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ORIGINAL ARTICLE

Investigating the effect of blade sweep and lean in one stage of an industrial gas turbine's transonic compressor

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Efficiency

Abstract In this paper, the simultaneous effects of the sweep and lean of the blades in one stage of a transonic compressor on its performance have been investigated. Then, with the help of numerical solution, fluid flows over these two modified geometries generated from the original sample were analyzed. Considering the applied constraints, the two generated rotor geometries have different geometrical characteristics; so that in rotor No. 1, the blade has a backward sweep and it is less affected by lean, while in the modified rotor No. 2, the blade has a forward sweep and it is more affected by lean. In the first sample, it is observed that the stage efficiency increases by 0.5% for operating design, while the stall margin reduces, and the chocking mass flow rate diminishes by 1.5%. Also regarding the second modified blade, the results indicate that the stall margin increases, the choking flow rate at the nominal rotational speed of the stage increases by 0.18% and the stage efficiency increases by 1%. The comparison of numerical results also shows that, in the first modified rotor, the pressure ratio of the stage diminishes by 0.01%; while in the second sample, the pressure ratio of the stage increases by the same amount. These results were then compared with the experimental results, showing a good agreement.

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1. Introduction

The most important goal in the operation of industrial gas turbine compressors is to achieve a higher performance. For this purpose, the blades should be designed and shaped so that they cause minimum flow losses and the boundary conditions are controlled. The second and most difficult challenge in minimizing the flow losses is to minimize the losses near the hub, which are caused by the secondary flow losses. Currently, in the three-dimensional engineering design of blades, blade sweep and dihedral blades are used for the control of secondary flows in order to reduce the corner stall and prevent low-momentum fluids from returning to the row of blades [1]. The mentioned phenomenon can only be predicted by solving the Navier–Stokes equation three-dimensionally. The amounts of flow rate, pressure and efficiency are strongly dependent on the simulation of the boundary layers of blade and casing wall, which may undergo separation. The reduction of flow and separation in the corner regions of blade and casing wall has the greatest influence on the exact solutions of the simulation.

It is important to investigate the behavior of flow in the first stage, because the flow rate of other stages is a function of the output flow rate of the first stage rotor, and because the loss and efficiency of other stages are strongly influenced by the kinematic and thermodynamic characteristics of the first stage.

1.1. Sweep and lean of rotor blades

The effects produced by the sweep and lean of transonic rotor have already been analyzed and explained in many research articles; however, a full and comprehensive definition of the effects of these changes on flow behavior and on overall rotor performance has not been presented. Bergner et al. [2] studied a forward-swept blade in a transonic compressor and realized that rotor-tip forward sweep extended the distance between the leakage vortex originating position and the tip shock so as to delay the leakage vortex breakdown thus resulting in a larger stall margin. Breugelmans and Sasaki [3,4] investigated the effects of blade curvature and sweep in a wind tunnel and found out that the forward sweep and the positive lean of blades are effective in delaying the corner stall and weakening the secondary flow. The findings of these researchers indicate that the use of forward-swept blades reduces the interaction between the shock wave and the casing wall boundary layer and is more effective in increasing the efficiency and the stall margin. Oyama et al. [5] associated the increase of efficiency with the reduction of entropy generation and attributed that to the change of blade geometry by means of lean and sweep. Hoeger et al. [6] studied the forward and backward-swept blades and discovered that the chocking flow rate diminishes in the backward-swept blades relative to the forward-swept blades. By analyzing the results of blade sweep and lean related to rotor 37. KwediKha [7] demonstrated that these changes, even if they occur in a small region in the vicinity of blade tip clearance, influence the whole aerodynamic flow. Nevertheless, these geometrical changes produced no significant improvement in the overall performance of the rotor. Also, by examining the lean and sweep of the blades of a transonic fan, Denton [8] confirmed that blade lean has a greater effect on the improvement of performance and reduction of efficiency than blade sweep.

