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Performance Enhancement of a Low Speed Axial Compressor Utilizing Simultaneous Tip Injection and Casing Treatment of Groove Type

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Abstract

Performance of a low speed axial compressor is enhanced through a proper configuration of blade row tip injection and casing treatment of groove type. Air injectors were mounted evenly spaced upstream of the blade row within the casing groove and were all aligned parallel to the compressor axis. The groove, which covers all the blade tip chord length, extends all-round the casing circumference. Method of investigation is based on solution of the unsteady form of the Navier-Stokes equations utilizing k-ω SST turbulence model. Extensive parametric studies have been carried out to explore effects of injectors' flow momentums and yaw angles on compressor performance, while being run at different throttle valve setting. Emphasis has been focused on situations near to stall condition. Unsteady numerical analyses for untreated casing and no-injection case for near stall condition provided to discover two well-known criteria for spike stall inception, i.e., blade leading edge spillage and trailing edge back-flow. Final results showed that with only 6 injectors mounted axially in the casing groove and at yaw angle of 15 degrees opposite the direction of the blade row rotation, with a total mass flow rate of only 0.5% of the compressor main flow, surprisingly, the stall margin improves by 15.5%.

Key words: Axial compressor, Stall inception, Tip injection, Casing treatment, Stall margin

1. Introduction

In any dynamic turbomachine there is always a small gap between rotary blades and surrounded stationary casing. This gap, in spite of tenth of millimeters, may impose influential effects on safe and stable operation of the turbomachine. Consequences of the blade tip leakage flow are so sophisticated and critical, specifically, while dealing with axial compressors.

Blade tip leakage flow has dominant effects on the rotating stall characteristics which may, under some circumstances, lead to ultimate compressor surge. In addition, tip leakage flow causes the losses in compressor to increase, and as a result, the rate of pressure rise to decrease [1-3]. All the above mentioned events may happen in both the low [4] and high speed [5] axial compressors. Spike stall is a common form of stall phenomenon. There are two criteria responsible for inception of this kind of stall which are introduced as leading edge spillage and trailing edge backflow. These criteria are observed in the blade tip region of an axial compressor while working at near stall condition [6].

Up to now, different active and passive control techniques for alleviation of undesirable effects arising from the tip leakage flow are presented. Casing treatment in its various forms is the most popular passive control method. Considering circumferential grooves in the compressor casing is among the methods which can significantly enhance the compressor stability while being designed and commissioned, properly (e.g., as executed in JT9D engine [7]). Shabbir et al. [8] conducted a numerical study to realize physical mechanism of flow in a casing circumferential groove and discovered that how it can augment the compressor stall margin.

The earliest idea of controlling the instabilities in axial compressors using active methods has been proposed by Epstein et al. in 1989 [9]. Injection of air at the blade tip region has been executed by Suder et al. [10] to enhance

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the compressor performance. They tested different configurations of injectors and discovered that stable operating range would be consistent with increase in the mass-averaged axial velocity via a proper injection. They found out that the best performance would be attained if the injectors are being choked at their exit section.

Nie et al. [11] have experimentally studied effects of tip injection on a low speed axial compressor performance. They investigated flow unsteadiness for various axial distances between injectors and blade tip leading edge. Geng et al. [12], through their numerical investigations, showed that how a proper tip injection can reduce or fully resolve the self-induced unsteadiness of the tip leakage flow.

A new idea of injecting the air jet in the axial compressor blade tip region has been proposed by Beheshti et al. in 2006 [13]. They suggested a grooved casing positioned over the blade tip region with a continuous circumferential slot for injection of the air flow. They studied effects of time delay between the air injection and compressor dynamic response. Injection of air upstream of the first rotor blade row of a multi-stage compressor was carried out by Hiller et al. [14], experimentally. They succeeded to increase the throttling capability of the compressor up to 40% by injection of air at the blade row tip region.

In the present study, a new idea is presented for performance enhancement of axial compressors. This new idea utilizes a proper combination of both the rotor blade tip air injection and casing treatment of groove type, simultaneously.

2. Model specification and numerical approach

An isolated rotor blade row of an axial compressor is considered for the present study. It consists of 12 blades with cross-sectional profiles of NACA-65 series type. Detailed specifications of the rotor blade row are introduced in Table 1.

