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Effects of different turbulence models in simulation of unsteady tip leakage flow in axial compressor rotor blades row

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Abstract

Characteristics of rotor blade tip clearance flow in axial compressors can significantly affect their performance and stable operation. It may also increase blade vibrations and cause detrimental noises. Therefore, this paper contributes to the investigation of tip leakage flow in a low speed isolated axial compressor rotor blades row. Simulations are carried out on near-stall condition, which is valuable of being studied in detail. In turbomachines, flows are non-isotropic and highly three-dimensional. The reason arises from the complicated structure of bound walls, tip leakage flows, secondary flows, swirl effects, streamlines curvatures and pressure gradients along different directions. Therefore, accurate studies on tip leakage flow would be accompanied by many challenges such as adopting suitable turbulence models. So, investigations are carried out numerically utilizing two well-known turbulence models of k-ε and k-ω-SST, separately. It is shown that the k-ε model yields poor results in comparison to the k-ω-SST model. To realize reasons for this discrepancy, turbulence parameters such as turbulent kinetic energy, dissipation and eddy viscosity terms at the tip clearance region were surveyed in detail. It is found out that estimation for eddy viscosity term is too high in the k-ε model due to excessive growth of turbulent kinetic energy, timescale, and lack of effective damping coefficient. This leads to dissipation of vortical structure of flow and wrong estimation of the flow field at the rotor tip clearance region. Nevertheless, k-ω-SST turbulence model provides results consistent with reality.

Nomenclature

C	Blade chord length at rotor tip	Q	Flow rate
C_z	Axial velocity	T	Rotor blade passing period
C_θ	Tangential velocity	U	Peripheral speed of blade
C_p	Static pressure coefficient $(p-p_r)/\ 0.5\rho u_t^2$	ε	Turbulent dissipation rate
$C_{p,rms}$	Root mean square of static pressure coefficient	ω	Specific dissipation rate (ε/k)
f	Frequency	T^*	Turbulent time scale (k/ε)
k	Turbulent kinetic energy	c_μ	Damping coefficient
L	Shaft power from torque and speed	ν_t	Eddy viscosity
P	Static pressure		

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1. Introduction

The extent of rotor blade tip clearance in turbomachines can highly affect their performances due to the characteristics of subsequent tip leakage flow. The unsteady nature of this flow may highly influence the stable operation of the turbomachine. It may also magnify blade vibrations and generate undesirable noises. Due to the importance of the tip leakage flow effects, many investigations have been focused on this subject by researchers, either experimentally or numerically. Although experimental measurements are the mainstream of studies of tip leakage flow, the recent trend is applying numerical methods based on solving Reynolds Average Navier Stokes equations as a complementary tool to the experimental results. The numerical simulation is drawing more attention for studying this type of flow, since it compensates drawbacks of experimental implementations.

Flow in turbomachines is highly three-dimensional and complex. Large gradients of pressure, velocity and angle in all directions occur in most of turbomachinery. The occurrence of tip leakage flow and interact with the mainstream added to its complexity [1]. Tip vortex core size, position of its centerline, intensity of vorticity, velocity field and turbulence intensity distribution are of the main parameters for numerical analyses of this type of flow. For concise exploring of this flow, employing a suitable turbulence models is essential.

In comparison with results obtained from Baldwin Lomax algebraic equations, Dunham studies on an axial compressor rotor blade row show better results of flow analyses obtained through the turbulence models with isotropic eddy viscosity (i.e., k-ε and k-ω models) [2]. Comparing with one/two-equation models, DNS, LES and RSM models demand heavy and long computational tasks. Taking into account the accuracy, computational time and costs, two-equation models are commonly applied to study the tip leakage flow structure and its effects on the turbomachine performance. What follows is a brief review of the literature regarding k-ε and k-ω turbulence models applied to axial flow compressors.

Hwang et al. [3] applied k-ε turbulence model to obtain performance characteristics of an axial compressor for mass flow rates at near stall conditions. This model was also used by Zhang et al. [4], Du et al. [5, 6], Tong et al. [7] and Geng et al. [8] in their analyses of tip leakage flow of centrifugal and axial compressors. Their results included location and trajectory of tip leakage vortex flow, unsteadiness caused by the tip leakage flow, fluctuating flow frequencies, and circumferential propagation of disturbances at near stall mass flow rates. The approach for controlling unwanted phenomena and delaying stall process were also studied by Geng [9] in another work using the same turbulence method. On the other hand, k-ω turbulence model has been under consideration by many researchers to study the tip leakage flow. In this respect, Menter et al. [10] presented industrial examples with the SST turbulence model during ten years. Ramakrishna et al. [11], for instance, used unsteady analyses of flows for studying characteristics of the stall and tip leakage flow in compressors with forwarding swept blades. Gourdain and Leboeuf [12] and Gourdain et al. [13] conducted their investigations to survey vortical structure of tip leakage flow and treatment on the casing and blade tip for controlling and modifying its structure when the stall occurrence is probable. Yamada et al. [14] and Yamada and Funazaki [15] studied the relationship between the tip leakage flow and rotating disturbances on an axial compressor using an unsteady numerical approach.

A recent trend of studies has been focused on the comparison between different turbulence models in numerical analyses of such flows. Glanville [16] utilized Baldwin-Lomax algebraic and Spalart-Allmaras models to study tip leakage flow in an axial compressor. He found out that losses occurred in the compressor and accuracy of the results are subjected to the turbulence model type. Turner and Jennions [17] conducted an investigation to compare turbulence models for steady analysis of flows in a transonic fan. Lui et al. [18] analyzed RANS turbulent models for simulating flows in isolated rotor blade rows of axial compressors at design condition. They used RSM, v2-f, SST, algebraic mixing plane, k-ε, and Spalart-Allmaras turbulent models. They find out that results obtained by RSM are more precise than the other techniques. Bardina et al. [19] performed a research to evaluate and validate four known turbulence models including Wilcox two-equation k-ω model, Launder and Sharma two-equation k-ε model, Menter two-equation k-ω and k-ε SST model, and Spalart and Allmaras one-equation model. Also, other specialists such as Ito et al. [20], Benini et al. [21], Calvert et al. [22] focused on diverse works such as flow on the blade edge and changes in the blade geometry with different turbulence model. Simoes et .al [23] showed the use of different turbulence models applied to steady CFD simulation of turbulent flow inside a rotor of an axial flow compressor [23]. Dailey [24] and Merz [25] surveyed performance of compressors through steady analyses using k-ε and k-ω turbulence models and found out that the k-ε model overestimates than actual values and vice versa for the latter.

