A 2-D NUMERICAL INVESTIGATION ON THE MODAL CHARACTERISTICS OF ROTATING-STALL WITH A VARIABLE-CASCADE-LENGTH APPROACH IN AN AXIAL COMPRESSOR

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Abstract The Rotating-Stall (RS) through a rotor-cascade of an axial compressor were numerically investigated with an unsteady two-dimensional finite-volume density based computer code. To validate the computer code, a test case was prepared and the good agreement of the compared results has given the adequate assurance of the code. The RS was incepted with a 40% reduction in flow coefficient and a 0.4% increase in the load coefficient from their normal operating values. The velocity traces from numerical probes, for all chosen number of blades had a periodic behavior, during RS effect. It was also concluded that the captured modal characteristics of RS may vary with varying the number of blades and this variation may cause various modes of RS. Moreover, in each mode, decreasing the number of blades may cause phase lag times in stall progress. In studied cases, approximately two modes and a transient mode were captured which had different rotational speeds and modal characteristics. Finally, a minimum required number of the blades for the RS inception studies (in spite of large numbers in previous works) was proposed, to intensify the probability of using shorter cascades in stability margin calculations of a compressor.

Key Words Rotating-Stall, Number of Blades, Rotor-Cascade, Finite-Volume, Axial Compressors

چکیده پدیده ″واماندگی دورانی″ در یک کمپرسور محوری به طریق عددی در چندین آرایش زنجیره ای مورد بررسی قرار گرفت. در این مطالعه از یک برنامه کامپیوتری حل معادلات جریان لزج دو بعدی به روش حجم های محدود استفاده گردید. برا ی تایید صحت عملکرد برنامه، علاوه بر تائیدات قبلی مراجع، یک مساله نمونه برای جریان های با جدایش احتمالی حل شد و نتایج انطباق خوبی با داده های آزمایشگاهی داشت. سپس مساله جریان ناپایدار در یک طبقه از یک کمپرسور محوری مورد بررسی قرار گرفت. نتایج ″واماندگی دورانی″ برای آرایش های یک زنجیره روتور تنها با تعداد پره متفاوت ارزیابی گردید. تمامی دنباله های سرعت محوری حاکی از رفتاری نوسانی بود. تغییر تعداد پره باعث نمود مودهای متنوعی از نوسانات سرعت محوری گردید. در این شرایط الگوی جریان در زمانهای معینی مطالعه شد و ویژگیهای گرداب های جریان ناپایدار و ارتباط آنها با نوسانات سرعت مطالعه شد. تغییر تعداد پره در مطالعه حاضر دو مد اصلی و یک مود میانی از واماندگی دورانی را نشان داد. در هر مود کاهش تعداد پره باعث عقب افتادن آغاز ″واماندگی دورانی″ گردید. در پایان معیاری از تعداد پره لازم در مطالعه عددی حاضر برای پیش بینی آغاز پدیده واماندگی دورانی بدون بررسی آثار نوسانی آن ارائه شد، تا نویدی بر کاهش محاسبات زمان بر و غیر ضروری با تعداد پره زیاد در هنگام تعیین محدوده کاری مطمئن یک کمپرسور ارائه گردد.

1. INTRODUCTION

The analysis of the flow field in unstable condition, especially in Gas-Turbine engines, has been based mainly on experimental observations and studies during last decay [1-3]. Recently, several computational fluid dynamics (CFD) codes have been developed. The axial compressors with their adverse pressure gradients in through flow direction, is the most critical component in Gas-Turbine engines from the viewpoint of flow instability phenomena. Today, the use of CFD tools is a standard practice in the study of the cascade flow within the stable operating range of a compressor, but the CFD approach still needs to be established as a sound prediction method for operation in the unstable region. Notable progress in use of CFD-type techniques for the calculation of instability effects was prepared by Sisto et al. [4] and Jonnavithula et al. [5]. They used a two-dimensional discrete vortex model with the separation point being obtained by an integral boundary layer. The evolution of stall is well predicted when compared with their experiment, although only up to six blade passages are used in the computation. Further, a numerical study was prepared by He [6] in a single stage of an axial compressor. The Navier-Stokes equations were discretized in space by finite volume method and integrated in time by using a four stage Rung-Kutta scheme and the second and fourth-order blended smoothing was adopted in both the stream wise and circumferential directions for numerical damping, and Baldwin-Lomax turbulence model was also adopted. Outa et al. [7], made a numerical simulation for stall cells in rotor-stator frame of a compressor, using viscous approach. Furthermore, a numerical study was prepared by Saxer et al. [8], for inviscid flow passing through axial compressors, and the history of stall vortices were investigated for a fifteen-blade passage of a single stage. The mass flow rate fluctuations were approximately similar to those obtained from their experimental observations.

