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A MODEL FOR NUMERICAL SIMULATION OF VARIABLE STATOR AXIAL FLOW COMPRESSORS

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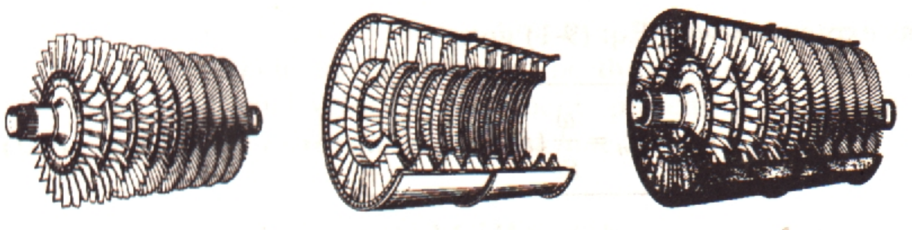
Abstract. *Axial flow compressors are usually found in gas turbines, both for aero and industrial applications due to their higher flow capacity, pressure ratio and efficiency when compared to the centrifugal counterpart. Due to the wide range of ambient conditions at which the gas turbines are required to work, compressor designers have to foresee its performance capability and possible points of malfunction. To explore all points of actual compressor operation it is required extensively and time-consuming tests, in addition to high costs. To cut time and costs, numerical simulations have been employed extensively, through simple to more complex mathematical models. Capability to predict the compressor map is required at the beginning, since is the map characteristics an indication of the success of the designed compressor. At design point compressor efficiency is optimized due to the alignment of flow with blade passage areas. At off-design point the flow misalignment at the various rows causes losses to increase sharply, therefore decreasing pressure ratio and efficiency. To bring the flow to alignment with the blade passages it is required to re-stagger the blades. To avoid mechanical complexities it is generally accepted to re-stagger only the stators. This work deals with a numerical approach to the simulation of an axial flow compressor equipped with variable stators. Improvement to the compressor performance is demonstrated with calculated compressor maps for different blade staggers, using a specially developed computer program to simulate virtually any axial flow compressor having variable stators.*

Keywords*. Compressor, Axial-flow Compressor, Gas Turbine, Variable Geometry, Performance improvement, Part-load.*

1. Introduction

An axial compressor is usually made of many stages each one composed by a rotor and a stator (rotating cascade and fixed cascade, respectively - Figure 1). The air properties at the stage outlet are calculated from the air properties at the stage inlet. Losses are calculated from cascade data (Abbott - 1959) or from data obtained from rig-tested compressors, for fixed geometry (Miller - 1971; Casey - 1987; Carter - 1948/1949; Barbosa - 1987/2001, NASA SP -36 - 1956; Saravanammuttoo - 2001; Mattingly - 1996) and for variable geometry (Walsh - 1998; Bobula - 1983; Serovy - 1968).



Rotor Stator Rotor and stator

Figure 1: Examples or rotor and stator blade rows of a 9-stage axial flow compressor.

The compressor is the component that most strongly influences the gas turbine performance, either for the peculiar characteristic of operation instability or for the high consumption of energy during the air compression

Flow properties at the stage outlet are calculated combining a rotor with a stator, thus whole compressor characteristics are calculated stacking the compressor stages: the inlet conditions of a previous stage are obtained from the outlet conditions of the previous stage. Friction, shock waves, secondary flow and levels of velocities (Mach numbers), influence the stage performance most.

.The high performance compressors, those that promote a high compression per stage, require a flow with high velocity in the blade channels. The flow velocity is associated with energy losses when the flow direction does not match the blade channels direction. Thus, at off-design operation the compressor efficiency can vary significantly, causing high losses and deterioration of the gas turbine performance as result and to decrease the losses in the compressor operation, use is made of variable geometry when the position of the stators (angles) is altered during operation in order to align the blades with the flow (Figure 2). Setting stator blades to a new position is equivalent to having a new compressor. Therefore, the same methodology developed for the compressor design and analysis would be enough for the evaluation of the compressor with varying geometry.

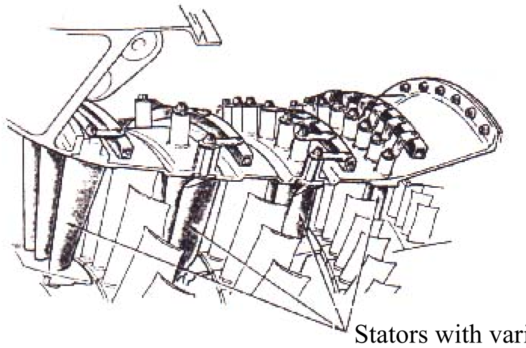


Figure 2: Compressor with variable geometry.

