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Efficient Performance Prediction of a one-and-half Stage Axial Compressor based on a Streamline-Curvature Throughflow Method and Stage Characteristics

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ABSTRACT

The blade modeling of axial compressor mostly adopts curved and swept design, which results in strong three-dimensional unsteady effects in the compressor. Experiment is the main method to obtain the flow field and characteristics of compressors, but it needs too much time and money. Compared with experiments, throughflow calculation and Computational Fluid Dynamics (CFD) save the cost in the design stage greatly. With the increase of compressor load, compressor blades become lighter and thinner, which leads to the blades deviating from the design profile. The manufacturing deviation may influence the aerodynamic performance and stability boundary of compressors. The key issue is how to predict the influence of manufacturing deviation on the compressor aerodynamic performance in the design stage. As the manufacturing deviations are random, it will occupy a huge amount of computing resources if we add deviations to CFD models directly. To solve this problem, we propose a method based on stage data, which is expected to predict the impact of manufacturing deviations on the aerodynamic performance of the compressor quickly. The reliable loss and deviation models based on stage data can predict the spanwise flow field parameters and performance of the compressor better.

INTRODUCTION

Experiments and numerical simulations are commonly used for studying the aerodynamic performance of fan/compressor. The streamline curvature method has the advantages of short convergence time and can quickly obtain compressor performance. It was proposed by Wu (Wu, 1952) first time in 1952, and has been continuously improved and developed by lots of scholars. The early streamline curvature method program was developed by Smith (Smith Jr, 1966), Novak (Novak, 1967), and Denton et al (Denton, 1978). Denton (Denton, 1978) pointed out that the accuracy of the method depends on the accuracy of the empirical model. Previous researchers have established a large amount of experimental data and obtained different empirical models.

The workload of studying the impact of manufacturing deviations on the aerodynamic performance of the fan/compressor will be extremely large. The experimental and computational costs are too high, and it is unacceptable. Although lots of empirical models have been developed, this article proposes the correlations of the loss model and deviation model based on the CFD calculation results, because it is diffecult for the traditional models to correlate the influence of manufacturing deviations with the aerodynamic performance of compressor. This method is hoped to predict the influence of manufacturing deviations on the aerodynamic performance of a specific compressor. The work of this article is to establish the framework of this method, and the influence of manufacturing deviations will be considered in the future work. The results calculated by the throughflow program show that the novel empirical model can reconstruct exactly the spanwise distribution of the loss coefficient and deviation angle.

METHODOLOGY

Streamline Curvature Method

Streamline curvature method became a critical tool in the turbomachinery design based on the throughflow theory and was widely developed and applied. To make the article concise, streamline curve method will be abbreviated as SLC. The computational domain in this article is a two-dimensional meridional projection of the cascade channel established on the S2 surface as shown in Figure 1. The computational domain consists of hub line, shroud line, streamlines, blade row leading and trailing edges. The intersections of streamlines and blade leading and trailing edges are the calculation nodes. The flow field can be obtained by solving the continuity equation, momentum equation, energy equation and state equation on the calculation nodes. The streamline curvature method has the advantages of clear physical concepts, low memory requirements, and short calculation time. To simplify the calculation, this article further simplifies the flow field, including steady flow, adiabatic, axisymmetric, and inviscid flow. The radial calculation station is set at the blade edge and the bladeless channel, and the simplified governing equation is as follows:

Complete radial equilibrium equation:

$$\frac{\partial V_m}{\partial n}=\frac{1}{V_m}\left(\frac{\partial I}{\partial n}-T\frac{\partial S}{\partial n}-\frac{V_\theta-\omega r}{r}\frac{\partial rV_\theta}{\partial n}\right)-V_m\left(\frac{\cos(\varphi+\phi)}{r_m}-\frac{\sin(\varphi+\phi)}{V_m}\frac{\partial V_m}{\partial m}\right) \quad (1)$$

Continuity equation:

$$m=\int_{r_{hub}}^{r_{tip}}2\pi r\rho V_m\cos(\varphi+\phi)dn \quad (2)$$

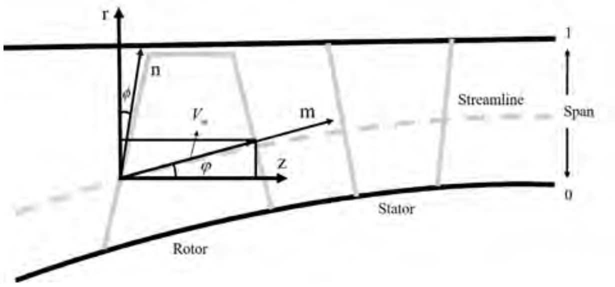


Figure 1 S2 Stream surface

The governing equations are solved by finite difference relaxation iteration. The program needs some inputs such as flow rate, rotating speed, boundary conditions and geometric information. The meridional velocity V_m is solved through the radial equilibrium equation from the root to the tip of the blade. The mass flow can be obtained by solving the continuity equation. If the flow rate error exceeds the target error of the given flow rate, modify the value of meridional velocity V_m and repeat the above solving process until the mass flow rate converges. The position of the streamline changes with the flow iteration, and the updated streamline will generate a new computational grid. The calculation starts at the inlet calculation station and ends at the outlet calculation station. When the streamline reaches the convergence condition, the aerodynamic parameters at each node and the compressor characteristics can be obtained.