The effect of blade sweep and lean on fan performance has been explained by Denton and Xu [9]. The effect of these geometrical changes has been well understood by analyzing the gradient of the applied pressure. This pressure gradient is very small compared to the pressure gradient between blades, and this is due to the larger bending of flow lines in sections between blades compared to the meridional surface. Because of the low pressure gradient in the vicinity of the casing wall, blade load cannot change rapidly in the wall-normal direction; as a result, pressure distribution is applied in a region further away from the casing wall, without considering the shape of the local profile. Therefore, blade sweep reduces the blade load at the leading edge and increases it at the trailing edge. For similar reasons, blade lean leads to pressure increase on the pressure surface. In the first research conducted by Denton, it was observed that when changes are created in the stacking line of a blade, the blade sections move in a constant pressure field. The important point regarding the shock waves is that the shock wave must impact the casing wall vertically. Thus, the effects of shock sweep near the casing wall and hub are inevitably lost. As the hub's boundary layer influences the flow behavior at the root of blade, the effect of blade tip clearance on rotor performance is also severe. The closest solutions in the investigation indicate that the displacement of the upstream shock over the hub is due to the strong viscosity effects, which produce a severe choking by the boundary layer near the hub. The anticipated viscosity effects are much greater in the cascade than in the compressor rotor; and this is due to the lack of flow contraction effects in the cascade, which produces more dispersion. Nevertheless, in both samples, it is observed that the shock wave has a strong tendency to remain normal to the casing wall, irrespective of blade geometry.

Numerical and empirical analyses by Hah et al. [10–12] in evaluating the performances of a rotor with radially stacked blades without blade sweep and a rotor with forward-swept blades show that the latter rotor enjoys a higher maximum efficiency and the stall margin in it increases considerably. Also, in a rotor with backward-swept blades, the efficiency increases compared to blades with no sweep, but the stall margin diminishes considerably. Different aspects were investigated to find out how blade sweep influences rotor performance; and the three-dimensionality of the shock wave is one of the underlying reasons. As Hah has explained, the shock wave must impact the casing wall vertically (a phenomenon known as the wall effect). This causes the shock wave to move towards the upstream of flow in the backward-swept blades and downstream of flow in the forward-swept blades; and usually the shock wave that moves downstream is more stable. In addition, in the lower sections of the blade, where the casing wall does not have an influence, blade sweep imposes a strong

Nomenclature

Mach	relative Mach number
w	weight function
η_{ref}	efficiency of initial geometry
P_{ref}	total pressure of initial geometry (unit: N/m^2)
$y+$	Y plus

effect on shock pattern and leads to the reduction of loss and thus the improvement of local efficiency.

Blade sweep also has a significant effect on the accumulation of fluid with low momentum near the casing wall in the blade tip clearance region; and this section of blade has a high influence on rotor instability. Yamaguchi et al. [13] realized that this instability is negligible for a forward-swept rotor blade, and it is due to the reduction in the radial migration of fluid particles into the boundary layer of the blade's suction surface (a secondary flow which is created due to a lack of balance between centrifugal forces and pressure gradient).

The experimental and analytical studies by Wadia et al. [14] show that the forward sweep of rotor blades has a significant influence on the improvement of efficiency and stall margin. The primary flow mechanism that improves the performance of forward-swept blades is the reduction of shock wave interaction with the boundary layer and also the reduction of fluid accumulation in the boundary layer over the suction surface due to the existence of a centrifugal force at blade tip clearance. This is the most basic effect of forward sweep of blades by changing the flow paths over different blade sections by directing the flow more towards the blade tip clearance and thereby reducing the load in this region. Low-velocity tests have indicated that the outcome of this change of geometry is the reduction of load at blade tip clearance. These tests have demonstrated that this load reduction at blade tip clearance with the forward sweep of blades has a positive effect on the blockage of tip clearance leakage. Computational fluid dynamics (CFD) analyses have shown that this characteristic endures even in the presence of strong shocks. The backward sweep of blades causes the shock wave to impact the casing wall vertically; especially, in a particular section, the bow shock is combined with the passage shock and produces a strong vertical shock which practically reduces the losses resulting from the shock. In this paper, the simultaneous effects of lean and sweep in blades on the performance charts of the transonic stage is investigated. Finally, the changes in the performance chart is discussed using the generated contours and figures.

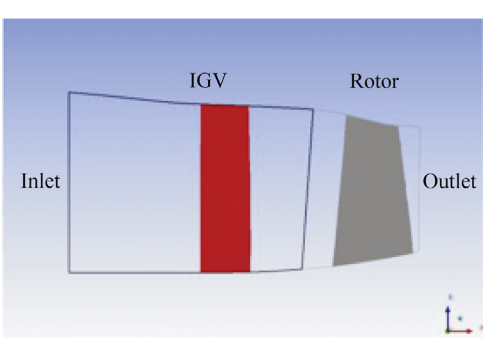

Figure 1 Meridional view of the transonic stage.