The literature review showed that unsteady numerical analyses are mainly adopted for surveying the structure of the tip leakage flows and consequent phenomena. An overview of the literature reveals that all studies are limited to the two-equation turbulence model which affects performance characteristics, but details of unsteady flow structure of tip leakage flow in near stall condition are under question. Moreover, the main turbulence parameters which cause a discrepancy in the results are not surveyed accurately. So, the necessity of unsteady study is more evident, especially when computation cost and time required by k-ε and k-ω-SST are taken into account.

The current paper reports numerical results of the tip leakage flow simulation obtained for an isolated rotor blade row of an axial compressor while operating at near stall condition. Initially, the steady analysis is conducted, and the consequent performance map is compared with available experimental results, which shows a close agreement. Then, using k-ε and k-ω-SST turbulence models, unsteady analysis of the tip leakage flow is executed for mass flow rates near to stall condition. Frequency analysis of static pressure signals at the tip region of the blade is also carried out for each turbulence model used through the analyses. Resultant flow fields obtained from the two models are compared with each other to evaluate their effectiveness for correct simulation of the tip leakage flow, particularly at the near stall condition. Finally, reasons for differences between the two models are explained.

2. Model specifications and numerical model

The test model is a low speed isolated axial compressor rotor blade row. This compressor was used by other researchers like Inoue et al. [26] and Furukawa et al. [27]. The compressor is comprised of 12 blades designed based on NACA-65 airfoil series. Geometric specifications of this rotor blade row are listed in Table 1 [26]. The Reynolds number based on the blade midspan chord length and the rotational speed of the rotor blade row are selected 3.77×10^5 and 1300 rpm, respectively. The full annulus of the compressor is split into three parts including 4 rotor blades each, as shown in Fig. 1. For generating the mesh system a multi-block structured grid type is employed. Each flow passage includes 74 streamwise nodes, 50 spanwise nodes, and 60 pitchwise nodes. The radial space between the blade tip and compressor casing is divided into 16 nodes. Fig. 1 shows the surface grid system generated on solid walls of the model. The density of grid close to the walls is set to keep $y^+ < 5$ which makes it possible to evaluate viscosity flux close to the walls without using wall function and only by considering the no-slip condition and adiabatic wall. The total grid structure includes 890000 cells.

Table 1. Model specifications.

Hub diameter (mm)	270
Hub to tip ratio	0.6
Tip clearance (% of chord)	1.7
Tip chord length (mm)	117.5
Midspan chord length (mm)	117.8
Solidity of rotor blades at tip	1
Tip stagger angle (deg.)	56.2
Midspan stagger angle (deg.)	47.2

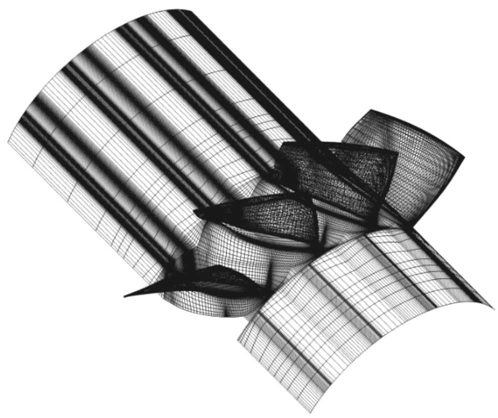


Fig. 1. Computational geometry and grid distribution on the blades and hub solid walls.

To obtain the blade row performance map, a steady analysis is executed, while the unsteady simulation is adopted for analyzing tip leakage flow at near stall condition. The well-known commercial flow solver package of Fluent has been used for the current study. This flow solver is a three-dimensional, viscous and time-accurate code. For the solution of governing equations of continuity and momentum, the computation method utilizes a finite volume scheme. Moreover, the usual simple algorithm is employed to couple the velocity and pressure fields.

Time discretization is also carried out using second-order implicit scheme to solve the unsteady equations. It should be noted that low accurate discretization methods may lead to excessive numerical diffusions; consequently may reduce the accuracy of the results. So, an upwind second-order model is employed for discretization of k, ε, ω and momentum equations.

Inlet velocity and flow direction are specified on the inlet boundary and the static pressure distribution is imposed at the outlet by means of radial equilibrium law. No-slip and adiabatic conditions are imposed all over the solid walls. In the current analysis, multiple reference frames are used. Consequently, the compressor hub is divided into the stationary and rotating parts, with respect to an inertial frame. For the repeating boundaries, the periodic boundary conditions are imposed. A sliding interface is used between different fluid zones. For the unsteady simulation purposes, the time step is chosen in such a way that one blade passing to be completed in 120 steps. The convergence criterion in numerical simulation process is the residual values of the main governing equations approaching to around 10^{-7}. Statistical steady state monitoring of the blade lift and flow variables at some selected points are also executed, as the complementary convergence criteria.

3. Performance curves and validation

The blade row performance map, in terms of stage loading coefficient and rotor blade row efficiency versus flow coefficient, is executed taking into account the tip leakage flow effects. Fig. 2 compares numerical results with those obtained through the experiments by Inoue [26]. Results are obtained utilizing both k–ε and k-ω-SST turbulence models, which coincide each other. This figure shows a good agreement between numerical and existing experimental results [26].

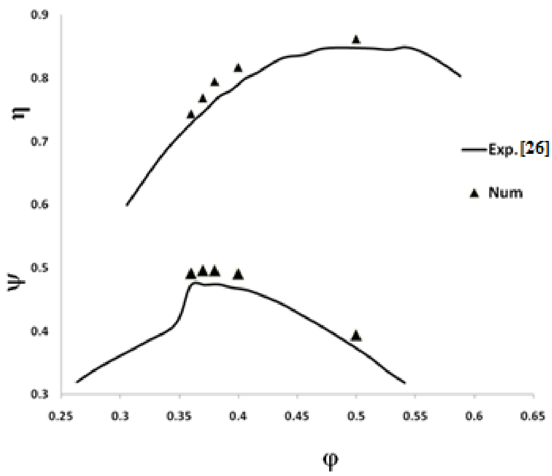


Fig. 2. Compressor blade row performance map

4. Tip leakage flow unsteady analyses at near stall condition

This section is donated to an unsteady analysis of tip leakage flow of blade row near to stall condition of the compressor. Furukawa et al. [27] showed that φ = 0.36 is near to stall condition, where both generation and breakdown of tip leakage vortex are observable in the compressor (Fig. 2). Thus, unsteady numerical analysis of the flow should be conducted for a precise simulation at this flow coefficient.