Program Structure

The calculation process of this program is shown in Figure 2, with the geometric parameters and initial values set in two input files. One file is used to give the coordinates of the flow channel, and another file contains initial values including boundary conditions, node number of axial and radial direction, iteration steps, and convergence error etc.

The program includes a two-layer loop mainly, and the inner loop solves the complete radial equilibrium equation to obtain the spanwise distribution of the meridional velocity V_m. The program can calculate the distribution of other aerodynamic parameters in the flow field based on the meridional velocity and empirical models. After the inner loop ends, the program will verify the meridian velocity using the flow rate. The prediction of the aerodynamic performance of the compressor is a forward problem, which requires a loss model and a deviation model to correct the flow field. The total pressure ratio of the rotor and the angular momentum of the stators are corrected by the deviation model. The loss model is used to correct the rotor efficiency or the total pressure recovery coefficient of the stators. In the outer loop, the program will update the streamline. When the iteration residual of the outer loop reaches the convergence standard, the program stops and the compressor characteristics and the final flow field are obtained.

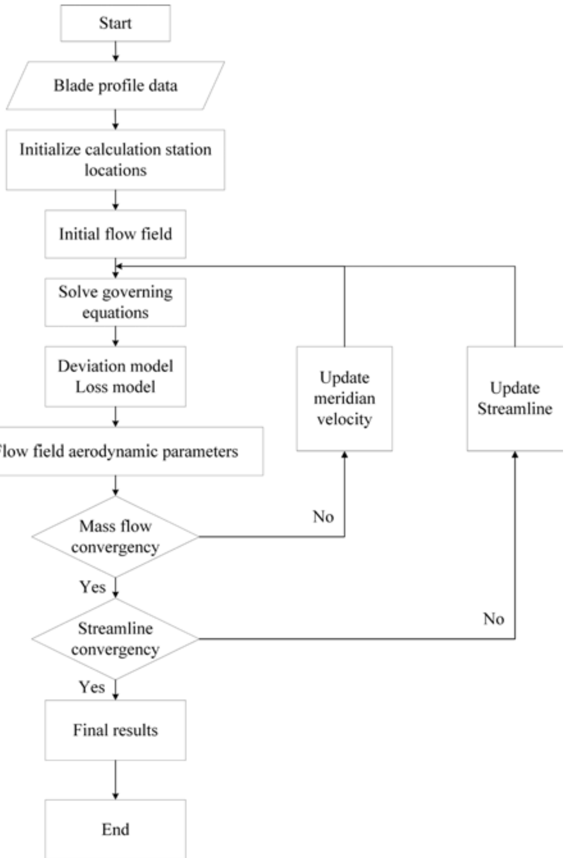


Figure 2 The flow diagram of the throughflow program

Empirical Model

Due to the inviscid and axisymmetric flow assumptions in the throughflow model, the rationality of the results solved by the streamline curvature method relies on accurate loss model and deviation model. Traditional empirical models need to consider the effects of three-dimensional and viscosity in real flow as much as possible. The designers have to try the empirical models by trial and error and obtain the most reasonable model combination. For example, the deviation model usually includes the minimum-loss deviation angle representing the performance of the design point, correction of the attack angle after falling from the minimum attack angle, the changing of deviation angle caused by the three-dimensional effects such as tip leakage flow and end wall secondary flow, etc. The total pressure loss model usually includes profile loss, shock wave loss, and secondary loss.

The aim of the throughflow program developed in this article is to predict the aerodynamic performance of a specific compressor quickly. The loss model and deviation model of the program are both based on the three-dimensional CFD results of the target compressor.

Research Objective

The object in this article is a 1.5 stage high-speed compressor with four rows of blades, including inlet guide vanes (IGV), rotors, stators, and outlet guide vanes (OGV). The main design parameters are shown in Table 1. The three-dimensional steady-state results of the compressor were obtained by Ansys CFX, using an O4H structure mesh and a single channel model, with a total of 2.09 million mesh nodes, as shown in Figure 3. The first layer grid height is 1×10^{-5}m, which can guarantee the mesh captures the flow state in the boundary layer. The turbulence model is $k-\varepsilon$ model.

