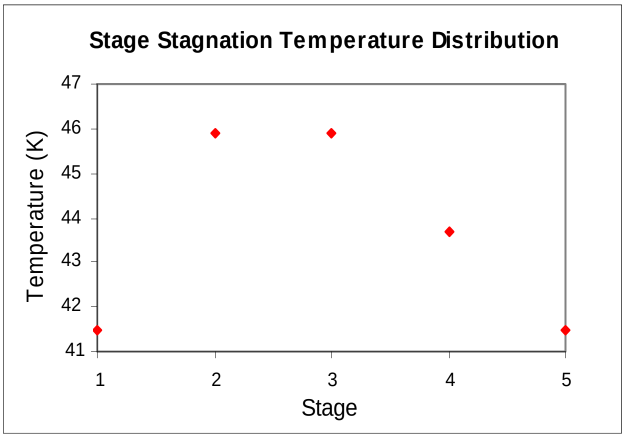

Figure 3. Stage temperature distribution.

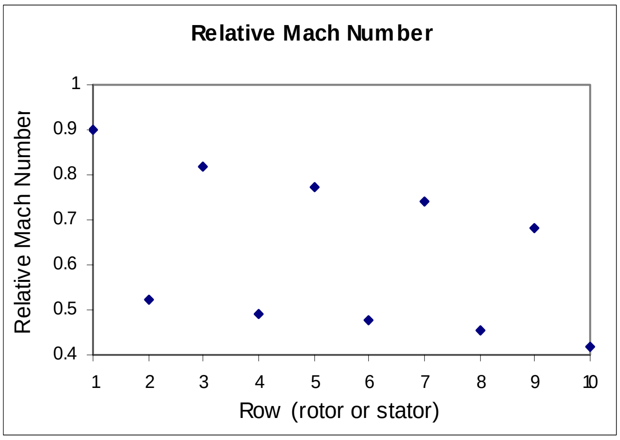

Figure 4. Relative Mach numbers.

Figure (4) shows the relative Mach number decreasing along the channel. The odd numbers in the absciss axis refer to the rotor rows and even numbers refer to the stator rows.

Curves of constant speed were calculated. At off-design the losses increase mainly due high loading.

At design it is considered that the rotational speed is 100%. As the rotational speed decreases the operation range of the compressor decreases, consequently high flow losses are verified to these low velocities.

The stall incidence angle i_{stall} shown on Fig. (5) is the incidence at which the losses are twice the minimum loss ($\omega_{p_{\min}}$). Deflection (ε) is the difference between the inlet and outlet relative flow angle Eq. (2). Maximum deflection is limited by stall.

$$\varepsilon = \alpha_1 - \alpha_2 \tag{2}$$

where, ε is the deflection angle and α stands for inlet and outlet relative flow angle.

Compressor choke is caused by high speed flow and is attained when Mach number at blade throat is close of 1.0.

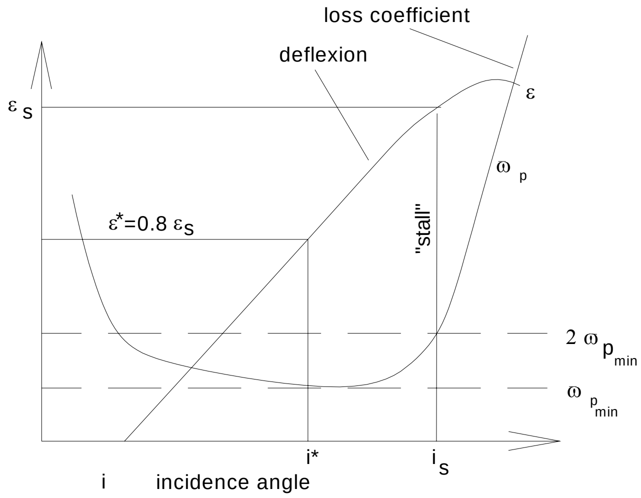

Figure 5. Incidence stall angle.

