Aerospace Science and Technology 54 (2016) 41–48

Contents lists available at ScienceDirect

Aerospace Science and Technology

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Effects of recessed blade tips on stall margin in a transonic axial compressor

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ARTICLE INFO

Article history:
Received 22 October 2015
Received in revised form 24 February 2016
Accepted 7 April 2016
Available online 13 April 2016

Keywords:
Stall margin
Recessed tip
Recess cavity
Axial compressor
Rotor 67

ABSTRACT

In this paper, a recessed blade tip was applied to a transonic axial compressor in expectation of increasing the stable operating range. The internal flow has been calculated by a commercial flow solver for different recess cavities on the blade tip and the predicted performances for each case have been compared with one another. It was found that the recessed blade tip with a proper depth and length is effective to increase the stall margin. The tip leakage flow became weak due to a strong vortex generated in the recess cavity so that it was pushed backward by the main flow. This is why the recess cavity can increase the stall margin without any reduction of the adiabatic efficiency at the design point. Additionally, based on the comparison of the circumferential velocity between the numerical and the theoretical results, it was found that the rim of the recessed blade tip can work as the labyrinth seal and consequently reduce the mass flow rate through the tip clearance.

1. Introduction

An axial compressor is a device to supply high pressure air to applications and is one of the major parts of a gas-turbine. A compressor is usually operated at the design point far from stall, because rotating stall causes more stress on the rotor blade. It is very important for compressor designers to avoid rotating stall in compressor operation so that much attention has been paid for finding an effective method to suppress it.

An effective method to control rotating stall in an axial compressor is the installation of axial slots on the casing. According to Takata and Tsukuda [1], skewed axial slots can improve stall margin when the orientation of the slots is opposite to the direction of the rotor revolution. It was observed in their experiment that the axial slots make the recirculation regions near the casing, in which the flow enters the rear part of the slots and emanates from the front part in the radial direction. Smith and Cumpsty [2] also observed the re-circulating flow pattern in their experiment. Greitzer et al. [3] showed in their experiments that casing treatments with slots are effective to increase stall margin in only a tip-critical rotor. Crook et al. [4] showed in their numerical simulations that the casing treatment with slots can reduce the blockage near the casing induced by the tip leakage flow. Gourdain and Leboeuf [5] found that the flow emanating from the axial slots pushes the tip leakage flow backward and provides a significant stall margin improvement.

Although circumferential casing grooves are less effective in stall margin improvement, circumferential grooves tends to exhibit less efficiency drop at the design point than axial slots. Wiseler and Beacher [6] found that a sloped trench, similar to a circumferential casing groove, provides a considerable efficiency gain without any negative effect on stall margin. Shabbir and Adamczyk [7] found that stall margin can be obtained by the radial transport of axial momentum and the increased turbulence intensity induced by the casing groove. Kim et al. [8,9] obtained an improvement in the peak efficiency and the stall margin simultaneously using an optimization for circumferential casing grooves. Haughton and Day [10], Sakuma et al. [11] and Choi [12] found that a single circumferential casing groove near the leading edge can push the tip leakage flow backward and delay the occurrence of rotating stall without any negative effect on the peak efficiency. Dinh et al. [13] applied a casing groove combined with blade tip injection and ejection to a transonic axial compressor and found that the combination is effective to improve efficiency and stall margin.

Recently, another passive method equipped with a recessed blade tip has been suggested by Gourdain and Leboeuf [5] for suppressing rotating stall in a subsonic axial compressor. It was found that the recess cavity increases turbulence intensity near the casing, which enables the flow overcome the adverse pressure. However, a recess cavity on the blade tip is not new in turboma-

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http://dx.doi.org/10.1016/j.ast.2016.04.009
1270-9638/

Nomenclature

C_c	Seal carryover coefficient	r_{seal}	Radius of seal clearance
C_r	Seal contraction ratio	SM	Stall margin
C_t	Seal throttling coefficient	SS	Suction side
LE	Leading edge	T	Static temperature
$\dot{m}_{seal}$	Mass flow rate through labyrinth seal	TE	Trailing edge
PR	Pressure ratio	Δ	Increment or decrement
PS	Pressure side	δ_c	Height of seal clearance
p	Static pressure	η	Adiabatic efficiency
p_0	Total pressure at inlet	ρ	Air density
R	Gas constant	φ	Normalized mass flow rate

Table 1
Geometric specifications.