Table 1. Rotor blade row specifications

Parameter	Value
Rotational speed (rpm)	1300
Hub diameter (mm)	270
Hub to tip ratio	0.6
Tip clearance/blade chord (%)	1.7
Tip chord length (mm)	117.5
Blade tip solidity	1
Tip stagger angle (deg)	56.2
Midspan stagger angle (deg)	47.2

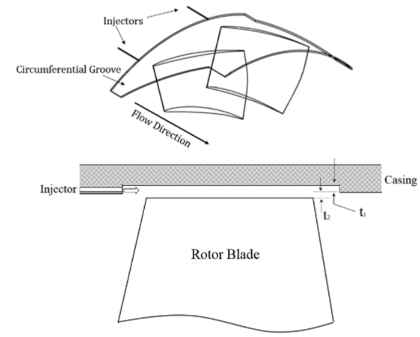

Fig. 1. Schematic drawing of rotor blade tip region including injectors and casing groove

Fig. 2. Mesh structure on solid surfaces and in blade tip clearance region

This rotor blade row has already been experimentally tested by some researchers like Inoue and Kuroumaru [15-17]. Their test results, in terms of general performance parameters, are used by the present authors to validate their numerical results.

As shown in Fig. 1, a proper configuration of the blade row tip injection and casing treatment of groove type is used in the present study. This groove of 2 mm depth (designated by t_1 in Fig. 1) has extended all around the casing circumference. It covers 30% of the blade tip chord length upstream and downstream of the leading and trailing edges, respectively (see Fig. 1). 12 air injectors, with 1.5 mm in their internal diameters, have been mounted evenly spaced upstream the blade row within the casing groove. The distance between the blades tips and the main casing (designated by t_2 in Fig. 1) is 2 mm.

As shown in Fig. 2, one third of the blade row circumference, including 4 blades, has been considered for the numerical analyses. Considering the rotational speed of the compressor, this number of blades allows the flow to experience all the events which may happen in this numerical domain.

Multi-block structured mesh system has been used for

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the numerical modeling purposes. The number of nodes distributed in each flow passage between two adjacent blades is 74, 50 and 60 in streamwise, spanwise and pitchwise directions, respectively. 32 nodes are radially distributed in the blade tip clearance region for a precise numerical simulation of the flow field. Grid independency test has been checked for both the smooth and treated casings. The number of grids in the entire solution domain, especially within the blade row tip region, was altered and its effect on the total pressure rise coefficient and efficiency was studied. Results for one flow passage are shown in Figs. 3a and b which suggest that the suitable numbers of grids for the smooth and treated casings are 293000 and 490000, respectively.

The commercial computational fluid dynamics (CFD) software of ANSYS-CFX 15.0 was used for the present flow simulations. Its two elements of ANSYS CFX-Pre and CFX-Solver were used for defining of boundary conditions and solving the governing equations, respectively. In addition, post processing of the results was executed by CFX-Post software.

Following boundary conditions are imposed to the flow domain.

- Flow velocity and its direction at the rotor blade row entrance,
- Smooth and adiabatic solid walls,
- Periodic boundary condition in circumferential direction,
- Radial equilibrium assumption along with average static pressure on the outlet boundary,
- Air mass flow rate and temperature at the injectors' inlets, and
- Interface boundary between stationary and rotary zones.

k-ω SST turbulence model has been used in this study.

The y^+ values of less than 5 for the regions adjacent to all the solid walls provided to simulate the flow within the thin boundary layers, precisely. A time step of 3.2×10^{-5} second, which is equivalent to 120 time steps for each blade passing period (i.e., 1440 time steps for each rotor revolution), has been considered for the numerical purposes.

Numerical simulations have been executed for different cases, including smooth and grooved casings with and without air injections in various operating conditions of the compressor. These extensive test cases are to discover effects of the injectors' momentums and yaw angles on the compressor performance, while being run at different compressor flow rates. Emphasis is focused on situations near to the stall conditions.

