Influence of Unsteady Effects on the Measurements in a Transonic Axial Compressor

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ABSTRACT

Interaction phenomena between rotor and stator are unavoidable in advanced compressors and their effects increase with the performances of the turbomachines.

Until now, it was not possible to quantify the interaction effects, but with the development of 3-D unsteady computation codes in a complete stage, it is possible to know, in detail, the flow field through the machine and to make evident and to explain the difficulties encountered in measuring the flow parameters.

A study has been conducted in this way at ONERA, on an axial transonic compressor stage. The computations have been made with a simulation of the losses; in this manner, the overall computed and measured performances of the compressor are the same.

A detailed analysis of the unsteady computation results makes evident, between rotor and stator, large variations of some parameters of the flow as a function of time but also as a function of the axial and tangential relative position of steady probes and stator blades. Unsteady measurements made on another transonic machine confirm the indications given by these computations.

NOTATIONS

I axial mark of radial section
J mark of tangential position
K radial mark of spanwise section
M Mach number
N speed of rotation
n number of blades
p static pressure
P\textsubscript{1} total pressure
Q mass flow
T\textsubscript{1} total temperature
U peripheral velocity
g absolute flow angle (with axis)
\eta efficiency
v hub to tip ratio
n total pressure ratio
\phi diameter

SUBSCRIPTS

0 far upstream
1 rotor inlet
2 stator inlet
3 stator outlet
o outer

SUPERSCRIPT

* mean value

1 - INTRODUCTION

The advanced compressors presently used on engines or in development are the location of important unsteady phenomena between fixed and rotating wheels. Up to now, it was very difficult to quantify these phenomena, to have an accurate idea of their field of action and to identify the corresponding perturbations on the measurements made, with the objective of determining the performances of these machines.

But, the recent development of three-dimensional computation codes for a compressor stage, has now made it possible to calculate and analyze in detail the unsteady flow field in a specific machine, particularly in the location of the space between the rotor and the stator of a one-stage transonic compressor. Through these computations, it has been possible to identify and explain some experimentally encountered difficulties and to show that some of the measurements techniques, that are currently used, are unsatisfactory.

In this respect, a study has been undertaken at ONERA on an axial transonic compressor developed in cooperation with CHINA (Chinese Aeronautics and Astronautics Establishment-CAE). After a together definition of the main characteristics of the compressor, ONERA used its EULER three-dimensional code with loss simulation for blade definition, then CAE manufactured the compressor and undertook the experiments at SARI (Shenyang Aeroengine Research Institute in Shenyang). Following these first tests, the computations were repeated with the same 3D code, with the aim of checking the global experimental performance.

A comparison between theory and experiment has not yet been made concerning the detail of the flow field, but the tendencies identified are confirmed by unsteady measurements made on another transonic compressor.
2 - COMPUTATION CODE

The computation of the viscous unsteady three-dimensional flow in a compressor stage, would require solving averaged Navier-Stokes equations associated with a turbulence model. Such codes, not completely operational, would lead to very long and very expensive computations. We therefore preferred to use the 3-D EULER code developed at ONERA (1985), with loss simulation (Meauze, 1990). This code leads to a reasonable computation time and has been validated in several cases.

This computation program solves Euler equations directly discretized in absolute physical space in cylindrical coordinates. A Mac-Cormack predictor-corrector finite difference scheme is used and the boundary conditions are treated using compatibility equations derived from the theory of characteristics.

In practice, the following are given to start the calculation:
- the total pressure distribution on the inlet to the computation domain, which includes the upstream surface boundary layers;
- the total temperature distribution on the inlet side of the domain;
- the meridian angle distribution at the beginning of the calculation domain;
- any prerotational distribution which may exist at the inlet to the domain;
- estimates of the relative upstream angle, the relative downstream angle and the meridian angle, only for initialization purposes.

As concerns the parameter that is used to vary the operating point, we generally use the static pressure, for which we set the value at the root, at the downstream edge of the computation domain, from which a simplified radial equilibrium is recalculated. Choking can also be simulated downstream of the domain by adding heat, characterized by a coefficient $Q_0$.

Concerning loss simulation, the code takes into account losses due to friction or diffusion on the casings and on the blades and losses due to the gap between the outer casing and the tip of the rotating blades.

