ISSN 1846-6168
UDK 621.51:533

NUMERIČKO ISTRAŽIVANJE UTJECAJA RADIJALNE ZRAČNOSTI NA PERFORMANSE KOMPRESORA S KOMBINIRANIM TOKOM

NUMERICAL INVESTIGATION OF INFLUENCE OF TIP CLEARANCE IN MIXED-FLOW COMPRESSOR PERFORMANCE

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Izvorni znanstveni članak

Sažetak: *Kod kompresora s kombiniranim tokom, protok rasipanja kroz radijalnu zračnost stvara vršni vrtlog rasipanja zbog interakcije s glavnim tokom te uzročno tok u prolazu rotora čini kompleksnijim. Različite veličine radijalne zračnosti uzrokuju različiti intenzitet smetnje glavnom toku. U ovom članku, numerička analiza provodi se korištenjem komercijalnog koda kako bi se istražio utjecaj radijalne zračnosti na glavni tok. Ocjenjene su performanse rotora kombiniranog toka sa četiri različite radijalne zračnosti između rotora i stacionarnog kućišta te su uspoređene s eksperimentalnim rezultatima. Krivulje performansa rotora dobivene su za različite parametre masenog protoka sa različitim radijalnim zračnostima pri konstrukcijskoj brzini. Rezultati pokazuju kako protok vršnog rasipanja ima snažno međudjelovanje s glavnim tokom i da pridonosi potpunom gubitku tlaka i smanjenju performansa. Smanjenje tlaka i performansi približno je linearno proporcionalno razmaku između rotora i stacionarnog kućišta. Kroz raspored vektora brzine, računalni izračuni otkrivaju kako intenzitet smetnja koje se stvaraju kod međudjelovanja protoka rasipanja i glavnog toka, ima priličan utjecaj na efikasnost. Kvantiteta protutoka je minimalna kad radijalna zračnost iznosi 0.5mm, a usporedno tome, kad je radijalna zračnost 0.75mm, ima značajan utjecaj na glavni tok kroz međudjelovanje s protokom rasipanja.*

Ključne riječi: *kompresor s kombiniranim tokom, radijalna zračnost, efikasnost, proto vršnog rasipanja, numerička analiza*

Original scientific paper

Abstract: *In mixed-flow compressor, the leakage flow through the tip clearance generates the tip leakage vortex by the interaction with the main flow, and consequently makes the flow in the impeller passage more complex. Different tip clearances generate different intensity of disturbance to main flow. In this paper, numerical analysis is performed using a commercial code to investigate tip clearance effects on main flow. The performance of mixed-flow impeller with four different clearances between impeller and stationary shroud are evaluated and compared with experimental results. The impeller performance curves are obtained for different mass flow parameters with different tip clearances at design speed. The results show that the tip leakage flow strongly interacts with main flow and contributes to total pressure loss and performance reduction. The pressure and performance decrement are approximately linearly proportional to the gap between impeller and stationary shroud. Though the velocity vectors distribution, the computed results reveal that the intensity of the disturbance generated by the leakage flow interacts with the main flow has rather a large influence over efficiency. And the quantity of backflow is minimum when the tip clearance is 0.5 mm, while the 0.75mm tip clearance, by contrast, has a considerable effect on main flow by the interaction with leakage flow.*

Keywords: *Mixed-flow compressor, tip clearance, efficiency, tip-leakage flow, numerical analysis*

1 INTRODUCTION

A mixed-flow compressor is favored for applications in small gas turbine engines as it provides smaller frontal area and higher thrust to weight ratio. Maintaining a gap (i.e., tip clearance) between the blade tip and stationary shroud is necessary to ensure the relative motion be-tween the rotor and stationary shroud in a mixed-flow compressor. However, the tip clearance provides a channel for fluid to leak from the pressure surface to the suction surface, which leads to a tip leakage flow, and the tip-leakage flow has a considerable effect on the stage pressure ratio and efficiency, Besides, the efficiency and reliability of the compressor depend to a great extent on flow behavior in its flow passage and flow near shroud (tip) gap. It is well known that the interaction between impeller and shroud has substantially influence over the flow field and performance of the compressor. It is therefore necessary to study and under-stand the complex flow field inside the flow channel of the mixed-flow compressor [1].