In previous CFD reports, any criteria about choosing the number of blades and its possible effect on stall initiation can not be found and the related modal characteristics were not studied.

The scope of this work is twofold. First, to study the effect of choosing the blade number in a rotor-cascade on modal characteristics of the Rotating-Stall (RS), and the second, calculating the minimum required number of blades in CFD studies for stability margin calculations in axial compressors. The latter, provides faster studies for determination the stable limits of operating conditions for the given cascade geometry.

2. GOVERNINIG EQUATIONS

For the numerical solution, the transformed governing equations in computational space were used in their coservative form, as following [9]:

$$\frac{\partial \overline{\mathbf{Q}}}{\partial \tau}+\frac{\partial \overline{\mathbf{E}}_E}{\partial \xi}+\frac{\partial \overline{\mathbf{F}}_E}{\partial \eta}=\frac{\partial \overline{\mathbf{E}}_v}{\partial \xi}+\frac{\partial \overline{\mathbf{F}}_v}{\partial \eta} \tag{1}$$

Regarding to two-dimensional approach in present work, the governing equations for relative frame is the same as those in absolute frame, because there is no Coriolis acceleration (no radial component of velocity). Additionally, the stagnation enthalpy is constant in the relative frame and as a result there is no energy transfer between the fluid and the blades in the relative coordinate. Consequently, the only treatment in stationary-cascade studies (if desired) is the using of absolute velocities in place of relative velocities in the solver. As evident, the absolute velocities are achieved by adding the rotation speed to relative velocities.

The computer code that is employed for present investigation is that, which was introduced by Farhanieh et. al.[10]

3. NUMERICAL PROCEDURE

Finite-Volume Formulation Reconsidering Equation 1, the time derivative is approximated by a first-order backward differencing quotient and the remaining terms are evaluated at time level n+1. Thus:

$$\begin{aligned}&\frac{\overline{\mathbf{Q}}^{n+1}-\overline{\mathbf{Q}}^{n}}{\Delta \tau}+\left(\frac{\partial \overline{\mathbf{E}}}{\partial \xi}\right)^{n+1}+\left(\frac{\partial \overline{\mathbf{F}}}{\partial \eta}\right)^{n+1}\\&=\left(\frac{\partial \overline{\mathbf{E}_v}}{\partial \xi}\right)^{n+1}+\left(\frac{\partial \overline{\mathbf{F}_v}}{\partial \eta}\right)^{n+1}\end{aligned} \tag{2}$$

Integrating Equation 9 over square **ABCD** shown in Figure 1, and using Green's theorem provides:

$$\begin{aligned}&\Delta \overline{\mathbf{Q}}+\frac{\Delta \tau}{\Delta \xi}(\overline{\mathbf{E}}_{Er}-\overline{\mathbf{E}}_{El})+\frac{\Delta \tau}{\Delta \eta}(\overline{\mathbf{F}}_{Et}-\overline{\mathbf{F}}_{Eb})\\&=\frac{\Delta \tau}{\Delta \xi}(\overline{\mathbf{E}}_{vr}-\overline{\mathbf{E}}_{vl})+\frac{\Delta \tau}{\Delta \eta}(\overline{\mathbf{F}}_{vt}-\overline{\mathbf{F}}_{vb})\end{aligned} \tag{3}$$

Since, in Equation (3), the flux vectors are evaluated in time step n+1; they can be expressed in terms of $\Delta \overline{\mathbf{Q}}$, by using the Taylor expansion and a first order approximation in time[9].