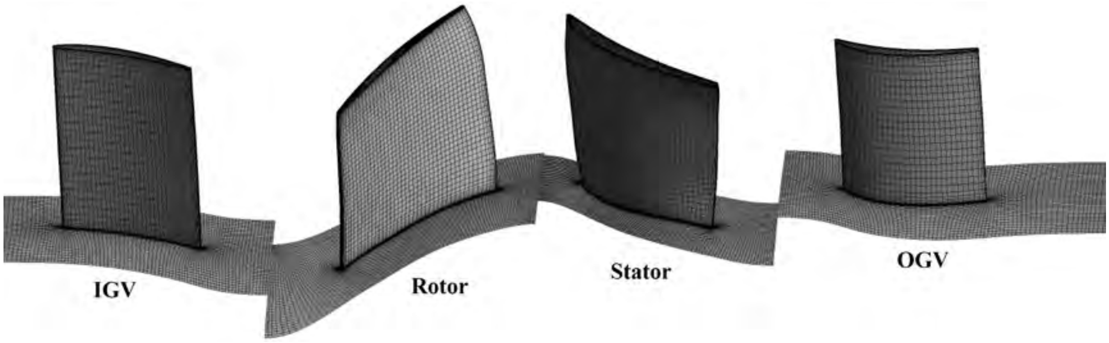


Figure 3 CFD computation mesh for the object compressor

Table 1 Design parameters of one-half compressor

Parameters	Value
Total pressure ratio	1.373
Adiabatic efficiency	0.860
Rotating velocity (r/min)	12000
Inlet tip-hub ratio	0.75
Design mass flow rate (kg/s)	10

The meridian computational domain of the compressor consists of 13 radial calculation stations and 13 streamlines, as shown in Figure 4. Four calculation stations are set upstream of the inlet guide vane and four calculation stations are set downstream of the outlet guide vane. The remaining five calculation stations are set on the meridian projection of the leading and trailing edges of the blade rows respectively.

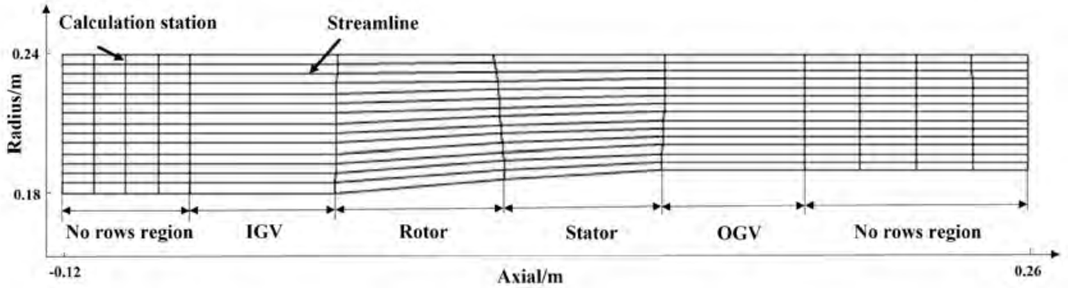


Figure 4 SLC computational domain of one-half compressor

Loss Model

There are complex secondary flow problems such as mixing and leakage in the cascade passage. Extracting the loss coefficient at the outlet of the corresponding element stage in the stage environment can reduce the complexity of modeling. The loss coefficient ω of the element stage is defined as:

$$\omega = \frac{P_{1av}^* - P_{2av}^*}{P_{1av}^* - P_{1av}} \qquad (3)$$

P_{1av}^* is the length average total pressure at the inlet of the element stage. P_{2av}^* is the length average total pressure at the outlet of the element stage. P_{1av} is the length average static pressure at the inlet of the element stage. In the meridional surface, there are 13 calculation stations in the radial direction. The loss coefficient extracted by this method includes the profile loss, secondary flow loss, shock wave loss and mixing loss caused by the three-dimensional effect in the flow field. This method simplifies the modeling process, and does not require a lot of trial and error to establish the required loss models for a specific compressor.

The inlet attack angle *i* of the element stage is defined as:

$$i = \beta_1 - \beta_{1k} \qquad (4)$$

β_1 is the inlet flow angle of the element stage (the angle between the inlet airflow direction and the axis), and β_{1k} is the inlet geometric angle of the element stage.

The inlet Mach number is defined as:

$$Ma = \frac{V_{axial}}{c} \qquad (5)$$

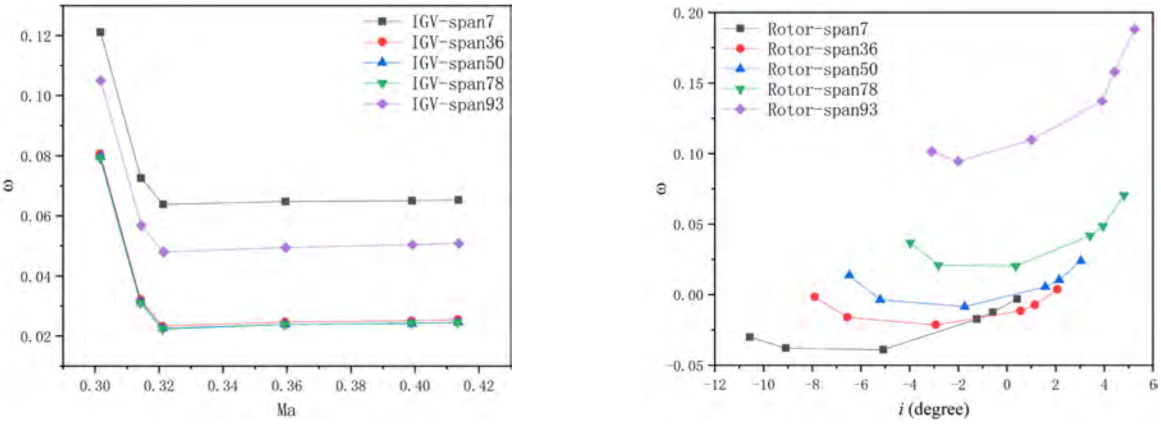
V_{axial} is the axial velocity at the inlet of the element stage, and *c* is the local sound velocity.