King and Glodeck [2] experimentally investigated a parallel cut-off mixed impeller with 0.89 mm (0.3500) frontal clearance to study the performance of the compressor. A very low value of maximum adiabatic efficiency, 0.76, is reported. Wilcox and Rabbins [3] tested an impeller pre-whirl vanes designed using Goldstein's method. The impeller has a maximum tip diameter of about 176 mm and has a peak pressure ratio of 3.7 with impeller adiabatic efficiency of 0.78, which is a very low value. Dallenbach [4] presented a method for aerodynamic design for centrifugal and mixed-flow compressors to achieve prescribed impeller blade-loading distri-butions for impellers with radial blade elements. He also presented experimental velocity-distribution results for 12 impellers including four mixed-flow impellers de-signed with this method. Only one impeller gives satisfactory blade loading. Experimental investigations were carried out by D Ramesh Rajakumar to study the effect of tip clearance (between impeller and stationary shroud) in a mixed-flow compressor stage. Two configurations, namely constant and variable clearance gaps, between impeller and stationary shroud were considered [5].

In a centrifugal compressor, the leakage flow through the tip clearance generates the tip leakage vortex by the interaction with the main flow, and consequently makes the flow in the impeller passage more complex by the interaction with the passage vortex [5]. The influence of the interaction of the tip leakage vortex with the shock wave on the performance of axial compressor is clari-fied by many studies [6, 7], Masanao Kaneko and Hoshio Tsujita [8] clarified the influences of the tip leakage flow on the behavior of secondary flow, the formation of shock wave and the loss generation in the transonic centrifugal compressor at the design condition by using the commercial CFD code, but that in the mixed-flow compressor has not been fully clarified yet.

The objective of this work is to determine the effect of different tip clearances at design rotational speed on the performance of a mixed-flow compressor stage, and the reason will be discussed. The objectives are achieved by using the commercial CFD code.

2 NUMERICAL METHOD

2.1 Physical model

Fig. 1 shows a schematic of the study object: the mixed-flow compressor which was designed by D Ramesh Rajakumar [5]. The design rotational speed is 39,836r/min, and the design pressure ratio and the design mass flow rate are 3.8 and 2.72kg/s, respectively. The impeller had 11 main blades with a cone angle of 60°. Table 1 lists the main parameters of the mixed-flow compressor.

Figure 1 Schematic diagram for the mixed-flow compressor

In the present study, the maximum thickness of the blade tip is 1.15 mm. Software of BladeGen is applied to design the mixed-flow compressor model. Software of TurboGrid is employed to establish fluid calculation area and generate the grid according to the practical size of mixed-flow compressor blade. Then the grid is imported to CFX to calculate the distribution of fluid field around the blade tip, the total pressure ratio (π_{12}) and isentropic efficiency (η_{12}). π_{12} is the ratio of the total pressure mass-averaged on the cross section at the impeller outlet to that at the inlet (=P_{t2}/P_{t1}). η_{12} is defined by

$$\eta_{12} = \left[(P_{t2}/P_{t1})^{(k-1)/k} - 1\right]/(T_{t2}/T_{t1} - 1) \times 100\% \quad (1)$$

Where T_{t1} and T_{t2} are the total temperature mass-averaged on the cross section at the impeller inlet and outlet, respectively, and κ is the specific heat ratio.

Table 1 Specifications of the mixed-flow impeller

	value
impeller inlet parameter	
Impeller inlet tip diameter d_{1it} (m)	0.156
Impeller inlet hub diameter d_{1ih} (m)	0.0625
Impeller rotational speed N (rpm)	39836
Hub-to-tip diameter ratio d_{1ih}=d_{1it}	0.4
Relative blade angle at tip β_{1it} (°)	61.8
Relative blade angle at hub β_{1ih} (°)	36.7
Impeller exit parameter	
Impeller exit tip diameter d_{2it} (m)	0.253
Impeller exit hub diameter d_{2ih} (m)	0.239
Impeller exit blade height f b_{2i}(m)	0.0141
Relative blade angle at tip β_{2it} (°)	58.5
Relative blade angle at hub β_{2ih} (°)	46.8
Absolute flow angle α_{2it} (°)	70
Number of blade Z_m/Z_s(Main/Splitter)	11/11