2. Geometry of the stage

Designed gas turbine is a two-shaft turbine consisting of 10-stage compressor and 3-stage turbine. The first two stages of this compressor are transonic stages with a blade tip clearance Mach number of 1.4 at the nominal rotational speed. The studied acoustic pass stage is the first stage before which there is a variable IGV (inlet guide vane). In the 3D analysis performed on this stage, the nominal rotational speed constitutes 96% of the rotational speed; and the operating performance of the compressor occurs at this rotational speed. The Figure 1 represents the meridional view of the transonic stage.

The geometrical specifications of the considered stage have been summarized in Table 1.

3. Modified stacking line

Modified the geometry of rotor blades is generally accomplished by making changes to the profile of blade sections or changing the stacking line of blade sections. In the present paper, the stacking line was modified by modifying the arrangement of blade section, without changing their profiles, and then the desired geometry was produced.

Since the goal of geometry change has been to increase the amount of efficiency while pressure ratio do not decrease, in the established final geometries, the effects of blade sweep and leas are observed together. These changes include the forward sweep of the blades and their leaning in the direction of rotor span.

The flow field for these geometries was solved and the results associated with each geometry were extracted. The equation which describes stacking line is defined as the three order functions ($a_1X^3+b_1X^2+c_1X+d_1$). The hub points are constrained. The stacking line can cause swept, either backward or forward, by changing the axial direction. Moreover, it can make changing in the camber of the rotor's blade possible. The combination of these changes leads to design of blades with both characteristic of sweep and lean simultaneously.

Since efficiency is very important in industrial compressors, the modifying process has been done to reach the best efficiency. Considering stage pressure ratio to be constant, the objective function is defined as below:

$$f = \min\left[w_1\left((1-\eta)/(1-\eta_{ref}\right) + w_2\left((1-P)/(1-P_{ref})\right)\right] \tag{1}$$

where η_{ref} and P_{ref} are the values corresponding to the initial geometry. w_1 and w_2 are weight functions which affect the sensitivity to the modified efficiency or pressure parameters. Weight function values show the effectiveness of each non-dimensional parameters, therefore, considering the greater importance of non-dimensional efficiency, we assume the values for w_1 and w_2 as following:

$$w_1 = 0.8, \qquad w_2 = 0.2$$

Finally, the purpose is minimizing the value for the defined function. This value can be obtained through changing stacking line and finding suitable coefficients assigned to its two-order equation.

Figure 2 illustrates changes in blade-sweep in the direction of rotor's rotation. As shown in this figure, changes in the generated rotor number 2 is greater than those of the number 1.

Sweep changes in the generated blades has been shown in Figure 2. The geometrical changes of lean has been shown in Figure 3.

As shown in figures the hub section has been constrained and by moving the stacking line in axial direction and also in the direction of rotor's rotation, geometries of the blades with sweep and lean characteristics are generated.

3.1. Final geometries of the generated blades

It should be mentioned that, in consideration of the changes of relevant constraints, the changes of blade lean forward or backward and the degree of lean towards or contrary to the compressor spin direction, the shapes of the modified geometries have been so designed as to prevent the impact with adjacent blades. The Figure 4 represents flow direction in the geometry of blades.

Table 1 Specifications of the compressor stage.

Number of IGV blade	48
Number of rotor blade	31
Number of stator blade	48
Rotation speed/(rev/min)	9700
Mass flow inlet/(kg/s)	75.5
Pressure ratio	1.4
Hub radius (rotor)/mm	262.85
Hub radius (stator)/mm	261.7
Shroud radius (rotor)/mm	422.75
Shroud radius (stator)/mm	423.3

3.2. Forming the grid

After creating the considered 3D geometry, the software program was employed to generate the computational grid by using the J/L/C grid topology and the boundary layer adjustments. In order to observe the boundary layer effects, the mesh structure around the blade became very fine; which by using the adjustments of the mentioned software, the value of $y+$ was considered as below 20 that it represents in Figure 5. In total, a mesh density of 2,292,898 was established for the whole stage geometry.

The grid independency test of various meshes have been summarized in Table 2.