In order to establish the relationship of each blade row, this article ultimately determines that the loss coefficient of the IGV is related to the inlet Mach number. The other three rows' loss coefficient is related to the inlet attack angle *i*.

$$\omega_{IGV} = f_{\omega}(Ma) \quad (6)$$

$$\omega_{R/S/O} = f_{\omega}(i) \quad (7)$$

The loss model's correlation method is polynomial fitting. The order of the correlations does not exceed four in case of overfitting. Figure 5 shows the relationship between IGV loss coefficient and Mach number, as well as the relationship between Rotor loss coefficient and inlet attack angle. The loss models are extracted for all 13 radial positions to ensure the accuracy and robustness of the throughflow program. As shown in Figure 5, the losses at the endwall of the IGV are relatively large, while the losses in the main flow region are similar to each other. The loss of the rotor is related to the inflow velocity. The loss coefficient at the blade tip is high and the parameter is low at the blade root, as the speed at the tip is the highest that always induces shock wave, resulting in a large loss coefficient. The flow velocity at the root of the blade is relatively low and the loss coefficient is small. The loss coefficient gradually increases with the increase of the spanwise.



(a) IGV loss coefficient correlation model (b) Rotor loss coefficient correlation model

Figure 5 Loss models of throughflow program

Deviation Model

Similar to the loss model, the deviation angles of the element stage in the compressor are different from the deviation angle characteristics of the two-dimension cascade experiment. As mentioned before, the deviation model is used to correct the total pressure ratio of the rotor and the outlet circulation $V_{\theta}r$ of the stator. Therefore, the accuracy of the deviation model will determine the reliability of the compressor characteristics calculated by the throughflow program. The deviation angle δ is defined as:

$$\delta = \beta_2 - \beta_{2k} \qquad (8)$$

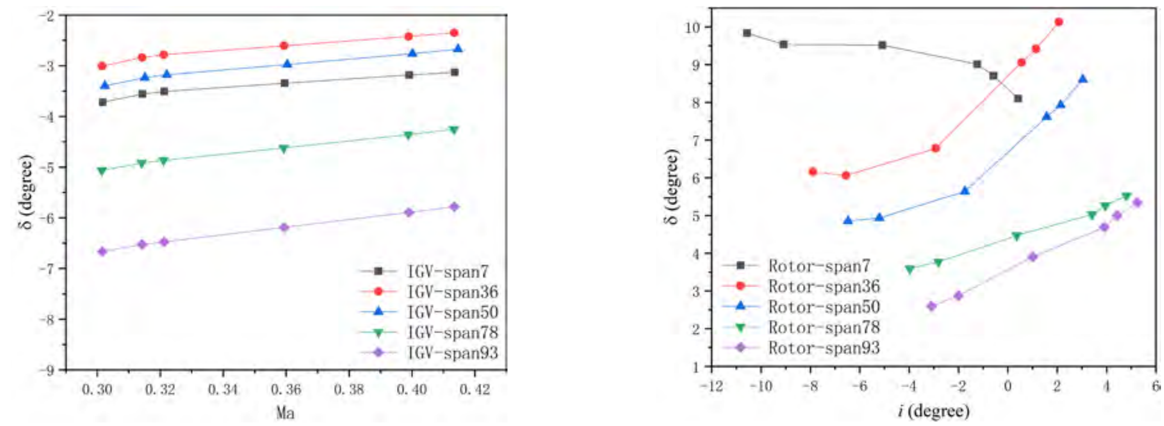
β_2 is the outlet flow angle of the element stage (the angle between the outlet airflow direction and the axis). β_{2k} is the outlet geometric angle of the element stage. Similar to the modeling approach of the loss model, the deviation angle of the IGV is related to the inlet Ma. The deviation angles of the other three rows of blades are related to the inlet attack angle *i*.

$$\delta_{IGV} = f_{\delta}(Ma) \quad (9)$$

$$\delta_{R/S/O} = f_{\delta}(i) \quad (10)$$

The deviation model's correlation method is also polynomial fitting, with a fitting order not exceeding four in case of overfitting. As shown in Figure 6, the deviation angle of the IGV and rotor exhibits a certain regularity. The deviation angle increases with the inlet Ma and the attack angles.

According to the discussion above, the loss model and deviation model based on the stage environment only need the loss coefficient and deviation angle at the corresponding position, as well as the attack angles and Ma from the CFD results. Compared with the traditional loss models and deviation models, this method reduces lots of workload and trial and error costs.