The incidence stall can be calculated using the curves used by Tomita (2003):

$$i_{stall} = f(\alpha_{2opt}; Clv_{opt}; \beta_1; \beta_2; i_{opt}; \theta; \delta_{opt}; \frac{s}{c}; \zeta) \quad (3)$$

where, α_{2opt} is the optimum relative outlet flow angle; Clv_{opt} is the optimum lift coefficient; β is the inlet and outlet blade angle; i_{opt} is the optimum incidence angle; θ is the camber; δ_{opt} is the optimum deviation angle (Carter´s rule); $\frac{s}{c}$ is the blade space-chord ratio; ζ is the blade stagger angle.

The incidence and deviation are calculated by Eq. (4) and (5):

$$i = \alpha_1 - \beta_1 \quad (4)$$

where, i is the incidence; α_1 is the relative inlet flow angle at the rotor or stator and β_1 is the inlet (rotor or stator) blade angle (leading edge).

$$\delta = \alpha_2 - \beta_2 \quad (5)$$

where, δ is the deviation; α_2 is the relative outlet flow angle at the rotor or stator and β_2 is the outlet (rotor or stator) blade angle (trailing edge).

Note that at the stator blades the relative flow coincides with the absolute flow.

To analyze if the incidence, deviation and losses are acceptable, the minimum loss criteria, from NASA SP-36 (Johnsen and Bullock, 1956), were implemented in the AFCC program. One important remark is that the minimum loss angles are not adequate for the nominal incidence or deviation due to high losses at low flows.

$$i_{ml} = K_i (i_0)_{10} + n\theta + (i_D - i_{2D}) \quad (6)$$

To account for high speed flows, if the relative Mach number is greater than unity, Eq. (6) is corrected as indicated in Eq. (7):

$$i_{ml}^{*} = i_{ml} + 0.5\left\{ a\tan\left[\frac{4\left(\frac{t}{c}\right)}{\theta} sen\left(\frac{\theta}{2}\right)\right]\right\}\left[1 + sen\left(\frac{\pi}{2}\frac{M_{n\,relative} - Mn}{1 - M_{n\,relative}}\right)\right] \quad (7)$$

The minimum loss deviation is:

$$\delta_{ml} = K_\delta (\delta_0)_{10} + m\sigma^b \theta + (i_c - i_{2D}) \left(\frac{d\delta}{di} \right)_{2D} + (\delta_c - \delta_{2D}) \tag{8}$$

where, *ml* stands for minimum loss condition; K_i is the correction factor; $(i_0)_{10}$ is the zero-camber incidence angle; *n* is the slope factor; $(i_D - i_{2D})$ is the deduced variation of average rotor reference incidence angle minus low speed two dimensional cascade rule reference incidence angle with relative Mach number; $\frac{t}{c}$ is the thickness-chord ratio; σ is the solidity; K_δ is the thickness correction for zero camber deviation angle; *m* is the slope factor for minimum loss deviation; the exponent *b* is function of inlet air-angle; $(\delta_0)_{10}$ is the zero-camber deviation angle at reference minimum loss incidence angle deduced from low speed cascade data for 10 percent thick and $\left(\frac{d\delta}{di} \right)_{2D}$ is the slope at reference incidence.

Figures (6) and (7) show the results calculated by AFCC. From Figs. (8) and (9), a comparison between nominal and minimum diffusion factor and the degree of reaction is straightforward.

The diffusion factor expresses the local diffusion at the blade suction surface in terms of velocities or angles. The limiting value for this factor at the design point is 0.45. For high values of diffusion factor around of 0.65 the compressor row (rotor or stator) has a premature stall and for small values around of 0.40 the boundary layer at the blade can be a problem followed by flow separation. These limits are only approximate. A complex design study can be necessary to know how each stage will behave. In the present design, the axial compressor diffusion factor is high in the last stage, acceptable when compared with the minimum loss diffusion factor.