3.3. Numerical solution

The software was used to numerically solve the flow in three-dimensional states. The compressor stage was solved in steady state; and due to the important effects of the casing wall and the inter-blade region, the *k–w* turbulence model was used. In comparing the *k–e* and *k–w* methods, the *k–w* model is more advantageous, because it considers the boundary layer effects as well as the effects of rotation and the losses resulting from compressibility. The boundary conditions include a pressure of 1.0 atm and temperature of 288 K at the inlet and the static pressure at the outlet. These boundary conditions apply to the average section of the guide vane inlet and rotor outlet. The good match between the results of this method and the experimental results confirms the validity of this approach. The computational domain was used to periodic boundary condition. The results of this method for the vicinity of hub well match the empirical results and indicate the losses of the casing wall. The static pressure boundary conditions were used at the inlet and the mean static pressure conditions were used at the outlet. The amount of turbulence was considered as 5% at inlet and the numerical solution was achieved by employing a high-resolution flow solver and using the first-order turbulence values to clearly demonstrate the effects of the shock wave. The approach to model between the stationary and rotational

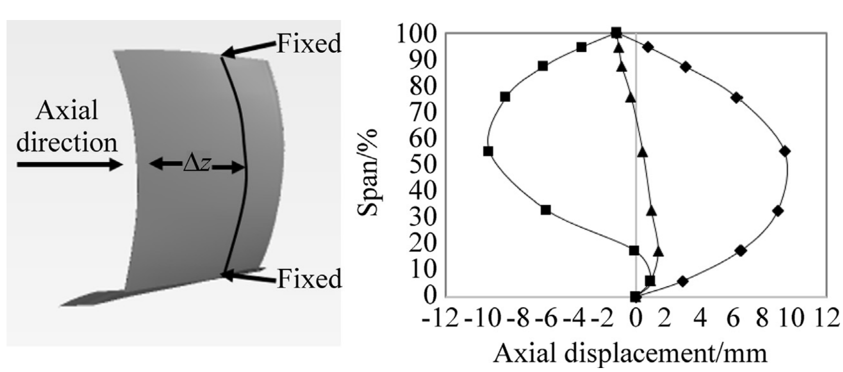

Figure 2 The geometrical changes of sweep.

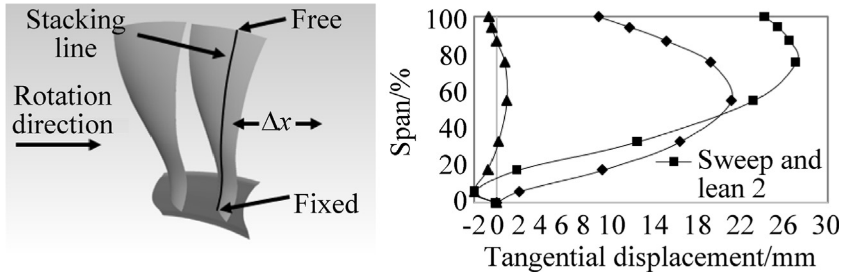

Figure 3 The geometrical changes of lean.

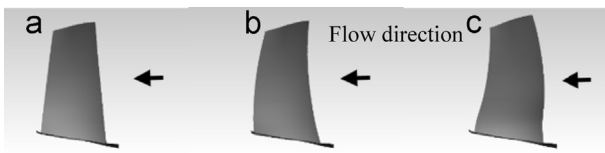

Figure 4 (a) Rotor prototype, (b) generated rotor No. 1, and (c) generated rotor No. 2.

domains is stage interface. Also, a physical time step of 10^{-4} and residual convergence value of 10^{-6} were used as well.

4. Discussion and analysis

In this section, two blade geometries obtained by modifying the rotor formed by radially stacked profiles are analyzed and compared with the original models. It should be mentioned that, with regards to the applied constraints, in the generated geometry of Blade No. 1, the amount of blade sweep is more than lean; and conversely, in Blade No. 2, blade lean is greater than blade sweep. Considering the overall flow effects, the 3D analysis was performed at 95% blade section; because due to the stage being an acoustic pass stage at lower blade sections, the shock pattern is not observed completely, and this is due to the low value of Mach number in these regions. This section is non-dimensional span-wise length starting from hub.

As is observed in the Figure 6, in the contours related to maximum efficiency, the bow shock in the cascades of the ordinary blade has the largest distance from the leading edge; while in the cascades related to the modified Blade No. 1, the distance between the bow shock and leading edge is minimum, and the highest efficiency belongs to this rotor. In comparing the contours related to the 95% section of three blades near the chocking state, in the contour of Blade No. 1, the influence of the bow shock in the inter-blade region has reached a minimum, and the bow shock wave has become dispersed and not impacted the suction surface of the blade, which is effective in reducing the overall loss of the shock wave. In the contour related to the initial rotor (middle contour), two shock waves can be clearly seen and the bow shock wave has fully impacted the suction surface of blade and has been drawn into the inter-blade region, resulting in flow separation on the suction surface of blade. This fact indicates that on a blade without sweep, the shock wave is expanded more relative to swept blades, which causes a greater loss of total efficiency in blades without sweep. However, in the chocking state of flow, both modified geometries experience more critical conditions in the downstream of flow (rotor outlet) relative to the original sample that represents in Figure 7.