A schematic drawing of an axial flow compressor as installed in a turbojet engine is shown in Figure 3. In the general configuration, the first row of blades, the IGV (Inlet Guide Vanes), imparts a rotation to the air to establish a specified velocity distribution ahead of the first rotor. The pre-rotation of the air is then changed in the first rotor, and energy is thereby added in accordance with Euler's equation. This energy is manifested as increases in total temperature and total pressure of the air leaving the rotor. Usually accompanying these increases are increases in static pressure and in absolute velocity of the air. A part, or all, of the rotation is then removed in the next stator, thus converting velocity head to static pressure and this stator sets up the distribution of air flow for the subsequent rotor row. The air passes successively through rotors and stators in such a manner to increase the total pressure of the air to the degree required in the gas turbine. As the air is compressed, its density is increased and the annular flow area is reduced to correspond to the decreasing volume. This change in area may be accomplished by means of varying tip or hub diameter or both. In this compression process, there are losses that result in an increase in the air entropy. Thus, in passing through a compressor, the velocity, the pressure, the temperature, the density, the entropy, and the radius of a given particle of air are changed across each of the blade rows.

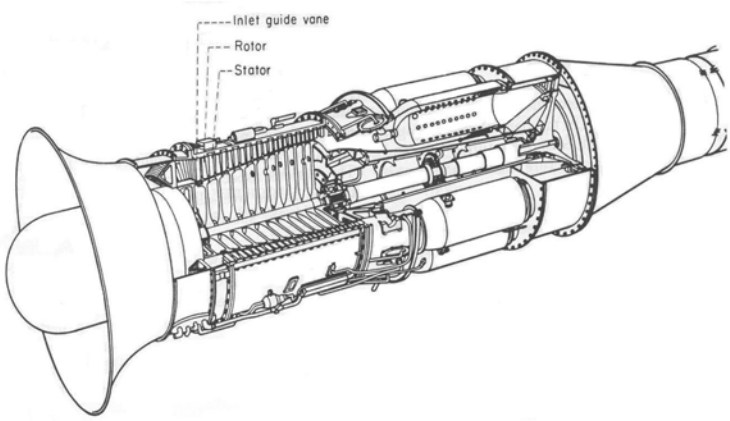


Figure 3: Axial flow compressor in turbojet engine.

The axial flow compressor is the principal type of compressor used in aircraft gas turbine engines. Although some of the early turbojet engines incorporated the centrifugal compressor, currently the axial compressor has a broader range of applications. This dominance is a result of the ability of the axial flow compressor to better satisfy the basic requirements of the aircraft gas turbine.

In general, the axial compressor has a high efficiency, high airflow capacity per unit frontal area, and high pressure ratio per stage. Because of the demand for rapid engine acceleration and for operation over a wide range of flight conditions, this high level of aerodynamic performance must be maintained over a wide range of speeds and flows. In Table 1, some examples of axial compressor evolution in the last decades are shown. Note that with the development of new technologies, it is possible to build a compressor with higher pressure ratio and smaller number of stages.

A compressor is a complex equipment, required to work at a wide range of flow pumping capacity. Therefore, constraints on both mechanical and aerodynamical matters must be taken into account.

Table 1: Axial compressor evolution.

Engine	Year	Thrust (kN)	Pressure Ratio	Number of stages	Pressure Ratio per stage
Avon	1958	44	10	17	1,145
Spey	1963	56	21	17	1,196
RB-211	1972	225	29	14	1,272
Trent	1995	356	41	15	1,281

2. Design-Point (DP)

Design point is the condition at which the cycle parameters are specified for the machine development. Generally at the design point the efficiency of the components is maximum and the efficiency of engine is maximized. Thus, the design point is chosen as the condition that the engine will work for a longer time.

The problem of designing axial compressors becomes the problem of accurately calculating the flow through the compressor blade rows. In order to be accurate and have the greatest range of applicability, these calculations should be based on the fundamental laws of motion as much as possible. At the same time, they should be of such a nature that they can be made readily with available computing techniques and equipment.

3. Off-Design Point (ODP)

It is the condition of engine operation at which some of its parameters are different from the adopted for the design, thus, varying the rotation speed or the air mass flow from the design point, the equipment won't have the same efficiency. If analyzed in a global way in the gas turbine, the turbine would transmit the same power to the compressor shaft, but this won't realize the same work. That is the reason the engine efficiency drops, due to the losses of each component efficiency.

Off-design performance is defined as the compressor performance at flow conditions and speeds other than those for which the compressor was specifically designed. The off-design analysis differs from the design case in that the compressor geometry is given and the objective is to find the compressor outlet conditions for a range of speeds and mass flows. If the compressor has variable geometry, the work range at off-design is larger and its efficiency and pressure ratio can be increased for each mass flow required, choosing appropriate stator blade settings.

The off-design point performance calculation is denominated “direct compressor problem”, whereas the original design is called “indirect problem”. The off-design analysis is one of the most difficult tasks for the compressor designers.

