A Theoretical Model to Predict the Stall Inception of an Aeroengine Compressor with Micro Tip Injection

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Abstract: Steady micro tip injection is experimentally validated as an effective method in improving the stall margin of an axial compressor. Because the total amount of injection flow is extremely small, the compressor response on the injected flow is inferred to be unsteady. A three-dimensional CFD simulation is performed, and it is found that the interface between the tip clearance flow and the main flow is periodically pushed backward inside the blade passage. In order to provide a quick prediction of the stall inception point, a simplified model for the calculation of the trajectory of the tip clearance vortex is proposed. A two-dimensional unsteady CFD calculation is used to reproduce the evolution of the formation of the tip clearance. It is found that this model could be further improved and used as a criteria of the stall inception. However, it is still required to find a method of modelling the effect of micro tip injection in this two-dimensional model.

Keywords: *Stall inception, Micro injection, Aeroengine compressor*

1. Introduction

In order to satisfy a safety and stable operation of aeroengine, the operating line of a compressor is generally designed to reserve a sufficient distance, which is the so-called stall margin, away from the stall boundary of the aeroengine compressor. Because the stall margin should consider any possible variance of operation point in service of an aeroengine. Especially, the stall boundary has an uncertainty, and nowadays there is no effective method of accurately predicting the stall boundary. In addition, the design operating point has to be chosen away from the possible stall boundary in order to keep a sufficient stall margin, i.e., the best efficiency is restricted by the stall boundary. Therefore, the best performance of a new compressor could be achieved only if the stall inception point is predicted in the design stage.

Therefore, different types of methods of stall margin improvement are utilized, for instance, tip injection and casing treatment. Casing treatment is a typical passive control of flow instability, and require to tailor the shroud or hub casing with the addition of circumferential grooves or axial slots. Generally the perforation ratio is up to more than 50%, and flow loss is generated because of the permanent interaction between the flow in the main stream and inside the grooves or slots.

In contrast, during recent years tip air injection attracts more and more research interest, because this method will not lead to a great perforation ratio at casing. A few small nozzles are installed around the annulus, and less flow interaction is induced. The main disadvantage of this method is caused by the energy loss of the injection flow. When the compressor is operating far away from the stall inception point, the injection valve can be closed, and the injection is activated only when the compressor is approaching the stall boundary. Therefore, in order to further decrease the efficiency loss, many research have been conducted by active tip injection [1-5] and micro tip injection [6, 7] to minimize the amount of the tip injection flow.

Pullan et al. [8] conducted a numerical study and summarized that there are two important stalling mechanisms which are related to the inception of rotating stall, i.e., tip leakage flow in blade tip region, and flow blockage and separation inside blade passages. So it is inferred that the stall margin improvement can be obtained in a specific compressor if the different stalling mechanism is suppressed.

The investigation of Nie et al. [7] on a low speed compressor showed that micro tip injection can postpone the forward movement of the flow interface between main flow and tip leakage flow. Actually, Vo et al. [9] stated that the spillage of the interface is the significant features when the compressor is approaching to stall inception of spike-type.

In conclusion, in order to make a prediction of the stall boundary, it is of great significance to compute the movement of the flow interface between tip clearance flow and the main flow. However, during the design stage of a new compressor, how to make a quick judgement on the position of the flow interface at blade tip is still an unanswered question. Firstly, this paper will make a theoretical analysis, and simplified the three-dimensional unsteady flow problem as an unsteady two-dimensional flow simulation. Secondly, a three-dimensional computation is also performed to verify the mechanism of stall margin improvement by the micro tip injection.

2. Simplified Theoretical and numerical Model of the Tip Clearance Flow

2.1 Theoretical model

Since a variety of complicated physical phenomenon with different time and length scales are involved in the tip clearance flow, it is not straightforward to simulate the detailed evolution of flow at blade tip. Therefore, this section firstly theoretically discuss the basic principle of the tip clearance flow, on the basis of which a simplified two-dimensional unsteady CFD simulation is conducted.

Chen et al.[10] summarized that the tip clearance flow in a compressor can be approximated as mainly inviscid, and flow compressibility plays a non-significant role in determining the tip clearance vortex formation. Other experimental and numerical study [11] also reveal the inviscid nature of the tip clearance flow. So the viscosity is neglected in the present paper, because focus is placed on qualitatively analysis the role of tip clearance flow in the inception period of AR. Additionally, in light of the basic idea of dual time step approach for unsteady flow simulation, the major physical parameters, which determine the steady flow filed, are sure to strongly influence the evolution of the corresponding unsteady flow field.

