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INFLUENCE OF VARIABLE GEOMETRY TRANSIENTS ON GAS TURBINE PERFORMANCE

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ABSTRACT

During the design of a gas turbine it is required the analysis of all possible operating points in the gas turbine operational envelope, for the sake of verification of whether or not the established performance might be achieved. In order to achieve the design requirements and to improve the engine off-design operation, a number of specific analyses must be carried out. This paper deals with the characterization of a small gas turbine under development with assistance from ITA (Technological Institute of Aeronautics), concerning the compressor variable geometry and its transient operation during accelerations and decelerations.

The gas turbine is being prepared for the transient tests with the gas generator, whose results will be used for the final specification of the turboshaft power section.

The gas turbine design has been carried out using indigenous software, developed specially to fulfill the requirements of the design of engines, as well as the support for validation of research work.

The engine under construction is a small gas turbine in the range of 5 *kN thrust /* 1.2 *MW shaft power, aiming at distributed power generation using combined cycle.*

The work reported in this paper deals with the variable inlet guide vane (VIGV) transients and the engine transients.

A five stage 5:1 pressure ratio axial-flow compressor, delivering 8.1 kg/s *air mass flow at design-point, is the basis for the study. The compressor was designed using computer programs developed at ITA for the preliminary design (meanline), for the axisymmetric analysis to calculate the full blade geometry (streamline curvature) and for the final compressor geometry definition (3-D RANS and turbulence models).*

The programs have been used interatively. After the final channel and blade geometry definition, the compressor map was generated and fed to the gas turbine performance simulation program. The transient study was carried out for a number of blade settings, using different VIGV geometry scheduling, giving indication that simulations needed to study the control strategy can be easily achieved. The results could not be validated yet, but are in agreement with the expected engine response when such configuration is used.

Keywords: *Gas Turbine Performance, Axial Flow Compres-*

sor, Variable Geometry transients, Transient Performance.

NOMENCLATURE

DP Design-Point.
$\dot{m}$ Mass Flow rate.
$\dot{m}_C$ Corrected Mass Flow, kg/s.
PR Pressure ratio
PR_{surge} Pressure ratio to surge margin
P_{ref} Reference pressure = 101325 Pa.
P_t Stagnation Pressure, Pa.
sm Surge margin
sfc Specific Fuel Consumption.
TIT Turbine Inlet Total Temperature.
T_{ref} Reference Temperature = 288.15 K.
T_t Stagnation Temperature, K.
VIGV Variable Inlet Guide Vanes.
η_{TGG} Gas Generator Turbine Isentropic Efficiency.
η_{th} Cycle Efficiency.

INTRODUCTION

The Center for Reference on Gas Turbine staff comes from the Technological Institute of Aeronautics (ITA) and from the Institute of Aeronautics and Space (IAE). ITA and IAE are an aeronautical engineering school and an aeronautical and space institute, respectively. They belong to the Department of Aerospace Science and Technology (DCTA). Presently they are developing a small gas turbine in the range of 5 kN (~ 1.2 MW shaft power), aiming at acquiring gas turbine know-how. Intention is to use such engine for distributed power generation in combined cycle.

The first part of the project is the development of the engine gas generator, which is undergoing vibration and lubrication tests, in preparation for the tests for performance measurement. All parts of the engine had been designed using indigenous software, for the sake of technology development. The gas generator may be used in a turbojet version. The turboshaft version is a free power turbine engine. The cycle parameters were chosen to compromise acceptable performance for both the turbojet and turboshaft versions. The major design parameters are, at standard atmosphere and temperature:

- ✓ Compressor pressure ratio: 5;
- ✓ Maximum cycle temperature: 1173 K;
- ✓ Fuel: kerosene;
- ✓ Thrust: 5 kN (turbojet version class);
- ✓ Shaft power: 1.2 MW (turboshaft version class).

The engine is required to run on other fuels, like ethanol and biofuels. The experience with alternative fuels can be traced back to the 1970s, when a well succeeded program was developed by the Aerospace Technical Center (CTA, now DCTA), to provide alternatives for the use of fuels derived from petroleum. At that time two biofuels, prosene and prodiesel, were developed and produced in pilot plants and were tested in gas turbines for both industrial and aero applications. An aircraft was adapted for flight-test a gas turbine running on the prosene biofuel, with excellent results.