The inviscid flux vectors on cell faces were evaluated by Van-Leer's Flux splitting scheme. To

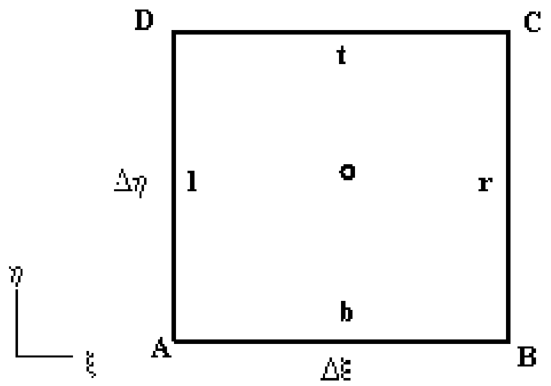

Figure 1. The finite volume cell.

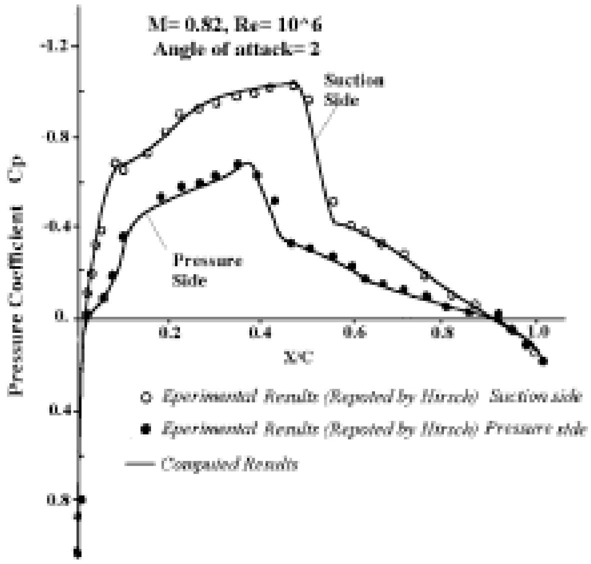

Figure 2. Computed and experimental pressure coefficients over a semi NACA0012 airfoil. Mach number is 0.82, the Reynolds Number is 10^6, and at an angle of attack 2°.

prevent the oscillatory behavior of the numerical results and to increase the accuracy, the Van-Leer's limiter was added to the flux splitting algorithm ([10]).

The second-order derivatives are evaluated by central difference approximation.

The multi-block technique used by Farhanieh et. al. [10] was employed for the numerical solution of the rotor cascades.

Numerical Boundary Conditions The inflow is subsonic and the static-temperature T_s, the axial and the tangential component of relative inlet velocity were taken from upstream. The static pressure P_s is set from the upstream in subsonic outflow.

For the solid walls, the no-slip boundary condition is employed.

For the upper and lower boundaries of both the rotor cascade, the periodic boundary condition was used to give the circumferential continuity of the cascade. The values in fictitious cells of the lower boundary are set to be the same values in the upper fictitious cells.

Each row of blades was split to multi zones for the multi-block technique and each block shares one or more boundaries with its surrounding blocks. Figure 2 shows the shared area of the two adjacent blocks.

The common walls of the rotor and stator-free passages shares their data in each time step as their adjacent wall boundary condition [9].

Grid Generation Each passage (between two blades) has an individual mesh, which is generated by mesh generator program using Partial Differential Equations (PDE) method. Clustering is available by related source terms as well as orthogonality. The mesh generated for each individual passage is considered as a single block, and the solver assembles them to prepare the complete area of solution by using the multi-block boundary condition. The resolution is chosen upon grid dependency studies. The studies show that the 75×41 resolution for each block has the minimum CPU time without any grid dependency of the results.

Turbulence Modeling An algebraic model, which is not written in terms of the boundary layer quantities and is very robust in separated regions, is the modified Baldwin-Lomax (BL) model ([9]). The BL model is employed by He [6] in numerical investigation of Rotating-Stall inception in a multi-blade cascade flow in an axial compressor, which the flow may have large scale separated zones. Moreover, the comparison of other turbulence models such as κ-ε with BL model, done by Bohn et al. [11], shows the adequate assurance of using BL model in cascade problems. Regarding a large

TABLE 1. Geometrical Characteristics of the Cascade.

