Circumferential Casing Treatment in a Transonic Fan

Hossein Khaleghi[1*] Martin Heinrich[2] Mohammad J Shahriyari[3]

[1] Department of Aerospace Engineering, Center of Excellence in Computational Aerospace Engineering, Amirkabir University of Technology, Tehran 15875-4413, Iran

[2] Institute of Mechanics and Fluid Dynamics, Technische Universität Bergakademie Freiberg, 09599 Freiberg, Germany

[3] Department of Aerospace Engineering, Center of Excellence in Computational Aerospace Engineering, Amirkabir University of Technology, Tehran 15875-4413, Iran

Abstract

This paper reports on a numerical simulation of circumferential groove casing treatment in a high-speed axial fan. Four circumferential grooves of the same geometry are located over the tip of a NASA Rotor-67 and unsteady calculations are performed from choke to near-stall. Results show that circumferential grooves reduce the incidence angle near the pressure surface at the blade leading-edge. Furthermore, the passage shock and the leakage flow are pushed rearward in the passage. It is found that circumferential grooves increase the momentum in the streamwise direction (fluid is absorbed by the grooves from their downstream part and is injected from their upstream section). The grooves also provide a flow path between the suction and pressure surface, leading to a reduction in the pressure difference between them. At the near-stall point the flow field near the grooves was found to be highly unsteady. Maximum unsteadiness was observed in the first upstream groove: the circulated mass flow rate changed as high as roughly 30 percent of its time-averaged value. As a result, in order to simulate circumferential groove casing treatment in compressors, unsteady computations are required.

Keywords

Circumferential Groove, Range Extension, Casing Treatment, Transonic Compressor, NASA Rotor-67

1. Introduction

The use of slotted or grooved casings over the tip of fan or compressor blades can substantially enhance the stable operating range in both low and high-speed compressors. Unfortunately, this is usually obtained at the expense of degradation in the compressor performance (i.e., mostly the efficiency and sometimes the pressure ratio). Different types and configurations of casing treatment have been tested or numerically simulated which are well summarized and discussed [1]. Circumferential groove casing treatment have been tested or numerically simulated by a number of researchers (e.g., [2-5]). Most of the numerical studies reported on circumferential groove casing treatment have used steady computations. Therefore, one of the main goals of the current work is to investigate the endwall unsteady flow field near the circumferential grooves.

2. Geometry

The numerical simulations were performed using the well-known NASA axial compressor Rotor 67. The rotor consists of 22 blades with a tip speed of 429 m/s at design point. The tip clearance is constant at 1.01 mm [6]. The rotor geometry is adopted by introducing four circumferential grooves in the casing as shown in Figure 1. The grooves are equally sized, with W = 12.5 mm and aspect ratio D/W = 3.2. Furthermore, L1 = 3mm and L2 = 5mm.

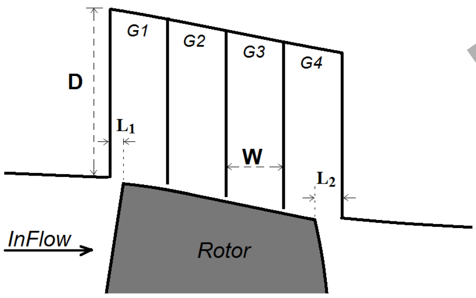


Figure 1. Casing treatment configuration

3. Results and Discussion

All of the unsteady computations in the current study have been performed at 100 % design speed. The static pressure at the outlet was gradually in-creased to throttle the compressor from choke to stall. The last stable operating point near stall is determined to be the point with the lowest flow rate where fully-developed periodically-repeated condition could still be obtained. Figure 2 shows a comparison between the computed (time-averaged) and measured performance curves of the rotor. The performance curves of the treated rotor have been already shown in Figure 2. As shown, the grooves have effectively increased the compressor safe operating range. Only negligible total pressure ratio loss in the compressor is observed. However, some efficiency decrease is introduced for the treated casing which is due to the rise in the total temperature ratio. This effect is caused by the flow recirculation within the grooves. Two different operating points are defined to compare treated and smooth casing. Operating point I is defined as near stall point of the smooth casing and operating point II is defined as near-stall point of the treated casing.

