SURGING TO CHOKING: A STUDY OF THE AXIAL CHANNEL OF AXIAL COMPRESSORS

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Abstract. *The importance of a well designed axial channel is discussed. High performance axial compressors characteristics are shown. Different axial channel shapes are compared due their capability of reaching higher values of pressure ratios. Surging and choking, two important parameters in the design of axial compressors, are described and discussed. A 5-stage axial compressor is studied; surging and choking conditions are determined through a proper algorithm and computational program whose output data make possible the assembly of $\alpha_1 \times M_1$ diagrams, important for the mass flow range determination.*

Keywords: *gas turbine, axial compressors, surge, stall, choke.*

1. INTRODUCTION

The use of high performance axial flow compressors is growing, especially in aero gas turbines. Those gas turbines operate under several regimes, requiring great operation flexibility of their compressors to accommodate air mass flows compatible with the flying conditions and the aircraft manoeuvres, and also, as an additional requirement, the efficient operation in a large range of air mass flow at all rotational speeds, requiring complex aerodynamic design, usually imposing the use of variable geometry in some stages.

The designer concern, since the first design stage, is the axial channel. If it is not well designed, it may make impossible the compressor development on test facilities, when fine adjustments are made to reach the design requirements. Along with the design of the blades and the passages between blades, the study of the axial channel needs to be detailed enough to predict with precision, the range of air flow in each rotational speed, the corresponding pressure ratios and efficiencies.

This work addresses two important aspects during the design of a high performance axial compressor for use in gas turbines: the numerical simulation of surge and choke at each compressor speed above idle.

2. HIGH PERFORMANCE AXIAL COMPRESSORS CHARACTERISTICS

Figure 1 (Tomita, 2003) shows an axial compressor of several stages. In it are indicated the rotors and stators, and the inlet director grid, usually known as IGV (inlet guide vanes). A standard stage of compression of axial flow compressors cannot raise enough the pressure, which requires several stages of compression, for large pressure discharges. Compressors with over ten stages of compression are common in gas turbines.

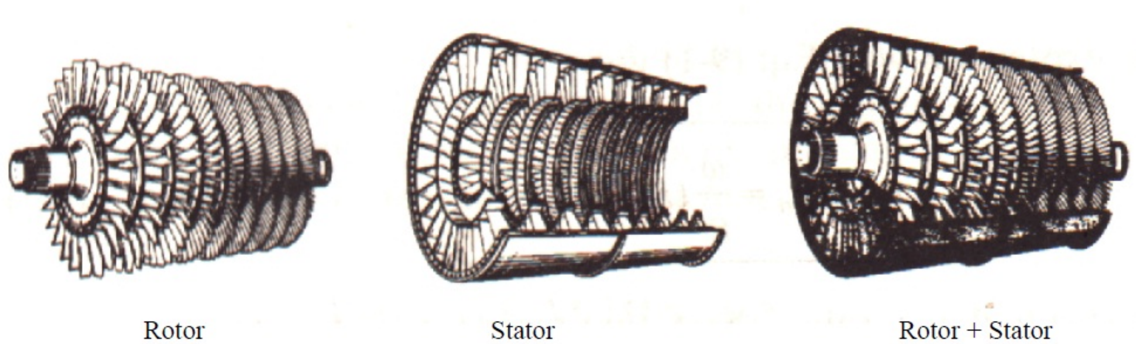

Figure 1. Example of high performance multi-stage axial compressor.

The characteristics of operation of axial compressors usually are graphs of pressure ratios and efficiencies against the mass flow, for each rotational speed. To take into account the compressibility effects, those variables are corrected (corrected mass flow and corrected rotational speed) or corrected and normalized with the design conditions.

Figure 2 (Bringhenti,Tomita and Barbosa, 2005) shows the most used form of performance maps of axial compressors. In it appear some constant corrected rotational speed curves and the corresponding efficiency curves. Usually the range of rotational speed is from the idle to the design rotational speed. Lower rotational speeds usually are presented in separated graphs, because they are not interesting during the operation of a compressor being part of a gas turbine.