Stagger angel of rotor blades	55°
Stagger angel of stator blades	35°
Rotor blade profile	NACA65-(A_{10})
Stator blade profile	NACA65-(A_{10})
Solidity	1.35

TABLE 2. The Stable Condition.

r_h / r_t	0.6
R	0.56
P_{in}	100000 Pa
T_{in}	300 K°
P_{exit}/ P_{in}	1.02589
β_1	62°
ψ	0.28
ϕ	0.4
V_{xin}	36 m/s
M_{in}	0.223
U_r	90 m/s
RPM	1240

TABLE 3. The Unstable Condition.

P_{exit}/Pin	1.02989
β_1	72°
Vx_{in}	22 m/s
ϕ	0.24
RPM	1240

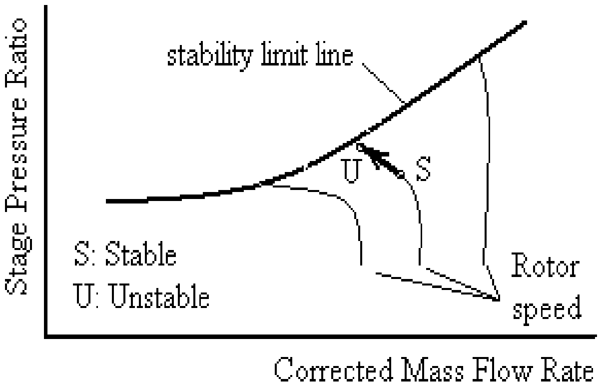

Figure 3. The thermodynamic path for reaching to RS condition from the normal operating point.

amount memory required in multi-blades studies, the BL consumes the least memory and CPU time with respect to higher-order turbulence models. Consequently, for present work the BL model is preferred.

4. DISCUSSION OF RESULTS

Code Validations Studies The validation of the employed numerical methods in the computer code and the performance of the described methodology were prepared by Farhanieh et al. [10].

To have more validations of the employed computer code, a numerical study was prepared for viscous-transonic flow over a NACA0012 airfoil with 2° angle of attack. The Reynolds number is set to 10^6 and the flow is turbulent. The inlet Mach number is set to 0.82. Figure 2, shows the computed results, which are in good agreement with the experimental results reported by Hirsch [12], in both sides of the airfoil and especially in shock sections.

This test case is prepared to examine the solver in viscous – turbulent flows with the existence of pressure gradients and separation areas. The grid

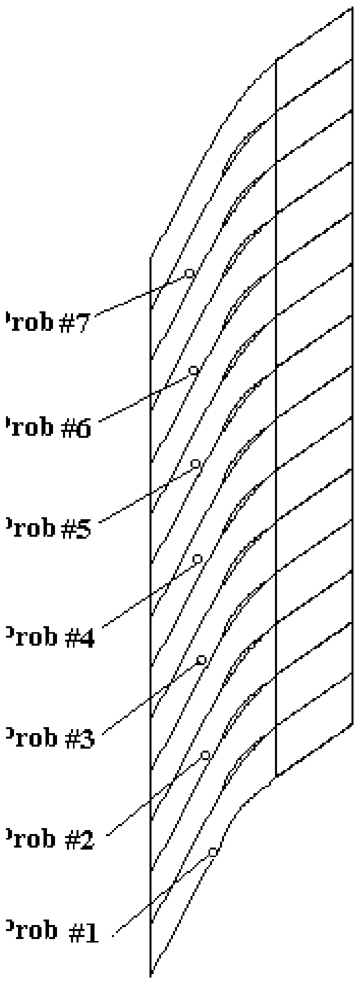

Figure 4. The 14-blade geometry and the location of the probes.

resolutions are chosen upon grid dependency studies, which are not discussed in present work.

Flow Through a Rotor of an Axial Compressor

The main study is prepared for flow through a rotor-cascade at the unstable operation point of an axial compressor.

The geometrical characteristics of the blades in the rotor-cascade are given in table 1, and the flow characteristics in the stable and unstable operating conditions are given in tables 2 and 3 respectively. The thermodynamic path for choosing the unstable operating point near the stability limit line is shown in Figure 3.