As mentioned earlier, k-ε and k-ω-SST turbulence models are applied for estimation of the eddy viscosity. It is worth mentioning that all turbulence models derived from k–ε including k–ε-Standard, k–ε-RNG and k–ε-Realizable models are also examined and similar results are obtained.

Based on experimental results of Inoue and Karoumaru [26] minimum pressures are observed along the centerline of the tip vortices. Therefore, it would be possible to find the location of tip leakage vortex and its trajectory by tracking the pressure field. Static pressure fields of the proposed model at 97% span measured from the blade root section are shown in Figs. 3 and 4 for k-ε and k-ω-SST turbulence models, respectively. Nine successive instants with a time interval of T/4 are considered for a total duration of two periods (i.e., 2T time interval). For a better understanding of the flow structure, streamlines are drawn in one moment. As shown in Fig. 3, the flow field does not change with time. In other words, k–ε turbulence model is not able to simulate the tip leakage vortex flow and capture its unsteady behavior. On the other hand, spots with low pressures are obvious in the contours obtained from k-ω-SST turbulent model (see Fig. 4).

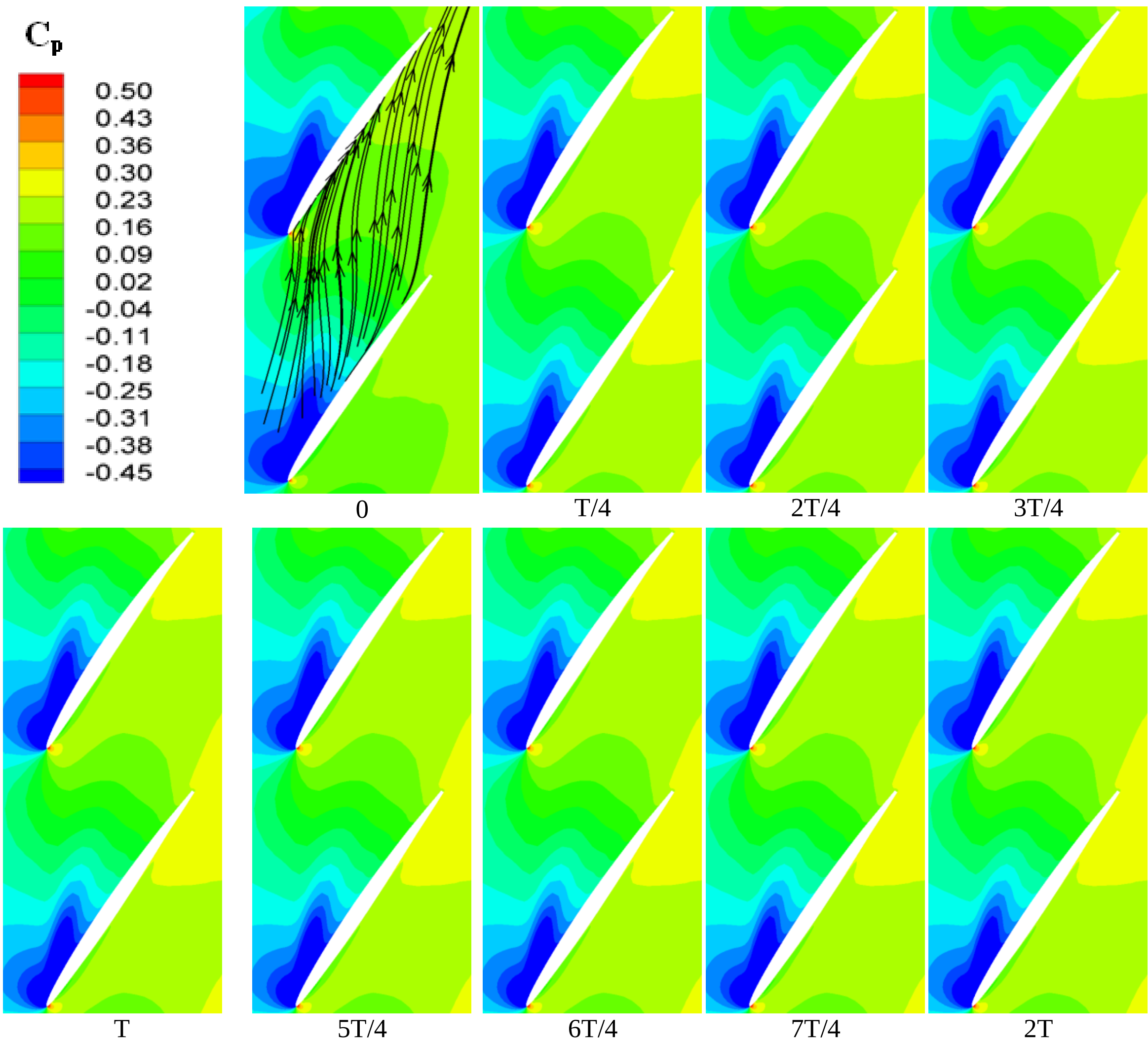


Fig. 3. Unsteady static pressure fields at 97% span using k-ε turbulence model.