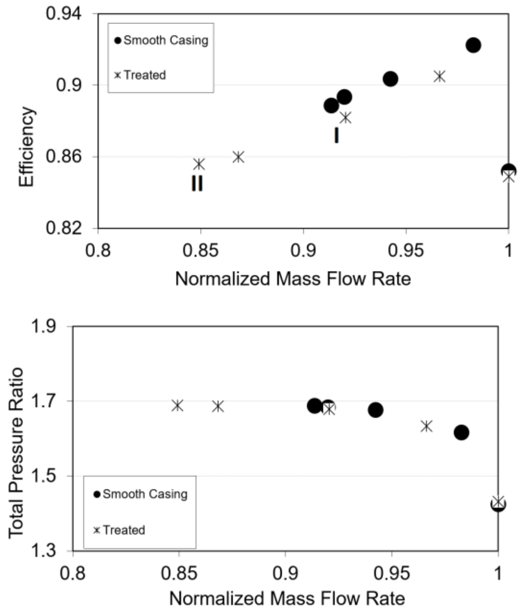


Figure 2. Compressor characteristic

Figure 3 shows the inlet and outlet mass flow rates (normalized by the corresponding annulus mass flow) at the groove-passage interface for a period of two blades passing. At operating point I, the time-averaged mass flow rates circulated in G1, G2, G3 and G4 are 1.2, 1.6, 1.29 and 1 percent of the annulus flow, respectively. These values increase to 2, 2.5, 2.1 and 1.24 at operating point II. The average amount of fluid circulating within the grooves is significantly larger at operating point II. Compared to that, only small fluctuations are observed in the circulated mass flow rate at operating point I. This suggests, that this operating point can be regarded as steady. However, the large oscillations in the absorbed and injected mass flow rate at the near-stall point suggest that the flow field near the grooves is highly unsteady at this condition. As seen, maximum unsteadiness occurs in G1: the circulated mass flow rate changes as high as roughly 30 percent of its time-averaged value.

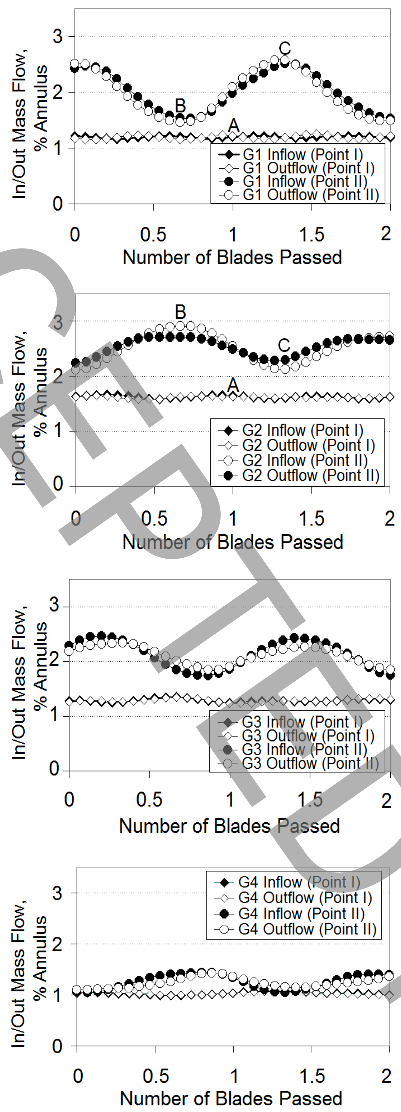
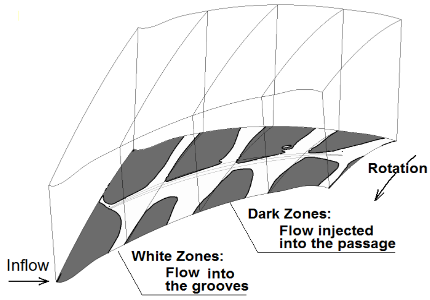
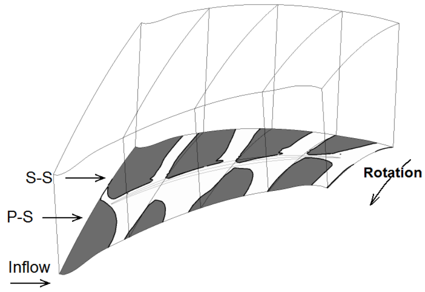