The results of Mach number contours for different blade sections indicate that the shock wave from the base to the tip of blade is moving towards the downstream; although, in numerical analysis, the displacement of shock can be roughly predicted. The contour related to the meridional view indicates that blade sweep influences the shock wave’s position and that in the backward swept and leaned blade, the shock leans forward and in the forward swept and leaned blade, this displacement has a lesser effect. Also, the amounts of corner stall and the low-velocity region stall in both modified blade samples have considerably improved compared to the primary model. Due to the favorable impact of blade curvature on the three-dimensional structure of the shock, the stage performance basically improves in the forward or backward swept and leaned blades.

The contour of the forward swept and leaned blade is at the midpoint between the backward swept and leaned blade and the primary blade, and the bow shock wave in this case has not completely touched the blade’s suction surface. The amount of chocking mass flow rate in Blade No. 1 has diminished considerably and this is due to the greater effect of blade sweep on the flowing fluid.

The contours related to Mach number distribution along the blade length indicate that the Mach number has been distributed more regularly on the span-wise flow surface of the generated blades. This is due to the different flow speeds at different blade spans as a result of changes in axial flow velocity with regards to the change of blade length, which causes the inlet Mach number to be different for every blade section. Now, if modifications are made to the geometry of the leading edge which are in proportion to the inlet velocity changes, the inlet Mach will be distributed more properly at different blade sections.

The diagrams related to blade surface loading demonstrate that, in the backward-swept blade (Blade No. 1), the variation of load on the pressure surface of blade has a more uniform distribution, and that Blade No. 2 bears more load relative to its original sample.

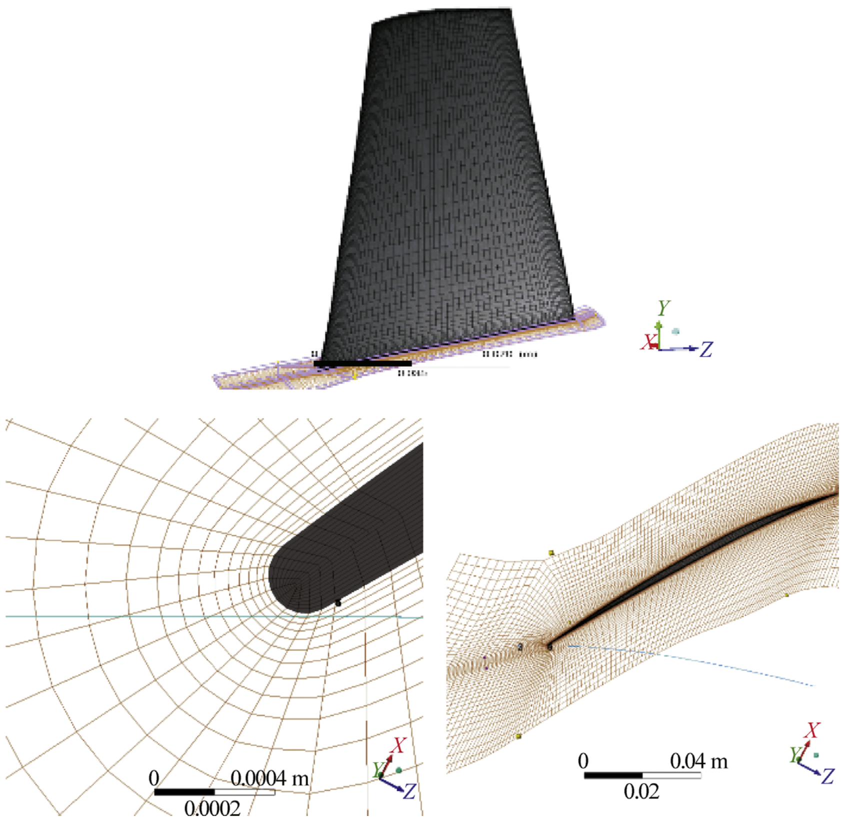

Figure 5 A view of meshed section of rotor geometry.

Table 2 Grid independency test of various meshes.