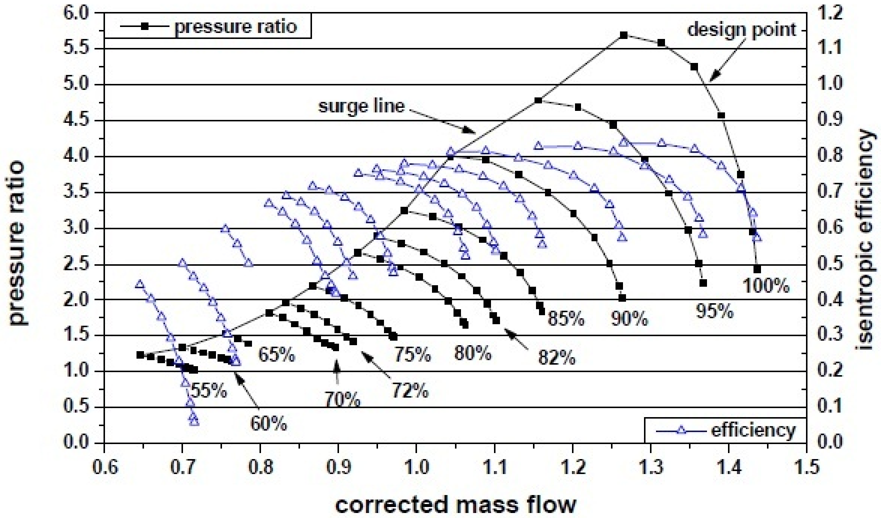

Figure 2. Compressor performance map.

3. AXIAL CHANNEL SHAPE

The shape of the axial channel results from the layout of the gas turbine in which it is installed. In general, given the requirement of compactness, the stage design requires a high transfer rate of specific work to the air, quantified by Eq. (1).

$$W = U(V_{1u} - V_{2u}) \tag{1}$$

where the indices 1 and 2 indicate, respectively, rotor inlet and outlet and u the projection of the absolute velocity on the tangential velocity U direction and V the flow absolute velocity. It is common the use of a velocity diagram, as shown in Fig. 3 (Saravanamuttoo, Rogers and Cohen, 2001).

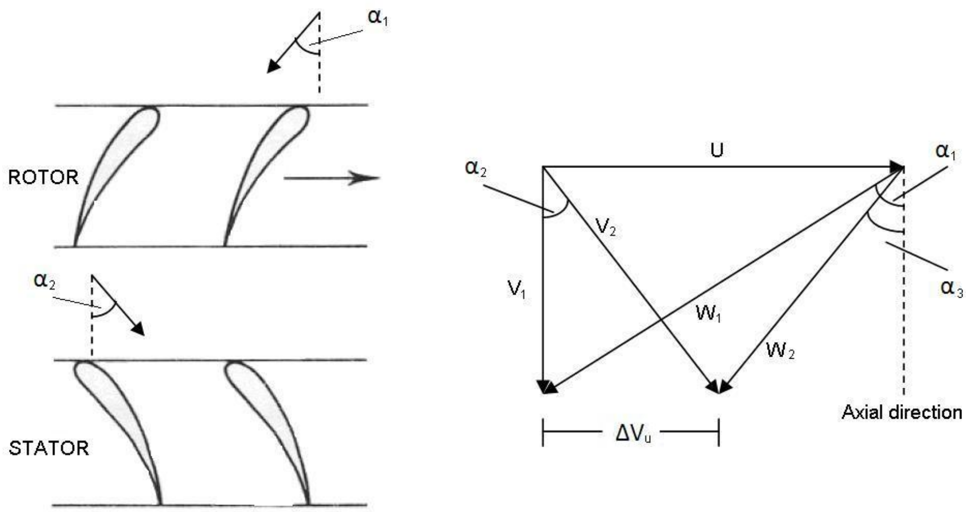

Figure 3. Velocity diagram.

The analyses of Eq. (1), along with the velocity diagram, indicate the large values of specific work are obtained with large values of peripheral velocity U and/or large values of the difference between the projections of the absolute velocities, at rotor inlet and outlet, on the U direction, i.e., $V_{1u} - V_{2u}$. Moreover, in axial machines the rotor inlet and outlet peripheral velocities are not so different. Today, due material limitations of what compressors are manufactured, U is upper limited (about 450 m/s). As the several rotors of a compressor are on the same shaft and with the same rotational speed, usually it is preferred to work with the maximum value of U possible. From this choice, the outer diameter of the several rotors is constant, resulting on the type of axial channel showed in Fig. 4 (Saravanamuttoo, Rogers and Cohen, 2001).