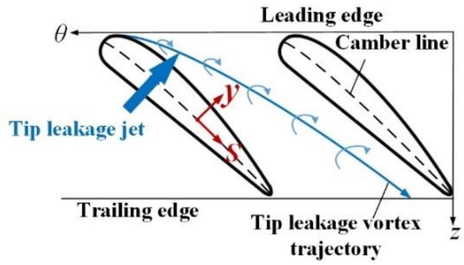

Fig. 1 Sketch of Tip Leakage Flow and the Curvilinear Coordinate

In order to find the main parameters of tip clearance flow, in the present paper we applied three-dimensional, incompressible, steady Euler equations in curvilinear coordinates, which is fixed on a rotor blade. As shown in Fig. 1, *s* and *r* represent the camber line (streamline) and radial coordinates, respectively, and *y* coordinate is normal to the camber line.

$$\frac{R}{R+y}\frac{\partial v_s}{\partial s}+\frac{\partial v_y}{\partial y}+\frac{v_y}{R+y}+\frac{\partial v_r}{\partial r}=0 \quad (1)$$

$$\frac{R}{R+y}v_s\frac{\partial v_s}{\partial s}+v_s\frac{\partial v_y}{\partial y}+v_r\frac{\partial v_s}{\partial r}+\frac{v_s v_y}{R+y}=-\frac{R}{R+y}\frac{\partial p}{\rho\partial s} \quad (2)$$

$$\frac{R}{R+y}v_s\frac{\partial v_y}{\partial s}+v_y\frac{\partial v_y}{\partial y}+v_r\frac{\partial v_y}{\partial r}+\frac{{v_s}^2}{R+y}=-\frac{\partial p}{\rho\partial y} \quad (3)$$

$$\frac{R}{R+y}v_s\frac{\partial v_r}{\partial s}+v_y\frac{\partial v_r}{\partial y}+v_r\frac{\partial v_r}{\partial r}=-\frac{\partial p}{\rho\partial r} \quad (4)$$

where *R(s)* is the radius of the curvature of the rotor camber line, *v* is the velocity, *p* is the pressure, ρ is the density. The streamwise velocity is decomposed into a uniform component and a small variation component,

$$u(s,y,r) = \bar{u}(s) + u'(s,y,r) \qquad (5)$$

The length scales in the streamline direction and other directions are the blade chord and tip clearance, respectively. If we suppose *R(s)* is much longer than the blade chord, and the tip clearance is much smaller than the blade chord, then the dominant terms of Eq. (1)-(4) can be formulated as followings.

$$\bar{u}\frac{d\bar{u}}{ds} = -\frac{1}{\rho}\frac{\partial p}{\partial s} \qquad (6)$$

$$ds = \bar{u}dt \qquad (7)$$

$$\frac{\partial v_y}{\partial y} + \frac{\partial v_r}{\partial r} = 0 \qquad (8)$$

$$\frac{\partial v_y}{\partial t} + v_y\frac{\partial v_y}{\partial y} + v_r\frac{\partial v_y}{\partial r} = -\frac{1}{\rho}\frac{\partial p}{\partial y} \qquad (9)$$

$$\frac{\partial v_r}{\partial t} + v_y\frac{\partial v_r}{\partial y} + v_r\frac{\partial v_r}{\partial r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} \qquad (10)$$

As stated by Chen et al. [10], it is found that the tip clearance flow is decoupled into two parts of flow filed, i.e., the cross flow takes the form of Eq. (6) and (7), and the through flow is described by Eq. (8)-(9). It is also concluded that the streamwise formation of the tip clearance flow can be considered as the evolution of a two-dimensional unsteady flow in the cross plane. In other words, the unsteadiness of a jet flow in two-dimensional plane can capture some main characteristic of the tip clearance flow. So a simplified URANS computation is conducted. It is obviously that the jet velocity is the most significant parameter for the dynamics of jet flow through a small tip gap.

2.2 Numerical Calculation

A four-stage high speed compressor at the Institute of Turbomachinery and Fluid Dynamics, University of Hannover is investigated. The nominal rotor speed is 17100 rpm, the design mass flow is 7.82 Kg/s, and the total pressure ratio is about 2.7. More details of this compressor is referred to Hellmich and Seume [12].

In the present paper, we focus on the tip clearance flow at the third rotor tip. Firstly a three-dimensional steady CFD calculation is performed for one single passage, and then the biggest velocity of the jet flow at the blade tip is obtained based on the circumferential mass-averaged value. Finally a simplified two-dimensional calculation is conducted to reproduce the trajectory of the tip clearance vortex.