Figure 3. Time histories of the mass flow rate into and out from each groove.

Figures 4(a) and 4(b) illustrate the time-averaged regions of positive and negative radial velocities at the rotor-groove interface. A positive radial velocity leads to the absorption of flow by the groove (indicated in white) while a negative velocity results in the injection of air into the passage (indicated in grey). The blade is located at the center of the passage. As seen in Figure 4(a), the white zones (flow into the grooves) are considerably larger in size near the blade pressure surface, and in contrast, the gray zones are larger in size near the blade suction surface. This suggests that air is absorbed by the grooves near the pressure surface and is injected into the passage near the suction surface. It is further obvious that the area of the white zones (entrance to the groove) is smaller than that of the gray zone for each groove. This suggests that the fluid entering to the grooves has greater radial velocity component than the leaving fluid. Throttling the compressor toward stall has increased the white zones (i.e., the area over which air enters into the grooves). Another observation in Figure 4 is that air is mostly absorbed by the grooves from the downstream part of each groove and is injected into the passage from the upstream part.



a) Inflow/Outflow regions- Point I



b) Inflow/Outflow regions- Point II

Fig. 4. Time-averaged inflow/outflow regions

4. Conclusions

Time-accurate numerical computations have been carried out to investigate the influence of circumferential grooves on the performance and operability of a transonic axial fan. Four grooves (having the same geometry) were placed over the tip of the blades and choke to stall computations were performed. The rotor overall characteristics and detailed endwall flow structure were studied which can lead to the following conclusions:

- The grooves effectively increased the compressor safe operating range at the expense of small penalty in the adiabatic efficiency.

- Results indicate that circumferential grooves have two simultaneous important effects on the flow field near the grooves. First is adding momentum in the streamwise direction (the endwall fluid is mostly absorbed by the grooves from their downstream part and is injected from their upstream section). The second function of the grooves is providing a flow path between the suction and pressure surface, leading to a reduction in the pressure difference between them (fluid is absorbed by the grooves mostly near the pressure surface and is injected into the passage near the suction surface).

- At the near-stall point, the endwall flow field was found to be highly unsteady. Maximum unsteadiness was observed in the first upstream groove: the circulated mass flow rate changed as high as roughly 30 percent of its time-averaged value. This is important because it shows that steady calculations might be inadequate in circumferential groove casing treatment.

5. References

[1] M.D. Hathaway, Passive endwall treatments for enhancing stability, (2007).

[2] D. Rabe, C. Hah, Application of casing circumferential grooves for improved stall margin in a transonic axial compressor, in: Turbo Expo: Power for Land, Sea, and Air, 2002, pp. 1141-1153.

[3] A. Shabbir, J.J. Adamczyk, Flow mechanism for stall margin improvement due to circumferential casing grooves on axial compressors, (2005).

[4] C. Hah, M. Mueller, H.-P. Schiffer, Study of convective flow effects in endwall casing treatments in transonic compressor rotors, in: Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2012, pp. 95-106.

[5] Y. Sakuma, T. Watanabe, T. Himeno, D. Kato, T. Murooka, Y. Shuto, Numerical analysis of flow in a transonic compressor with a single circumferential casing groove: influence of groove location and depth on flow instability, Journal of Turbomachinery, 136(3) (2014).

[6] A.J. Strazisar, J.R. Wood, M.D. Hathaway, K.L. Suder, Laser anemometer measurements in a transonic axial-flow fan rotor, (1989)