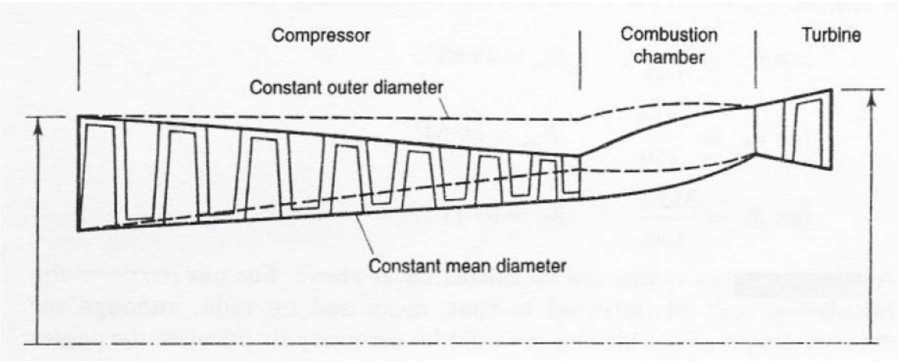

Figure 4. Axial channel shape.

In some cases, to simplify the design of the disks on which the rotor blades are fixed, the inner diameter is fixed even if they lose capacity to transfer energy to the fluid due to the lower value of U. There are designs that require that the rotor blade mean height be constant (Fig. 4) and others that the rotor blade mean height be variable.

Clearly, the compressor with axial channel with constant outer diameter is the most compact, because its stages can develop the greatest pressure ratios.

Whatever the type of axial channel chosen, one must ensure that the shape of the channel does not impose barriers to air flow under conditions of unfavorable gradients, favoring the boundary layer separation and the consequent loss of performance.

Thus, during the phase of design of the several stages of compression, attention is required because, both the surfaces of the blade hub and case must facilitate the progression of the flow, not causing separation. Figure 4 shows an axial channel with the smooth curves of the hub and the case, requirement for good performance.

4. IMPORTANT PARAMETERS IN DESIGN OF AXIAL COMPRESSORS

4.1 The choking concern

Observing the point 1 on Fig. 5 in the design corrected rotational speed curve it is seen that the mass flow is maximum and the pressure ratio is minimum, resulting in low density and high flow speed. The flow velocity increases significantly, to the point of the channel between the blades work choked. Analyzing what happens in the passage between two consecutive blades, it is verified that choke appears as the streamtubes choke. The stream-surface will adapt to the new flow conditions, until the last streamtube becomes choked. Figure 6 shows streamtubes formed by streamlines of a compressor axial channel.

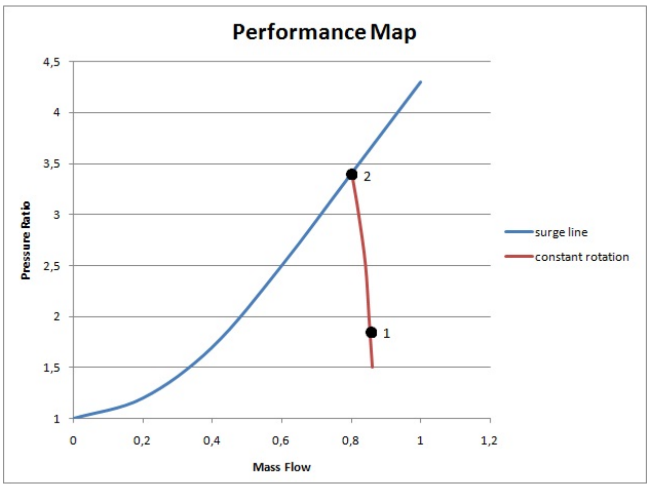

Figure 5. Constant rotational speed curve where the compressor surging and choking positions are indicated.