ENGINE DESIGN AND ITS MAIN COMPONENTS

For the definition of the design parameters, high fidelity gas turbine performance computer programs [1-9], developed at ITA, have been used. The compressor, combustor and turbine performance maps were calculated and interactively used in the engine performance program.

Details of the compressor and the engine are shown in Figures 1 and 2, respectively. The compressor, which is a transonic five-stage axial, has been designed interactively using a mean-line program [2], a streamline curvature program [1] and a CFD computer program [10] in-house developed.

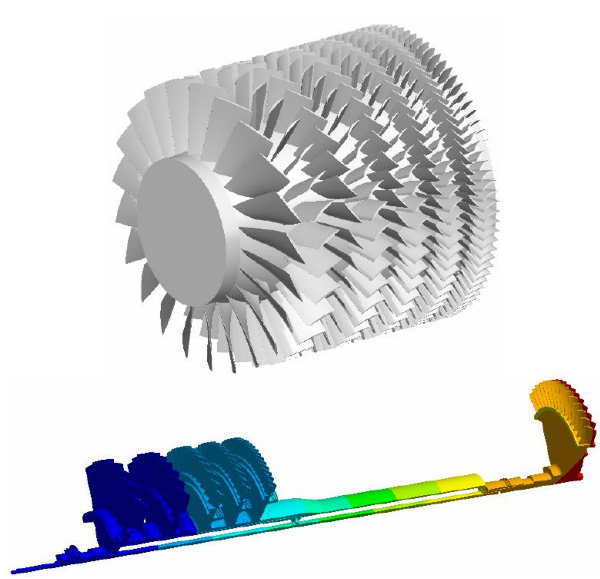

FIGURE 1. CAD VISUALIZATION OF THE DESIGNED COMPRESSOR AND A SECTION OF ITS ROTATING GROUP.

Figures 3 and 4 show the compressor performance maps indicating the pressure ratio and isentropic efficiency versus corrected mass flow, respectively. The lines of constant non-dimensional speeds, their corresponding percent values and surge line are also shown. Corrected mass flow is given by Equa-

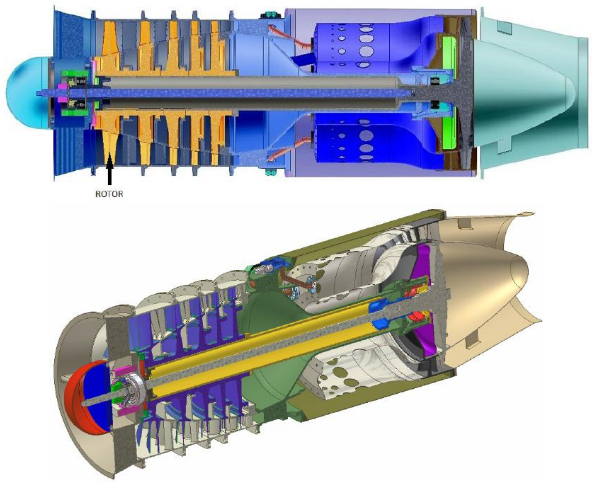

FIGURE 2. VIEWS OF THE GAS GENERATOR IN A TURBOJET VERSION.

tion 1.

$$\dot{m}_C = \dot{m}\frac{P_{ref}}{P_t}\sqrt{\frac{T_t}{T_{ref}}} \tag{1}$$

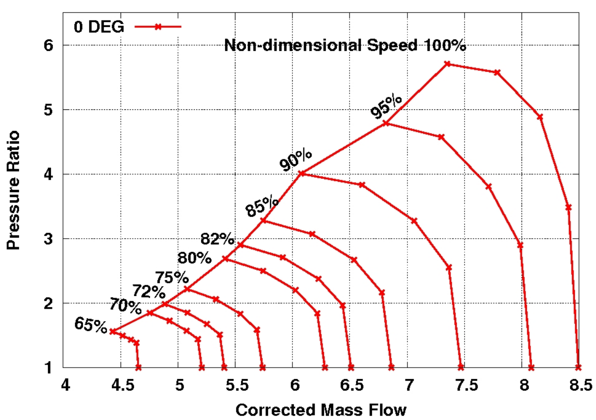

FIGURE 3. FIVE-STAGE AXIAL-FLOW COMPRESSOR PRESSURE RATIO MAP.