Number of blades		22
Rotational speed (rpm)		16,043
Mass flow rate (kg/s)		33.25
Pressure ratio		1.63
Relative tip speed (m/s)		429
Running tip clearance (cm)		0.061
Inlet tip relative Mach number		1.38
Rotor aspect ratio		1.56
Rotor solidity	Hub	3.11
	Tip	1.29
Hub/tip ratio	Inlet	0.365
	Outlet	0.478

chinery. It has been widely used in axial turbines, which is also called as 'squealer tip', to reduce the total pressure loss and the heat load on the rotor tip near the leading edge [14–18]. Jung et al. [19] applied the recess cavity to a centrifugal compressor and achieved an improved efficiency in stable operating range. In their numerical study, the recessed blade tip is ineffective to increase stall margin in a centrifugal compressor.

However, a mechanism for a recess cavity in an axial compressor has not been fully understood and consequently there is no guideline for installing the cavities on the rotor tip. An extreme caution is required to install the cavities in an axial compressor, because the rim of each cavity crumbles too easily. In this work, different recess cavities with various heights and lengths have been tested using a numerical method and a detailed flow analysis attempted to find an underlying mechanism of the recess cavities.

2. Test configuration

2.1. Geometric specifications and test cases

3D numerical flow simulations were conducted to analyze the effects of a recessed blade tip on the performance in a transonic axial compressor, Rotor 67 (Table 1). The schematic diagram of the compressor is shown in Fig. 1 and the flow properties along the span have been measured at two aero-stations upstream and downstream of the rotor by Strazisar et al. [20]. The mass flow rate and the pressure ratio at the design point are 33.25 kg/s and 1.63 respectively with the rotational speed of 16,043 rpm. The number of the rotor blade is 22 and the hub–tip ratio varies from 0.365 at the leading edge to 0.478 at the trailing edge. Although the design tip clearance size of this rotor is 0.101 cm, the observed running tip clearance size is 0.061 cm according to Jennions and Turner [21] so that the running tip clearance size has been applied to the tip gap in this work. The size of the running tip clearance is about 0.38% of the blade span at the leading edge.

Firstly, the recess cavities on the blade tip with different depths but the same length were tested to find the optimal depth of the

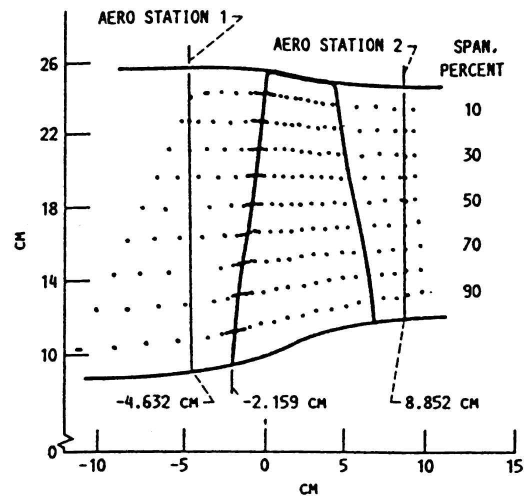

Fig. 1. Schematic diagram of compressor (Strazisar et al. [20]).

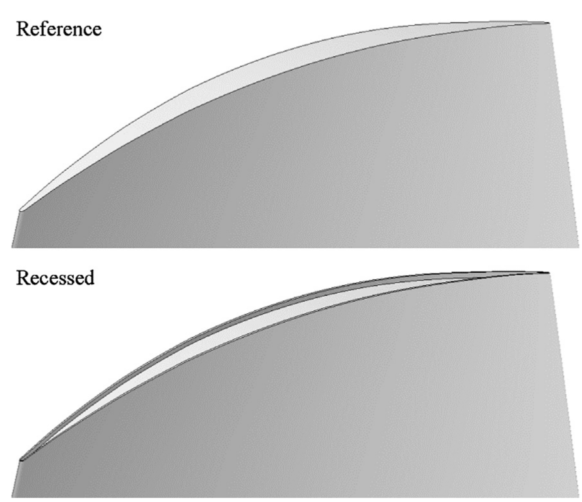

Fig. 2. Different tip configurations.

cavity. The depth was changed from 0% to 100% of the tip clearance height as shown in Fig. 2, while keeping the length of the cavity the same as the blade chord. After obtaining an optimal value of the depth, the position and the length of the cavity were changed to evaluate which parts of the cavity are important in improving the stall margin of a transonic axial compressor.

2.2. Numerical methods and computational grid

A commercial flow solver, Ansys CFX-16, was used to compute the steady flow in the transonic axial compressor with dif-