Grid number	Element count	Efficiency	Pressure ratio
1	1136798	0.9015	1.647
2	1749510	0.9027	1.644
3	2706639	0.9044	1.644
4	3709277	0.9042	1.644
5	4827975	0.9043	1.644
6	5605219	0.904	1.641

The reduction of load in the forward-swept blades is the result of forward displacement of flow over the blade section and, with a radial equilibrium, it leads to an identical increase of pressure with a higher dynamic head. Although it can be said that for both the swept and ordinary blades, the reduction of load at the blade tip clearance leads to limited stability in fluid flow.

Another reason for the decline of chocking flow rate is most probably the effect of blade shape on downstream flow. In the modified Blade No. 2, the shock wave has a normal shock instead of several shock, the advantage of this model is the increase of the chocking flow rate by 0.3 kg/s and efficiency by 1% relative to the ordinary rotor model, which is due to the higher influence of blade lean relative to blade sweep [8].

The charts which are related to relative Mach number, velocity and flow angle indicate a similar trend with the prototype rotor so these trends are acceptable. Figure 8 shows that the changes of relative Mach number assigned

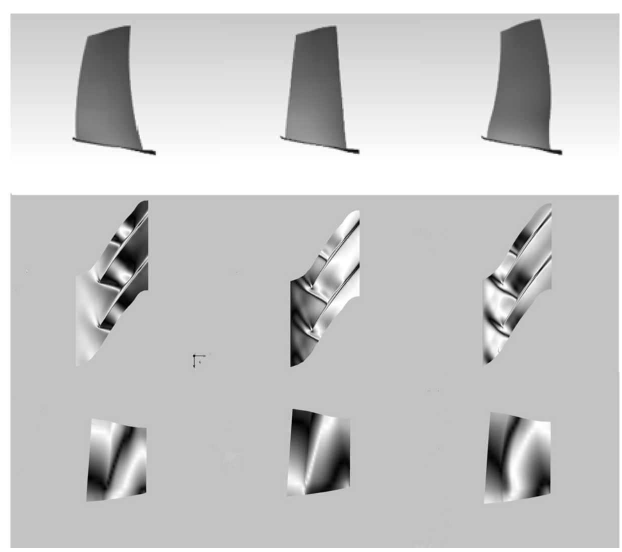

Figure 6 Contours related to maximum efficiency of the 95% blade section.

Figure 7 Contours related to the choking state of the 95% blade section.

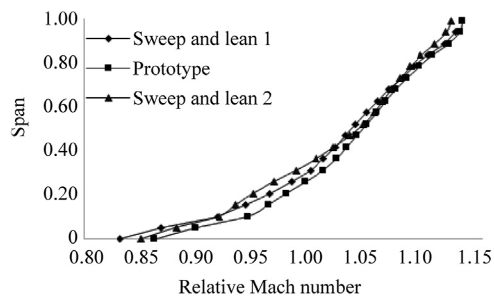

Figure 8 Relative Mach number at leading edge for max efficiency.

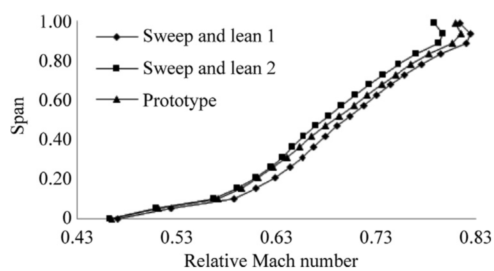

Figure 9 Relative Mach number at trailing edge for max efficiency.

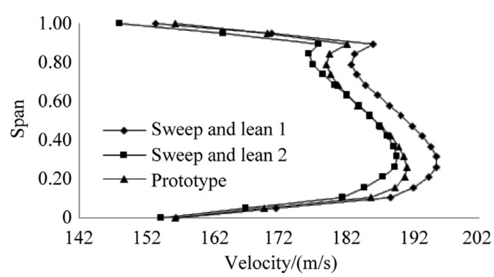

Figure 10 Velocity meridional at streamwise direction for max efficiency.

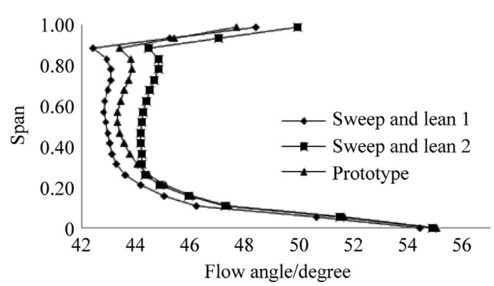

Figure 11 Axial relative flow angle at streamwise direction for max efficiency.