4. Prediction of Off-Design Performance

A typical axial compressor performance map is shown in Fig. 4.

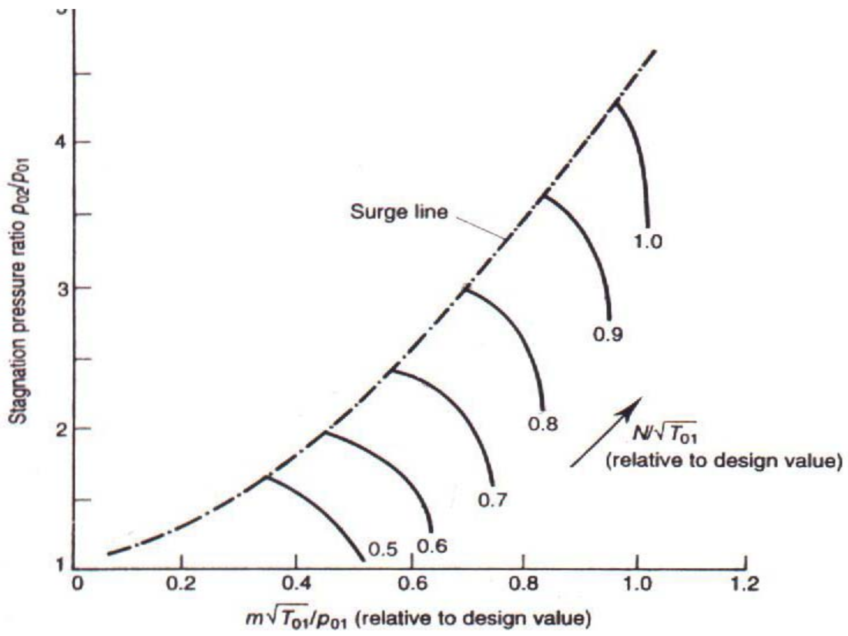


Figure 4: Axial compressor map.

The constant rotation curves are limited by the minimum mass flow where there is stall (or surge) and by the maximum mass flow, at which choke occurs.

5. Method Used

The stage-stacking method may be used as a research tool to investigate compressor off-design problems or to estimate the performance of an untested compressor. This method is also useful for determining the effects of interstage bleed and variable geometry on compressor performance.

In this method, overall compressor performance for a range of speeds and mass flows may be estimated. The performance of each stage of the multistage compressor is obtained and presented so that its performance is a function only of its inlet equivalent mass flow and rotational speed. For adopted values of compressor mass flow and speed, the first stage performance yields the inlet equivalent flow and rotational speed to the second stage. A stage-by-stage calculation through the compressor gives the individual stage pressure and temperature ratios, and efficiency can be calculated for the adopted values of compressor mass flow and rotational speed.

Thus, the mean line technique is applied using the stage-stacking method. It consists of calculations based on the mean streamline of the axial compressor channel. The flow properties as temperature, pressure, velocity and the dimensions of the equipment, including the blading angles, are determined at half blade height, this fact, of just using one streamline makes necessary the use of empiric correlations originating from experimental data. The method, when simulated in computer, presents fast numeric convergence and sufficiently accurate results for a first analysis of the compressors performance, allowing obtaining the compressor operating characteristics through the construction of its map.

6. Inlet Guide Vanes

The Inlet Guide Vanes (IGV) is a blade row located in front of the first rotor row. The use of the IGV is related with the inlet flow in the first rotor, because the IGV will determine the flow incidence angle in the leading edge of the following blade.

Generally, in compressors design that have uniform and parallel inlet flow to the rotation axis, the IGV use is not considered. When IGV is used the flow receives a pre-swirl in the rotor inlet.

There are IGVs with variable geometry used to extend the compressor operation range in the off-design condition, this IGV type is denominated VIGV (Variable Inlet Guide Vanes).

7. Variable Geometry

When the compressor is operates at off-design, the efficiency decreases due to rotational speed and mass flow variations, consequently decreasing the pressure ratio. In many cases this occurs due to the flow separation in the blades passages, so that an alternative is to vary the stators stagger angle (re-stagger) to align the flow with the blades. Hence, the compressor map is different for each variation of the stagger. In practice, for each re-stagger distinct compressors are obtained.

Variable inlet guide vanes have been also effectively used as means of improving engine acceleration characteristics. Adjustable guide vanes and stators provide a potential technique to improve the flexibility of high pressure ratio compressor operation without sacrificing design point performance. However, the additional weight and engine complexity and its control system will add an extra cost to the compressor. Adjustable inlet guide vanes and stator blades have been used to alleviate part-speed performance problems and to improve engine stability.