The degree of reaction is defined as the relation between increase of static enthalpy at the rotor divided by the increase of total enthalpy at the stage. At the design point a good value for this parameter is around of 0.50.

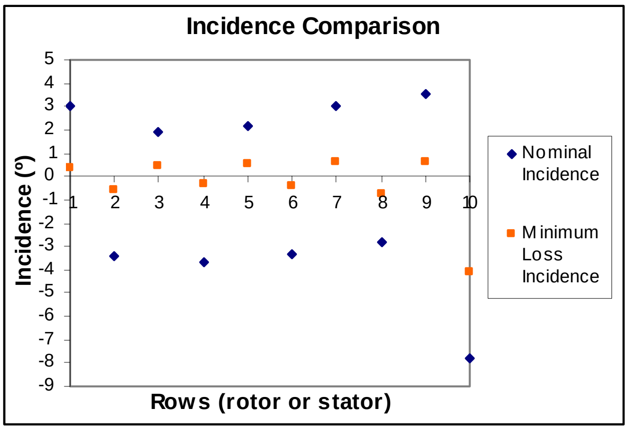

Figure 6. Nominal and minimum loss.

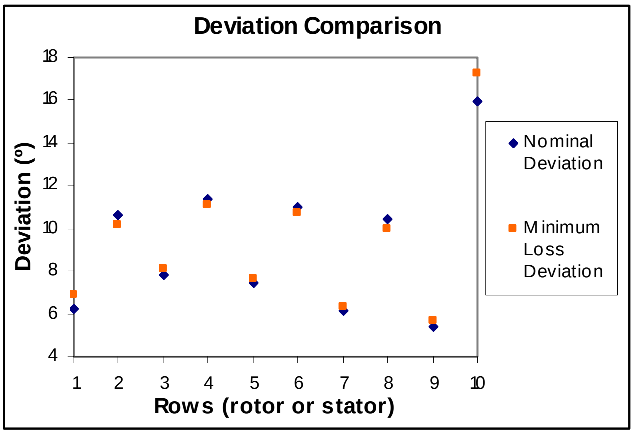

Figure 7. Nominal and minimum loss.

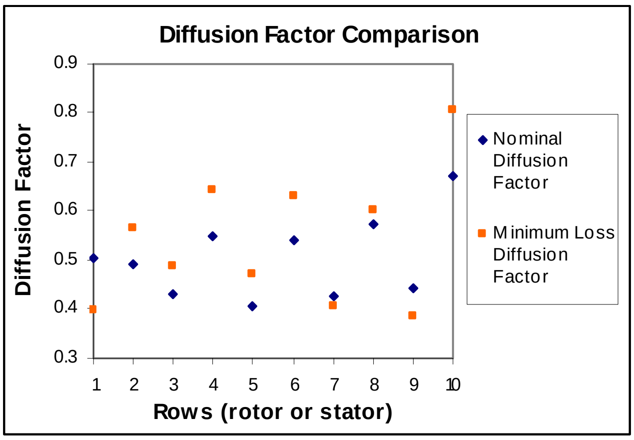

Figure 8. Nominal and minimum loss diffusion.

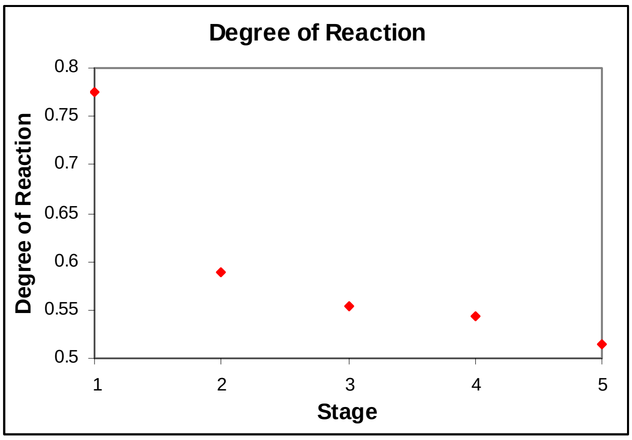

Figure 9. Degree of reaction.