3. Results and discussion

Performance curves, in terms of the total pressure rise coefficient and efficiency versus flow coefficient (φ), are extracted from the numerical calculations and results are shown in Fig. 4. Experimental results of Inoue et al [17] are superimposed in this figure for comparison; which show

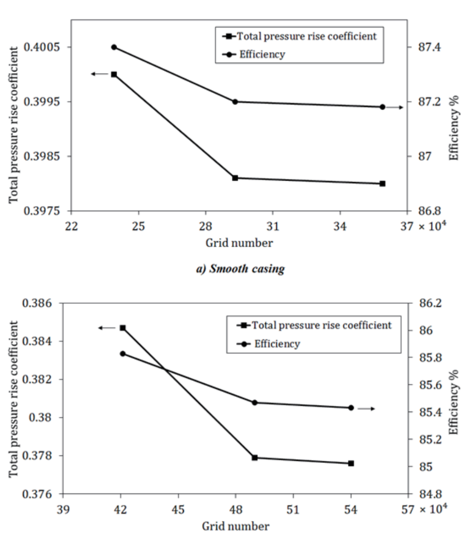

Fig. 3. Grid independency results for smooth and treated casings

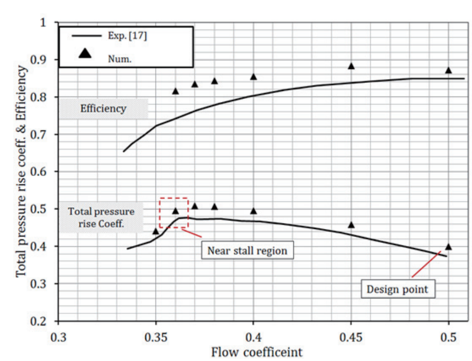

Fig. 4. Performance curves of the compressor

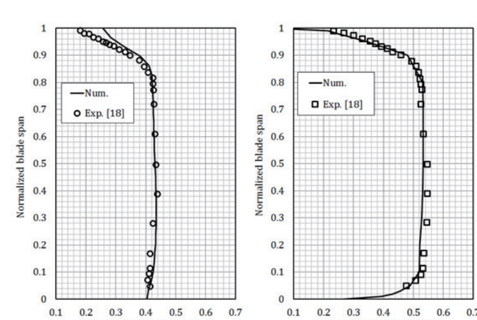

Fig. 5. Variations of flow properties over the blade span

reasonable agreement. However, for a better validation of the numerical scheme undertaken in the present research work, results of spanwise distributions of the total pressure rise and the exit axial velocity are compared with those of available experimental data of Furukawa et al. [18], too. These results are presented in Fig. 5 for the compressor operating at the design condition. The axial velocity is normalized with respect to the rotor blade tip speed.

Initially, no-injection cases are studied. Then, axial injection at the blade row tip region is tested. Finally, effects of the injectors' yaw angle and their numbers are investigated.

3.1 No-injection results

In the no-injection case the compressor mass flow rate was gradually reduced up to the stall inception instant, where the instabilities start to grow and eventually may lead to formation of the stall cells precursor. There is always a low pressure zone along the centerline of the blade tip vortex flow (see for example Inoue et al. [15]). This low pressure zone starts from one blade leading edge region and extends towards the adjacent blade pressure surface. Fig. 6 shows the pressure distribution in the flow passage region at 97.5% blade span for the near stall condition (φ=0.37). The trajectory of the tip leakage vortex flow centerline is illustrated in this figure by a dashed line. Results are presented for four sequential time instances. Duration of one blade pass (i.e., blade passing period) is designated by letter T in Fig. 6. The tip vortex flow breaks down at a position where it experiences some local high pressure region. This phenomenon (i.e., vortex break-down) has happened at the end of the dashed lines shown at different instances in Fig. 6. The extension of the tip vortex flow can still induce a local low pressure area on the pressure side of the adjacent blade.

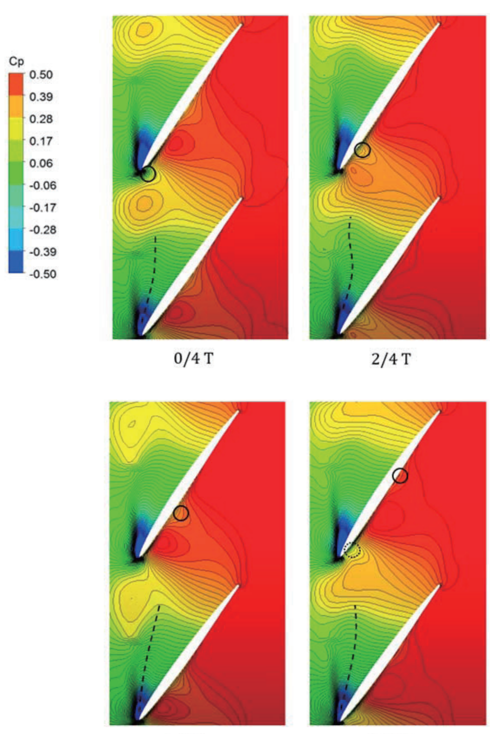

Fig. 6. Transient static pressure coefficient contours at 97.5% blade span for no-injection case, φ=0.37