8. Compressor Air Bleed

Compressor discharge bleed has been used effectively to alter the matching of the compressor and turbine at part speed and to avoid compressor stall. This technique, however, does not eliminate the potential of blade vibrations instigated by rotating stall. Interstage air bleed in the multistage compressor has the potential of effectively altering the matching of the inlet and exit stages and thus appreciably reducing the speed at which rotating stall is encountered, as well as improving the complete compressor stall or surge margin at intermediate speeds.

The use of interstage bleed will add some weight and complexity to the engine but, in general, it offers improvement at part speed, concerning compressor stall or surge margin and to blade vibration problems resulting from rotating stall.

9. The Numerical Simulator - AFCC

A modular computational program, written in FORTRAN language, named AFCC (Axial Flow Compressor Code), has been developed (Tomita - 2003) that is capable to design and do the off-design performance calculation of variable geometry stators, validated from experimental results.

The AFCC, has two different modules. The first module does the design of an axial compressor (DP) and the second module does the performance analysis of an existing or newly designed compressor (ODP).

The structure developed for the AFCC may be summarized as shown in Fig. 5.

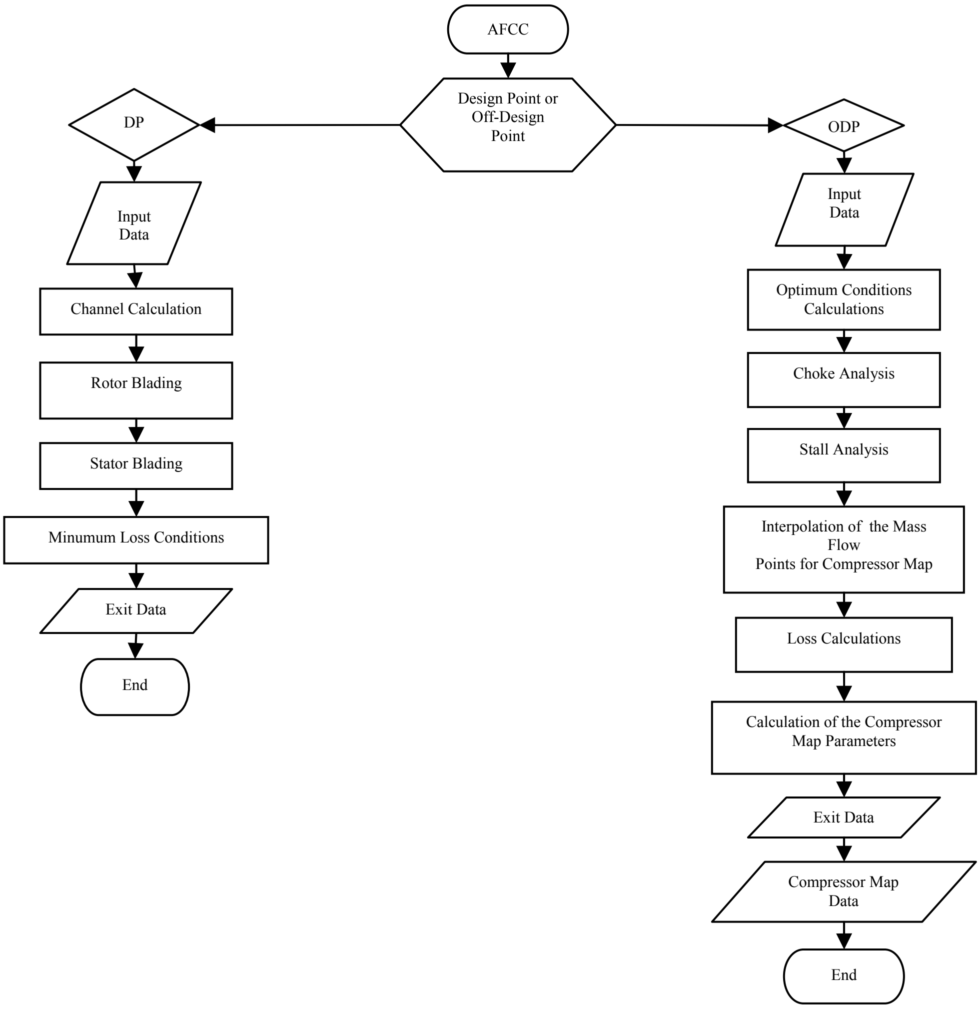


Figure 5: Scheme of the AFCC in the flowchart representation.

Note that the first module (DP), deals with the "indirect problem" and the second one (ODP) is the "direct problem" case.

10. A Study of the Variable Geometry Influence:

The variable stator setting is used to align the blades to the flow in order to reduce losses for better compressor performance.