As surge margin was set to 15% at design (DP), simulation results indicate that the engine corrected speed can be lowered to 82% before it vanishes. The surge margin definition used in this work is defined by Equation 2.

$$sm = \frac{PR - 1}{PR_{surge} - 1} \tag{2}$$

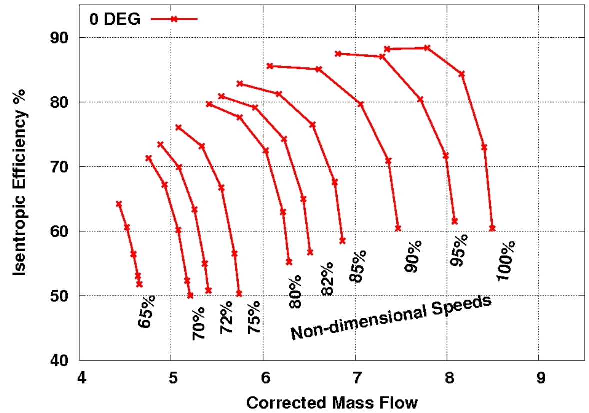

FIGURE 4. FIVE-STAGE AXIAL-FLOW COMPRESSOR EFFICIENCY MAP.

where sm is the surge margin, PR is pressure ratio at the considered operation point and PR_{surge} is the pressure ratio at which the compressor reaches the surge at the same corrected non-dimensional speed of the considered operation point.

The design mass-flow and pressure ratio were confirmed with the steady-state CFD program calculations at the design point.

Spatial integration within the CFD program used a second-order centered spatial discretization scheme with artificial dissipation to avoid numerical instabilities. Time-integration used a second-order five-steps Runge-Kutta scheme. Turbulence used the Spalart-Allmaras turbulence model for the eddy viscosity calculation.

Variable time-step and implicit residual smoothing are incorporated to the CFD code and used to accelerate convergence. To set the initial condition for starting the CFD program the resultsof streamline curvature program were used. The boundary conditions used for the calculations are: stagnation conditions and flow angles at inlet; static pressure at outlet; periodicity; no-slip at walls; mixing-plane at inter-blade rows. The relative frame of reference is used. Figures 5 and 6 show the computational domain and the mesh generated on the blade surfaces.

Figures 5 shown the compressor mass-flow history monitored during the numerical iterations.

The calculated mass-flow and pressure-ratio were 7.9 kg/s and 4.9, respectively, which were considered in agreement with the design values. The pressure ratio is calculated based on mass-averaging. A more refined analysis of the flow properties will be made when this compressor is submitted to tests in a newly built engine test facility. Fig. 8 shows the compressor manufactured rotor that will undergo development tests in the near future. Figure 9 shows the manufactured single-stage axial-flow turbine rotor (blisk).

The combustion chamber is of annular type. It has been de-

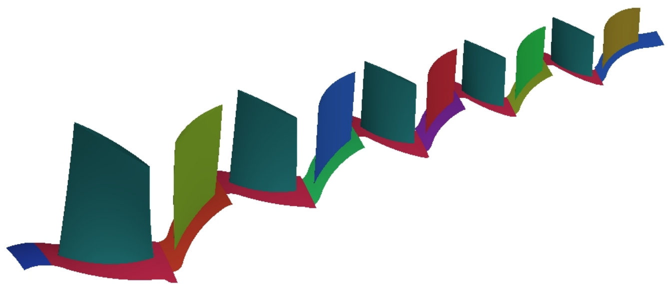

FIGURE 5. COMPUTATIONAL DOMAIN OF THE FIVE-STAGE AXIAL-FLOW COMPRESSOR.

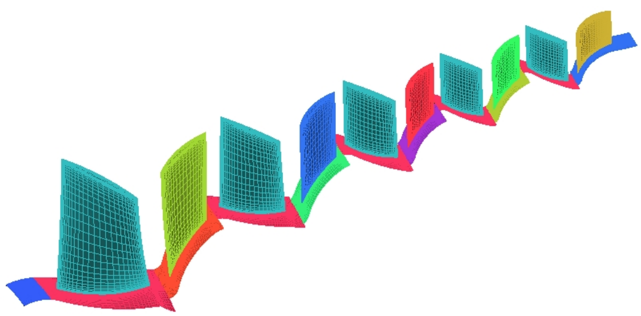

FIGURE 6. O-GRID WITH HEXAHEDRON ELEMENTS ON THE BLADE SURFACES.

signed for kerosene and diesel. Details are shown in Fig.10.

