Chapter 24
Effect of Rotor Tip and Stator Hub Clearance on Aero Performance in Axial Flow Compressors for Aero Gas Turbine Engines

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Abstract Gas turbine engines are widely used as power plants. Absence of reciprocating and rubbing parts in gas turbine engine helps in lower consumption of lubrication oil and provides higher power to weight ratio with high reliability, apart from reduction in balancing and vibration problems. Turbine inlet temperature and efficiency of components are two main factors which are responsible for gas turbine engine performance. Compressor is one of the main component which directly influences the efficiency of the gas turbine engine. The tip clearance causes leakage of flow which interacts with the main passage flow and the wall boundary layer region causing performance deterioration in compressors. The efficiency and compressor operating envelop gets impacted with the tip leakage flow. In this study, a steady state analysis has been carried out to understand the combined effect of radial clearance on rotor and the stator hub on the aerodynamic performance of intermediate stage axial flow compressor through three dimensional viscous analysis using ANSYS CFX-19.2. The analysis was carried out for 30 different tip and hub clearance configurations varying from 0 to 6% of rotor tip axial chord. Out of 30 configurations only 11 combination of hub and tip clearance were selected meeting the design requirement of surge margin. The study concluded that out of 11 configurations, the combination having 1.5% rotor tip clearance and 3.0% stator hub clearance resulted into higher efficiency of 3%, higher pressure ratio of 4% and surge margin improvement from 15 to 18.3% a gain of about 22%. It is also observed that the stator performance is better with hub clearance for the configuration under investigation.

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24.1   Introduction

The axial flow compressors are preferred choice mainly for gas turbine power plants and specifically in aircraft applications. The performance of the compressor dictates the operating envelope of the gas turbine engines. Achieving a wide operating range and maximize efficiency of compressor poses a challenging task to the designer. The operating range in axial flow compressor is defined by choke point and stall point within which the compressor can perform efficiently. The tip leakage flow has certain effect on the operating range which will significantly effects the performance of the compressor. Various methodology were studied to understand the tip leakage flow physics due to radial clearance at the tip and hub of rotor and stator blades respectively by Donghyun [1]. Mitesh et al. [2] presented the effect of casing treatment for stability enhancement of a single stage transonic axial flow compressor. For higher tip gap the trailing edge separation was dominated and was responsible for compressor stalling. Baojie et al. [3] showed that the formation of tip leakage vortex makes the flow unstable and initiate the stalling process with loss in performance. Subbaramu et al. [4] investigated the effect of radial clearance and rotor solidity on the secondary flow interaction in a compressor stage. The study showed that increase in radial clearance will reduce the stall margin and efficiency. Gong et al. [5] showed that due to presence of tip gap, the leakage vortex was observed at the downstream of the blade row and flow disturbance near the tip. Xicheng et al. [6] studied various linearly variable radial gaps and uniform tip gaps by using numerical methods. The results showed there is considerable effect on the performance of the compressor due to various tip gap configuration. Songtao [7] observed at the design condition, the leaked flow travels through blade passage and interrupts the flow on the adjacent blade. At choke and stall condition tip circulation moves downstream along streamwise direction causing flow turbulence and large wakes. John et al. [8] showed the loss in the flow rate measurement in a multi stage axial flow compressor due to tip leakage flow. The study showed that the loss in the flow measurement is to the tune of 7% due to the reverse flow caused by tip leakage on the suction surface of blade. Most of the studies are limited to radial clearance for the rotor alone the combined effect of radial clearance on rotor as well as stator is not studied in detail. The emphasis of this study is to evaluate the combined effect of radial tip clearance on rotor and hub clearance on stator on aerodynamic performance of an intermediate stage axial flow compressor.

24.2   Methodology

The steady state 3D CFD analysis is carried out to study the combined effect of tip clearance at the rotor and hub clearance at the stator on the aerodynamic performance parameters using Ansys CFX 19.2. The computational methodology followed for the analysis process is explained below.