Figure 1 shows the computational mesh which corresponds to 34 300 points. The calculations are made taking into account the loss scheme and arranging the empirical coefficients so as to check the global experimental results: pressure ratio and mass flow. In the case described, the gap at the blade tip was zero because the experiments were made with an abradable materials in the outer casing around the rotor.

In the computations, the experimental total radial pressure distribution at the rotor inlet is also taken into account.

The values of the coefficients are classical and realist ($C_f = 0.002$ f.e.) and they are imposed constant whatever the working point on the performance characteristic curve.

Table 1 compares the performance characteristics:
- target;
- deduced from design computations;
- obtained experimentally;
- deduced from adjusted computations;

<table>
<thead>
<tr>
<th></th>
<th>Target</th>
<th>Design computation</th>
<th>Experiment</th>
<th>Adjusted computation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow, kg/s</td>
<td>37.5</td>
<td>38.2</td>
<td>37.6</td>
<td>37.5</td>
</tr>
<tr>
<td>Stage pressure ratio</td>
<td>1.90</td>
<td>1.95</td>
<td>1.995</td>
<td>2.01</td>
</tr>
<tr>
<td>Isentropic stage efficiency</td>
<td>0.85</td>
<td>0.85</td>
<td>0.90</td>
<td>0.88</td>
</tr>
</tbody>
</table>

The target performance characteristics are, by security, lower than the design ones because at the time of the compressor definition, the code had not been validated. The experimental results are better than the target ones and the adjusted computations give very similar results.

In the case of a complete stage where the number of rotor blades is different from the number of stator blades, it would theoretically be necessary to compute the flow field in all of the channels of the two wheels, which is practically impossible on existing computers.

Some simplifying assumptions are necessary and, as analyzed in detail (Fourmaux, 1987, 1988, Le Meur, 1988, Meauzé, 1987, Povinelli, 1986, Rai, 1987), the best approach consists of carrying out the unsteady calculation on a restricted number of blades, $n_1$ for the first wheel and $n_2$ for the second, choosing the numbers so that the ratio $n_1/n_2$ is as close to reality as possible.
There is still a problem at the interface between the two wheels: the relative pitch specific to each wheel must be preserved in order to maintain the blade loading. There are two solutions:

a) to modify the geometry of one wheel by changing for example the chord so as to obtain the same angular spacing with the actual relative pitch. This solution, used by several authors, is acceptable in 2D but becomes unrealist in 3D where it is necessary to keep the exact geometry.

b) to keep the true geometry and angular spacing of the blades (as we do) but the problem of the periodicity appears. The "solution" consists to apply the periodicity every $n_1$ channels for the first wheel and every $n_2$ for the second.

The corresponding angular sectors calculated for each wheel are then similar although not strictly identical.

A further simplifying assumption is then needed: for each radius of the interface, we assume that the gradient of the various aerodynamic quantities (pressure, velocity, flow angle) remains tangentially proportional to the angular sector of the domain considered. There is no physical justification for this aside from the fact that it does correspond to the exact solution when the true number of blades is taken into account in the computation.

A systematic 2D analysis, made where an exact solution can be found by the space-time techniques (Hodson, 1984) shows that this simplification is satisfactory as long as it is not exaggerated by considering only one channel in each row. For practical applications, it often suffices to consider one channel in one wheel and two in the other. We feel that this approach provides a good approximation of the amplitude of the unsteady effects, but obviously not their frequency since the technique mentioned above introduces a pseudo-period specific to the simplification considered. In the case presented here, the ratio $n_1/n_2 = 0.587$, value not too far from 0.50.

3 - COMPRESSOR

The compressor studied is an axial transonic one-stage machine developed in cooperation by ONERA and CAE.

The main characteristics are indicated in Table 2 and Figure 2 shows a meridional section of the compressor.