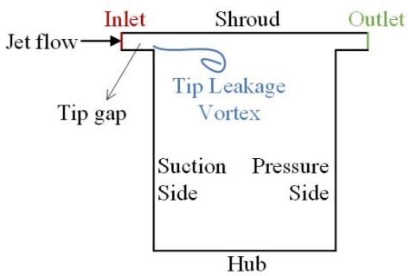

Fig. 2 Sketch of a Simplified Two-dimensional Jet Flow

Figure 2 shows a sketch of the simplified two-dimensional model of the tip clearance flow. The jet flow velocity at the inlet is prescribed as the relative speed at a point A in the strongest reverse flow region in the tip gap of R3. The pressure at the outlet is prescribed as the static pressure at the same point A. As shown in Fig. 3, the total number of the discretised nodes on this plane is approximately 40 thousand.

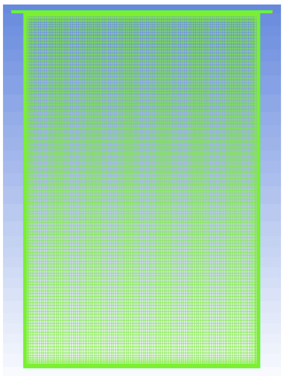

Fig. 3 Distribution of discretized nodes for unsteady CFD

In the unsteady flow simulation, the physical time step T_p is prescribed as 5×10^{-7} s, and 100 inner iteration steps are calculated in each physical time step in order to obtain a physically convergent solution. Fig. 4 displays the time evolution of the contour of the static pressure on the plane. An obvious TLV marked by yellow color is moving downstream. Then the trajectory of tip clearance flow can be drawn, and the compressor is approaching to stall inception if the trajectory is aligned with the leading edge of the rotor. However, it is still required to consider the effect of micro tip injection.

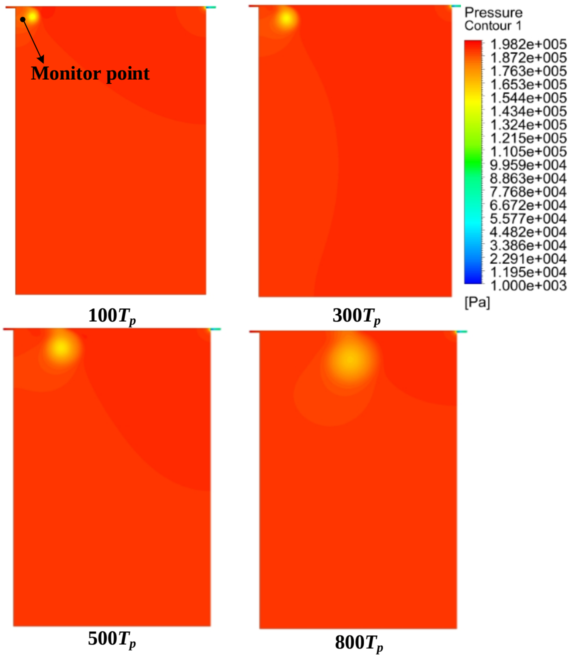

Fig. 4 Time evolution of the contour of the static pressure on the plane

3. Numerical Simulation of a Single Rotor with Micro Tip Injection

In order to clarify the unsteady mechanism of stall margin improvement by the micro tip injection, a three-dimensional CFD simulation is conducted on the first rotor of a low-speed, three stage axial compressor test rig (see Fig.5 [7]). The design mass flow and rotational speed are 2.6 Kg/s and 2400 rpm, respectively. More details about this compressor is referred to the work of Nie et al. [7] and Tong et al. [13], which showed that micro tip injection can lead to stall margin improvement of more than 4.0%.

Fig. 5 Low-speed three- stage compressor test rig (Nie et al. [7]).

The steady, three-dimensional flow field of one single passage is computed for this rotor. The solution algorithm is on the basis of central discretization, 4-stage Runge-Kutta scheme, and Spalart-Allmaras turbulence model. Local time stepping technique and multi-grid are used for efficient convergence.

One injector, which is modelled as a small cuboid (see Fig. 6), is installed in front of the rotor. The right side of the injector is the outlet of tip injection, and all the other sides of the cuboid are prescribed as solid boundaries. The distance between rotor leading edge and injector outlet is 8.6% chord. Constant total temperature and total pressure with axial incoming flow velocity are prescribed at the outlet of the injector and the inlet of the main passage.