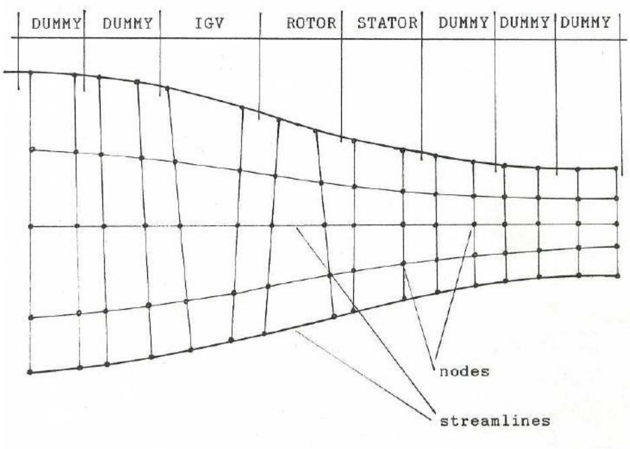

Figure 6. Streamtubes.

These operating conditions result in high losses in the channel, what is not advisable in a compressor working in a gas turbine. Therefore, during the compressor design it is necessary the evaluation, row by row, of the maximum mass flow (choke mass flow). Once detected a choked row, the compressor is also considered choked because that row limits the air flow all over the compressor. If necessary, this row must be redesigned to accommodate increased flow. The process of setting the row to allow the maximum required flow is iterative and requires, indispensably, the calculation of the flow in all the rows of the compressor, since the conditions of flow at the row inlet depends on what happens in the previous row. Among the procedures for determining the blockage (choke) of the flow in the row, Mckenzie's procedure (Mckenzie, 1997) was adopted and is described below.

The inlet flow width, delivering to the throat, is $s\cos(\alpha_1)$ so that the inlet Mach number would equal the throat Mach number when $T_h/s\cos(\alpha_1) = 1$.

$T_h/s\cos(\alpha_1)$ varies only with the working fluid inlet relative angle (α_1) when blade geometry is already fixed and it is usually lower than unity. When $T_h/s\cos(\alpha_1)$ is higher than unity it means that the throat passage is higher than the flow width making choke impossible and an increase of flow speed could cause a supersonic acceleration, as occurs downstream of the throat in a convergent/divergent nozzle.

For $T_h/s\cos(\alpha_1)$ less than unity the variation of α_1 can lead the Mach number to a limit before choke is achieved. This limit is given by the maximum inlet relative Mach number or just choking Mach number (M_{ch}). Figure 7 (Mckenzie, 1997) shows the blade passage throat width and Fig. 8 (Cumpsty, 1998) shows a schematic of choked flow in a blade passage: inlet and outlet flow subsonic.

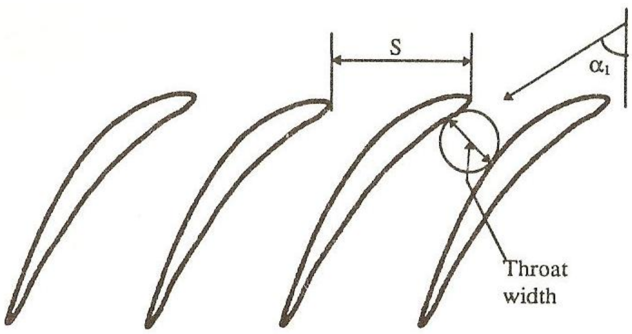

Figure 7. Throat width.

The computational implementation of the adopted procedure requires that all the compressor geometry have already been determined, that is, they must have already been calculated, at the row inlet and outlet: the flow angles, incidences and deviations to determine the blade inlet and outlet angles; the blade aerodynamic profiles, the row stagger angles, the space between blades and the space between rows, and an axial channel well designed. Thus, all information necessary for the mechanical design of the rows. This project also requires a process of optimization of the compressor geometrical parameters and, consequently, enough knowledge of the designer. The design of the compressor is not treated in this work, but the reader can see NASA (1965) and Barbosa (1987) to become aware of issues associated to the design.

After the compressor design process, it is time to determine the off-design characteristics. If the compressor geometry is variable, any changes in the geometry results in off-design operation.