(a) IGV deviation correlation model (b) Rotor deviation correlation model

Figure 6 Deviation models of throughflow program

RESULTS AND DISCUSSION

Design Points and Overall Characteristic Verification

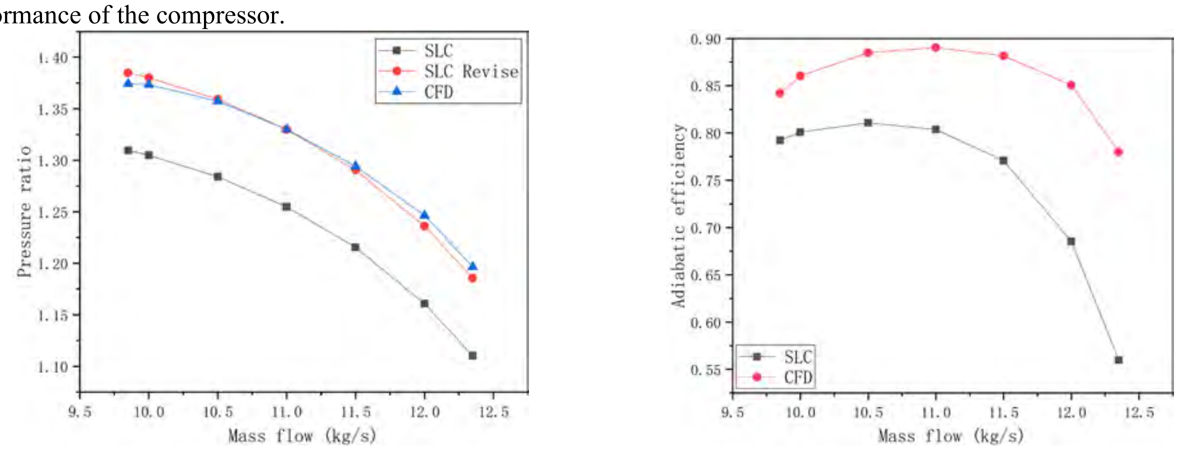
The results of the throughflow program are compared with the CFD results at the design point, verifying the accuracy of the empirical models. The comparison results are shown in Table 2. CFD represents the results calculated by the Ansys CFX at the design point. SLC represents the results calculated by the streamline curvature method, and ERROR represents the error between the results calculated by the streamline curvature method and the CFD method, respectively. As shown in the table, the error of the total pressure ratio is 4.99% and the error of the efficiency is 6.91% at the design point. The error of total pressure ratio is less than 5%, which is acceptable in the design stage. The ultimate purpose of this program is to predict the aerodynamic performance and stability boundary of the compressor. Efficiency is not a key indicator to focus on, so it is acceptable that the efficiency has a certain error.

Table 2 Characteristics verification under design condition

	π	η
CFD	1.373	0.860
SLC	1.305	0.801
ERROR	4.99%	6.91%

The total pressure ratio and efficiency characteristics of the compressor are calculated at the design rotation speed for verifying the reliability of streamline curvature method. The obtained characteristic curves are shown in Figure 7. The loss model and deviation model established based on the CFD results are more rigorous, as the total pressure ratio and the efficiency calculated by the streamline curvature method are generally lower than the three-dimensional CFD results. The total pressure ratio calculated by the streamline curvature method can be corrected by adding a coefficient B, because the overall trend of total pressure ratio is totally consistent with the CFD results.

Although the prediction of the throughflow program cannot fully match the CFD results, it is still a critical tool in the initial design stage. The throughflow program can achieve convergence in only a few seconds, and the three-dimensional CFD case often takes several hours under the same hardware conditions. Therefore, the throughflow method has a natural advantage in quickly predicting the impact of manufacturing deviations and other instability factors on the aerodynamic performance of the compressor.



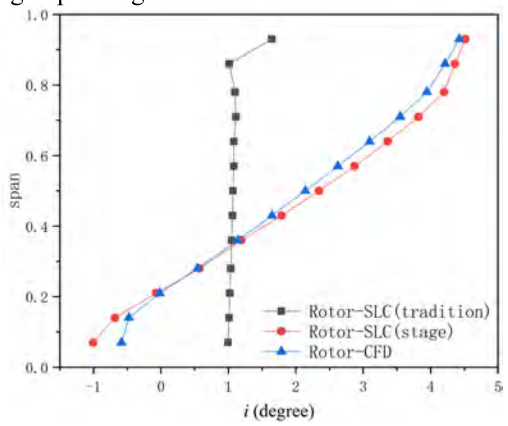
(a) Total pressure ratio (b) Adiabatic efficiency

Figure 7 Characteristic diagram of one-half stage compressor

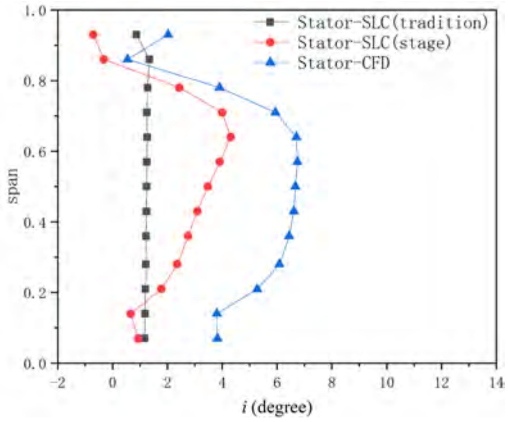
Spanwise Results under Undesigned Conditions

In off-design conditions, the reliability of the deviation model is particularly important, as the inlet boundary conditions of downstream blades are directly determined by the upstream outlet boundary conditions, which is called stage matching. If the deviation angle of the upstream row calculated by deviation model differs from the actual deviation angle a lot, it will cause the inlet attack angle of downstream blade to exceed the available attack angle range, and the solver will divergence. To verify the accuracy of the novel empirical models under off-design operating conditions, the operating point with a flow rate of 9.85kg/s, which is close to the stall boundary, is selected for specific analysis. The spanwise distribution of attack angles and deviation angles of blade rows are shown in Figure 8 and Figure 9, respectively.