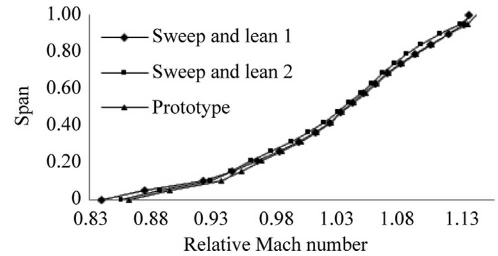

Figure 12 Relative Mach number at leading edge for choked flow.

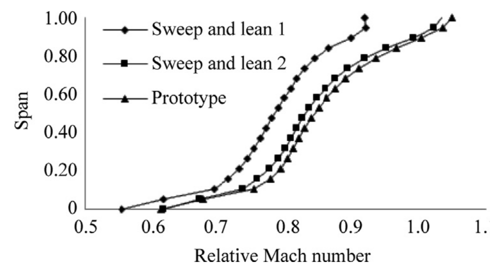

Figure 13 Relative Mach number at trailing edge for choked flow.

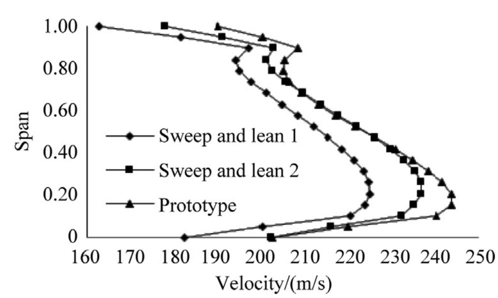

Figure 14 Velocity meridional at streamwise direction for choked flow.

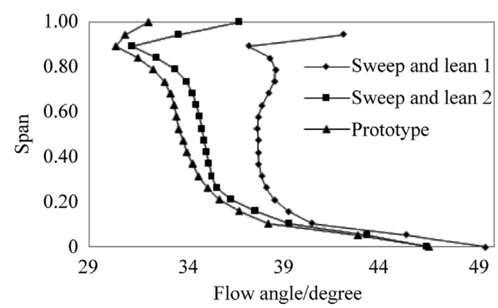

Figure 15 Axial relative flow angle at stream-wise direction for choked flow.

with the maximum efficiency state are approximately the same for all the three rotor and relative Mach number is greater than 1 after the section of 50% till the blade tip. Moreover, as shown in Figure 9 the value for the relative Mach number at the trailing edge is lower than 1 in the maximum efficiency state. This phenomena shows that some part of dynamic pressure has converted to static pressure. Figure 10, which shows the relative velocities variations from leading edge to trailing edge indicates that the maximum velocities is due to the rotor number 1 which the influence of sweep is greater than lean in it. The minimum velocity is due to the generated rotor number 2. In Figure 11 as we proceed to higher sections, the difference of flow relative angel in the blade surface increases which can be explained by the appearance of shock wave in the high sections of the blade and forcing different conditions to the flow due to different shapes of the rotor's blade in three aforementioned geometries. In Figure 12, since the

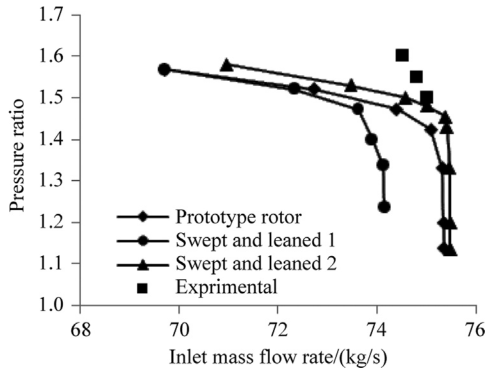

Figure 16 Pressure ratio map.

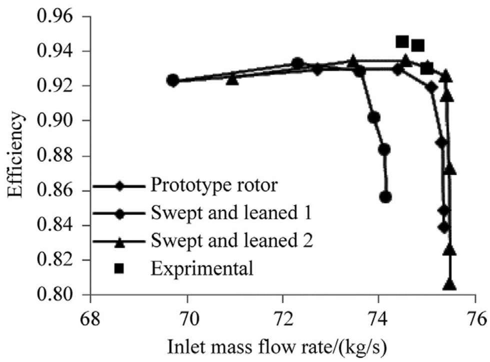

Figure 17 Performance map.

shock wave moves to the trailing edge of the blade, it has a little influence on the relative Mach number conditions in the leading edge. In contrast, in the trailing edge of the rotor the charts describing the relative Mach number have gone far away each other (Figure 13). Figure 14 shows the relative velocity in the state of choked flow. In the Figure 15 variation of relative flow angel in the choked state is shown and it can be seen that the maximum relative exit angel is due to the generated rotor number 1.

