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Research paper

Aerodynamic design of a multi-stage industrial axial compressor

Jinhuan Zhang, Zhenggui Zhou*, Hui Cao, Qi Li

Jiangsu Province Key Laboratory of Aerospace Power Systems, Nanjing University of Aeronautics and Astronautics, Nanjing 210016, China

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A B S T R A C T

Axial compressors are widely used in industrial applications; high efficiencies and large operating ranges are main performance requirements for their aerodynamic designs. In this study, an aerodynamic design method for multi-stage axial compressors is proposed. The design of the first stage is based on the S1/S2 stream surface theory, and a repeated stage design method is proposed for subsequent stage designs. And in the first stage design, a multi-point optimization design is used to design the rotor and stator blade profiles for both high efficiencies and large operating ranges. A five repeated stage axial compressor with a large mass flow rate has been designed by this design method. Analysis results of the rotor and stator three-dimensional flow fields indicate that the flow is primarily limited to the S1 stream surface and the two-dimensional design method is applicable; the flow in subsequent stages possesses similar characteristics as the flow in first stage. In addition, the designed five-stage axial compressor has a total pressure ratio that approaches the design goal and a stability margin that exceeds the design goal. The novelty of this study includes two parts, one is the usage of multi-point optimization method for rotor and stator blade design, and the other is the repeated stage design method.

1. Introduction

Leading international manufacturers of industrial axial compressors include MAN Turbo and Siemens in Germany; Mitsubishi and Kawasaki in Japan; and GE Dynamics and Ingersoll-Rand in the United States. Industrial axial compressors must operate across a relatively wide mass flow rate range and offer high efficiency [1,2]. Because of their small stage pressure ratio, the passage flow can be described using the S1/S2 stream surface theory [3]; the theory subsequently formed the basis of aerodynamic design method of axial compressors. Early axial compressor designs adopted standardised blade profiles from the National Advisory Committee for Aeronautics (NACA) 65 and C4 series [2,4]. As aerodynamic design methods advanced, alternative blade profiles emerged, including control diffusion profiles [5–7] and profiles designed using inverse problem and blade optimization design methods [8–14].

In the blade optimization design methods, the optimization algorithm are combined with the computational fluid dynamics (CFD) method for the blade design, and the design process is less dependent on artificial experience. Buche, Guidati and Stoll [8] carried out an optimization design of a large-scale axial compressor to widen its operating range; results indicated a 15% increase in operating range following optimization. Comparatively, Sieverding [9] applied a breeder genetic algorithm (BGA) to optimize NACA 65 blade profile designs at various blade spans, with the objective of minimizing flow loss and maximizing the operating range for an axial compressor. These efforts resulted in significantly lower total pressure loss coefficients for the optimal blade profiles compared with the original blade profiles and an expanded operating range. Similarly, Lotfi, Teixeira and Ivey [10] optimized C4 blade profile designs (commonly used in axial fans) using a genetic algorithm (GA) and the CFD software, CFX-TASCflow, to determine the flow fields. The results indicated a 0.3% increase in optimal cascade efficiency. Zhou [11] used a complex optimization design method for the NACA 65-A10-12-10 blade profile, which significantly reduced the separation area at the profile's trailing edge. With concurrent objectives to decrease noise and increase efficiency, Li and Wang [12] adopted a similar complex design optimization method for an axial fan and Marinić-Kragić, Vučina and Milas [13] optimized 3D shape of fan vanes for an efficiency increase and a noise reduction. Kim JH, Kim JW and Kim KY [14] employed a hybrid multi-objective evolutionary algorithm (MOEA) to optimize an axial-flow ventilation fan for improving the efficiency and total pressure ratio, and the results show that the efficiency increased by 1.8% and the total pressure rise increased by 31.4%.

The present study developed a repeated stage aerodynamic design method for an industrial multi-stage axial compressor. The first stage design was based on the S1/S2 stream surface theory, subsequent stage designs replicated the first stage design. A multi-point

* Corresponding author.
E-mail address: zzgon@nuaa.edu.cn (Z. Zhou).