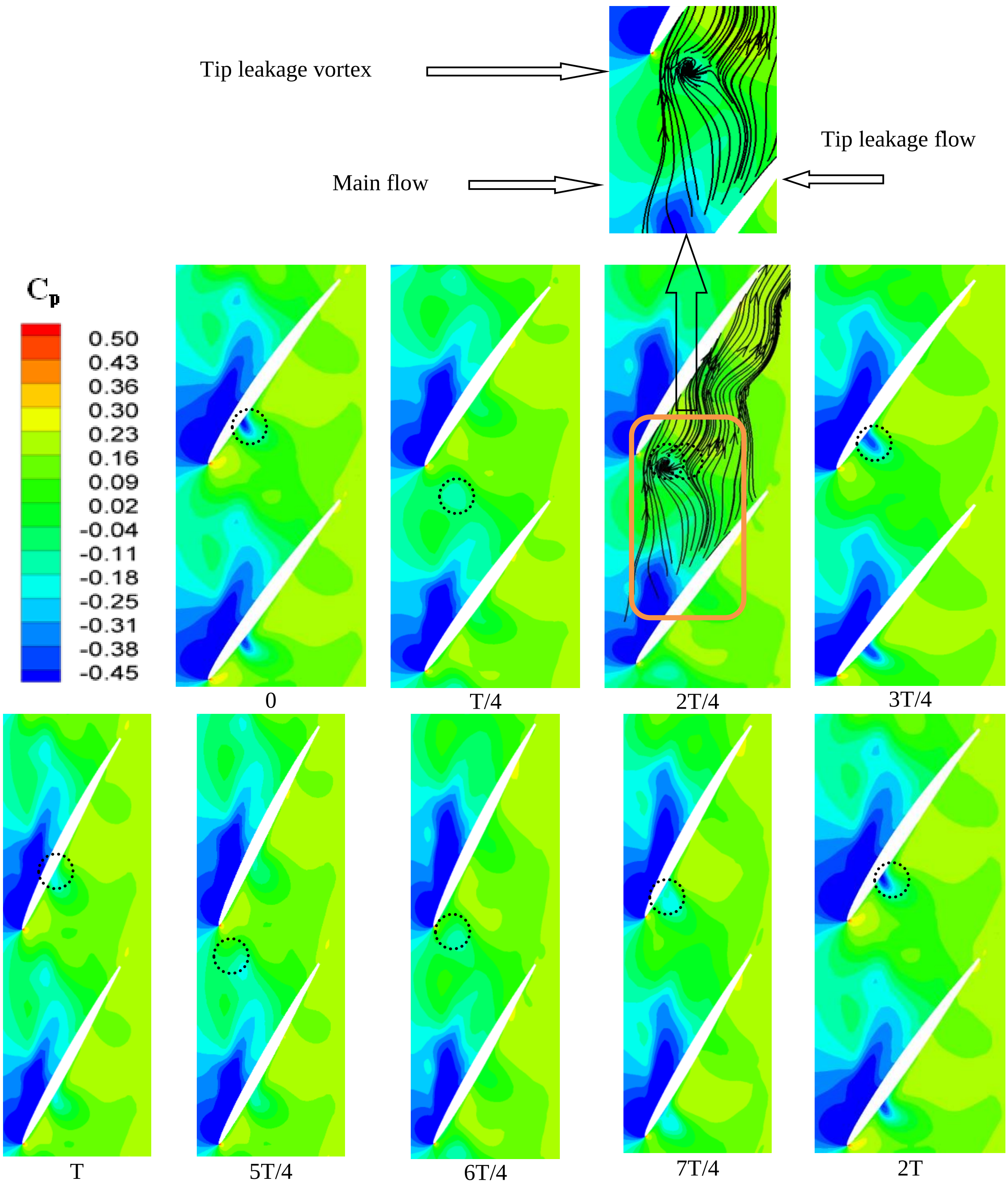


Fig. 4. Unsteady static pressure fields at 97% span using k-ω-SST turbulence model

Streamlines pointing these low-pressure spots are shown for 2T/4 instant. The interaction between the tip leakage flow and main flow leads to tip leakage vortex formation. However, the streamline crosses the passage in the k-ε model without formation of any vortex structure. The low-pressure spot is designated by a dashed line circle. It can be detected from Fig. 4 that the low-pressure zone moves from the blade leading edge towards the pressure side of the adjacent blade. It impinges on the pressure surface of the adjacent blade to downstream of the passage and eventually dissipates. Evidently, time interval T denotes the duration between the tip vortex formations from the leading edge until its breakdown. These results reveal the fact that the tip vortex flow behaves periodically and breakdown frequency is nearly the same as the blade passing frequency.

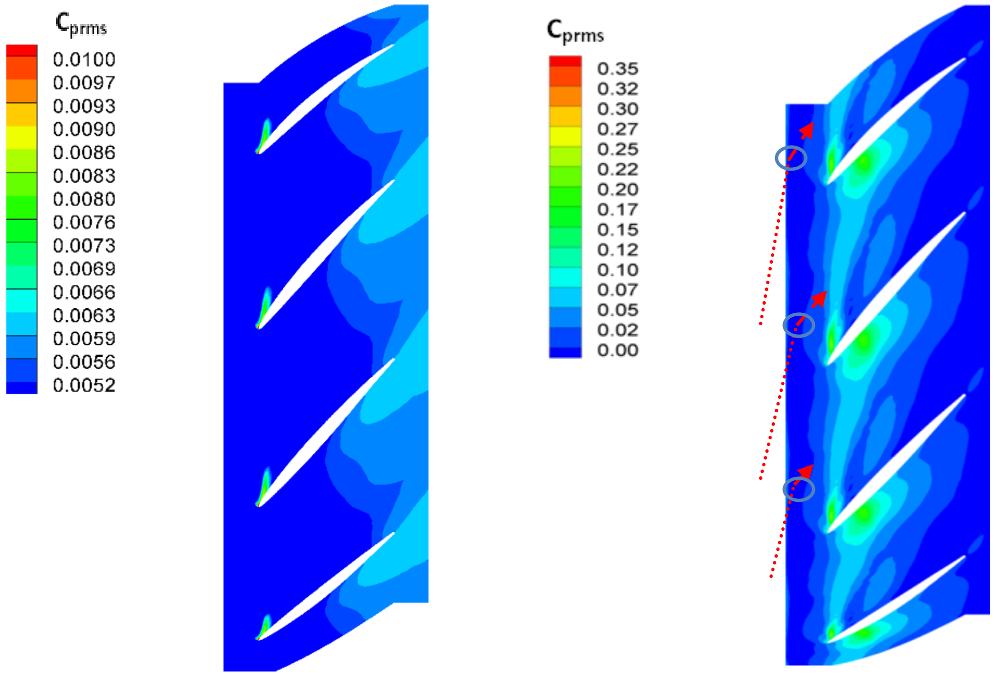
To quantify the unsteady behavior of the tip leakage flow, the root mean square (RMS) values of the static pressure coefficients are calculated, and the final results are shown in Fig. 5. These results are presented at 97% span stream surface for both the k-ε and k-ω-SST turbulence models.