At this stage, the geometry (dimensions, aerodynamic profiles, axial channel, etc.) is fixed and the flow characteristics

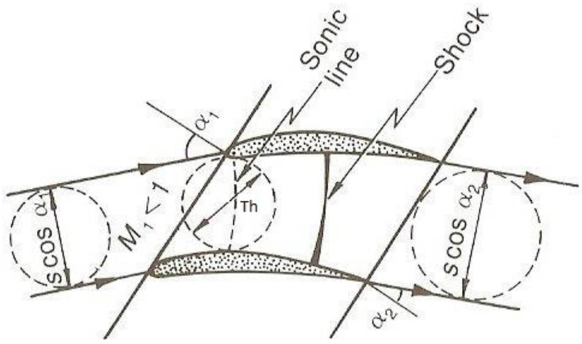

Figure 8. Choked flow.

in the fixed geometry passages are calculated. A new calculation procedure must then be established. Details of this procedure and techniques available can be found in NASA (1965) and Barbosa (1987). In this work, the determination of the compressor choking condition was performed using the following algorithm:

- Start from the design mass flow.
- Increase the mass flow of 1%.
- Evaluate the compressor performance, row by row, for this new mass flow, using off-design point calculation technique.
- Verify if there is some choked streamtube in any of the rows: if more than half of total streamtubes are choked for the same row, the row is considered choked and, consequently, the compressor too.
- Return to the previous mass flow rate, increase the mass flow about 0,1% and then repeat the performance calculation until one row is found choked. The previous item is repeated as many times as necessary to obtain the choking limit, each time dividing the increment in mass by 10, this is, 0,01%, 0,001%, etc.

The mass flow rate being reached with the required precision, the characteristics of the flow for each flow are recorded, especially those related to the inlet flow angles at the rows (α_1) and their respective inlet relative Mach numbers, M_1. Then, $\alpha_1 \times M_1$ graphs can be plotted for each row.

The $\alpha_1 \times M_1$ diagram is very important for the study of compressor performance, because it indicates how near the compressor operation point, installed in a gas turbine, is to the choking point. Similar information is given about the compressor stalling condition. The $\alpha_1 \times M_1$ curves (Fig. 9) (Wall, 1971) can be plotted for each streamline, but the ones referred to the blade mean height streamline are the usually utilized for the compressor performance preliminary analyses.

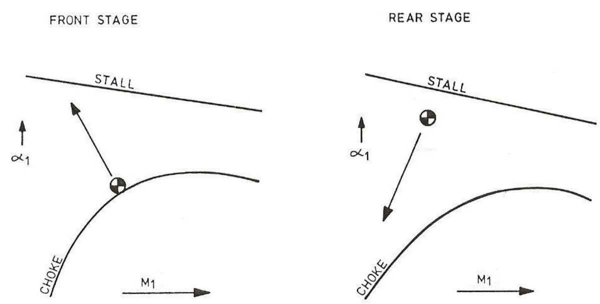

Figure 9. $\alpha_1 \times M_1$ diagram.

4.2 The surging concern

Similarly to what has been shown in the previous item, point 2 in the curve of constant design rotational speed, it is seen that the mass flow is minimum and the pressure ratio is maximum. At this point the flow is already detaching, further decrease of mass flow, resulting on compressor surging, must be avoided. In gas turbines usually the compressor operating point is chosen as close as possible to surge, because the efficiencies are highest. It becomes, thus, important the surge mass flow determination, because the compressor must not operate at this point. Usually a surge margin is fixed. Figure 10 (Bringhenti and Barbosa, 2004) is used for the sake of easing the definition of surge.

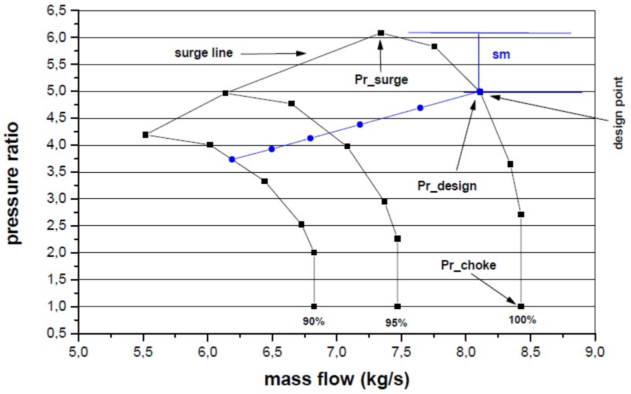

Figure 10. Surge margin indicated on a compressor map.