24.2.1 CFD Code Validation Study

Computational fluid dynamics (CFD) has emerged as a most powerful tool for determining the flow physics inside the casing for turbo machinery applications. Prior to testing of the actual components complete evaluation of the design can be done using CFD. A careful validation should be done with the standard test case before adopting any CFD software. Hence an assessment of Ansys CFX tool was carried out to check the suitability of the tool for the turbo machinery application. For assessment of the CFD tool used in this study, a validation study was done by considering NASA stage 37 test case. It is a single stage transonic compressor and the details of geometry and boundary conditions are mentioned in the paper by Reid and Moore [9]. A detailed analysis is carried out using Ansys 19.2 CFD tool by Dalbanjan and Sarangi [10] and the outcome of analysis is compared with experimental results of NASA stage 37. The study showed pressure ratio and massflow are within 1% error band, but efficiency is predicted slightly higher compared to test data. The comparison of CFD analysis and experimental data is shown in the Figs. 24.1 and 24.2.
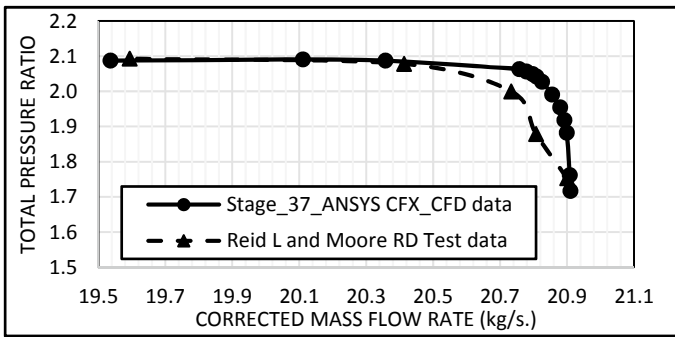

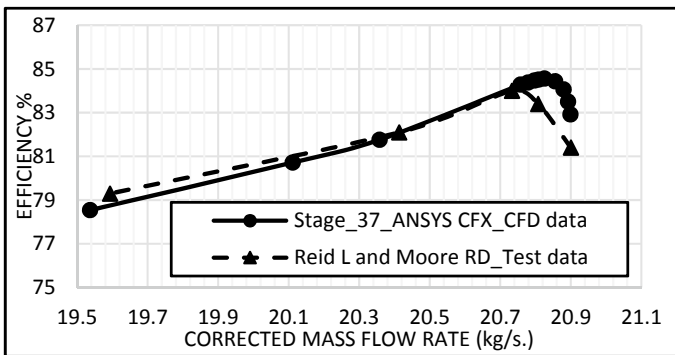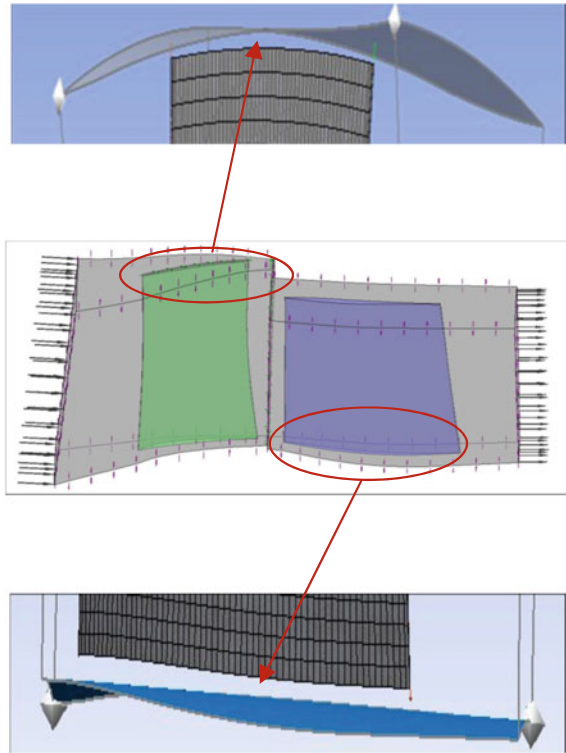**Fig. 24.1** Mass flow rate versus pressure ratio for NASA stage 37

Fig. 24.2 Mass flow rate versus efficiency for NASA stage 37

Fig. 24.3 Rotor tip
clearance. Geometric model
of rotor and stator domain.
Stator hub clearance

24.2.2 Solid Modeling of Blade Geometry in Flow Domain

A 3D geometric modeling using the 3D modeling software is carried out. The
geometric model consists of inlet and outlet domain, rotor blade and stator blade.
The domain at inlet and outlet is extended so that the flow at inlet of rotor and outlet
of stator is developed and settled. Single rotor and stator domain is considered for
analysis so that the computational time can be reduced. The next step is to discretize
the entire domain in to number of elements. Figure 24.3 shows the rotor and stator
geometric model. Figure 24.3 shows the rotor tip clearance and stator hub clearance
respectively.