### Table 2.

<table>
<thead>
<tr>
<th>Target characteristics</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Stage pressure ratio</td>
<td>$n = 1.90$</td>
</tr>
<tr>
<td>Mass flow : kg/s</td>
<td>$Q = 37.50$</td>
</tr>
<tr>
<td>Isentropic efficiency</td>
<td>$\eta_{is} = 0.85$</td>
</tr>
<tr>
<td>Outer diameter at rotor inlet : mm</td>
<td>$\phi_0 = 700$</td>
</tr>
<tr>
<td>Hub to tip ratio</td>
<td>$v = 0.70$</td>
</tr>
<tr>
<td>Peripheral velocity at rotor inlet : m/s</td>
<td>$U = 403$</td>
</tr>
<tr>
<td>Nominal speed : rpm</td>
<td>$N = 11 000$</td>
</tr>
<tr>
<td>Number of rotor blades</td>
<td>$n_1 = 37$</td>
</tr>
<tr>
<td>Number of stator blades</td>
<td>$n_2 = 63$</td>
</tr>
</tbody>
</table>

4 - RESULTS OF UNSTEADY COMPUTATIONS

The computations were made for only one operating point, close to the maximum efficiency point (about 12 % marging). In order to obtain a satisfactory periodic solution, several periods of rotor displacement are necessary. The next period is then divided in ten constant parts. The analysis of these ten computations is presented below.

In order to facilitate interpretation of the results, Figure 3 shows the different marks used and specifies two sections ($I = 73$ and $I = 77$) located on either side of the interface. Finally, the axial gap between the rotor outlet and the stator inlet over the local rotor chord is 17 % at the hub and 22 % at the tip.

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**Fig. 2 - Meridional section of compressor.**

**Fig. 3 - Marks used for computations.**
4.1 - Total Pressure Variation Downstream of the Rotor

Among the interesting parameters to be analyzed, the rotor pressure ratio \( \frac{P_{i2}}{P_{io}} \) (total pressure \( P_{i2} \) downstream of the rotor versus upstream total pressure \( P_{io} \)) is the first. However, in order to not overload the presentation, we have limited the figures to the domains where the unsteady phenomena are most marked. Figure 4 shows the pressure ratio variation versus time for several relative locations with regard, circumferentially, to the stator \( (J' \text{ variable}) \), that would be indicated by a probe located at \( I = 77 \), not far from the stator inlet plane; the figure corresponds to a section located at the hub \( (K = 2) \). One can see that, at \( J' = 10 \), for a probe located approximately between two stator blades, the total pressure \( P_{i2} \) varies by about 22\% during one period; this is a very important level of variation and it is not sure that a steady probe really indicates the mean value of the total pressure.

Moreover, and this is also very important, a probe located at \( J' = 2 \), just in front of a stator blade, would be subject to lower fluctuations and would indicate a mean value \( \frac{P_{i2}}{P_{io}} = 2.06 \) whereas the same probe located at \( J' = 10 \), would give a mean value of 2.11. The margin between the two values is not negligible.

At mid span, the variation of the mean total pressure \( P_{i2} \) relative to \( P_{io} \) on a stator pitch and at mark \( K = 10 \), is similar to the results obtained by Epstein (1989) as indicated in Figure 5.

![Fig. 4 - Rotor total pressure distribution versus time at various positions with respect to stator.](image)

Fig. 5 - Mean rotor total pressure versus stator pitch at mid-span

Moreover, and this is also very important, a probe located at \( J' = 2 \), just in front of a stator blade, would be subject to lower fluctuations and would indicate a mean value \( \frac{P_{i2}}{P_{io}} = 2.06 \) whereas the same probe located at \( J' = 10 \), would give a mean value of 2.11. The margin between the two values is not negligible.

In this case, there is a problem concerning the measurement of the mean pressure ratio. Strictly speaking, it would be necessary to cover a stator pitch with a fixed unsteady probe or to use a rotating probe. To use the stator as a probe strut, to measure the total pressure \( P_{i2} \) is not a good solution when the space between the rotor and the stator is small.

Outside this hub domain, we observe the same phenomenon at \( K = 4 \); then the fluctuations decrease as shown in Figure 6. This figure shows the pressure ratio variation as a function of time and of span \( (K \text{ variable}) \), concerning a probe located between two stator blades \( (J' = 10) \).

4.2 - Absolute Angle Variation

The absolute angle \( \alpha_2 \) formed by the flow with the axial direction at the rotor exit, is also an interesting parameter to analyze.

Figure 7 shows the variation of the angle \( \alpha_2 \) as a function of time that would be indicated by a probe located at \( I = 77 \), not far from stator inlet plane, for several relative locations with regard to the stator \( (J' \text{ variable}) \). The figure corresponds to a section located near the hub \( (K = 4) \). One can see that the angle fluctuations are very large, about 20 degrees, and that they are independent of the relative
probe-stator position. However, a probe located in front of a stator blade \( J' = 2 \) would indicate a mean flow direction \( \alpha_2 = 60 \) degrees, whereas the same probe, located between two blades \( J' = 10 \), would indicate another mean value \( \alpha_2 = 62 \) degrees. The difference is not negligible and we are not sure that a steady probe really gives the mean value of the flow angle. Unsteady probes would also be very useful for the measurement of this parameter.