The rotor study is done for 2, 4, 7, 9, 10, 11, 12 and 14 blades in rotor-cascade. The blades are

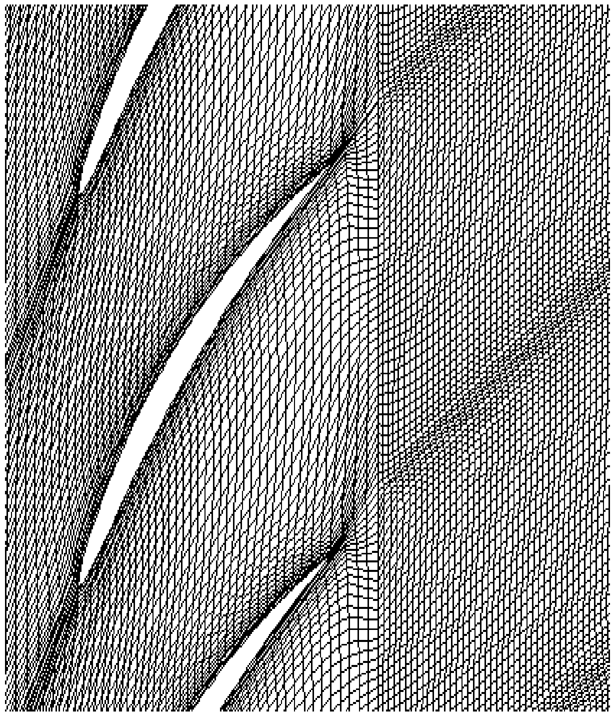

Figure 5. The sample enlarged mesh generated for rotor-cascade studies.

numbered from bottom to top. Near the leading edge of the odd blades, a numerical probe is located to indicate the axial velocity trace during the RS phenomena. To provide a clear imagination about the computational area, Figure 4 illustrates a 14 blade cascade (28 blocks). The solid lines show the boundaries of each block, and 7 probes are located near the leading edge of the blades.

The total number of finite volume cells is equal to: Number of blades $\times 2 \times 75 \times 41$. Figure 5 shows an enlarged section of the computational area.

The velocity traces for RS condition of the 14-blade are shown in Figure 6.

For clarity, the time traces from each probe, shown in Figure 6 are shifted by a constant interval, and the time is computed from the time of imposing the unstable conditions. It is noticed that this calculated RS pattern seems to be different from that commonly observed in experiments, where a single RS cell almost always comes into existence during a stall inception process. From the modal-wave point of view, the single-cell pattern should be associated with the first-order modal wave

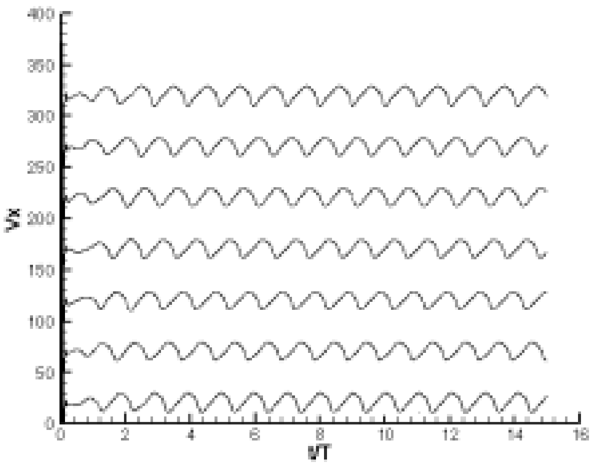

Figure 6. Velocity traces for 14-Blade geometry. The traces are shifted with constant interval for clarity.

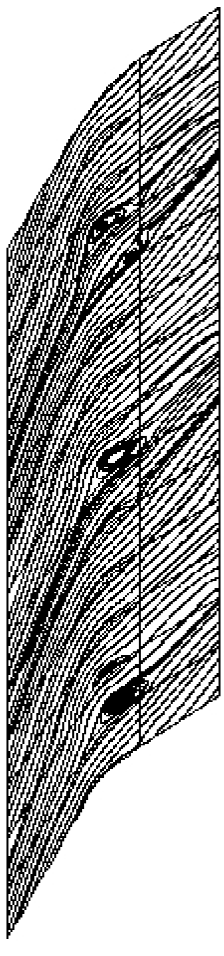

Figure 7. The unstable streamlines for 14-blade geometry at time 2.83 rotor revolutions.