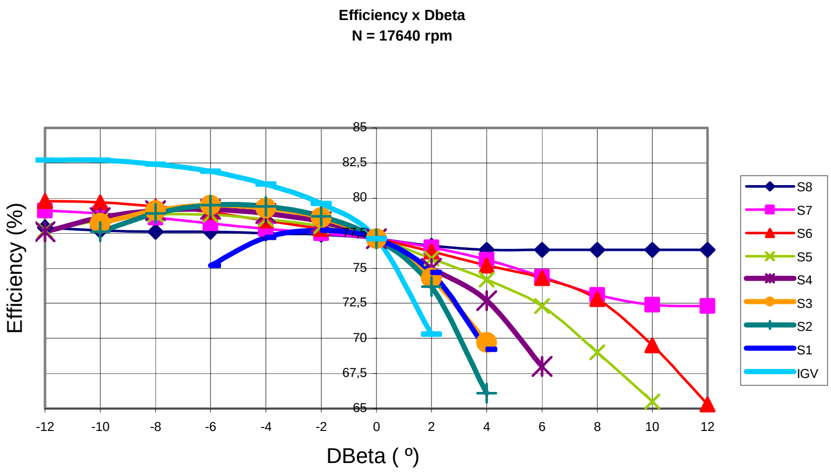


Figure 6. Variable Geometry influence on the compressor efficiency - compressor with 8 stage.

Stator blades are considered pivoted at midway in the camber line. Blade inlet and outlet angles are altered both ways, thus increasing or reducing flow.

In what follows, Δβ is the angle at with the stator stagger is turned.

Compressor geometrical data are recalculated after the blade setting is altered by an angle Δβ. From these new geometrical data the performance is calculated as for off-design condition and all flow properties are altered.

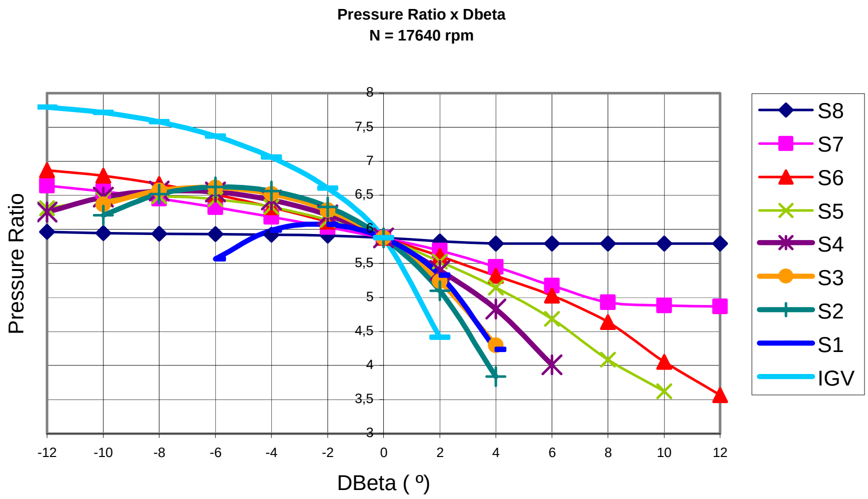


Figure 7: Variable Geometry on the compressor pressure ratio.

Figures 6 and 7 show the performance curves for several blade settings for the 8-stage compressor turning at 17640 rpm. The curves represent the variations of the stator staggers at each blade row and they show that the front variable rows have the most influence in the pressure ratio and efficiency than last ones.

This trend may give indication of the number of stator rows that would have blade repositioning in the search for better performance at part load. Thus, the less the number of compressor's variable rows the less the cost, the weight and the size of the equipment.

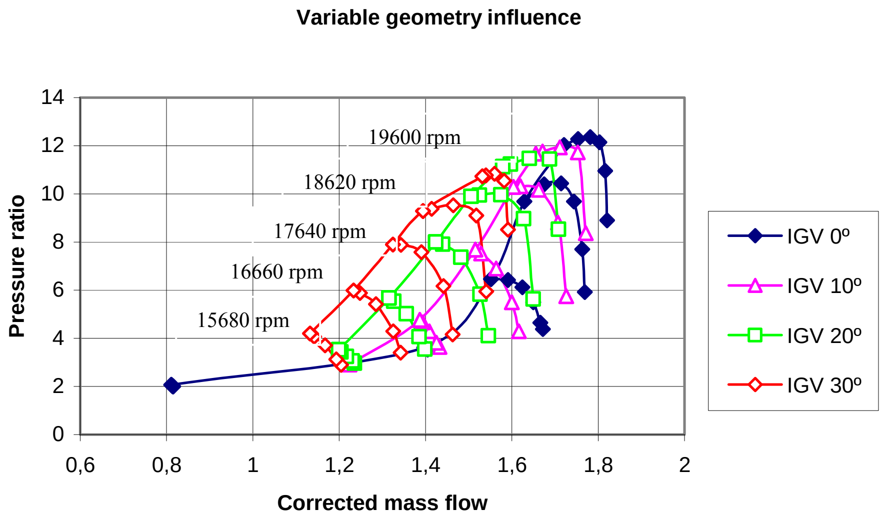


Figure 8: Variable IGV angles and its influence in the pressure ratio.