After the compressor has been designed, its performance was calculated at different rotational speeds. The design rotation speed is 25650 rpm. Figure (10) shows the performance curves (map) of the compressor designed using the AFCC computer program:

- Constant speed lines: $\dfrac{N}{\sqrt{T_{t0}}}$;
- Corrected mass flow: $\dot{m}\dfrac{\sqrt{T_{t0}}}{P_{t0}}$;

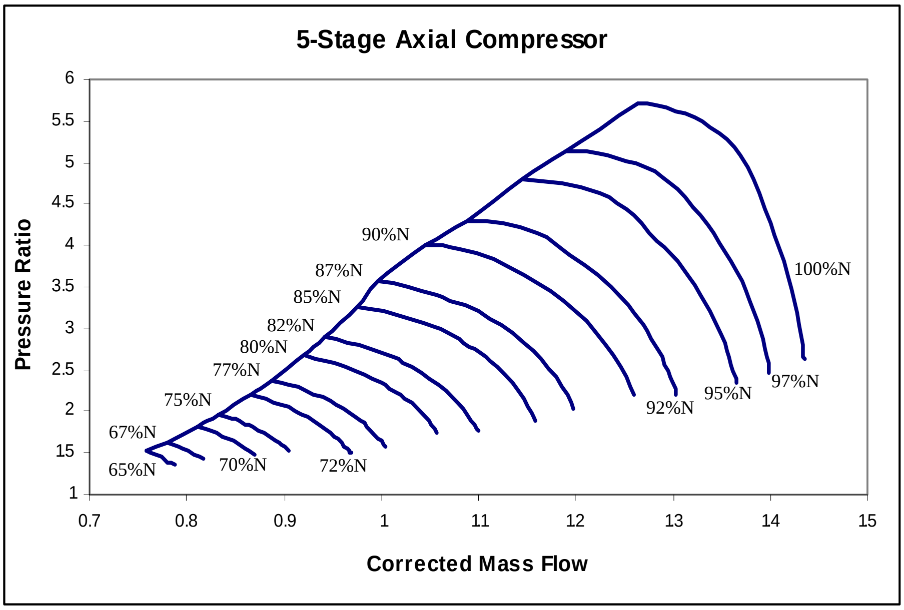

Figure 10. The 5 stage axial flow compressor map operation.

At low speeds stall and choke limits are close. To increase the surge line (stability) and to increase the operation ranges it is common to use variable geometry. Figure (11) shows the result of adding variable Inlet Guide Vane (IGV): efficiency improvement at part-load operation. The changes are satisfactory in terms of stability, mainly at low speeds variable IGV improves the operation range and efficiency as indicated in Fig. (12).

