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Blade bowing effects on radial equilibrium of inlet flow in axial compressor cascades

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Abstract The circumferentially averaged equation of the inlet flow radial equilibrium in axial compressor was deduced. It indicates that the blade inlet radial pressure gradient is closely related to the radial component of the circumferential fluctuation (CF) source item. Several simplified cascades with/without aerodynamic loading were numerically studied to investigate the effects of blade bowing on the inlet flow radial equilibrium. A data reduction program was conducted to obtain the CF source from three-dimensional (3D) simulation results. Flow parameters at the passage inlet were focused on and each term in the radial equilibrium equation was discussed quantitatively. Results indicate that the inviscid blade force is the inducement of the inlet CF due to geometrical asymmetry. Blade bowing induces variation of the inlet CF, thus changes the radial pressure gradient and leads to flow migration before leading edge (LE) in the cascades. Positive bowing drives the inlet flow to migrate from end walls to mid-span and negative bowing turns it to the reverse direction to build a new equilibrium. In addition, comparative studies indicate that the inlet Mach number and blade loading can efficiently impact the effectiveness of blade bowing on radial equilibrium in compressor design.

1. Introduction

To control the secondary flows in turbine machines, the concept of blade bowing was published in the 1960s by Deich et al.[1] Since then, experimental and numerical investigations of bowed blades were carried out in both compressors and turbines.[2–11] Blade bowing has been proved to be an effective design method to control corner separation and reduce secondary flow loss in design.

The experimental studies of Breugelmans[2] and Shang[3] et al. showed that blade bowing and lean strongly influenced the development of the secondary flows in rectilinear compressor cascades. For bowed blade, the corner stall was reduced at both suction/end-wall surface corners, while flow losses were increased at the mid-span. Weingold et al.[4] showed that bowed stator generated radial forces on the flow within the blade

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passage, which reduced the diffusion rate in the suction surface corner, and substantially delayed or eliminated the formation of corner separation. Fischer et al.[5] studied the effects of strongly bowed stator vanes on the performance of a four-stage high-speed compressor. Results showed that the bowed stators eliminated the corner stall at hub and leaded to the increase of overall static pressure rise, total pressure rise, and overall efficiency, when the compressor was highly loaded. Chen et al.[11] proposed a partial bowed rotor blade for a newly designed high loaded axial fan. Numerical studies presented that the separated flow was reduced, the efficiency of the fan with bowed blades was enhanced by 1.44%, and the static pressure rise was increased by 11% under design condition.

As mentioned above, most of the former studies focused on the improvements of the corner flow and total performance achieved by using bowed blades while the inlet flow variation was neglected. Although the change of the stacking line in bowed blades may lead to complicated interaction of different flow parameters, the inlet flow condition is believed to be the key factor as the element profile remains the same. However, in recent years, some researchers have noticed the inlet flow migration induced by sweep and blade bowing. The experimental results of McNulty et al.[12] showed that more flows were pulled towards the blade tip in forward swept rotors where higher inlet axial velocity was discovered. As a result, the leakage flow blockage and the aerodynamic loading were reduced in the tip region. Gallimore et al.[13] presented the computational fluid dynamics (CFD) results of a single stage axial compressor, which indicated that both the velocity profile and the incidence angle changed near the leading edge (LE) with the application of sweep and dihedral. The numerical study of Ramakrishna and Govardhan[14] clearly reported that lower incidences were received in a subsonic axial compressor as the blades were forward swept, leading to the mass flow rate variations. However, the authors considered the stagger angle as the key factor of this phenomenon. Gabriele et al.[15] described the pressure distribution upstream the blade passage in a straight cascade with the lean angle of 20°. He found that the pressure levels at two end walls were quite different when the blade was positively leaned. A radial pressure gradient upstream the LE was observed and thrust the flows from hub to tip. Tan et al.[6–8] studied the effects of blade bowing on the aerodynamic performance of highly loaded turbine cascades and showed that the incidence angles were different in the cascades with different bow angles, but the mechanism of the inlet flow change was not described.