2.2 Computational model and boundary conditions

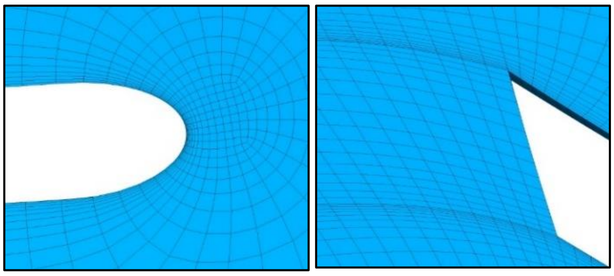
The CFD (computational fluid dynamics) software CFX is used to simulate the internal flow of mixed-flow compressor with the assumption that the flow is the steady state compressible flow. The $\kappa - \varepsilon$ turbulence model is widely employed for simulating complex flows, including swirling flow, secondary flow and boundary layer separation under adverse pressure gradient and this model properly reflect the effects of transient flow and streamline curvature; additionally, the simulated results are verified to agree well with experimental results [10, 11]. Thus, the $\kappa - \varepsilon$ turbulence is chosen to solve the three-dimensional steady Reynolds-averaged equations in this study. And air is considered as an ideal gas. The CFD solution adopted SIMPLE method. The boundary condition of the inlet are total temperature and total pressure, and mass flow of the outlet. The MRF (multiple reference frame) model is selected to couple the rotating impeller and stationary shroud [11]. The blade wall is set to the no slip boundary condition, and Counter Rotating Wall is used for shroud. The calculation can be reliably considered to have converged when the residuals of the parameters and velocity in all directions are less than 10-4, the volume flow rate difference between the inlet and outlet reaches 10-5 simultaneously.

2.3. Mesh generation and independence verification

Structured high-resolution hexahedral meshes are used for the discretization of the one-passage model (see Figure 2a). ANSYS Turbogrid 15.0 with ATM optimized option was used for grid generation of rotor domains and tip clearance. For the surrounding of the blades, an O-Grid is built up to give a satisfactory resolution for the boundary layer near them and the value of y^+ is approximately 25. Therefore, near-wall mesh resolution is acceptable. While for the blade passage, an H-Grid is applied. The detailed grid information near the leading and trailing edges of main blades are enlarged in Fig.2b for better views. The numbers of elements in the streamwise, pitchwise and spanwise directions are 380, 88 and 56, respectively. The number of cell in the tip clearance region for TC is 5 in the spanwise direction.

To eliminate the effect of the mesh number on simulated results, the mesh independence is verified using the mixed-flow compressor with eleven groups of different mesh numbers, the tip clearance is 0.9-mm. Fig. 3 indicates that when the mesh number exceeds 90 thousand, the simulation results for both the total pressure ratio and the isentropic efficiency are almost maintain unchangeable. Thus, the mesh number of 90 thousand meets the requirements of computation accuracy and is selected for the mixed-flow compressor in the present investigation.