Surge margin is defined as:

$$sm = \frac{Pr_s - Pr_{DP}}{Pr_s - Pr_{ch}} \tag{2}$$

where P_r is the pressure ratio and the subscripts s, DP and ch indicate, respectively, surge, design point and choke. The fixed value, for design point, of sm depends on the type of application of the gas turbine, usually related to aircraft manoeuvres or loads imposed to the engine.

The surge mass flow determination is done using the same procedure for the choking mass flow determination, but decreasing 1,0%, starting from the design mass flow. The diffusion factor (Eq. (3)) or the equivalent diffusion factor (Eq. (4)) (Aungier, 2003) indicate if stall was reached, according to NASA (1965) and Barbosa (1987). There is strong correlation between surge/stall and values above 2.2 for equivalent diffusion factor or 0.6 for diffusion factor. These factors are defined as:

$$D = \frac{(W_{max} - W_2)}{W_1} \tag{3}$$

$$D_{eq} = \frac{W_{max}}{W_2} \tag{4}$$

The diffusion factor well correlates the effects of the diffusion in the channel between the blades with the flow separation (NASA, 1965) and, consequently, with the compressor stall.

Likewise the choking case, during the design phase it is necessary to verify, with precision, the surge mass flow in all rows. If in any row the diffusion factor, or the equivalent diffusion factor, reaches values equal or greater than 0.6 or 2.2, respectively, it indicates that the flow detached and, so, the compressor reached the stall and further decrease in mass flow will make it reach surge.

Similarly to the one plotted during the choke mass flow determination, the $\alpha_1 \times M_1$ diagram is plotted related to the surge mass flow. The graphs are overlapped, obtaining, for each streamline, the change in the inlet relative flow angle as a function of the inlet relative Mach number. Thus, for each streamline, it can be obtained a graph as exemplified in Fig. 11. Usually the graphs referring to the blade mean height are chosen for preliminary analyses.

The compressor is considered well designed if the operating points are not too close to the limits of surge or choke. If in any row this is not verified, the designer must conveniently change the design of the row channel in question. Usually this procedure is hard-working, because the change of any row geometric parameter influences the flow of other rows, what could approach the operating point of the other rows to point of surge or choke.

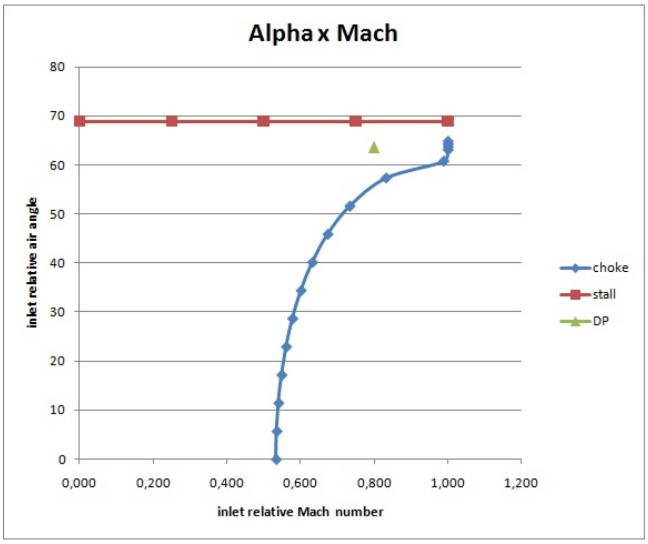

Figure 11. $\alpha_1 \times M_1$ graph for an axial compressor row at the mean height.

5. CASE STUDY

It was studied a 5-stage axial compressor for 8.2 kg/s of air, pressure ratio of 5 and isentropic efficiency of 85%. Figure 12 shows details of the rotor. It is a high performance transonic compressor with DCA blade profiles.