24.2.3 Grid Generation and Grid Convergence Study

Grid independence or convergence study ensures the results are independent of mesh
size. Grid independence study is carried out by considering four different grid size
configurations. H- type topology is selected for flow domain and near blade surface
O- type topology is used. In the tip and hub clearance region around 10 elements with

Fig. 24.4 Mesh domain for rotor and stator

**Fig. 24.5** Grid size versus normalised massflow rate
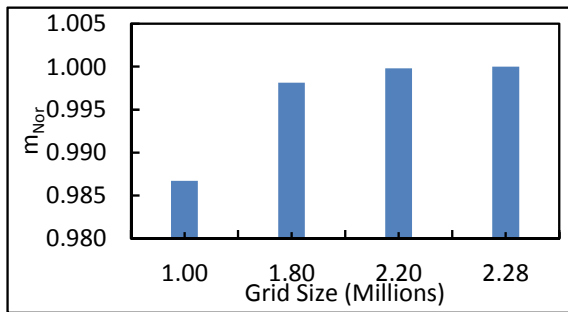
proper interface alignment and non matching grid topology is considered. The mesh distribution is done circumferential, stream and span wise direction on the rotor and stator blade surface, but near casing wall higher grid density is used to maintain y + < 2. Minimum angle is kept less than 165 degree whereas maximum angle and determinant value is greater than 20 degree and 0.6 respectively is maintained to achieve required grid quality. For the analysis k-ω with SST turbulence viscosity model along with high resolution turbulence numeric and advection scheme is selected. Mesh for both rotor and stator stage domain is generated as shown in Fig. 24.4. Grid independence study is performed to arrive at the final grid size. Four different cases is considered by varying number of grids in rotor and stator domain and their performance parameters are shown in Figs. 24.5 and 24.6. It can be seen that from the case-2 onwards the variation in performance parameters are negligible. Thus for further analysis case-3 is selected with grid size of 2.20 million elements.

24.2.4   Boundary Condition

Selection and application of appropriate boundary condition plays crucial role to achieve the reliable analysis results. At inlet total pressure and total temperature is

Fig. 24.6 Grid size versus normalised total pressure ratio
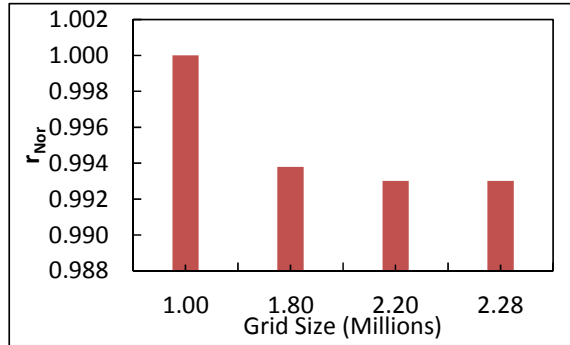
applied, at the interface between rotor and stator stage mixing model is selected. The static back pressure is applied at the stator exit. No slip wall boundary condition at blade hub and casing is specified.

24.2.5 Solution Methodology

Various combination of tip and hub clearance values ranging from 0 to 6.0% of rotor tip chord is analyzed. A total of 30 different combinations are considered for analysis. 11 Combinations out of 30 are selected which were meeting design surge margin requirement. The 11 combinations are shown in Table 24.1. The simulation is carried out using Ansys CFX-19.2 solver. Compressor characteristics are generated by keeping the speed constant and at the stator outlet increasing the static back pressure. Near stall point the static back pressure is increased in smaller steps of 1 kPa and the residual convergence level for all the parameters is specified as 1×10^{-6}.