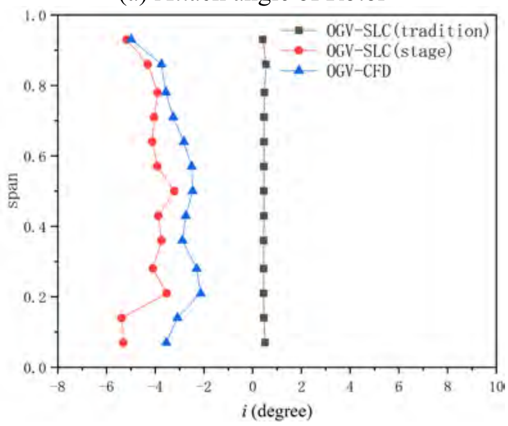
Figure 8 and Figure 9 compare the attack angles and deviation angles calculated by the Lieblein's deviation model (Lieblein, 1960) and the novel deviation model. As shown in the figures, Lieblein's deviation model reconstructs the flow angle spanwise distribution hard. While the novel deviation model can accurately calculate the outlet deviation angle under off-design operating conditions.



(a) Attack angle of Rotor

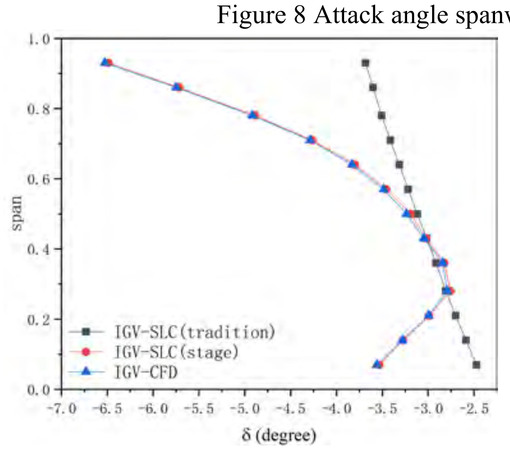


(b) Attack angle of Stator

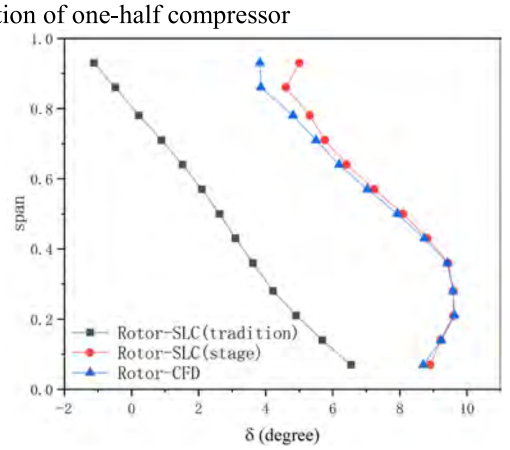


(c) Attack angle of OGV

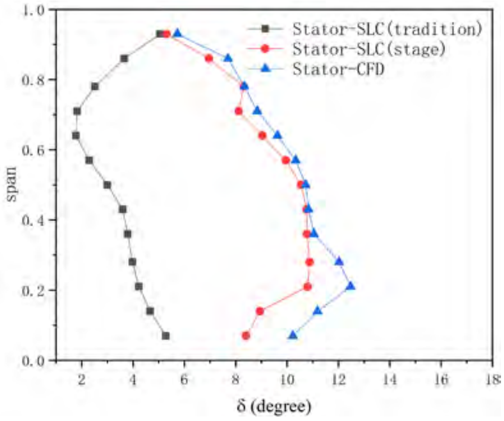
Figure 8 Attack angle spanwise distribution of one-half compressor