(a) The mesh of one-passage at 60% span



(b) Grids near leading edge and trailing edge of main blade

Figure 2 The mesh of the mixed-flow compressor

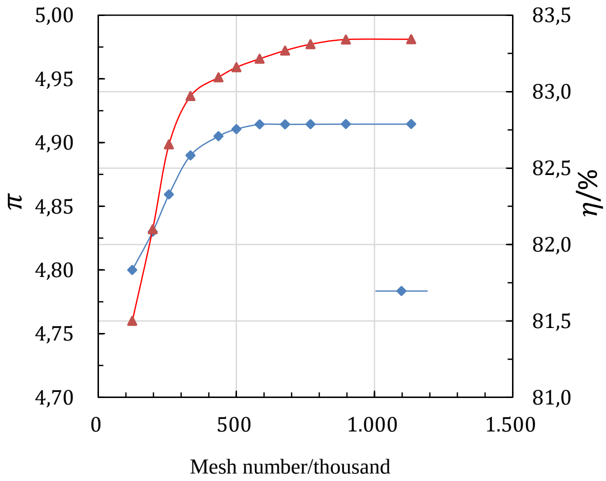


Figure 3 Independence verification of mesh number

2.4. Comparison of the mixed-flow compressor performance

Before change the tip clearance, the simulated and experimental performance curves of the mixed-flow compressor with 0.9mm tip clearance at 65% of design speed are compared for Mass flow parameter of 0.9~1.3, as shown in Fig. 4. Where

$$\text{Mass flow parameter}=m\sqrt{\frac{T_{01}}{T_{01\,ref}}}\Big/\frac{P_{01}}{P_{01\,ref}}. \qquad (2)$$

The experimental performance curves were reported in the reference [5]. The calculated total pressure ratio and isentropic efficiency curves are reasonably close to the experimental data, showing a slight shift toward low mass flow parameter. The difference between numerical values and experiments is within an arguably acceptable range. In particular, under the design mass flow rate $q_m = 2.72\ kg/s$ and the relative errors are less than 2% and 4%. Therefore, the present numerical results are considered to be reliable.

CFX is used to simulate the internal flow characteristics of the mixed-flow compressor with different blade tip clearances, 0.5mm, 0.75mm 0.9mm and 1.15mm, respectively.

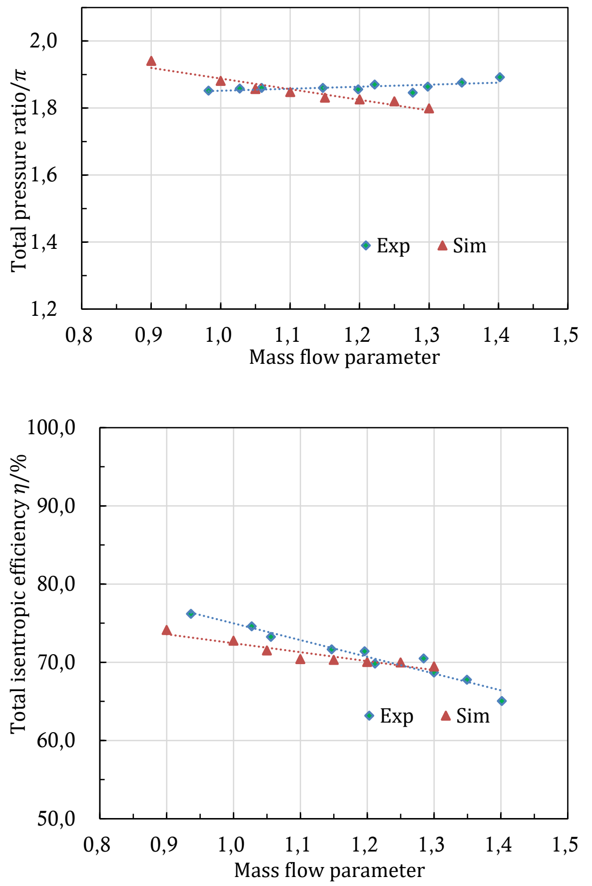


Figure 4 Comparison of simulated and experimental performance curves

3 RESULTS AND DISCUSSIONS

3.1 Performance characteristics of different tip clearances

The performance of the compressor estimated from the CFX analysis is shown in Fig.5. The total to total pressure ratio is plotted against mass flow parameter for four tip clearances at design speed.