Outside this domain, close to the hub, there are no major changes in the flow angle fluctuations as indicated in Figure 8. One sees only a small decrease in the fluctuation level that is between 12 and 14 degrees from mid-span to tip.

4.3 - Total Temperature Variation Downstream of the Rotor

Figure 9 shows the variation of the rotor downstream total temperature \( T_{12} \) versus the upstream total temperature \( T_{10} \), as a function of time, for several relative positions with regard to the stator \( J' \) variable), that would be indicated by a probe located at \( I = 77 \), not far from the stator inlet plane; Figure 9 corresponds to a section at the hub \( K = 2 \). One can see that the temperature fluctuations are relatively small (7.5 to 8 %), but in fact, these fluctuations represent about 30 degrees. At mid-span, on a stator pitch, the mean value of the ratio \( T_{12}/T_{10} \) is similar to the results obtained by Epstein (1989) as indicated on Figure 10. Outside this hub domain, Figure 11 shows a decrease in the level of the fluctuations from hub to mid-span then an increase from mid-span to tip.
4.4 - Mach Number Variation Downstream of the Rotor

Figure 12 shows the variation of the rotor downstream Mach number $M_2$, as a function of time, for several relative positions with regard to the stator ($J'$ variable) and in a section located at the hub ($K = 2$). One can see very large velocity fluctuations (21 to 26 %). A probe capable of correctly measuring this Mach number would indicate a mean value $M_2 = 0.782$ when located behind a stator blade and another mean value $M_2 = 0.854$ when placed between two stator blades. These very large discrepancies are due to $P_{12}$ total pressure fluctuations and to $P_2$ static pressure variations as well, as indicated on Figure 13. Outside the hub domain, the Mach number fluctuations rapidly decrease as shown on Figure 14; near the tip, they are practically null.

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**Fig. 9 - Rotor total temperature distribution versus time at various positions with respect to stator.**

**Fig. 10 - Mean rotor total temperature versus stator pitch at mid-span.**

**Fig. 11 - Rotor total temperature distribution versus time at various spanwises.**

**Fig. 12 - Rotor Mach number distribution versus time at various positions with respect to stator.**

**Fig. 13 - Rotor static pressure distribution versus time at various positions with respect to stator.**
5 - EXPERIMENTAL ANALYSIS

No measurements have yet been made in the rotor-stator space and it is not possible to give a comparison between theory and experiment. However, unsteady measurements have been made on another transonic compressor (Huard, 1990) and it seemed interesting to present them. Figure 16 shows the total pressure $P_{12}$ minus the atmospheric pressure $P_a$, as a function of time, measured near the hub in the rotor-stator space; the magnitude of the pressure fluctuations is about 10%. Compared to the computation results of Figure 4 (relative to a different machine), one notices similar phenomena. The discrepancy in levels can be explained by the fact that the computation corresponds to a section located near the stator leading edge (high interaction) while the measurements correspond to a section located near the rotor trailing edge (low interaction). Figure 17 shows the flow angle measurements versus time. One can see 15 degrees of fluctuation which is not very different from the computation results of Figure 7. The lower level of variation can also be explained as for total pressure variation.

6 - CONCLUSION

If it is well known that standard aero-probes give erroneous indications downstream of a rotating blade row, the detailed analysis of the computation results of a 3D EULER code with loss simulation, applied to a transonic compressor, showed that there can be significant variations of the flow parameters circumferentially in front of the front of the stator blades. In addition, the relative probe-stator location can have a non negligible effect and it is not advisable to use fixed probes on the stator blades at several radii.
The influence of the rotor-stator distance and of the number of blades has not yet been studied; however, we have observed that the unsteady effects are practically nil on the rotor blades.

With the aim of making accurate measurements, it would be desirable to use rotating probes, wherever possible or unsteady probes such as the one used on the second compressor.

However, an important problem remains to be solved: how to measure unsteady temperature?

Much work still has to be done on this subject.

REFERENCES


Keywords : Compressors - Viscous flow - Three-dimensional - Euler code