The rotating group was assembled to undergo vibration and lubrication tests. Figures 11 and 12 show the hardware in a test rig during the vibration and lubrication tests. For such tests, the blades were removed and replaced by equivalent inertias in order to reduce the power consumed in the rig. The natural frequencies calculated during the design by [11] have been confirmed by the tests.

The turboshaft version of the engine will consist of the gas generator and a free power turbine. For that engine, efficient work at part load may require variable geometry. The turboshaft figures are not available yet. Figure 2 shows the gas generator on turbojet version.

This work deals with the performance prediction of the *turbojet* with variable geometry, including the effects of the mechanical transients of the variable stators settings. A VIGV, not present in the turbojet version, is incorporated for the simulation. Initially, only the VIGV transients are studied.

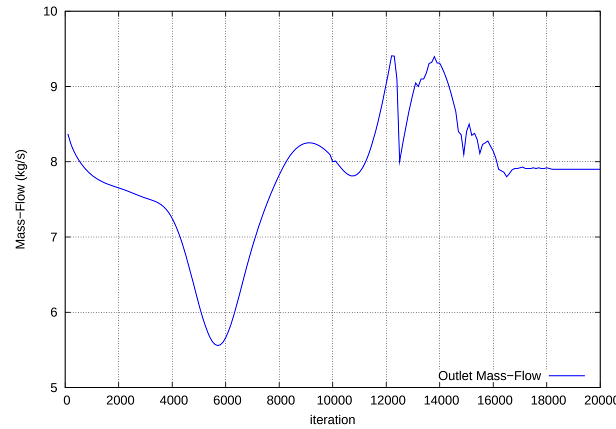

FIGURE 7. COMPRESSOR MASS-FLOW CONVERGENCE HISTORY.

FIGURE 8. MANUFACTURED COMPRESSOR ROTOR.

ENGINE TRANSIENT PERFORMANCE

The engine transients are calculated using the gas turbine performance computer program. The algorithms and the numerical implementation of the transient models resulted in a robust computer program [8], which is able to perform the transient performance for both shaft and volumes [7]. The verification was achieved using standard step functions for TIT. Large variations of either TIT or fuel flow may cause the compressor to surge, what is captured by the program, during simulation of accelerations. [8,9].

The engine and cycle parameters used for the study are shown in Table 1.

FIGURE 9. MANUFACTURED TURBINE ROTOR.

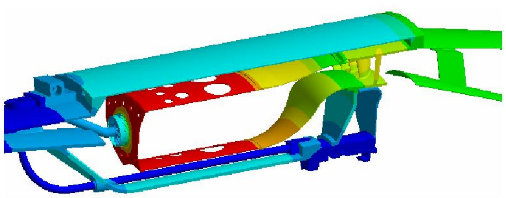

FIGURE 10. ANNULAR COMBUSTOR.

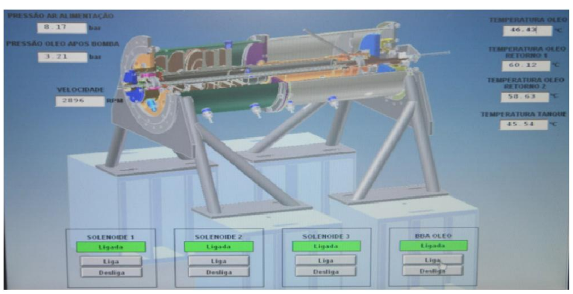

FIGURE 11. DETAILS OF THE DATA ACQUISITION DISPLAY USED DURING THE ROTATING GROUP TEST.

COMPRESSOR CHARACTERISTICS WITH VARIABLE GEOMETRY.

A computer program that performs the simulation of a real engine must be capable of simulating the actions that must be taken to keep stable the engine operation. One of such actions is to vary the VIGV setting. The transients of the VIGV variable geometry were taken into account.