IGV stagger repositioning may alone account for the greatest performance improvement at reduced flow. Figures 8 and 9 show the performance maps of the 8-stage compressor with variable IGV. Large improvement with respect to the surge line may be achieved varying the IGV stagger.

Blade setting is certainly beneficial at the lower speeds. As indicated in Figure 8, surge line moves upwards and the operation work range between choke and stall points increases.

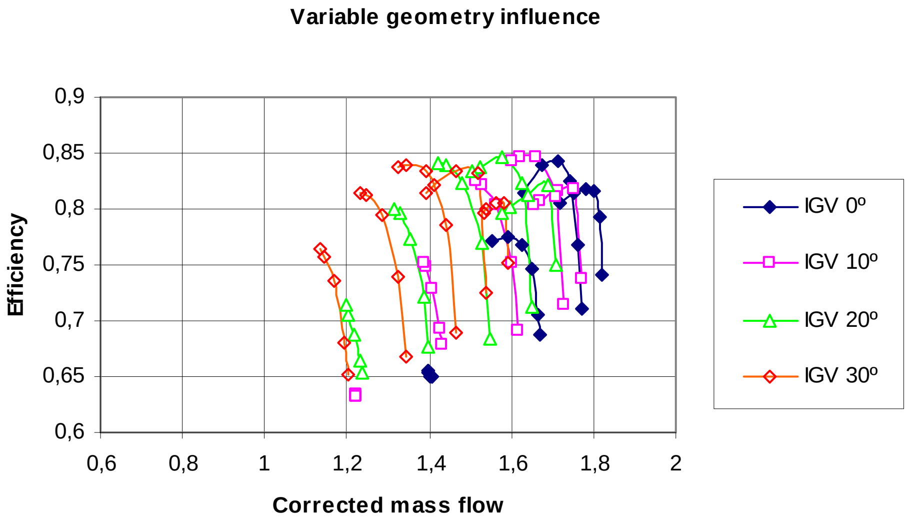


Figure 9: Variable IGV angles and its influence in the efficiency.

Figures 10 and 11 show the constant speed line of 19600 rpm for several IGV settings. Deleterial performance degradation would occur since air mass flow range would decrease, together with pressure ratio and efficiency.

It can be seen that, at a fixed speed, the compressor performance is reduced when the IGV stagger angle increases, showing the influence of the incidence on the rotor.

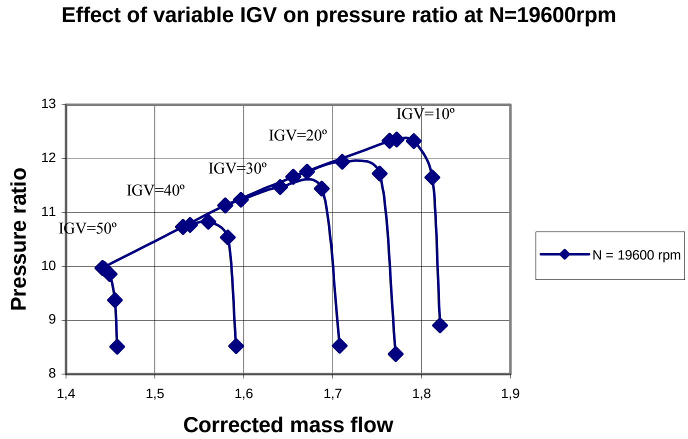


Figure 10: Effect of variable IGV on pressure ratio at N = 19600 rpm.

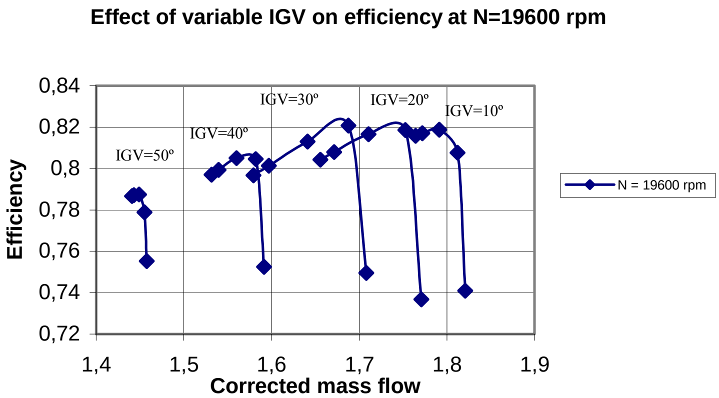


Figure 11: Effect of variable IGV on efficiency at N = 19600 rpm.