The results of experiment can only be achieved for mass flow rate, pressure and temperature and there was not enough facilities to achieve other characteristics. The results of numerical solution in Figure 16 show that quantity of different points is achieved by considering changes in exit pressure. If the back pressure of rotor is reduced, shock wave moves toward trailing edge. Finally there is no change in mass flow rate by reducing back pressure and in this condition flow is choked. This process has been done for all three geometries without changing in configurations. In prototype geometry, the maximum of mass flow rate is 75.2 kg/s and the maximum pressure ratio is also 1.57. In rotor number 1 the maximum pressure ratio is the same but the maximum mass flow rate is reduced by 1.1 kg/s but in rotor number 2, beside a little improvement in pressure ratio, the max on mass flow rate is increased by 0.3 kg/s. In Figure 17 the efficiency of rotor number 2 is improved by one percent in comparison with the prototype rotor. Rotor number one also experiences 0.5% increase in efficiency.

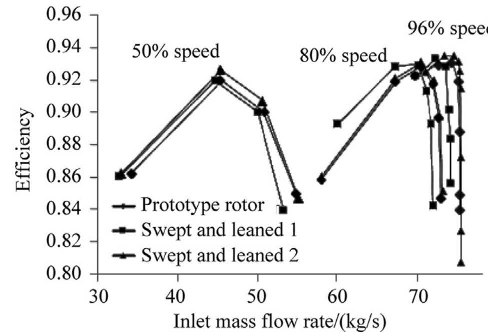

Figure 18 Performance map for various speed.

5. Conclusion

Shock is a flow phenomenon that can have the highest impact on the local performance of a rotor. In view of the Mach number contours, it is obvious that the shock pattern is influenced by blade curvature. Considering the shock pattern of the upper sections of blade, it is clear that the pattern of shock has a considerable curvature which certainly helps reduce the aerodynamic losses of the shock and thereby increase the overall efficiency of the rotor. On the other hand, in the impact with the suction surface, the downstream flow is mostly affected and the risk of boundary layer separation is increased. For the backward-swept blade, the shock is mostly slanted from the hub to the blade tip. As a result, changing the curvature of the leading edge has a significant effect on major changes in the passage blade region.

In the near-stall conditions, by looking at the relative Mach contours, it is observed that the generated shock is in a region much farther from the region of maximum efficiency; thus, due to the reduction of flow rate, the angle of incidence is larger.

With the improvement of rotor blade geometry, which had been achieved by changing the stacking line profile of the blade, the observed shock patterns at critical blade sections, in a state of stall in the modified Blade No. 1, are less distant from the leading edge. In the case of maximum stage efficiency, the shock wave sticks to the leading edge, and this is due to the backward blade sweep. By influencing the downstream flow, backward blade sweep affects the next rotor stages as well; while forward blade sweep affects the upstream flow. Blade lean is clearly more effective than blade sweep and it increases both the stage efficiency and the chocking flow rate. In fact, the lean of the blade bends the pressure surface of the blade, which increases the pressure applied on fluid and to some extent prevents the fluid from deviating towards the upper sections of blade, which is effective in reducing the inter-blade flow losses. Although in previous articles, the effects of blade sweep and lean have been demonstrated for a specific operating range, the results obtained in this paper have proved the improvement of efficiency and chocking flow rate for the rotational speed range of 50%–100%, and has shown that the highest improvement of efficiency belongs to the rotational speed of 96%, (Figure 18). Therefore is seems that the combined sweep and lean of rotor blades is more effective in improving the performance and that it improves both the chocking flow rate and efficiency in a vast operating range of a compressor stage. A more general conclusion is that the simultaneous effect of blade sweep and lean on the increase of efficiency and stall margin occurs in a more extensive range relative to the design point and reduces the sensitivity of stage efficiency to flow rate variation. It seems that the resulting improvement is due to the combined effect of blade sweep and lean. The contours related to the meridian view indicate that the generated shock follows a more regular trend in the modified blades, which is affected by the blade geometry. The positive influence of blade sweep and lean on the improvement of blade performance has been confirmed through many experiments. However, in most of the previous investigations on the blades of acoustic pass compressors, the effects of blade sweep and blade lean had been separately considered.

Finally, it should be emphasized that, for reducing the shock-induced losses, it is important to consider the three-dimensional structure of the shock.

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