24.3 Results and Discussion

The simulation is carried out at design speed and the constant speed characteristics are generated for various tip and hub clearance. Figure 24.7 shows the corrected mass flow rate versus total pressure ratio for various combination of tip and hub clearance configuration from 0 to 6.0% of rotor tip chord. It can be seen that the zero % clearance is giving highest mass flow due to absence of tip leakage flow. However the presence of circular vortex at the hub and tip of rotor in the flow passage causes the total pressure ratio to drop. Thus the zero clearance may not be the ideal or best performing condition. The results obtained shows that the combination having rotor tip clearance of 1.5% and stator hub clearance of 3.0% yields maximum performance parameters like pressure ratio and efficiency and wider operating envelope. It is also

Table 24.1 Showing tip and hub clearance for rotor and stator

Sl. No	Rotor tip clearance (%)	Stator hub clearance (%)	Case name
1	0.0	0.0	R0.0 + S0.0_%RCLR
2	1.5	0.0	R1.5 + S0.0_%RCLR
3	1.5	1.5	R1.5 + S1.5_%RCLR
4	1.5	3.0	R1.5 + S3.0_%RCLR
5	3.0	3.0	R3.0 + S3.0_%RCLR
6	3.0	4.5	R3.0 + S4.5_%RCLR
7	3.0	6.0	R3.0 + S6.0_%RCLR
8	4.5	4.5	R4.5 + S4.5_%RCLR
9	4.5	6.0	R4.5 + S6.0_%RCLR
10	6.0	3.0	R6.0 + S3.0_%RCLR
11	6.0	6.0	R6.0 + S6.0_%RCLR

seen that the massflow and pressure ratio deteriorates as the rotor and stator clearances are increased. The drop in massflow is around 1%, but the gain in peak pressure ratio from (0,0) clearance to the (1.5%, 3%) i.e. the combination of 1.5% tip clearance in rotor and 3% hub clearance in stator is 4.0%.

Figure 24.8 shows corrected mass flow rate versus efficiency. It is observed that for all the possible combination, the case of rotor tip clearance of 1.5% and stator hub clearance of 3.0% gave maximum efficiency compare to zero tip and hub clearance. The efficiency with the zero clearances is lower by 3 points due to formation of circular vortex at the hub and tip region creating higher losses. Further increase in the tip clearance, drop in efficiency by 6 points is observed.
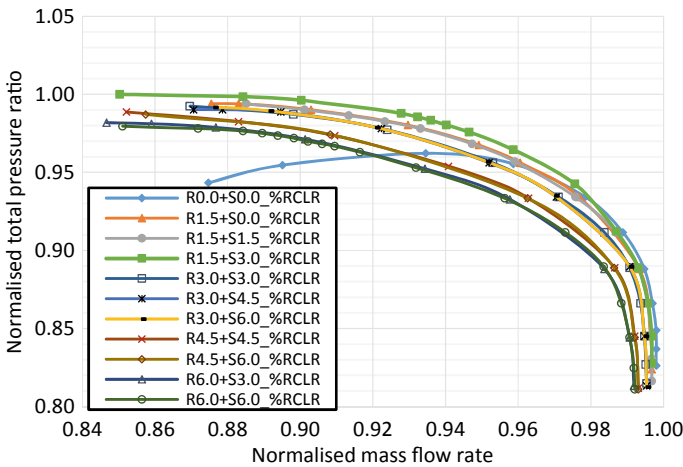

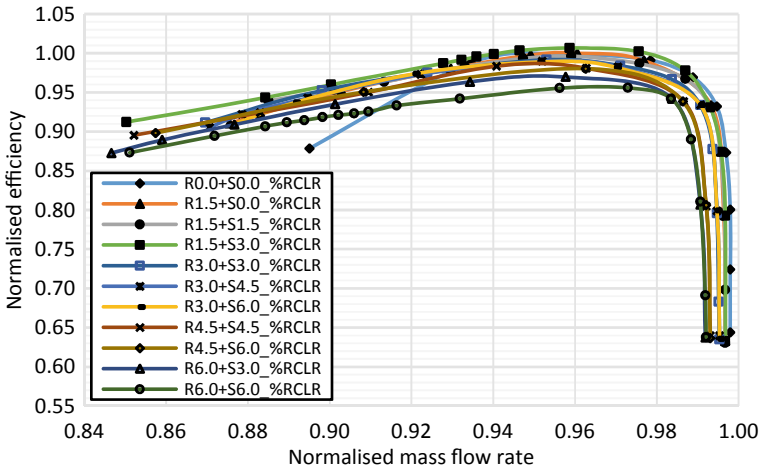**Fig. 24.7** Normalised corrected mass flow rate versus total pressure ratio