Through the designing of multistage compressors with the application of bowed and swept blades, it is very common to have blade loading variation along the span, which results in the different requirements of the incidence characteristic. So the question worth researching is how blade bowing and sweep induce the inlet flow variety and how to use this effectiveness to obtain the maximum benefits of the aerodynamic performance. On account of the complicacy of 3D flows, dimensionality reduction is preferred by some researchers as an effective approach to understand the mechanism of blade bowing and sweep quantitatively.

Based on this idea, Gui et al.[16] firstly discovered the relevance between the inlet flow migration and the circumferential fluctuation (CF) in axial fans/compressors with sweep using circumferentially averaged method. He attributed the incidence angle change and the sweep characteristics to the new radial equilibrium established by the inlet CF in the subsonic compressors/fans. The previous works of Chang et al.[17] proved the effects of the inlet CF on the inlet flow migration in different swept cascades. The study indicated that blade sweep did affect the inlet radial equilibrium and the inlet flow field varied mainly due to the combined effects of the radial pressure gradient and the CF source components.

As another three-dimensional blading concept, blade bowing is convinced to have significant influence on the blade inlet flow like sweep does in axial compressors, and the inlet CF should play an important role in the rebuilding of the inlet radial equilibrium. This paper presents a numerical study of simplified cascades to investigate the mechanism of the inlet equilibrium variation in bowed cascades. The circumferentially averaged method introduced in Ref. 16 is also used here to obtain the inlet CF source.

2. Blade inlet circumferential fluctuation

Lots of factors may contribute to the inlet CF in compressors, such as the inlet distortion and the wake interference from upstream. But for a single blade row with uniform incoming flow, the inlet CF can be considered as the extending of the CF within the blade passage. The circumferential pressure fluctuation acts on the flow as inviscid blade force inside the blade passage. The inviscid blade force turns to zero before the LE, but the pressure fluctuation remains nonzero. Fig. 1 presents the pressure distribution of a single blade row and it can be noticed that the inlet flow is circumferentially nonuniform. So the inlet CF is an inherent characteristic of the geometrically asymmetric blade rows, and it can be represented by the CF source items obtained with the circumferentially averaged method.

For inviscid flow in rotational cylindrical coordinates, the radial, circumferential and axial components of the Navier-Stokes momentum equations are expressed as follows, eliminating volume force:

$$\begin{cases} \frac{\partial(r\rho w_r^2)}{\partial r} + \frac{\partial(\rho w_r w_u)}{\partial \varphi} + \frac{\partial(r\rho w_r w_x)}{\partial x} = \rho(w_u + \omega r)^2 - r\frac{\partial p}{\partial r} \\ \frac{\partial(r\rho w_r w_u)}{\partial r} + \frac{\partial(\rho w_u^2)}{\partial \varphi} + \frac{\partial(r\rho w_u w_x)}{\partial x} = -\rho w_r(w_u + 2\omega r) - \frac{\partial p}{\partial \varphi} \\ \frac{\partial(r\rho w_r w_x)}{\partial r} + \frac{\partial(\rho w_u w_x)}{\partial \varphi} + \frac{\partial(r\rho w_x^2)}{\partial x} = -r\frac{\partial p}{\partial x} \end{cases} \tag{1}$$

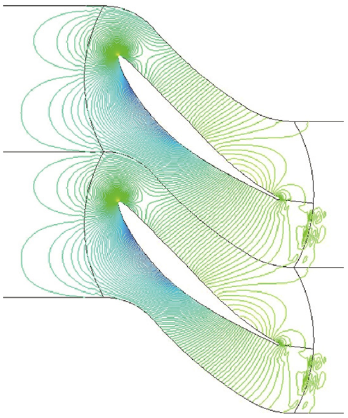

Fig. 1 Pressure distribution of single blade row.

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