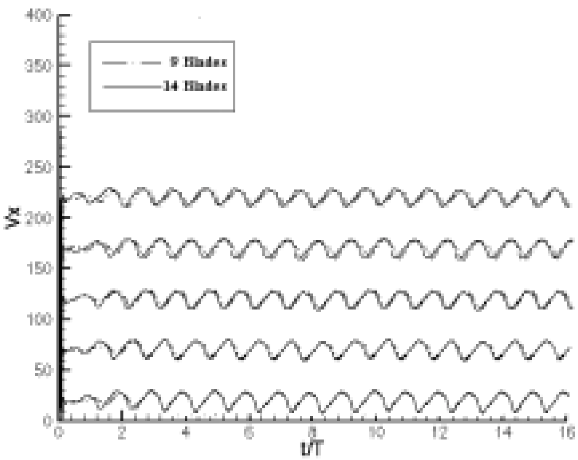

Figure 8. Comparison of velocity traces of 9 (dash dot lines) and 14 (solid lines) blades.

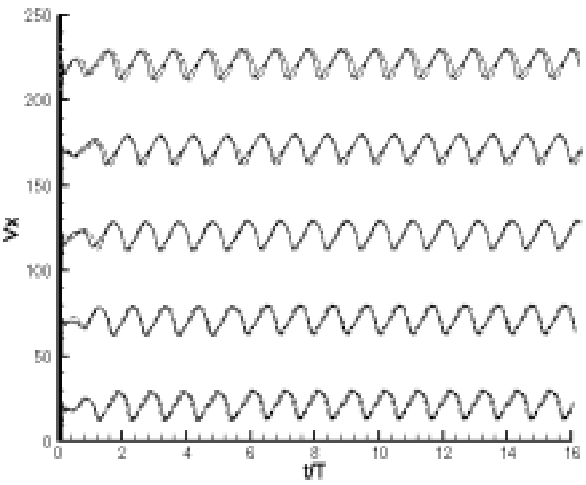

Figure 9. Comparison of velocity traces for 10 (dashed lines) and 14 (solid lines) blades.

with its wave-length being the circumference. In the present case, no indication of the first-order modal-wave was observed. As He [6] studied, the reason is that the background noise in calculations was not big enough to trigger the first modal wave to grow.

The streamlines of 14-blade study is shown in Figure 7. The deep cells are happened every five blades. A similar modal behavior was also nearly

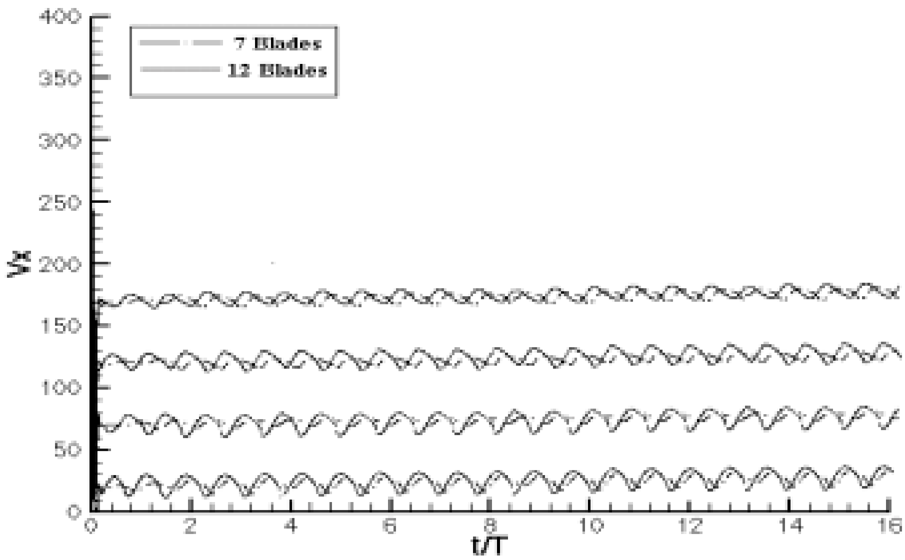

Figure 10. Comparison of velocity traces of 7 (dash dot lines) and 12 (solid lines) blades.

happened in 9 and 10-blade studies. The velocity traces of 14 and 9 blades were compared with each other in Figure 8 that illustrates the similar modal characteristics of the velocity traces. The studies show the same results for of 14 and 10 blades traces. In the same modal condition (mode A) for 9, 10 and 14 blades cases, the distance between every two deep cells is always 5 blades, and the rotating speed of the stall propagation is about 97% of rotor speed. In Figure 9, the 10 and 14 blades traces were compared with each other.