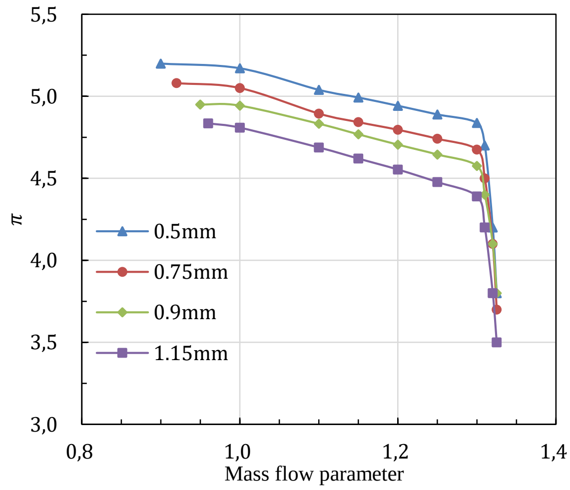


Figure 5 The total pressure ratio of the compressor

It is observed from Fig.5 that at design speed and at various clearances compressor choke mass flow parameter varies from 1.28 to 1.33 whereas surging mass flow parameter is varies from 0.9 to 0.95. Hence compressor at design speed has 10% to 15% surge margin for four tip clearances. The increase in choke mass flow parameter for higher tip clearance is due to increase in inlet area with same impeller tip diameter. Surge occurs earlier in the impeller with higher clearance as the flow has large tendency to separate with cross flow in the clearance spaces interacting with the main flow.

The total isentropic efficiency of the impeller was calculated. The variation of impeller total isentropic efficiency for different tip clearances at design speed is shown in Fig.6.

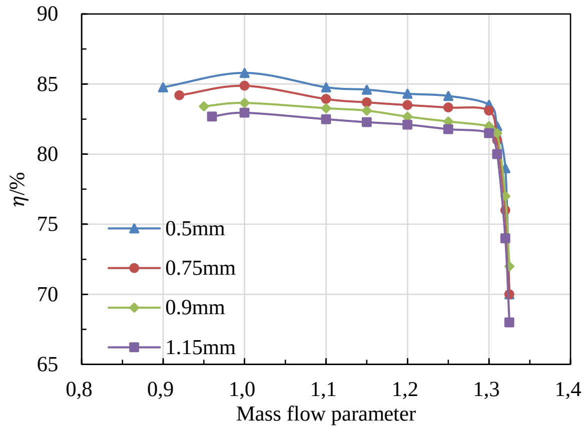


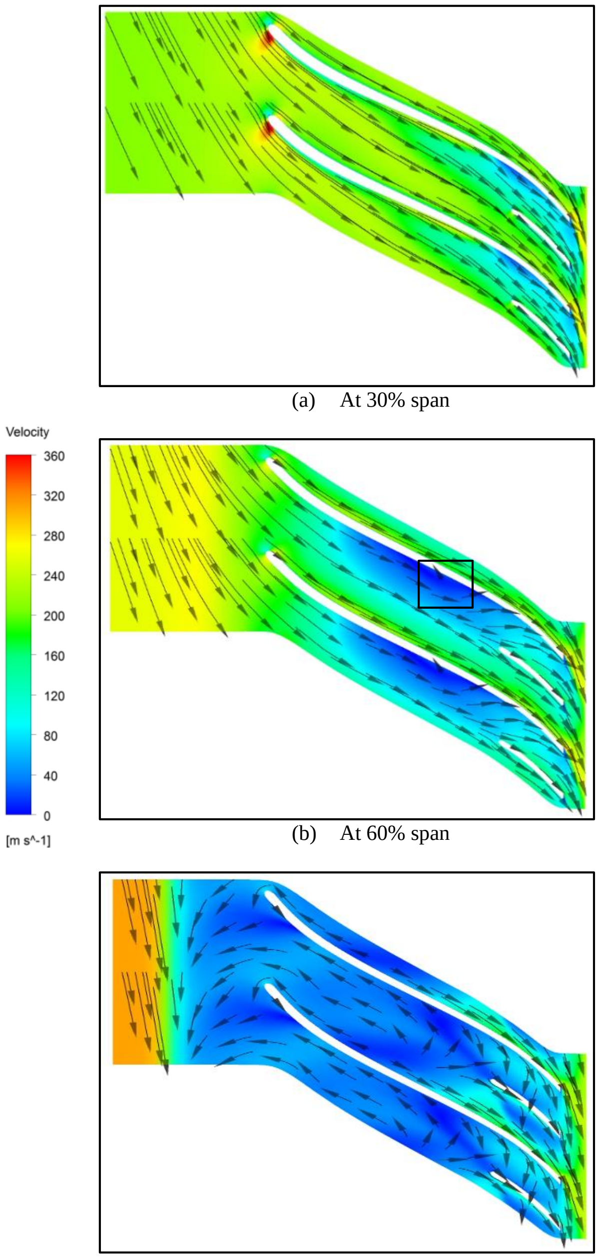
Figure 6 The total isentropic efficiency of the compressor