Figures 12 and 13 may be used for a similar analysis described above, but for lower rotation speed (16660 rpm). In this case, the compressor performance variation is minimal since the action of increasing the IGV stagger acts in the direction of decreasing the mass flow, therefore resulting in better flow alignment with the blades. Variable geometry is important at part speed operation.

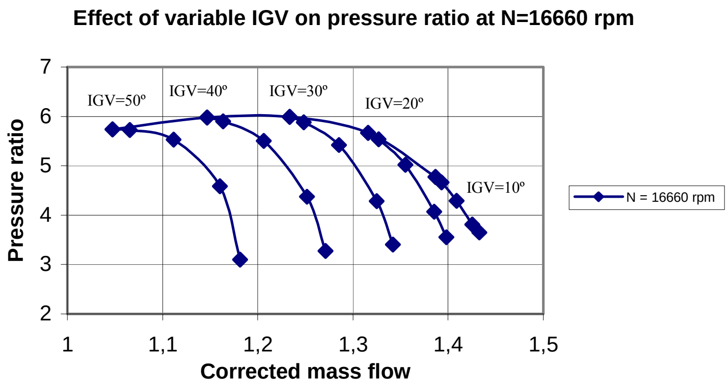


Figure 12: Effect of variable IGV on pressure ratio at N = 16660 rpm.

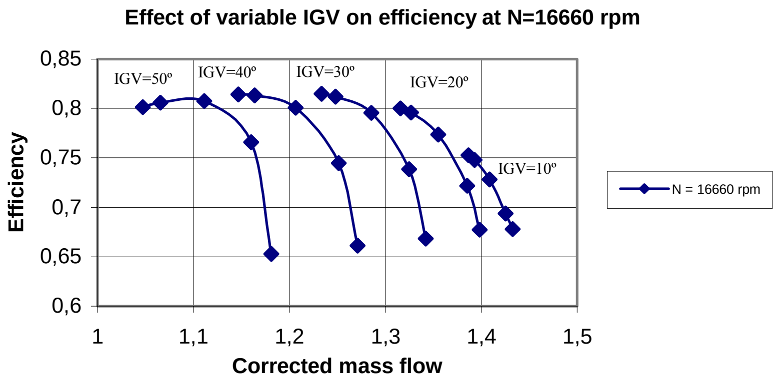


Figure 13: Effect of variable IGV on efficiency at N = 16660 rpm.

However, compressor performance improvement would be achieved provided the IGV are set at convenient angles at each constant speed line. Figure 14 shows the maps for IGV settings according to Table 2.

Table 2: Variable IGV and the respective speed.

Speed (rpm)	IGV (Δβ)
19600	0º
18620	10º
17640	20º
16660	30º
15680	40º

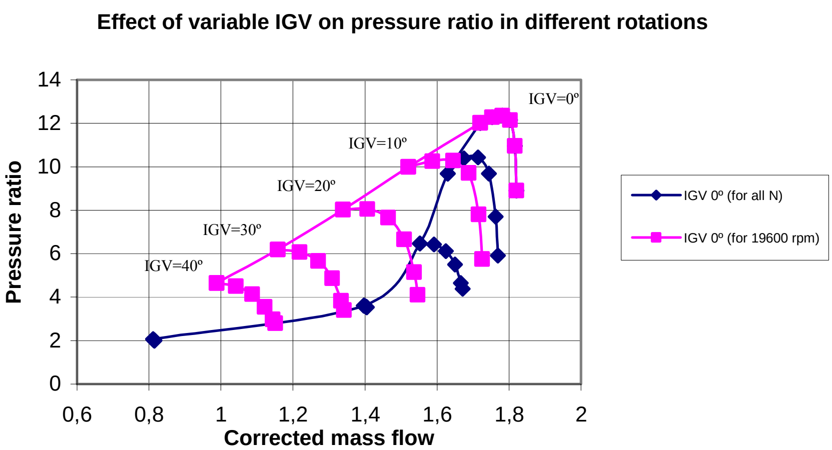


Figure 14: Effect of variable IGV on pressure ratio in different rotations.

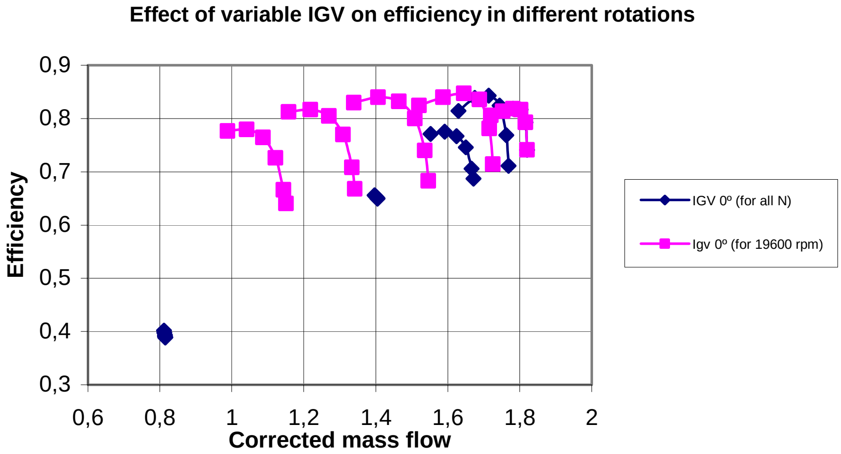


Figure 15: Effect of variable IGV on efficiency in different rotations.