For the numerical simulation, several compressor maps were synthesized, one for each position of the VIGV ($0°$, $-5°$, $-10°$, $-15°$, $-20°$ e $-25°$).

Figures 13 and 14 show the compressor maps for some of the indicated VIGV settings. On Figures 13 and 14 only maps for VIGV angles of $0°$, $-15°$ and $-25°$ are shown, as well as a

FIGURE 12. DETAILS OF THE ROTATING GROUP ON TEST.

TABLE 1. ENGINE AND CYCLE PARAMETERS FOR ANALYSIS.

Parameters	values
Mass flow (kg/s)	8.1
Compressor pressure ratio	5.0
Maximum cycle temperature (K)	1173.0
Thrust (kN)	5.2
sfc ($kg/h/kN$)	102.5
Compressor isentropic efficiency	0.86
Combustor chamber pressure loss	0.05
Combustion chamber efficiency	0.99
Gas generator turbine isentropic efficiency	0.87
Mechanical efficiency - gas generator shaft	0.99
Exhaust gas temperature (K)	857

reduced number of lines of constant corrected speed.

Specially developed computer programs were used for the syntheses of the compressor maps [2, 12].

ENGINE TRANSIENT PERFORMANCE WITHOUT VARIABLE GEOMETRY

For the deceleration study, the transient simulation was performed varying the maximum cycle temperature from the design point value of 1173 K down to 823 K. For the acceleration study, the temperature is risen from 823 K to the design temperature of 1173 K. For this study, the VIGV setting is kept constant ($0°$), Fig. 15.

Figure 16 indicates the hypothetical positions of the running lines (deceleration and acceleration) when the VIGV setting is unchanged, entering the unstable region (crosses the surge line) both during deceleration and acceleration.

Figure 17 shows the variation of compressor efficiency for

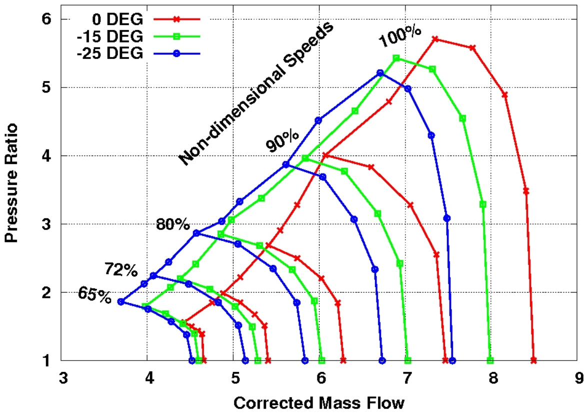

FIGURE 13. FIVE STAGE AXIAL-FLOW COMPRESSOR PERFORMANCE CHARACTERISTICS WITH VARIABLE GEOMETRY.

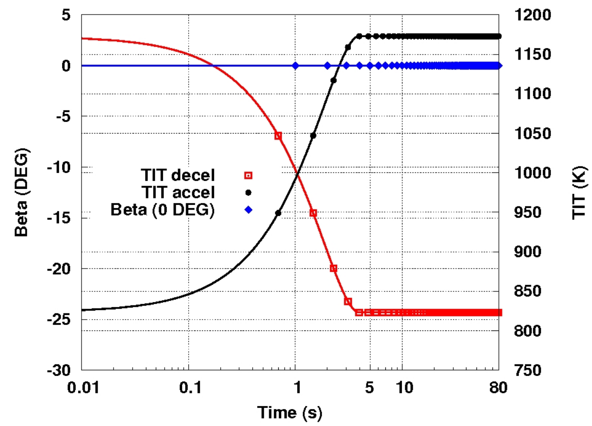

FIGURE 15. TIT AND BETA SCHEDULES FOR TRANSIENT PERFORMANCE WITH VIGV SET TO ZERO DEG.