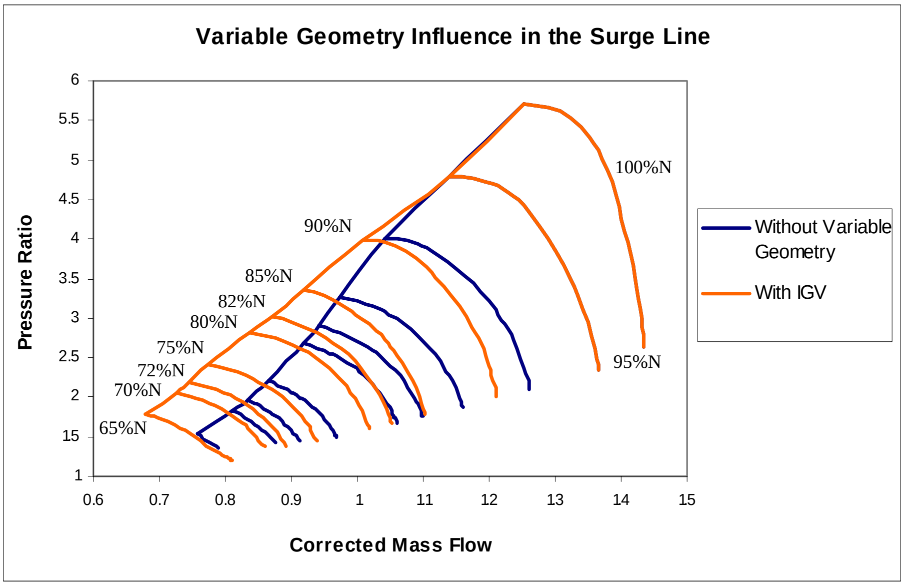

Figure 11. Axial flow compressor with variable geometry.

For the 5-stage compressor under study, only variable IGV was considered. Variable stators may be also installed aiming at moving upwards the surge line. The stators equipped with variable geometry are more expensive and heavier due to the complexity of the additional systems. Only the frontal stages require variable stators.

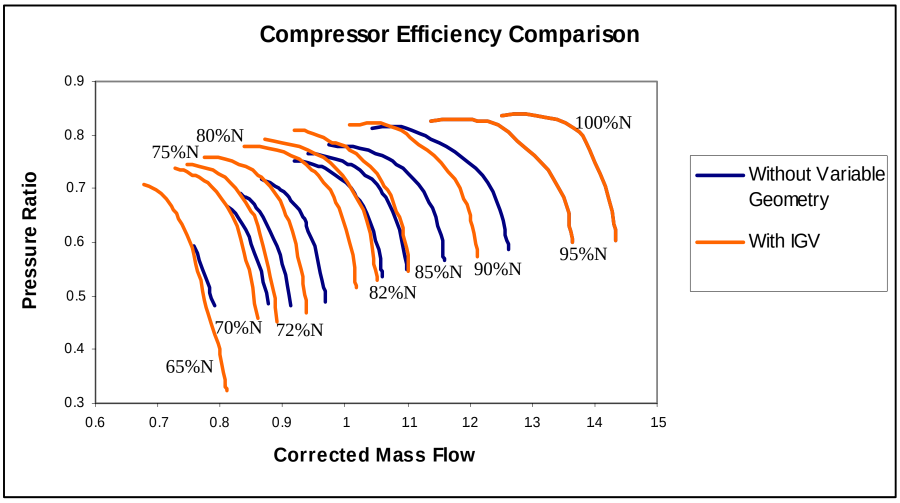

Figure 12. Axial compressor efficiency improvement using IGV.

The overall loss analysis shows that at low speeds the losses increase due to bad matching, as indicated in Fig. (13). The model used to calculate these losses depends on several parameters as indicated by Eq.(9):

$$loss = f(C_d; M_{n0}; M_{n1}; M_{nc\,opt}; \alpha_1; \alpha_2; i; \varepsilon) \tag{9}$$

where, C_d is the drag coefficient; M_{n0} is the inlet absolute Mach number at the blade; M_{n1} is the inlet relative Mach number at the blade; $M_{nc\,opt}$ is the optimum Mach number at the blade inlet; α_1 and α_2 is the inlet and outlet relative flow angle, respectively; i is the incidence angle and ε is the deflection. The calculations follow the procedures given by Tomita (2003).

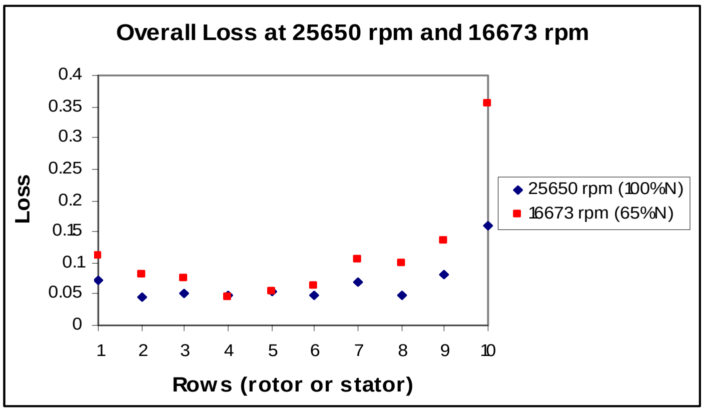

Figure 13. Compressor loss analysis.