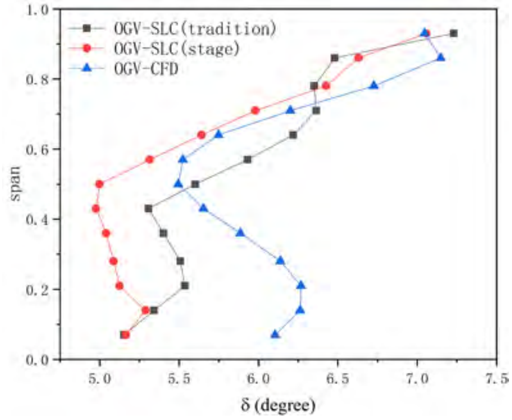
(a) Deviation angle of IGV



(b) Deviation angle of Rotor



(c) Deviation angle of Stator

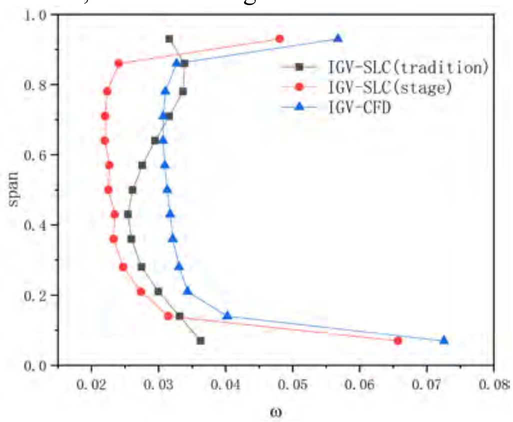


(d) Deviation angle of OGV

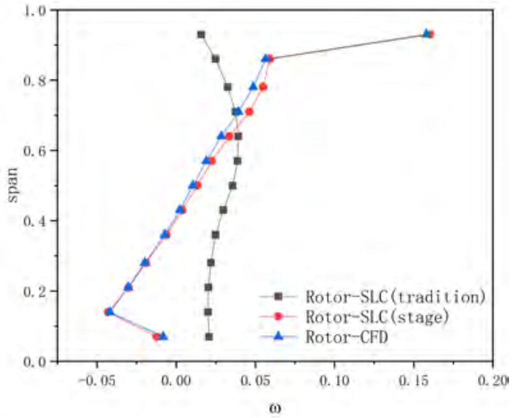
Figure 9 Deviation angle spanwise distribution of one-half compressor

The spanwise distribution of the total pressure loss coefficient is shown in Figure 10. The traditional loss model introduces profile loss (Lieblein and Roudebush, 1956) and secondary flow loss (Grieb, 1973) in this paper. As shown in the Figure 10, the traditional loss model lacks the ability to reconstruct the spanwise distribution of the loss coefficient. As the velocity near the blade tip of the rotor is transonic speed just like Figure 11(a), a shock wave forms at the leading edge of the profile, which causes shock loss inevitably. However, as the traditional loss model does not consist of a shock wave loss model, it is hard to predict the loss on the rotor blade tip. The novel loss model can reconstruct the rotor's loss coefficient spanwise distribution accurately.

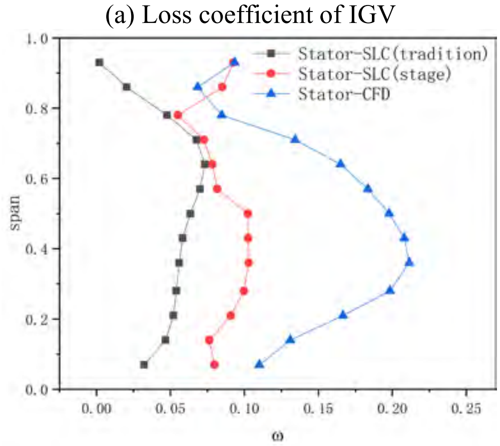
Figure 11(b) shows that severe boundary layer separation appears at the suction surface of the stator's middle profile. The air flow is severely mixed at the outlet of the element stage. The novel loss model can predict the distribution law of the stator loss, and the mixing model needs to be added later to correct the novel loss model.