In the previous observations of He [6], the effect of back-ground disturbances and the modal variety of rotating stall were studied with forced imposed perturbations in inlet stagnation pressure. In the present work, the variety of RS modal characteristics was achieved with varying the number of blades. The velocity traces from 7 and 12 blade studies were compared with each other in Figure10.

The velocity traces is approximately similar and has nearly the same frequencies, but they are apparently different from mode A. this new modal shape is called, mode B. It is considerable that, in each mode the variation of the blade number only affects on stall initiation times and may cause phase differences with respect to chosen geometries. In Figure 11, the streamlines of 12-blade geometry were shown, and it seems that the deeps cells were

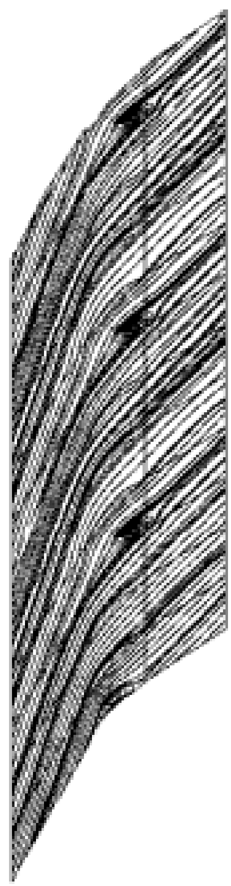

Figure 11. The unstable streamlines of 12- blade geometry after 2.83 rotor revolutions.