The figure demonstrates the general behavior of the unsteady tip vortex flow including its intensity, trajectory, and size. It is clear that using k-ε model gives no signs of the vortex formation. While the contour for k-ω-SST shows that the unsteady pressure fluctuations originate from the blade leading edge and move towards the pressure side of the adjacent blade along the dotted line, as shown in Fig. 5. This line actually coincides with the time-averaged tip leakage vortex trajectory [26]. In other words, the tip vortex flow fluctuates around this line. It can be detected from the figure that it impinges the opposite blade and moves downstream.

5. Frequency analyses

Unsteady nature of the blade tip vortex flow can be investigated through surface pressure studies. Some monitoring points are selected close to the blade tip in this respect. Static pressure signals are picked up at each time interval after the final convergence of the numerical analysis. In this study, the signals are collected for a flow coefficient of 0.36, i.e., near stall condition. These monitoring points are distributed within 10% to 50% of the blade chord length measured from its leading edge in the streamwise direction (designated by P1 to P4). Investigations have been carried out for the pressure sides of all 4 blades.

Fig. 6 shows the results for point P1 (at 10% of the chord length) for each blade. While the results for points P1 to P4 on one blade are presented in Fig. 7. The blue lines belong to the signals, obtained from the k-ω-SST model, and the red lines belong to signals from the k-ε model. Clearly, static pressure results obtained through the k-ε turbulence model show no significant oscillations. But considerable fluctuations are obvious in the pressures results utilizing k-ω-SST model.



(a) (b)

Fig. 5. Unsteady fluctuations of static pressures at 97% span stream-surface; (a) k-ε and (b) k-ω-SST

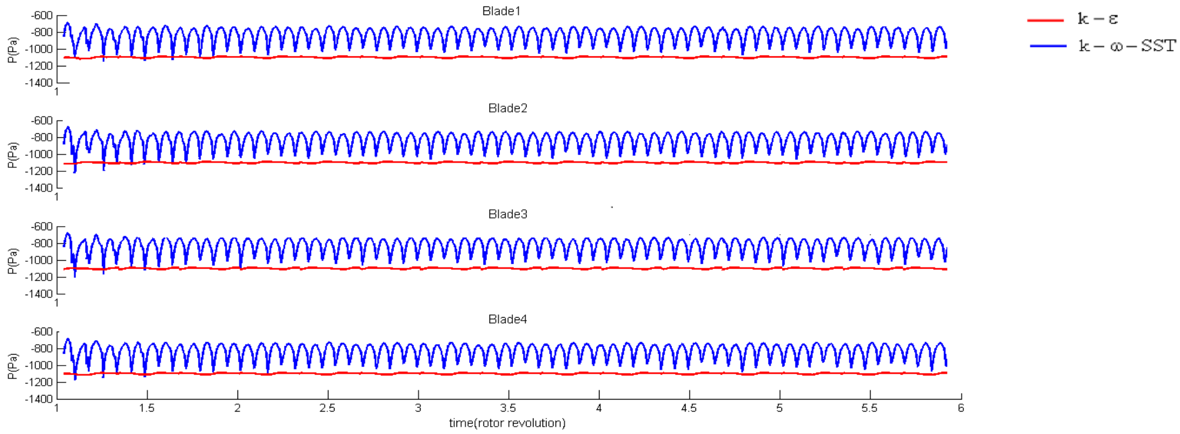


Fig. 6. Static pressure signals at point P1 (10% chord) of each blade.

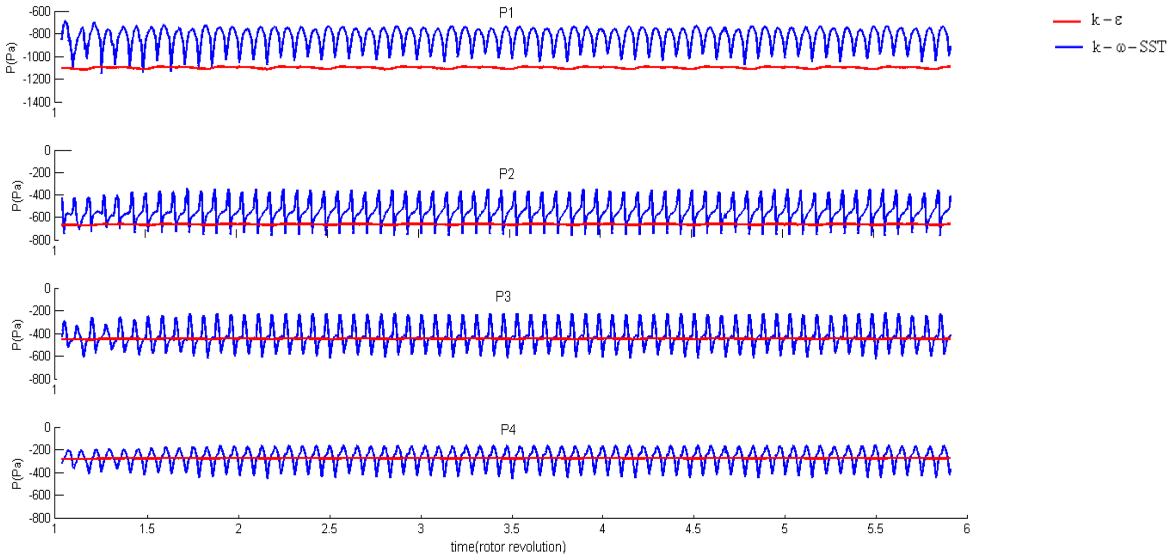


Fig. 7. Static pressures signals at points P1– P4 (10% - 50% chord) on one blade.

The frequency spectrum of the pressure signals at points P1 to P5 for one blade obtained by the two proposed turbulence models is shown in Fig. 8. As can be seen in the spectrum, results of k-ω-SST model predict a dominant frequency of 1.1 times the blade passing frequency (i.e., 286Hz) with the highest amplitude. The fact that the monitoring points are fixed on the blades and rotated with them implies that the blade passing frequency would not be visible at the frequency analysis. Therefore, the existing frequency would be the tip leakage vortex frequency (from generation time to dissipation time). This is consistent with the results extracted from Fig. 3, where the period from generation to dissipation time at this mass flow rate is T.