Fig. 24.8 Normalised corrected mass flow rate versus efficiency

Figure 24.9i shows variation of normalised total pressure versus normalised span for the rotor and stator blade for three different configuration i.e. R0 + S0%, R1.5 + S3.0% and R6.0 + S6.0% at design point. From the rotor plot it can be observed that up to 50% of blade span, there is no difference of total pressure. However after 50% span improvement in total pressure was observed for the R1.5 + S3.0% case. This higher total pressure helps to get better pressure ratio for the stage. Figure 24.9ii shows variation of normalised total pressure versus normalised span of the stator. The plot shows the stator tip is affected by the rotor tip clearance swirl flow and the hub clearance effect is not significant as compare to rotor tip clearance. The average total pressure for case R1.5 + S3.0% is higher compare to R0 + S0% and R6.0 + S6.0% configuration. The tip leakage vortex is up to 10% of span for the R6.0 + S6.0% configuration. This is the reason for lower efficiency at the higher tip clearance.

Figure 24.10i plot shows spanwise variation of relative Mach number at rotor exit at design point for the tip clearance of various configuration. It is observed from the plot that at the tip Mach number shows higher value for R1.5 + S3.0% compare
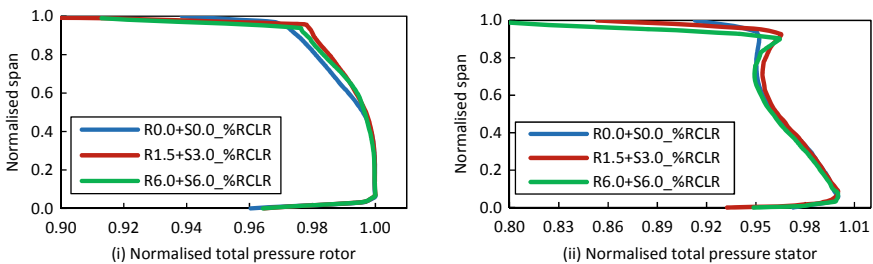

Fig. 24.9 Variation of total pressure across span of the **i** rotor and **ii** stator
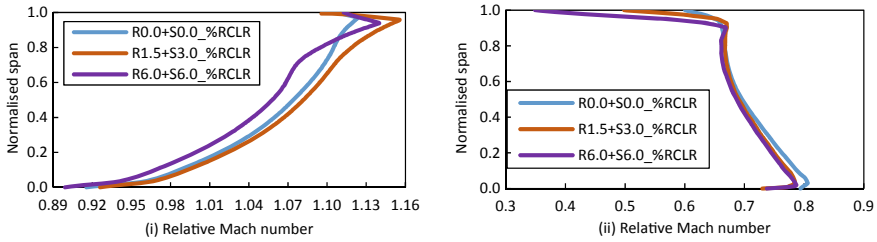
Fig. 24.10 Spanwise variation of relative Mach No. **i** rotor exit and **ii** stator exit

to R6.0 + S6.0% combination. The rise in the relative tip Mach no is 3% compare
to R0.0 + S0.0% condition. Figure 24.10ii shows spanwise variation of relative
Mach number at stator exit at design point for the tip and hub clearance of various
configuration. It is observed from the plot that at the hub Mach number is reduced for
R1.5 + S3.0% compare to R0.0 + S0.0% combination. The reduction in the relative
Mach number is 2.5% compare to R0.0 + S0.0% condition.

Figure 24.11i shows the blade loading plot for rotor at mid span for the tip and hub
clearance of various configuration for design point. Rotor is performing better up to
60% of span but due to higher tip clearance the drop in static pressure is observed.
Figure 24.11ii shows the blade loading plot for stator at mid span for the tip and hub
clearance of various configuration at design point. The effect of tip and hub clearance
is not very dominant on the blade loading parameters of stator. The blade loading is
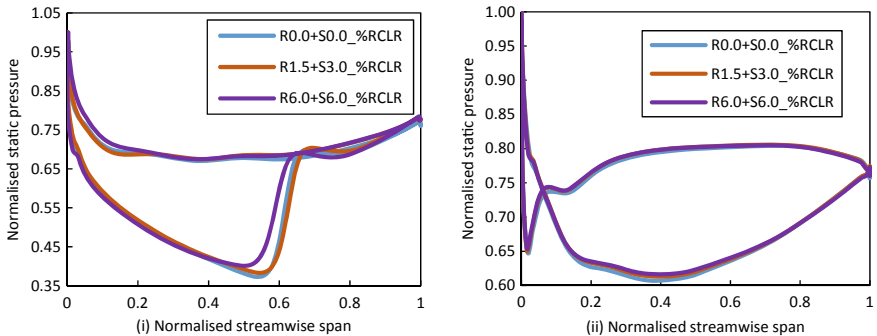more or less uniform throughout the stator blade and is performing as designed.