This area is shown by a small circle in Fig. 6 for all the instances. The position of this low pressure zone shifts towards the blade trailing edge under the influence of the main flow, where it ultimately leaves the blade passage and sheds downstream. Exceeding the blade passing period, the same sequence happens again. The dashed circle shown in the 6/4T frame of Fig 6 is the new position of the induced low pressure zone due to the tip leakage vortex flow and its oscillatory trajectory. This type of intermittent creation and dissipation of the low pressure zone at the blade tip region is so-called "self-induced periodic fluctuations" [12].

Three dimensional views of the instantaneous streaklines in the blade tip region are shown in Figs. 7a and b for the design and near stall conditions, respectively. For operation at the near stall condition, the tip vortex formation, its break-down process and also its trajectory direction towards the pressure side of the adjacent blade are all distinguishable in Fig. 7b.

3.2 Injection results

Injectors were aligned in parallel to the compressor axis at the rotor blade tip region (see Fig. 1). Unsteady static pressure coefficient results are shown in Fig. 8 for different instances of one blade passing period at 97.5% blade span. Injection flow rate through the whole injectors is 0.5% of the compressor total mass flow rate.

Comparing these results with those shown in Fig. 6, one can realize that the trajectory of the tip leakage flow

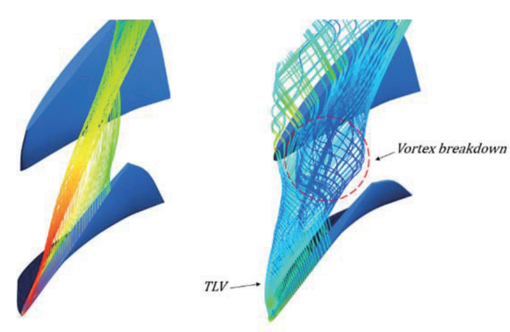

Fig. 7. Tip leakage vortex streaklines at the blade tip region

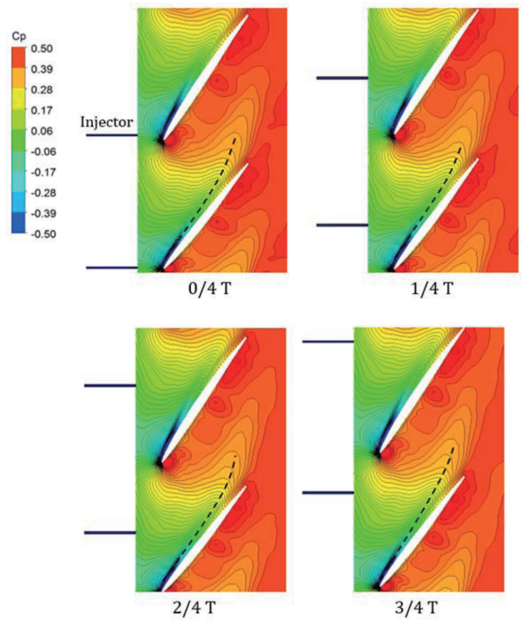

Fig. 8. Transient static pressure contours at 97.5% blade span for 0.5% injection case

(recognized by tracking the lowest pressure zone) is inclined more towards the blade suction surface for the injection case. It means that the blockage made to the main flow is reduced, and consequently, aerodynamic performance of the blade row will be improved.

Compressor performance curves in terms of the total pressure rise versus the flow coefficient are shown in Fig. 9 for injections of 0.3% and 0.5% of the compressor total mass flow rate. Results of smooth casing with no-injection are also superimposed in this figure for comparison. It can be easily detected from this figure that the air tip injection causes the stall margin to increase, remarkably. More injection is associated with more enhancement in the stall margin. Because, the blockage of the main flow has reduced and consequently the average axial velocity has grown up. The relative change in the stall margin is calculated through the

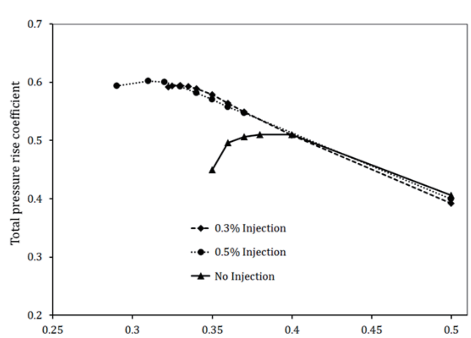

Fig. 9. Compressor performance curves for no-injection and injection case

following equation [7].