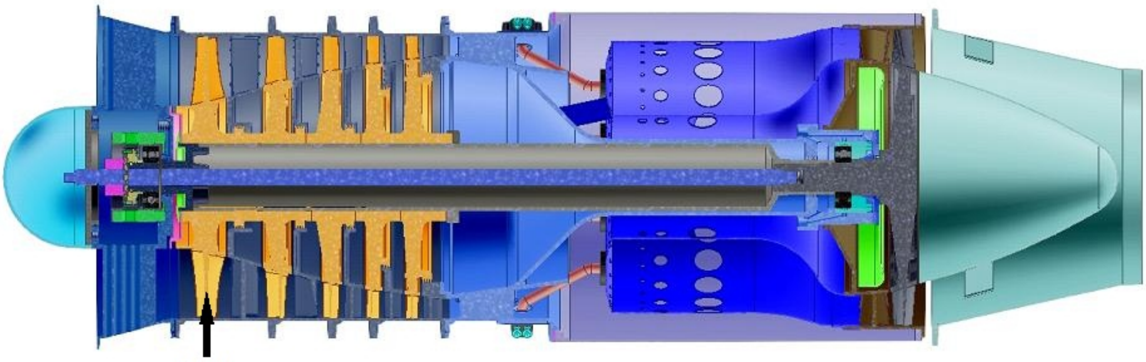

Figure 12. 5-stage axial compressor rotor in a turbojet engine.

The compressor design was done using a computational program of high technological content, developed at ITA. It uses the streamline curvature method. The designer can fix any number of streamlines. In this study, the design was done with 5 streamlines. Table 1 is an extract of the program output results of interest to assemble the $\alpha_1 \times M_1$ diagrams, where i and j are the row and streamline, respectively, and when M_{ch} is "ZERO" means that there is no possibility of choke ($Th > s\cos(\alpha_1)$), as explained before. Notice that the values of diffusion factor have not reached their limits; it means that stall was detected at some other row.

As an example, Fig. 13 to 15 show the graphs $\alpha_1 \times M_1$. The range of operation can change as explained before, and also the operating points can be replaced for reaching some operational characteristic, if required.

Table 1. Output data necessary to the assembly of the $\alpha_1 \times M_1$ diagram.

	i	j	Th	s	α_1	M_{ch}	M_1	D	D_{stall}
DP	4	1	0.013740	0.0252	-52.71	0.83	0.92	0.445	0.6
	4	2	0.019426	0.0355	-60.36	0	1.17	0.499	0.6
	4	3	0.020231	0.0396	-62.97	0	1.28	0.513	0.6
	4	4	0.019981	0.0426	-64.71	0	1.37	0.528	0.6
	4	5	0.020948	0.0453	-66.05	0	1.45	0.543	0.6
STALL	4	1	0.013740	0.0252	-53.09	0.84	0.88	0.503	0.6
	4	2	0.019426	0.0355	-60.73	0	1.14	0.570	0.6
	4	3	0.020231	0.0396	-63.30	0	1.26	0.579	0.6
	4	4	0.019981	0.0426	-65.01	0	1.35	0.587	0.6
	4	5	0.020948	0.0453	-66.34	0	1.42	0.601	0.6
CHOKE	4	1	0.013740	0.0252	-50.90	0.80	1.02	0.464	0.6
	4	2	0.019426	0.0355	-58.80	0	1.26	0.502	0.6
	4	3	0.020231	0.0396	-61.45	0	1.37	0.508	0.6
	4	4	0.019981	0.0426	-63.24	0	1.46	0.514	0.6
	4	5	0.020948	0.0453	-64.63	0	1.53	0.525	0.6

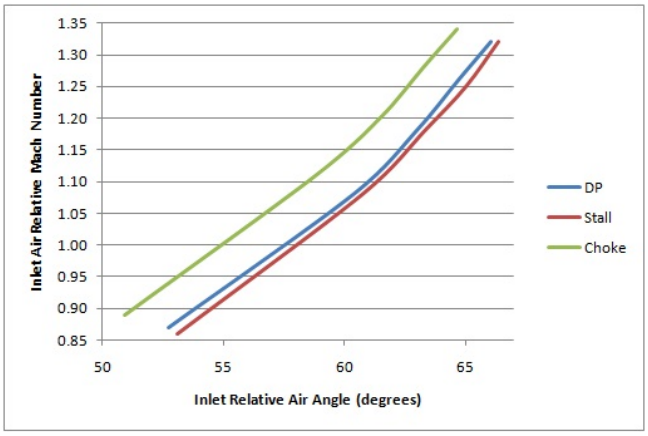

Figure 13. First rotor $\alpha_1 \times M_1$ graph of a 5-stage axial compressor.