Fig. 24.11 Blade loading plot for i rotor and ii stator at mid span

24.4 Conclusions

A 3D steady state viscous CFD analysis is carried out to investigate the sensitivity of tip and hub clearance on the aerodynamic performance of a intermediate single stage axial flow compressor using ANSYS CFX 19.2. The analysis is carried out for 30 different tip and hub clearance configuration varying from 0 to 6% of rotor tip axial chord. Out of 30 configurations only 11 combinations of hub and tip clearance are selected which were meeting the design requirement of surge margin. Out of 11 configurations the combination having rotor tip clearance of 1.5% and stator hub clearance of 3.0% yields 3% higher efficiency, 4% higher pressure ratio and surge margin improvement from 15 to 18.3% offering gain of 22%. The study also showed that the stator performs better if hub clearance is provided and helps to improve efficiency without disturbing the blade loading. The characteristics plots like corrected mass flow rate versus pressure ratio and corrected mass flow rate versus efficiency are generated for various tip and hub clearances at design speed. The study concluded that the zero tip clearance is not yielding higher pressure ratio and highest efficiency. It is also observed that the rotor tip clearance plays a dominant role compared to stator hub clearance in improving or deteriorating the engine performance.

References

1. Donghyun, Y.: Progress in large-eddy simulation of a rotor tip clearance flow. In: 12th Annual DOD HPCMP User Group, Conference, Austin, TX (2002)
2. Goswami, M., Alone, D., Inoue, M., Kuroumaru, M., Yoshida, S., Minami, T., Yamada, K. Furukawa, M.: Flow behavior in a single stage transonic axial flow compressor with diminishing to large rotor tip clearances in the presence of axisymmetric circumferential grooved casing treatment. In: Proceedings of the Asian Congress on Gas Turbines, ACGT2016-03, IITB, Mumbai, India (2016)
3. Liu, B., Yu, X., Wang, H., Liu, H., Jiang, H., Chen, M.: Evolution of the tip leakage vortex in an axial compressor rotor. In: Proceedings of ASME turbo Expo GT2004-53703, Power for Land, Sea, and Air, June 14–17. Vienna, Austria (2004)
4. Subbaramu, S., Nagpurwala, Q.H., Varpe, M., Narahari, H.K.: Effect of tip leakage and secondary flow interaction on the performance and stability of a transonic axial compressor stage. In: Proceedings of the Asian Congress on Gas Turbines, ACGT2016–133, IITB, Mumbai, India (2016)
5. Gong, H.L., Je, H.B.: A numerical study on the structure of tip clearance flow in a highly forward swept axial–flow fan. In: Proceedings of ASME FEDSM (2002)
6. Jia, X., Wang, Z., CAI, R.: Numerical investigation of different tip gap shape effects on aerodynamic performance of an axial–flow compressor stator. In: Proceedings of ASME TURBO EXPO 2001, New Orleans, Louisiana, USA, 2001-GT-0337 (2001)
7. Songtao, W., Zhongqi, W.: The tip and hub leakage flow of a repeated two stage compressor. In: Proceedings of ASME TURBO EXPO 2002, Amsterdam. The Netherlands, GT-2002–30437 (2002)
8. Trevor John, W., Susan Prahst, P.: 3-D digital piv measurements of the tip clearance flow in an axial compressor. In: Proceedings of ASME TURBO EXPO 2002. Amsterdam, The Netherlands, GT-2002–30643 (2002)

9. Reid, L., Moore, R.D.: Design and overall performance of four highly loaded high speed inlet stages for an advanced high pressure ratio core compressor. In: NASA TP 1337 (1978)
10. Dalbanjan, M.S., Sarangi, N.: An effect of tip clearance on aero performance in axial flow compressors for aero gas turbine engines. In: Int. J. Mech. Prod. Eng. Res. Dev. **9**(4), 769–776 (2019)