$$\Delta\varphi_{\mathrm{stall}} = [(\varphi_{\mathrm{stall}})_i - (\varphi_{\mathrm{stall}})_s]/(\varphi_{\mathrm{stall}})_s \qquad (1)$$

Where, $(\varphi_{\mathrm{stall}})_i$ and $(\varphi_{\mathrm{stall}})_s$ are stall flow coefficients for injection and no-injection cases, respectively. Table 2 shows values of the relative stall margin improvements for the proposed test cases. Surprisingly, with an injection of as small as only 0.5% of the compressor main flow rate, the stall margin enhances by 15.5%. This attractive result proves that how a limited local injection can influence the flow structure within the whole compressor annulus. Flow structure results of the rotor blade tip region at the near stall point are shown in Fig. 10. A rectangular window is superimposed in this figure located at 40% of the blade chord length measured from its leading edge. Flow patterns for three cases of the smooth casing with no-injection and two injection cases are shown in Fig. 11 within the above window. Effectiveness

Table 2. Stall margin gain for different conditions

Test Case	Stall flow coefficient	Stall margin augmentation
No injection	0.37	-
0.3% Injection	0.33	10.8%
0.5% Injection	0.3125	15.5%

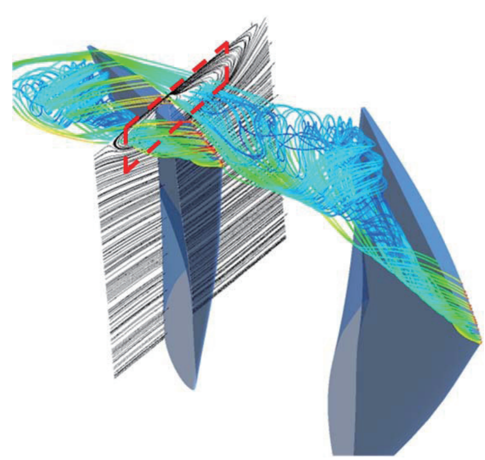

Fig. 10. Flow pattern in blade tip clearance region at near stall condition

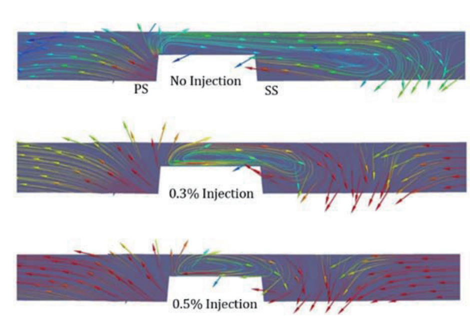

Fig. 11. Flow structure on 40% blade chord plane for no-injection and injection cases

of the air injection at the blade row tip region can be easily observed in this figure. Tip injection causes the original tip vortex cross sectional area to reduce, considerably.

Figure 12 shows instantaneous axial velocity coefficient contours at φ=0.36 for smooth casing and injected cases. Contours are displayed in a plane located at 50% of blade chord length measured from its leading edge. Negative axial velocities clearly demonstrate existence of the reversal flows within the passage, and consequently, the blockages made to the main stream. Results of three instances of the blade rotations (N) for no injection case are shown in Fig. 12a. As can be detected from this figure, the low speed region moves around the compressor annulus; indicating occurrence of the rotating stall phenomenon. As shown in Fig. 12b, after 0.3% injection the flow pattern improves considerably.

Contours of the relative total pressure rise coefficient at the stall condition (φ=0.36) are shown in Fig. 13 at 97.5% blade span.

Comparing results of no-injection and injected cases one can conclude that the former case is associated with more reduction in the total pressure rise .In addition, injection has caused the flow spillage at the blade leading edge region to reduce in comparison to the no-injection case. As already