As a result, the surge line has been moved upwards significantly at the same time the air flow range increased, thus conferring the compressor with improved performance.

Figures 16 and 17 show the influence of variable geometry at several stators rows, where VG1, VG2 and VG3 are the different maps used to compare the variable stators influence. In this study the variable geometry was included up to the 4th stage. Marginal performance improvement would be achieved with variable stators at the remaining stages. The rotation speed used to this analysis is the same of the Table 2.

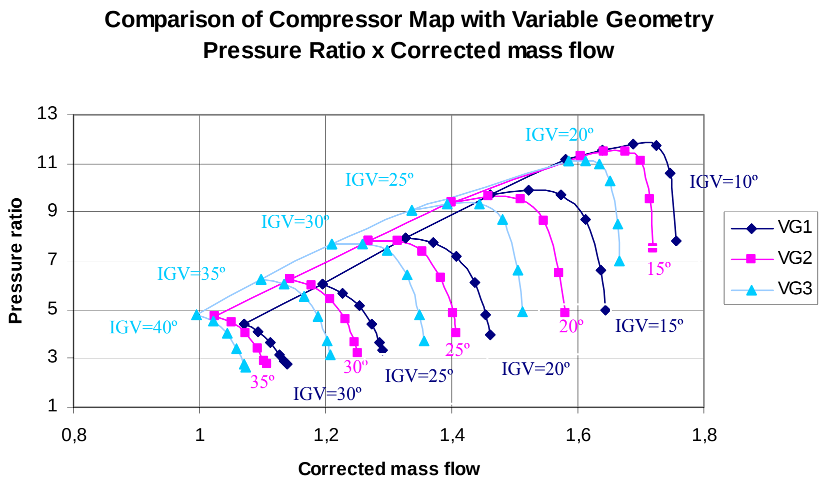


Figure 16: Compressor map with variable geometry (pressure ratio x corrected mass flow).

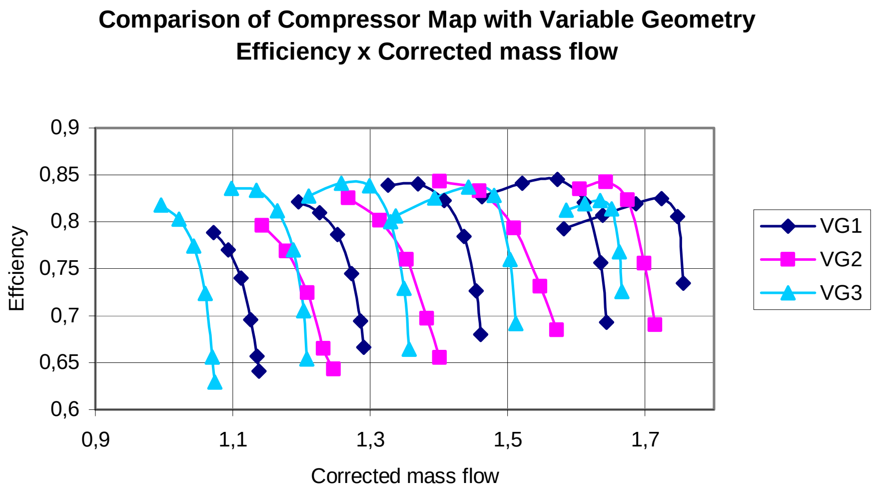


Figure 17: Compressor map with variable geometry (efficiency x corrected mass flow).

11. Conclusion

A technique to assess axial compressor performance has been developed that is able to predict the compressor performance for different IGV and stator settings. A computer program has been implemented that is able to design and do the off-design analyses of multistage axial flow compressors using the stage stacking method and mean line data. The program has been validated against numerical and experimental data from a 8-stage axial compressor and tested against available data from several other compressors. The results from the numerical simulation show that the computer program is able to confirm major compressor operation characteristics like the surge line upward movement with IGV closing at part speeds, a characteristic that is desirable when the compressor is part of a gas turbine.

The ability to predict the compressor performance is an important tool during the compressor design. Development cost and time may be significantly reduced since the undesirable compressor characteristics could be discovered at an early stage of the project.

12. Acknowledgement

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