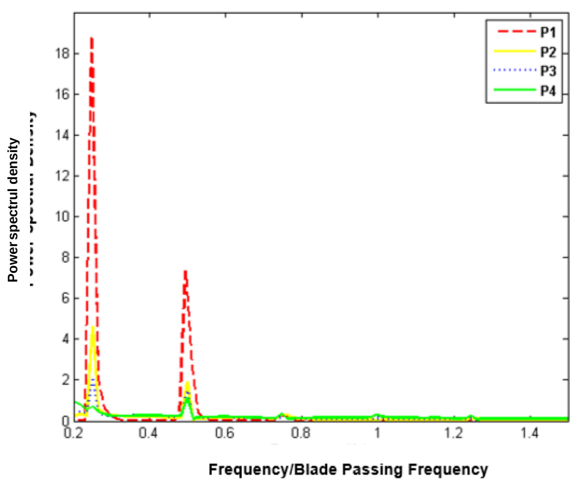
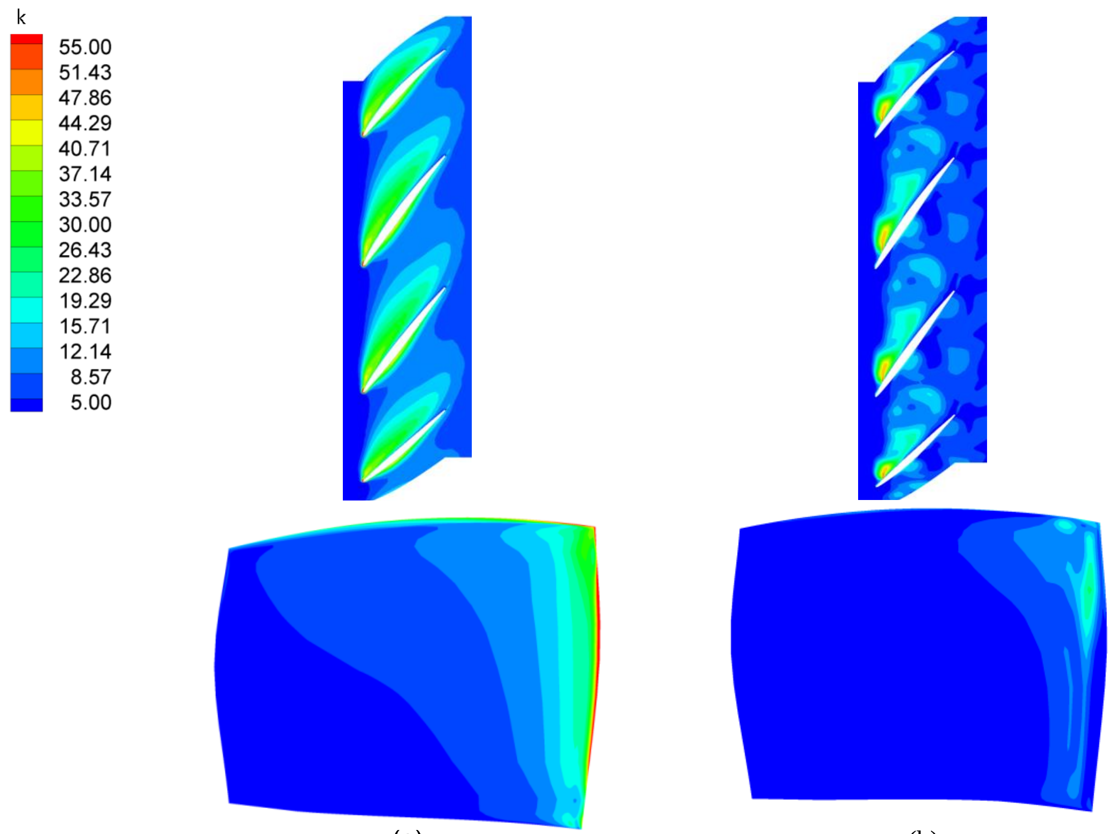


Fig. 8. Frequency spectrums in the rotor blade row tip region at points P1 to P4, (a) k-ε, (b) k-ω-SST.

That is, the frequency of the phenomenon is the same as the blade passing frequency. Frequency analyses of the signals of different points along the blade chord based on the k-ω-SST model show that the dominant frequency remains unchanged for all points, whereas the amplitude of frequency is changed. This is not the case for the k-ε turbulence model.
Dominant frequencies in the signals from this turbulence model have very low energy spectrum, which is apparently due to the nature of the turbulent flow rather than periodic phenomena caused by the tip leakage flow. That is to say that the pressure signal analyses hint merits of the k-ω-SST turbulence model and incapability of the k-ε model for identifying the unsteadiness nature of the tip leakage vortex flow. Evidently, in the k-ε turbulence model, turbulent kinetic energy values for the regions starting from the suction surface of the blade, which are occupying about half of the passage, are higher than other passage regions.

6. Comparison of turbulence models for unsteady for unsteady flow analyses

In turbulence models, some complicated equations are reduced to algebraic equations. For instance, in the k-ε turbulence model, k and ε are solved by these equations without any limitation on k, ε, or μ_T. The efficiency of different turbulence models, while dealing with unsteady tip leakage flow of axial compressors, is investigated by surveying parameters such as turbulent kinetic energy, dissipation rate, turbulent time scale and eddy viscosity. It is found that lack of any limitation for turbulent parameters may lead to non-physical results. Contours of turbulent kinetic energy at 97% span in the radial direction and on the blade surfaces obtained by both turbulence models are illustrated in Fig. 9. For the sake of logical comparisons, the contours of both models refer to the same time.



(a) (b)

Fig. 9. Turbulent kinetic energy at 97% span and on the blade surface; (a) k-ε and (b) k-ω-SST.

In comparison with k-ω-SST, distribution of k on the blade surfaces also points out higher values of the parameter especially at leading edges of the model. One explanation for the increase in k, as mentioned by Launder and Kato [28], is the fast strain rate caused by the formation of a stagnation point. However, Medic and Durbin [29] found later that it is not the stagnation point that should be under the account, but it is the high strain rate that should be highlighted.

Another effective parameter of the turbulent flow obtained by turbulent transfer equation is the rate of dissipation. This parameter directly affects eddy viscosity. Fig. 10 shows contours of dissipation rate of turbulent kinetic energy (ε) in both the proposed turbulence models. Clearly, the value obtained by k-ω-SST is higher than that of k-ε. Therefore, it can be concluded that higher turbulent kinetic energy in k-ε (Fig. 9) is not dissipated.