An straightforward application of this design procedures was to produce compressor maps that were supplied as data input for the GTAnalysis (Bringhenti, 2003; Bringhenti and Barbosa, 2004) computer code, for the calculation of the gas turbine performance.

4. Conclusion

A technique to design a multi-stage axial flow compressor and evaluate its performance based on channel mid-height flow (Casey, 1987) was used. The compressor map has been calculated and major foreseeable problems analyzed. After the calculation of a map that would satisfy the application in an actual engine, detailed compressor design may be carried out. Next step is to use the streamline curvature program (Barbosa, 1987) to check for the flow at other streamlines, in addition to the mid-height.

5. Acknowledgement

The authors thank CNPq (Conselho Nacional de Pesquisa) for the support to this research.

6. References

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

Bobula, G.A., Soeder, R.H., Burkardt, "Effect of Variable Guide Vanes on the Performance of a High-Bypass Turbofan Engine". Journal of Aircraft, Vol. 20, nº 4, April 1983, p. 306 – 311.

Bringhenti. C., "Variable Geometry Gas Turbine Performance Analysis". Thesis PhD, ITA – 2003.

Bringhenti, C. and Barbosa, J. R., "Performance Study of a !MW Gas Turbine", 2004, ENCIT Paper CIT04-0089.

Carter, A.D.S., "The Low Speed Performance of Related Aerofoils in Cascade" – NGTE, R.55 September 1949.

Carter, A.D.S. and Hughes, Hazel P., "A Note on the High Speed Performance of Compressor Cascades" – NGTE, M.42 December 1948.

Casey, M.V., A Mean-Line Prediction Method for Estimating the Performance Characteristic of an Axial Compressor Stage. Sulzer, Switzerland, C264/87.

Denton, J. D., "Loss Mechanisms in Turbomachines". IGTI 1993, Journal of Turbomachinery vol. 115/621.

Mattingly, J. D., "Elements of Gas Turbine Propulsion". McGrawHill, 1996.

Miller, D.C. and Wasdell D. L., "Off-Design Prediction of Compressor Blade Losses". Rolls-Royce, Bristol C279/87. March 1987.

Jansen W. and Moffatt W. C., "The Off-Design Analysis of Axial-Flow Compressors". ASME Paper nº 66 – WA/GT – 1 1966.

Johnsen, I.A. and Bullock, R.O., "Aerodynamic Design of Axial Flow Compressors". NASA SP-36, 1956.

Saravanamuttoo, H.I.H., Rogers, G.F.C., Cohen, H., "Gas Turbine Theory". Longman, 2001.

Serovy, G.K., Kavanagh, P., "Considerations in the Design of Variable Geometry Blading for Axial-Flow Compressor Stages". AGARD CP – 34, paper nº 10. September 1968.

Tomita, J.T., "Numerical Simulation of Axial Flow Compressors". Tese MC, ITA – Abril 2003.

Tomita, J.T. and Barbosa, J.R., "A Model for Numerical Simulation of Variable Stator Axial Flow Compressors". 17º COBEM – Congresso Brasileiro de Engenharia Mecânica, 10-14/11/2003, São Paulo, SP, ID-0239.

Walsh, P.P., Fletcher, P., "Gas Turbine Performance". Blackwell Science, 1998.

Wu, C.H., "Survey of Available Information on Internal Flow Losses Through Axial Turbomachines". NACA RME50J13, January 1951.

7. Copyright Notice