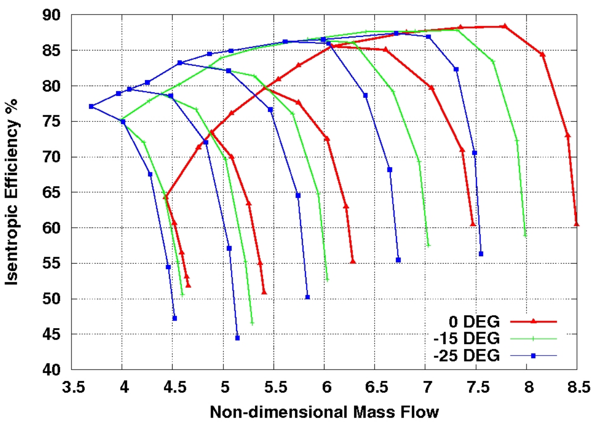

FIGURE 14. FIVE STAGE AXIAL-FLOW COMPRESSOR PERFORMANCE CHARACTERISTICS WITH VARIABLE GEOMETRY.

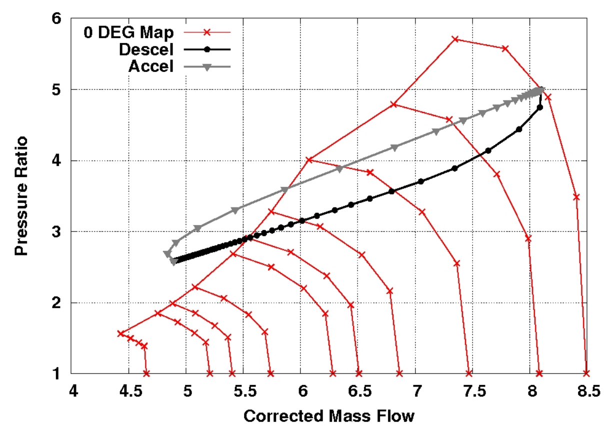

FIGURE 16. OPERATING LINE (HYPOTHETICAL) USING FIXED GEOMETRY UNDER TRANSIENT SIMULATION, DRAWN OVER THE CONSTANT SPEED LINES.

the case of fixed VIGV (fixed geometry), inidicating compressor efficiency deterioration from the design value of 85%, down to 81,25%. Part of the deceleration and the subsequent acceleration would occur outside the stable region. Indicating tha VIGV is necessary.

ENGINE TRANSIENT PERFORMANCE WITH VARIABLE GEOMETRY

The above simulations were repeated, but now taking into account variation of VIGV settings during the transient simulation. In this case it was considered that VIGV angle variation was parabolic, reaching the maximum (acceleration) or minimum (deceleration) in five seconds, according to the schedule indicated in Fig. 18. It may be noticed that both the transient VIGV and TIT settings are very gentle.

The same deceleration and acceleration conditions previously indicated were used for this simulation, namely TIT varying from 1173 K down to 823 K and from 823 K up to 1173 K.

Table 2 shows the major parameters used in the simulation with variable geometry. Time steps were 0.02 s, duration of TIT and VIGV angle steps was 5 s and de duration o transient simulation was 60 s, enough to reach stabilized engine rotational speed. TIT and variable geometry schedules were set to parabolic, with the variation of the VIGV proportional to TIT.

The variations of TIT and of the VIGV settings were chosen aiming at verification of the computer program robustness as well as the influence of the variable geometry transients during the engine transients. Other schedules, including strategies for engine control, will be reported separately.

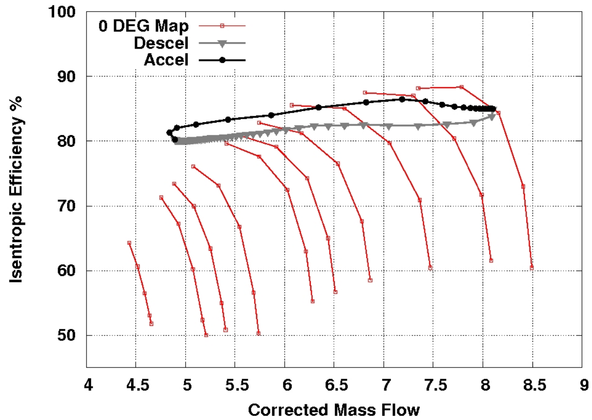

FIGURE 17. OPERATING LINE (HYPOTHETICAL) FOR THE FIXED GEOMETRY UNDER TRANSIENT SIMULATION, DRAWN OVER THE EFFICIENCY LINES.