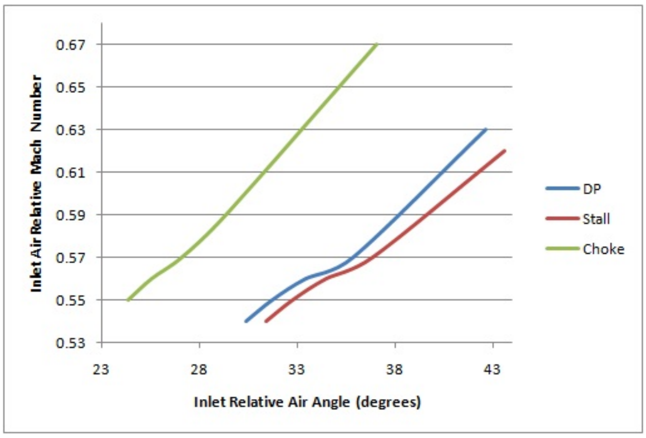

Figure 14. First stator $\alpha_1 \times M_1$ graph of a 5-stage axial compressor.

6. COMMENTS AND CONCLUSIONS

Figure 15 shows that the design point, for this particular streamline, is out of the limits, but as seen in Tab. 1, $M_{ch} = 0$ for the blade mean height (i = 4 and j = 3), so there is no problem, because choke is not possible.

Tables 2 and 3 refer to the detection of stall and choke by the program. Stall was first detected in the tip of the last rotor, and choke was detected at first, second and fourth streamlines of the last rotor. Notice that choke is detected when

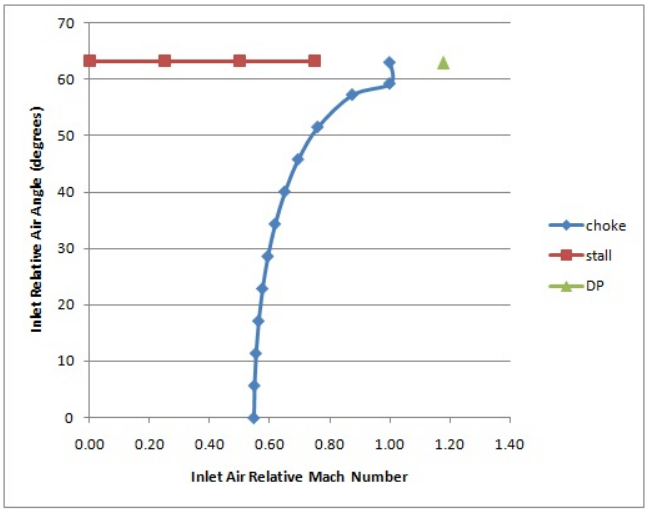

Figure 15. First rotor $\alpha_1 \times M_1$ graph at the blade mean height of a 5-stage axial compressor.

Table 2. Stall detection.

	i	j	D	D_{stall}
STALL	12	1	0.446	0.600
	12	2	0.505	0.600
	12	3	0.543	0.600
	12	4	0.573	0.600
	12	5	0.600	0.600

Table 3. Choke detection.

	i	j	M_{ch}	M_1
CHOKE	12	1	0.74	0.78
	12	2	0.83	0.84
	12	3	0.90	0.88
	12	4	0.91	0.91
	12	5	0	0.93

more than half streamlines are choked and that the fifth streamline can not choke, because $M_{ch} = 0$.

The computational program for high performance axial flow compressor design includes loss models for DCA profile at the design point. Loss models for the study at off-design point were also implemented to simulating off-design mass flows.

The analyses of the obtained results for the inlet flow angles and respective inlet relative Mach numbers indicate that the compressor has good characteristics at off-design point operation.

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8. Responsibility notice