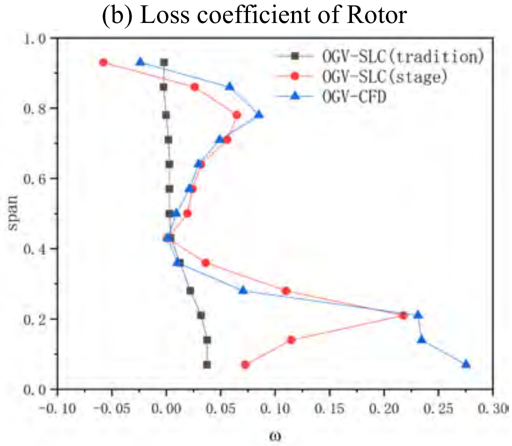
(a) Loss coefficient of IGV



(b) Loss coefficient of Rotor

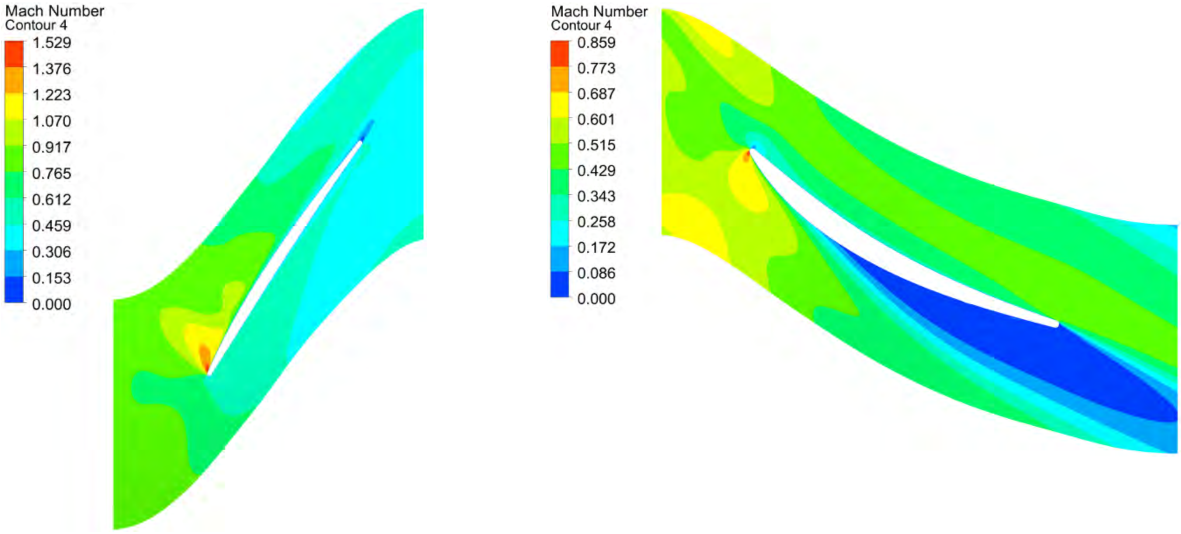


(c) Loss coefficient of Stator



(d) Loss coefficient of OGV

Figure 10 Loss coefficient spanwise distuibution of one-half compressor



(a) Rotor(span93) (b) Stator(span50)

Figure 11 Contour of Mach number

CONCLUSIONS

This paper proposes a novel modeling method for the streamline curvature method The data used for modeling is from the three-dimension CFD results. The accuracy of the novel modeling method has been verified. The conclusions are as follows:

(1) The loss coefficient and deviation angle of the IGV can be related with the inlet Mach number when the air enters the inlet axially. The other three rows' empirical models can be related to the attack angle with a polynomial correlation correspondingly, which can be used to correct the loss coefficient and deviation angle at the outlet.

(2) The deviation model and loss model are more stringent, because the total pressure ratio and efficiency characteristics calculated by the throughflow program are generally lower than the three-dimensional CFD results. The characteristic lines calculated by the throughflow program is consistent with the CFD results. The total pressure characteristic line can be corrected linearly. In addition, the throughflow program only takes more than ten seconds to calculate an example, which is much lower than the time required for RANS. Therefore, it is suitable to provide reference for designers in the early stages of design.

(3) The ability of traditional loss model and deviation model to reconstruct aerodynamic parameters' spanwise distribution is limited, and the prediction accuracy of the endwall is low. The novel empirical models developed in this article can better capture the loss coefficient and deviation angle. At near stall condition, the boundary layer of the stator's middle profile is seriously separated, leading to severe mixing effect. The stator's loss model developed in this paper accuracy is little poor, and the mixing model needs to be added to correct the novel loss model.

(4) The method developed in this article can quickly predict the aerodynamic performance of a specific compressor. This method is hoped to assess the impact of manufacturing deviations on the aerodynamic performance of specific compressors.

NOMENCLATURE

V	=	velocity
I	=	total enthalpy
S	=	total entropy
T	=	temperature
ω	=	angular velocity
r	=	radius
n	=	quasi-normal direction
m	=	streamline tangential direction
φ	=	angle between streamline and axial direction
ϕ	=	angle between quasi-normal direction and radial direction
ρ	=	air density
IGV	=	inlet guide vane
OGV	=	outlet guide vane
ω	=	loss coefficient

Subscript/supscript

m	=	streamline tangential direction
n	=	quasi-normal direction
rip	=	blade tip
hub	=	blade root
1	=	inlet
2	=	outlet
k	=	geometry
av	=	averaged
*	=	stagnation quantities
axial	=	axial direction
θ	=	meridional direction

P	=	Pressure
i	=	Attack angle
δ	=	Deviation angle
β	=	Flow angle/geometrical angle
Ma	=	Mach number
π	=	Total pressure ratio
η	=	Adiabatic efficiency
c	=	Sound velocity

References

Denton, J. (1978). Throughflow Calculations for Transonic Axial Flow Turbines, *Journal of Engineering for Power,* 100(2), pp. 212-218

Lieblein, S. (1960). Incidence and Deviation-Angle Correlations for Compressor Cascades, *Journal of Basic Engineering,* 82(3), pp. 575-584

Lieblein, S. and Roudebush, W. H. (1956). Theoretical Loss Relations for Low-Speed Two-Dimensional-Cascade Flow, *NACA Tech*, pp. 1-45

Novak, R. (1967). Streamline Curvature Computing Procedures for Fluid-Flow Problems. *Journal of Engineering for Gas Turbines and Power,* 89(4), pp. 478-490

Grieb, H. R. . (1973). Secondary Flow Loss in Axial Compressor. NATO: Advisory Group for Aerospace Research and Development.

Smith JR, L. (1966). The Radial-Equilibrium Equation of Turbomachinery, *Journal of Engineering for Power,* 88(1), pp. 1-12.

Wu, C.-H. (1952). A General Theory of Three-Dimensional Flow in Subsonic and Supersonic Turbomachines of Axial, Radial, and Mixed-Flow Types. *Transactions of the American Society of Mechanical Engineers,* 74 (8), pp. 1363-1380.