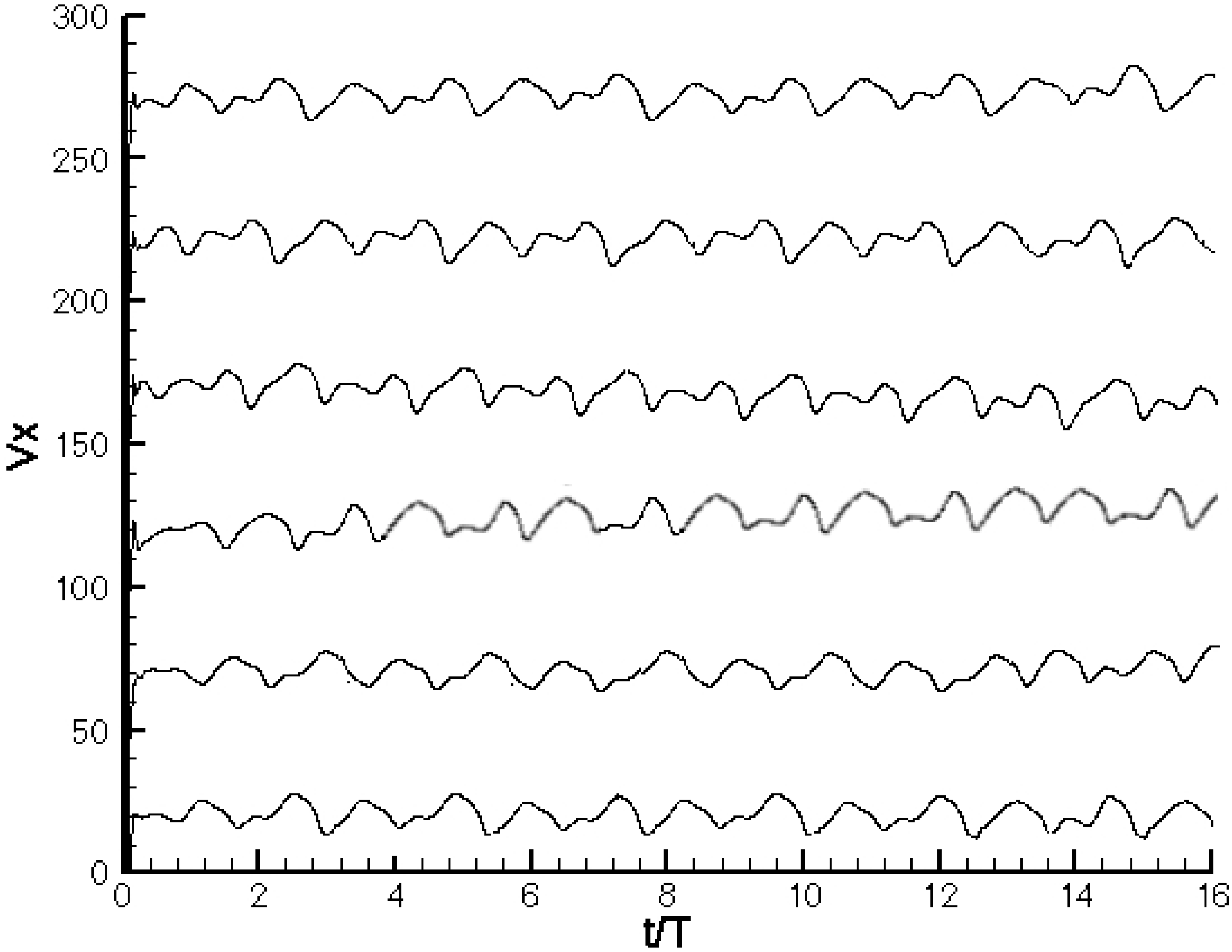

Figure 12. The velocity traces for 11-blade geometry at unstable condition.

happened every 4 blades.

The stall propagation in mode B is about 67% of the rotor speed. On the other hand, observing the velocity traces of the 11 blades geometry in Figure 12 shows that between two major modes A and B there is probably a transient mode during RS modal changes from mode A to mode B. However, other geometries e.g. a complete stage with various number of blades or rotor-cascade with more number of blades may show further modal characteristics of the RS.

Moreover, In present CFD studies and the modal characteristics a minimum 7 blades seems to be adequate to predict the stall inception without any consideration on modal shape of RS.

5. CONCLUDING REMARKS

The rotating stall effect was captured for the flow through the rotor-cascade of an axial compressor by varying the operating point to the unstable condition. The results give the following remarks:

1. All velocity traces show a periodic behavior as previous reports (He [6] and Saxer [8]), for any chosen number blades in RS phenomena.
2. Comparing velocity traces gives there are approximately at least two major modes for RS in selected geometries.
3. In each mode velocity traces has nearly the same frequency, and the blade number just

affects on complete stall initiation time and the phase lag times.

4. The reduction of blade number in each mode may often increase the stall initiation time.
5. The study prepared for 11-blade geometry show a transient modal characteristic between mode A and mode B.
6. The velocity trace frequency in mode B is approximately 1.5 times greater than that in mode A.
7. In all cases the RS effect was captures but with various modal shapes. As an important result for CFD investigators, one may conclude that choosing at least 7 blades in present work may be adequate for calculating the stability margin in off-design studies of the axial compressor. This result may help to prevent excess computational efforts for complete numerical simulations of the axial compressor.

The minimum limit of 7 blades is claimed to be minimum required number for all geometries, but it can show that a limited number of blades can be adequate for stall numerical predictions (in spite of previous works with large blade numbers) without any expecting of consideration on modal shapes.

6. NOMENCLATURE

τ time in transformed coordinate time step
ξ horizontal axis of transformed coordinate
ρ density
β_1 inlet angle of relative velocity
ψ load coefficient
ϕ flow coefficient
η vertical axis of transformed coordinate
$\overline{\mathbf{E}}_E$ inviscid transformed flux vector
$\overline{\mathbf{E}}_v$ viscous transformed flux vector
r_t tip radius
t time in physical coordinate
T_s static temperature
Tin inlet temperature
V_x Axial velocity in stage

Subscript

ξ partial derivative with respect to ξ
η partial derivative with respect to η
b index of bottom face flux
h hub for blade radius
in inlet condition
l index of left face flux
r index of right face flux
t index of top face flux
t total state for thermodynamic properties

Superscript

n previous time level
n+1 current time level

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