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Nomenclature

c_1	weight coefficient of total pressure loss coefficient at design point
c_2	weight coefficient of total pressure loss coefficient at the 1st off-design point
c_3	weight coefficient of total pressure loss coefficient at the 2nd off-design point
c_4	weight coefficient of static pressure ratio
c_5	weight coefficient of flow turning angle
f	objective function value
S1	blade to blade surface
S2	hub to tip surface
ω_1	total pressure loss coefficient at design point
ω_2	total pressure loss coefficient at the 1st off-design point
ω_3	total pressure loss coefficient at the 2nd off-design point
π_s	static pressure ratio
$\pi_{s,obj}$	objective static pressure ratio
$\Delta\beta$	flow turning angle, degree
$\Delta\beta_{obj}$	objective flow turning angle, degree

optimization method was used to design the rotor and stator blade profiles of the first stage for both high efficiencies and large operating ranges, and a repeated stage design method is proposed for subsequent stage designs. Using this method, a five-stage axial compressor with a large mass flow rate was designed. In this study, the novelty includes two parts, one is the usage of multi-point optimization method for rotor and stator blade design, and the other is the repeated stage design method.

2. The aerodynamic design method of the axial compressor

This study developed an aerodynamic design method based on the S1/S2 stream surface theory, which includes S2 stream surface through-flow design, blade profile design, stack of blade profiles and 3D flow field analysis. The design process is charted in Fig. 1. In step stack of blade profiles, the designed 2D profiles are stacked radially to form a 3D blade, and in step 3D flow field analysis, the flow field constructed by the 3D blade is simulated to exam aerodynamic performance and analyse flow characteristics. The steps S2 stream surface through-flow design and blade profile design are detailed in the following Sections 2.1 and 2.2. As shown in Fig. 2, the S1 stream surface is a blade to blade surface and the S2 stream surface is a hub to tip surface.

The 3D flow field in a compressor blade passage can be approximated as 2D flow fields in the S1/S2 stream surfaces. Flows in the 2D stream surfaces are governed by the momentum, continuity and energy equations. The flow near the blade hub and tip regions is strongly three-dimensional. It is in these regions, the main sources of errors occure when the S1/S2 surface theory is used.

2.1. S2 stream surface through-flow

Rotor and stator inlet and outlet velocity triangle distributions were determined by the S2 stream surface through-flow, which defines boundary conditions for the 2D blade profile design. The streamline curvature method and classical loss model [15–17] were used to determine the through-flow.

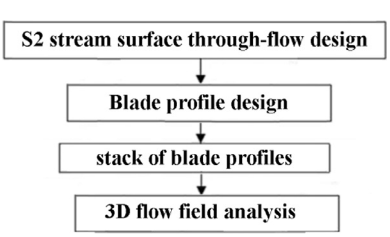

Fig. 1. Design process of the first stage.

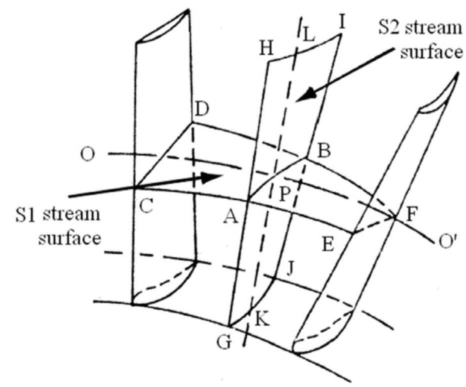

Fig. 2. S1/S2 stream surfaces.

2.2. Blade profile design

The estimated S2 stream surface through-flow guided the design of the 2D blade profiles in the S1 stream surface. The blade profile design occurred in two steps: (I) initial designs used the NACA 65 standardised profiles because of their effective low-speed performance and extensive application in subsonic axial compressors [2,4]; then (II) a multi-point optimization method was used to improve the initial profiles. Fig. 3 shows the optimization design process. And the optimization design

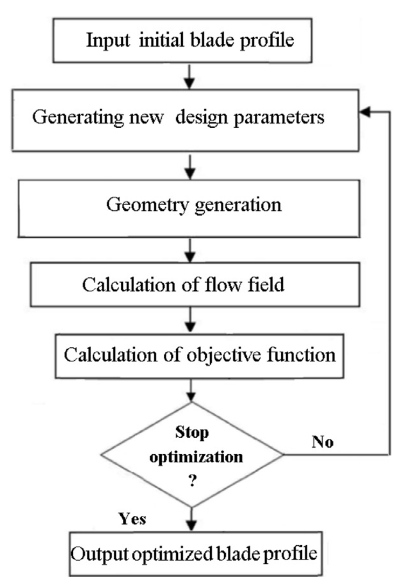

Fig. 3. Optimization design process.