The peak efficiency of the impeller for 0.5mm tip clearances is around 85%. However, the peak efficiency drops marginally with the tip clearance increases. For the maximum tip clearance the impeller peak efficiency is 82.95%. The impeller efficiency at the larger mass flow parameter is smaller than the value at a lower mass flow parameter. This is because the incidence angle for the main blade is largely negative with a possibility of flow separation on the blades. With the mass flow parameter increased, the impeller efficiency decreased sharply due to choking. Tip leakage flow strongly interacts with mainstream flow and contributes to pressure loss and efficiency reduction. It is observed that the pressure ratio and efficiency drop are nearly linear proportion to the tip clearance.

3.2 Distribution of flow field at different spans

The relative velocity vectors and streamlines can show vortex structures of the flow field perfectly. And in principle, the velocity vectors should tangent to blade surface in ideal circumstances. Fig. 7 illustrates the distribution of the relative velocity vectors on the blade-to-blade surface at the different spans with 0.9mm tip clearance.

As shown in Fig. 7a, the airflow which the direction of velocity from the leading edge to trailing edge is relatively uniform, and this is what we want to see. While a small amount of backflow flow is grew along the suction surface of the main blade at 60% span (Fig. 7b), and other most flow is acceptable. With the increase of span, a large amount of reflux is formed in flow channel and the flow at 90% span becomes complex and disorder (Fig. 7c), which will lead to the decrease of the efficiency of compressor and is unacceptable. This is due to the fact that the tip leakage flow induced by the high loading at the leading edge of the main blade is suddenly attenuated by the interaction with the main flow. Through above analysis, the intensity of the disturbance have rather a large influence over efficiency and should be focused closely.



(a) At 30% span

(b) At 60% span

(c) At 90% span

Figure 7 Velocity vectors distribution at different spans

3.3 The influence of different tip clearances on flow field

The relative velocity vectors at different tip clearances for design speed are plotted in Fig. 8. At 30% span, the velocity vectors with different tip clearances are similarly distributed and all comparatively uniform. However, with the increase of span, the direction of relative velocity vectors change in local region. For the different tip clearances, the span which appeared backflow for the first time is different, so is the region.

As for four tip clearances, the backflow primarily emerges is 0.75 mm tip clearance at 45% span, while 0.5mm tip clearance which the early backflow shows at the leading edge of 77% span exhibits acceptable distribution of flow field. For the significative phenomenon, the reason may be the leakage and intensity, when the tip clearance is small, the leakage is little but intensity is tremendously high, while the leakage increase and intensity decrease slowly with the enlargement of tip clearance. The span where the backflow begins to take shape increases gradually with the increment of tip clearance. At 90% span, a large amount of backflow is all appeared in flow channel for four tip clearances. In contrast, the quantity of backflow is minimum when the tip clearance is 0.5 mm.

With the support of the flow physics, the authors cautiously suggest that 0.5 mm tip clearance can be accept completely, 0.75 mm should be avoided.

4 CONCLUSION

Simulation was carried out for mass flow parameter ranging from 0.9 to 1.35 at 65% design speed. The total pressure ratio and isentropic efficiency are achieved. The calculated curves are reasonably close to the experimental data, the present numerical results are considered to be reliable.

The study reveals that the total-to-total pressure ratio and isentropic efficiency decrease with the increment of tip clearance. The drop in impeller efficiency is more outstanding with the change of tip clearance. 0.5mm tip clearance provides better performance in terms of pressure ratio and efficiency than those of other three tip clearances.

The loss generation in the mixed-flow centrifugal compressor with the tip clearance is mainly caused by the backflow on the passage of the impeller due to the interaction of the tip leakage flow from the tip clearance of the main blade with the main flow, and the interaction enhances with the tip clearance increases to 0.75mm, then weakens with the tip clearance increases to 1.15mm.