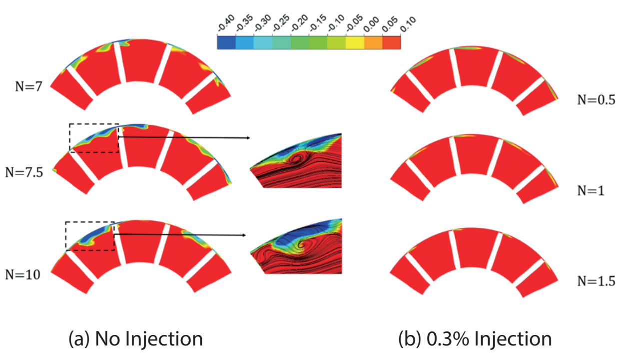

Fig. 12. Instantaneous normalized axial velocity at 50% blade chord, φ=0.36

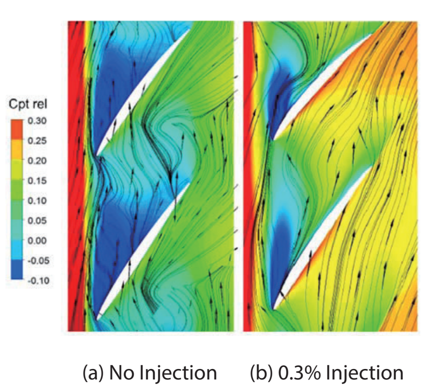

Fig. 13. Relative total pressure coefficient contours at 97.5% blade span at the stall flow coefficient, φ=0.36

mentioned, blade leading edge spillage is one of the essential criteria responsible for occurrence of a special kind of stall phenomenon known as the spike stall [6].

3.3 Effects of injectors' number and yaw angles

In order to study effects of the number of the injectors, flow simulations were executed for two cases of 6 and 12 injectors. The total mass flow rate for the above number of the injectors was the same and equal to 0.3% of the compressor total mass flow rate. These results are tabulated in Table 3. In fact, at a fixed total mass flow rate, 6 injectors transmit more jet momentum into the stream and as a result, cause the tip leakage flow structure to be improved. This process is significantly effective during the near stall condition while the average axial velocity of the flow at the blade tip region reduces; which causes the blockage to the main flow and reversal flow zone to be magnified. In fact, the injected air with relative high speed and its interaction with the tip clearance flow are the main sources for alleviation of these undesirable effects.

Table 3. Stall margin gain for different number of injectors

Test Case	Stall Flow coefficient	Stall margin augmentation
No injection	0.37	-
12 Injectors (0.3% Injection)	0.33	10.8%
6 Injectors (0.3% Injection)	0.325	12.2%

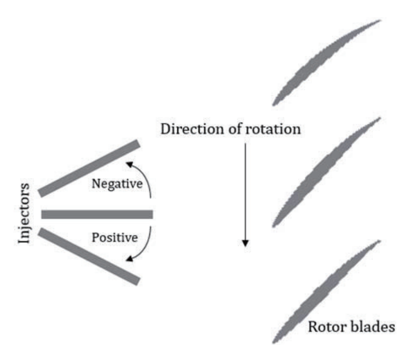

Fig. 14. Definition of the injectors' yaw angle relative to the rotor blades row

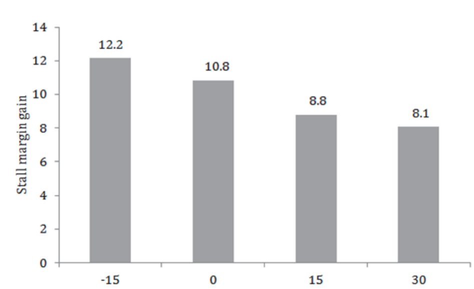

Fig. 15. Stall margin augmentation for different air injection yaw angles

The injectors yaw angle was changed between -15° to +30° with 15° interval in order to find the appropriate situation. Negative and positive yaw angles of the injectors are introduced in Fig. 14. Results of the injectors yaw angles are shown in Fig. 15.

The total mass flow rate of the injectors was fixed for all the angles and was equal to 0.3% of the compressor total mass flow rate. Twelve injectors were considered in this study. As can be observed in this figure injection at an angle of -15˚ produces the maximum augmentation in the stall margin of about 12.2%.

4. Conclusion

The main conclusions drawn from the current research work can be categorized as follows.

- Injection of proper amount of air at upstream of rotor blade row can enhance compressor performance significantly, especially in near stall conditions.
- Blade tip air injection can overcome leading edge spillage which is responsible for occurrence of undesirable spike stall phenomenon.
- Number of air injectors can considerably affect compressor performance, especially while dealing with a fixed total injection mass flow rate. In the present study, 6 injectors caused better performance than 12 injectors due to their higher momentum transmitted to the main stream.
- Blade tip injection in opposite direction of blade row rotation is associated with more beneficial effects in aerodynamic performance of axial compressors.

Acknowledgement

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