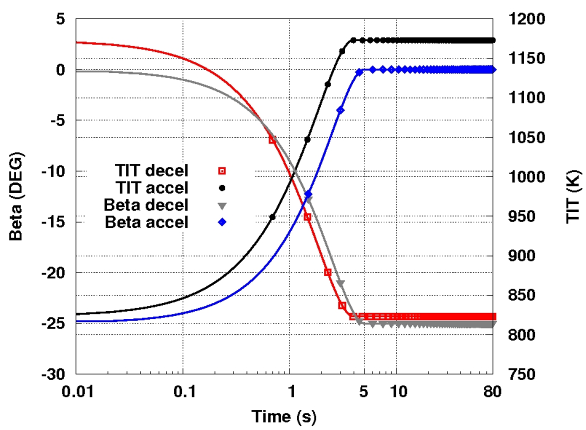

FIGURE 18. TIT AND BETA PARABOLIC SCHEDULES FOR TRANSIENT PERFORMANCE SIMULATION.

Results of the Variable Geometry Transient Simulations

Figure 19 shows the running lines drawn over the compressor map during variable geometry transients. With change of the VIGV settings, it can be seen that the compressor operates in the stable region and most of the time with acceptable surge margin. Surge margin may be dealt with by choosing adequate settings of both TIT and VIGV. Research for optimal settings is being carried out and must be the object of other publications.

Figure 20 shows the engine running line on the compressor efficiency map, from which it can be seen that the compressor operates stably near maximum efficiency, in a wide range of speeds.

Figures 21 and 22 show the variations VIGV angle and surge margin for the fixed and variable geometry transients simulation, respectively.

It can be observed that, during the simulation of engine

TABLE 2. ENGINE AND CYCLE PARAMETERS FOR TRANSIENT ANALYSIS.

parameters	values
Time step (s)	0.02
Moment of Inertia (kgm^2)	0.0125
Variation of maximum cycle temperature (K)	350.0
Duration of the variation of TIT (s)	5.0
Duration of the transient during acceleration (s)	60.0
Duration of the transient during deceleration (s)	60.0

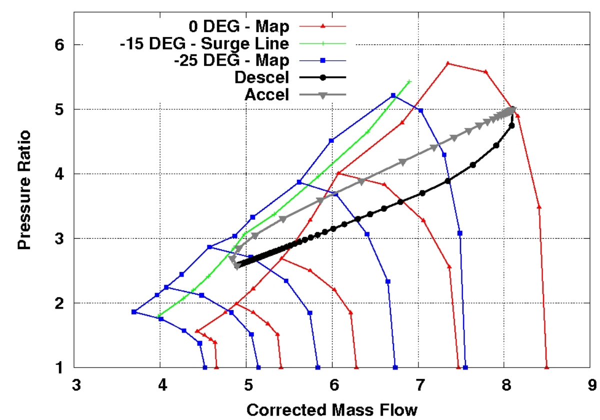

FIGURE 19. RUNNING LINE - TRANSIENT SIMULATION WITH VARIABLE GEOMETRY - COMPRESSOR PRESSURE RATIO.

deceleration with fixed geometry, the compressor would operate during the first 17 seconds with positive surge margin, after which it would (hypothetically) operate in the unstable region (negative surge margin); for the case of acceleration, initiated at the corrected speed of 77%, it would operate in the unstable region during the first five seconds, with negative surge margin (unstable). This is a kind of behavior expected in the literature [13-17]. [13]- [18].

Simulation of deceleration with variable geometry shows the compressor always operating with positive surge margin. For acceleration, during 3 seconds (between two and five seconds) it would be required to operate in the unstable region. This occur because the strategy of VIGV change (parabolic like TIT) is adequate to show influence and effectiveness in surge margin control of gas turbines, but is not adequate to effectively control the stability of engine simulated here.