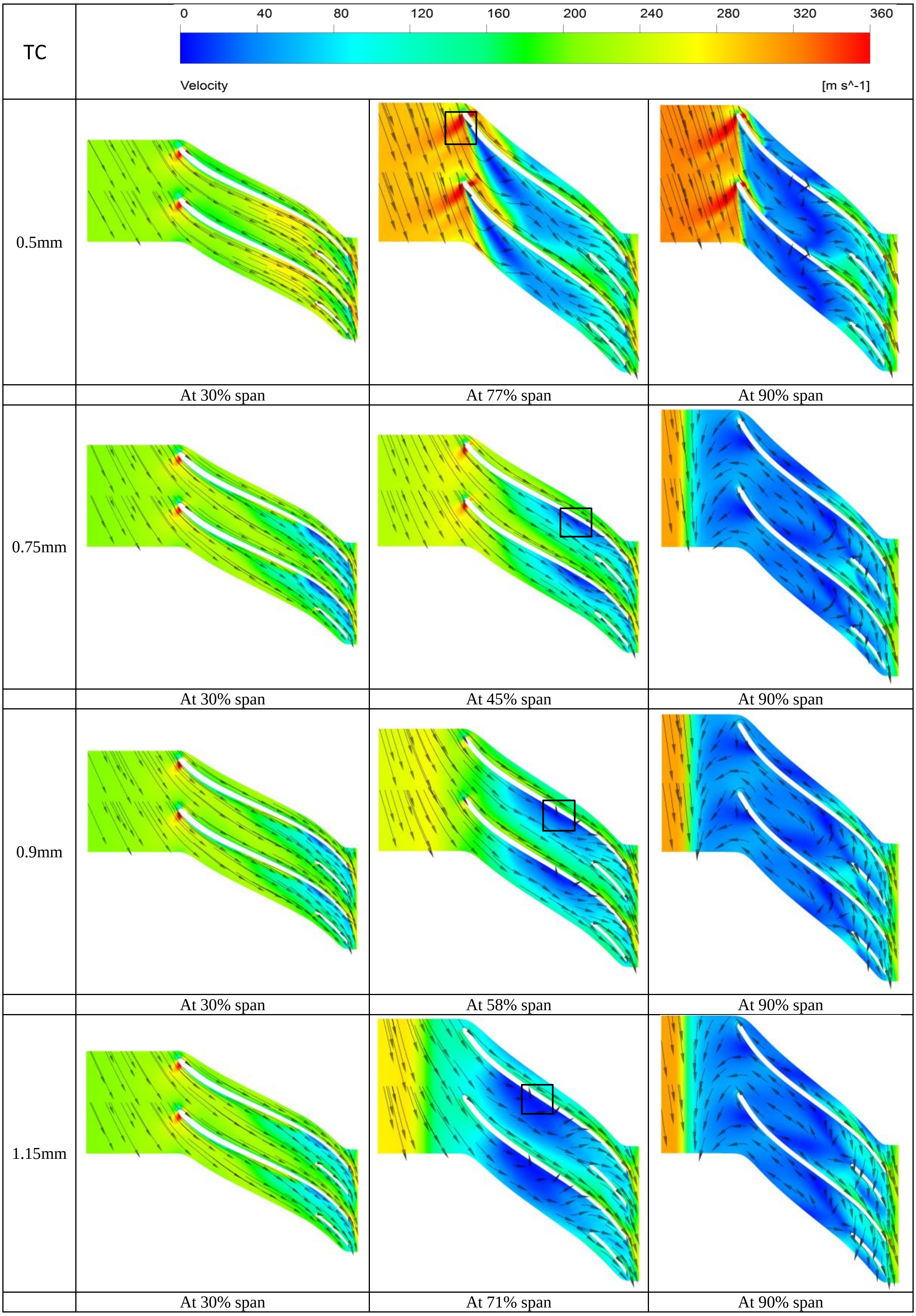


Figure 8 Relative velocity vector plot at different tip clearances for design speed

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