Excessive increase in turbulent kinetic energy and reduction in dissipation rate of the energy cause the turbulent time scale (T^*) of k-ε to exceed from than that of the k-ω-SST model. This fact can be detected from Fig. 11. Eq. 1 clarifies that the increase in turbulent kinetic energy and time scales are followed by the increase in the eddy viscosity. Fig. 12 shows eddy viscosity at 97% radial span and on the blade surface obtained from the two proposed turbulence models.

$$\upsilon_t = c_\mu k T^* \qquad (1)$$

Consequently, the turbulent viscosity can unreasonably grow in certain positions. Following equations show how eddy viscosity can be obtained by two turbulence models. According to Eq. 2, in addition to dependence of eddy viscosity on k and ε parameters, it is dependent on cμ, too.

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon} \qquad (k-\varepsilon) \qquad (2)$$

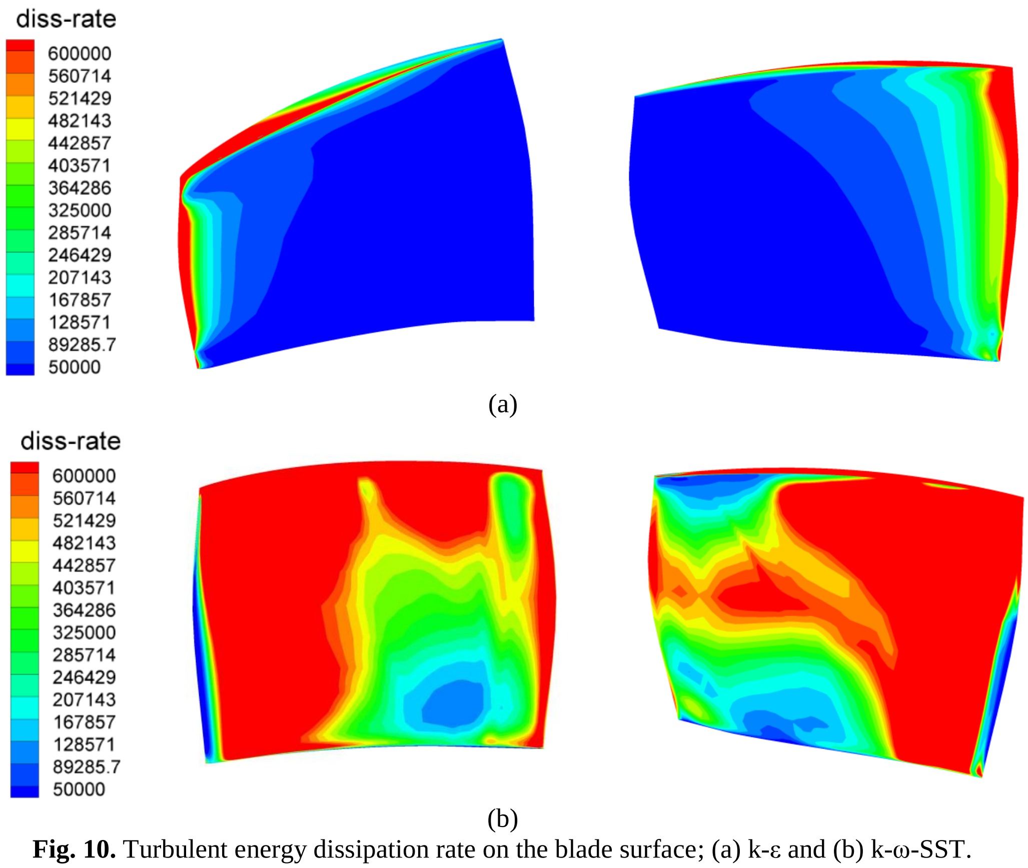


Fig. 10. Turbulent energy dissipation rate on the blade surface; (a) k-ε and (b) k-ω-SST.

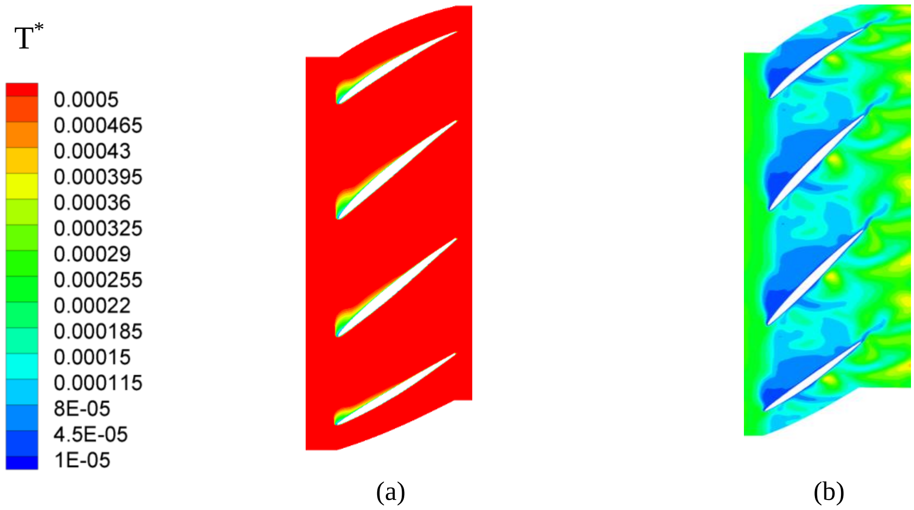


Fig. 11. Turbulent time scale at 97% span; (a) k-ε and (b) k-ω-SST.