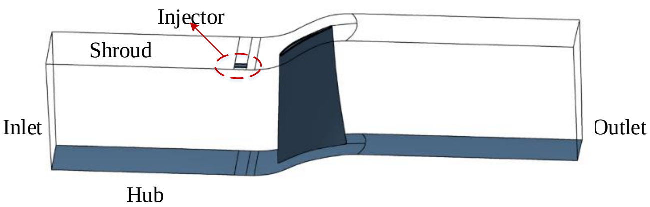

Fig. 6 Schematic of single rotor with tip injector (Liu et al. [14]).

The radial penetration of injector center line compared to shroud is approximately 0.80% span (about double tip gap size). To illustrate the relative size of radial height of the injector compared to the boundary layer thickness on the casing just upstream of it, Fig. 7 below shows the radial distribution of nondimensional absolute velocity, and the two green solid lines shows the span-wise height of injector.

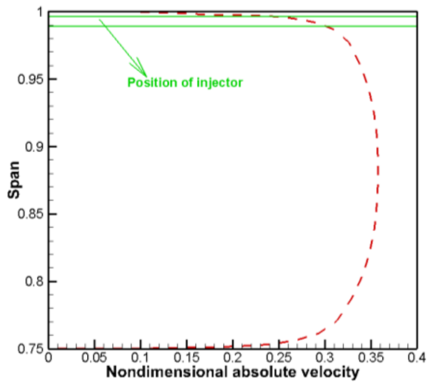

Fig. 7 Radial distribution of nondimensional absolute velocity.

The distribution of axial velocity below the blade tip slightly is shown in Fig. 8, it is found that high speed injection of 60 m/s can be generated at the injector outlet. The mass flow of tip injection for each injector is 0.13% of design mass flow of the rotor, if four injectors are used around the annulus.

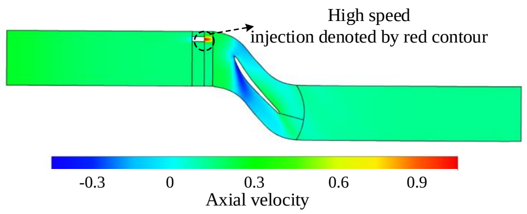

Fig. 8 Contour of axial velocity slightly below blade tip (Liu et al. [14]).

The radial distribution of relative total pressure in front of rotor leading edge (FRONT) and behind the trailing edge (BEHIND) are compared in Fig. 6. It is found that the relative total pressure in most of the spanwise region is almost not influenced by the micro tip injection, and there is only a very small increase of relative pressure near the tip gap. So the micro injection does not significantly modify the radial load distribution. It is therefore inferred that the high speed injection influences the unsteady evolution of the stall inception. An unsteady CFD simulation on the full annulus is required to verify this conclusion, though it is very time-consuming.

In this investigation, the boundary of a contour of negative axial velocity at blade tip region is used to clarify the interface between tip clearance flow and main flow. Fig. 9 shows that the spillage of the tip leakage flow is influenced by the high speed injection, although the change of the relative total pressure is just slightly influenced by the micro injection. Some researchers [15] [16] reported that a compressor starts to stall until the area of flow field, which is under the influence of tip injection, is not stable. So it is concluded that the high speed micro injection can postpone the stalling process, since the trajectory of tip clearance flow in some specific passage is pushed backward when the injector is just in front of the blade passage.

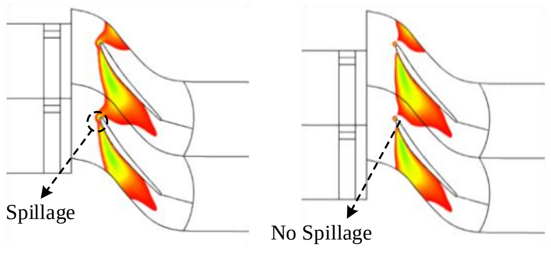

Fig. 9 Contour of negative axial velocity at blade tip.

4. Conclusions

Two methods are used in this investigation in order to provide a prediction of the stall margin improvement of an axial compressor by micro tip injection. Since the total amount of injection flow is extremely small, the compressor response on the injected flow is inferred to be unsteady.

In order to provide a quick prediction of the stall inception point, a simplified model for the calculation of the trajectory of the tip clearance vortex is proposed. A two-dimensional unsteady CFD calculation is used to reproduce the evolution of the formation of the tip clearance. It is found that this model could be further improved and used as a criteria of the stall inception. However, it is still required to find a method of modelling the effect of micro tip injection in this two-dimensional model.

A three-dimensional CFD simulation is performed, and it is found that the interface between the tip clearance flow and the main flow is periodically pushed backward inside the blade passage. An unsteady simulation of the full annulus is needed in the future work to clarify more details about the flow mechanism of stall margin improvement.

Acknowledgments

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