During this acceleration in the first two seconds, the VIGV angles variation from $-25°$ to $-5°$ is not enough to maintain pos-

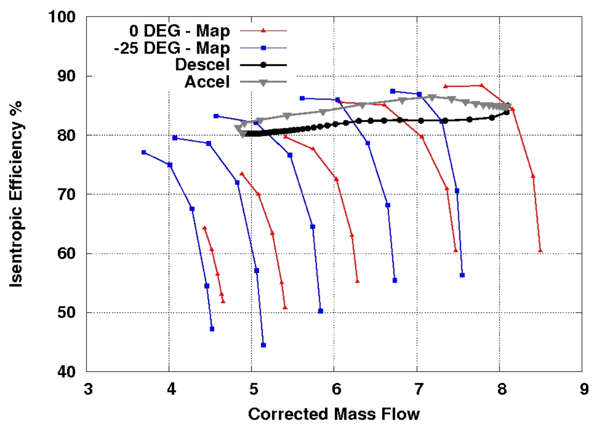

FIGURE 20. RUNNING LINE - TRANSIENT SIMULATION WITH VARIABLE GEOMETRY - COMPRESSOR ISENTROPIC EFFICIENCY.

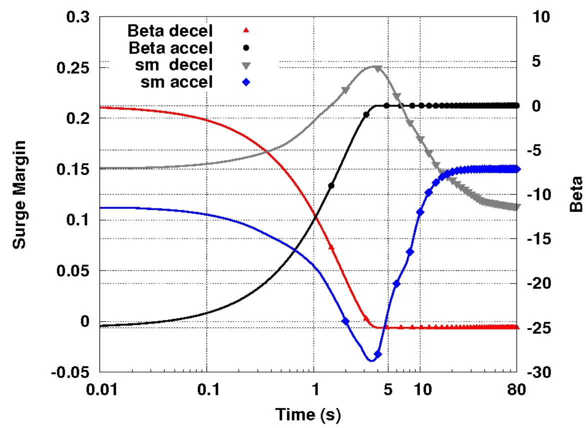

FIGURE 22. SPEED AND SURGE MARGIN VARIATIONS DURING VARIABLE GEOMETRY TRANSIENTS WITH VIGV.

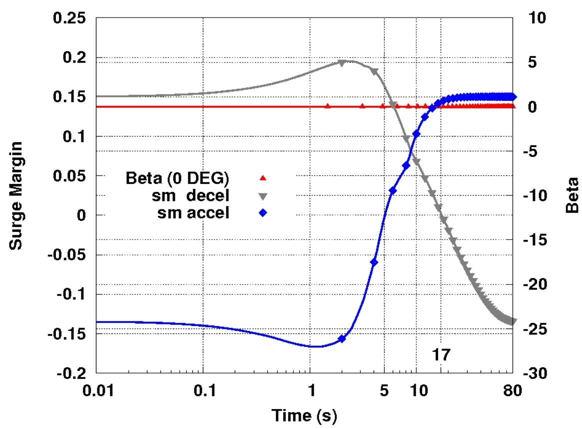

FIGURE 21. SPEED VARIATION AND SURGE MARGIN DURING TRANSIENTS WITHOUT VARIABLE GEOMETRY.

itive surge margin for the temperature variation, in the same time interval, from 823K to 1050 K. This condition is maintained for three seconds until VIVG setting is -0.7° and the temperature is 1172 K.

Therefore, the variable geometry eliminated the instability during deceleration and reduced the instability during acceleration. Improvement in the stability is obtained with a more efficient control strategy, what is not reported in this article.

DISCUSSION AND CONCLUSION

The compressor design and CFD programs were developed and used for the design of a five-stage transonic axial flow-compressor with VIGV, allowing the study of the gas turbine during transient operation, with and without variable geometry.

A more refined study and analysis will be made when this $\sim 1.2\,MW$ gas turbine and its components (compressor, combustion chamber and turbine) have been rig-tested. Validation can only be made based on experimental data, but they are not yet available. Nevertheless, all the computer programs have already been validated and the results obtained from calculations are in agreement with what is reported in the literature. Unfortunately, the turbomachinery community has very limited access to experimental data, details of the hardware (usually proprietary), as well as full description of the mathematical models and methodology used, so that verification and validation could be made.

The results of the simulations show that the implemented algorithms developed for the variable geometry transients during the engine transients can be used for performance calculation of engines with compressor variable geometry. In addition, the transients of the variable geometry can also be captured, making possible more realistic simulations of transients. These computer programs may become a reliable tool for the evaluation of engine control strategies.

Although the simulations reported here are not sufficient to define the control strategy for the engine, the results of the calculations can be used for the design of the control system, with reduction of time and cost.

ACKNOWLEDGMENT

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