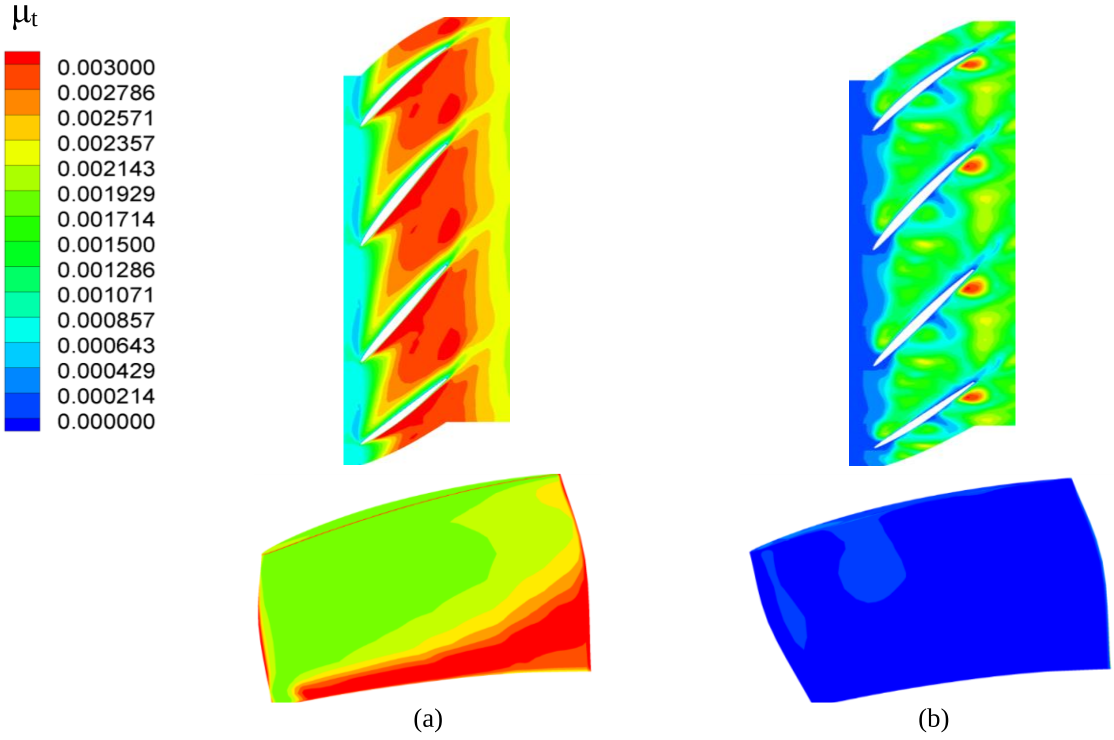


Fig. 12 Turbulent viscosity at 97% span and on the blade surface, (a) k-ε, (b) k-ω-SST.

In the k-ε turbulence model, the parameter cμ is a constant coefficient and is therefore unchanged even in situations where the turbulent kinetic energy grows excessively. Subsequently, eddy viscosity increases under these conditions, which is far from the reality. This fact can be detected by referring to Fig. 12. It should also be noted that although cμ factor is not always constant in realizable k-ε model (Eq. 2.3), it results in non-physical turbulent viscosity for computational domains including both rotating and stationary zones such as multiple reference frames and rotating sliding meshes. This lies with the fact that mean rotation effect is taken into account in the definition of viscosity. This model is more preferred than the other models when the single rotating frame is under consideration. However, due to the nature of this modification, the model must be used for multiple reference frames and rotating sliding mesh system with more cautious. In the present investigation, in spite of using the realizable k-ε model similar results are obtained as those of k-ε model.

$$c_\mu = 0.09 \qquad (Standard) \tag{2.1}$$

$$c_\mu = 0.085 \qquad (RNG) \tag{2.2}$$

$$c_\mu = \frac{1}{(A_0 + A_S \frac{kU^*}{\varepsilon})} \qquad (Realizable) \tag{2.3}$$

The cμ parameter in k-ω-SST is not constant. Turbulent viscosity equation (Eq. 3) showed that where the local Reynolds number is low, and/or where the solution of the governing equations result in inaccurate and excessive increase in turbulent kinetic energy, decrease in dissipation and increase in strain rate, the maximum value of ($1/\alpha^*$) and ($SF2/\alpha 1\omega$) must be adopted. Thus inverse of this value would be used for inserting damping effect on the eddy viscosity.

$$\mu_t = \rho \frac{k}{\omega} \frac{1}{\max[\frac{1}{\alpha^*}, \frac{SF2}{\alpha_1 \omega}]} \quad (k - \omega - SST) \tag{3}$$

where

$$\alpha^* = \alpha^*_\infty \left(\frac{\alpha^*_0 + \mathrm{Re}_t / \mathrm{R}_k}{1 + \mathrm{Re}_t / \mathrm{R}_k} \right)$$

$$\mathrm{R}_k, = cte\ ,\ \alpha^*_0 = cte,\ \mathrm{Re}_t = \frac{\rho k}{\mu \omega}$$

$$F_2 = \tanh(\phi_2^2),\ \phi_2 = \max[2 \frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho y^2 \omega}]$$

The model also overcomes limitations in multiple reference frames and rotating sliding meshes. Thus, in k-ω-SST turbulence model, estimation of eddy viscosity would be more close to actual value. Since the turbulent transfer equation and eddy viscosity are coupled, any error in μ_t value or k, ε and ω parameters would be accompanied by reciprocated effects on each other. Therefore, in the k-ε turbulence model, when k and ε are obtained unreal and left aside without any modifications on the damping coefficient, the value of μ_t also increases virtually. This incorrect cyclic computational system continues without any modifications. That is, the model overestimates eddy viscosity term near the solid walls and within the non-equilibrium flows. This eventuates in high diffusion and dissipation of tip leakage vortex flow. On the other hand, in the k-ω-SST turbulence model, correct assessment of the turbulent values (i.e., k, ε, and T*) and applying viscosity-damping coefficients result in correct estimation of the turbulent viscosity. This, in turn, causes to simulate the tip leakage vortex flow and its inherent unsteady nature more precisely.

7. Conclusions

Effects of different turbulence models on unsteady numerical analysis of tip leakage flow of axial compressor rotor blades are studied at near stall conditions. Results show considerable effects of turbulence model type on unsteady analysis of tip leakage flow. This is in consistent with the results obtained by other researchers. Even at near stall condition, where the tip leakage vortex flow is powerful, the k-ε turbulence model cannot capture this phenomenon. Moreover, no evidence of pressure fluctuations with a considerable amplitude and tip leakage frequency are found. On the other hand, for equal setting, k-ω-SST turbulence model produced acceptable results for tip leakage flow compared with those of the k-ε model. k-ω-SST results are obtained without any extra grid generation and consumed computational time and cost. More detailed survey of the complex structure of the flow and turbulent parameter values at the tip region reveal that the increase in turbulent kinetic energy and timescale eventuates in higher eddy viscosity near to tip clearance zone in the k-ε turbulent model. This leads to dissipation of vortical structure of the flow and pressure fluctuations close to the tip region. However, using k-ω-SST turbulence model is inherently associated with the modification of k and ε values and decrement in the eddy viscosity. Thus, the resultant flow field would be associated with acceptable physical interpretations. Present investigations clarify that the modifications made on the turbulent transfer equations and eddy viscosity term including variable damping coefficients in k-ω-SST model produce results better than those of k-ε model while dealing especially with vertical flows.

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