Flow Rate and Axial Gap Studies on a One-and-a-Half-Stage Axial Flow Turbine

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Abstract Flow in a turbine stage is complex, and improving performance is a major challenge. Hence, it is still the topic of concern in the gas turbine community. Flow rate and axial gaps are of the few important parameters that affect the performance of a turbine. Present work involves the computational study of a one-half stage axial flow turbine with axial gaps of 15 and 50% of the average of the rotor and stator axial chords. For each axial gap, analysis is done at three flow coefficients, namely 0.68, 0.78 and 0.96. The turbine components nozzle, rotor and stator are modeled for both the axial gaps. Each axial gap requires distinct modeling and grid generation of fluid domain consisting of all the components. Mid-span pressure distribution of the stator for the design configuration is compared with the experimental results and found to be in good agreement. Pressure, entropy, Mach number and TKE distributions along with torque and efficiency are analyzed for both the configurations. From inlet to outlet of the stage, variations of parameters are plotted in contours and *x*–*y* plots. Flow is visualized clearly in mid-chord contours, and mass average values are considered at each position. Flow impingement, presence of wakes, stagnation and saddle points are observed. Entropy drop across the stage is higher for 50% gap. Rotor torque and efficiency decreased with increased axial gap. Trends are changing with flow rate. Results thus specify that the turbine performance is reliant on flow rate and axial gap.

Keywords Axial flow turbine · Axial gap · Flow coefficient · Pressure distribution · One-and-a-half stage · Turbine performance

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1 Introduction

Axial turbine contains stators and rotors subsequently positioned in the stream pathway. Stationary guide vanes accelerate the flow in the required direction, and moving blades convert the pressure and kinetic energy of the fluid into mechanical work on the shaft. Improvement in the efficiency of the gas turbine engines would save much of the running cost. There are many parameters that can affect the performance of a turbine. Axial gap and flow rates are among those few important considerations. Normally, a single-stage turbine consists of nozzle and rotor, respectively. One-and-a-half stage includes downstream stator as well. Only, few works have been carried out on one-and-a-half-stage axial flow turbines. Earlier, Morphis [1] studied the performance of a one-and-a-half-stage axial turbine with various tip clearances. Radial variation of flow angles and other parameters for the square tip rotor and leakage flow losses were studied. The performance of a low-speed one-and-a-half-stage axial turbine with varying rotor tip clearance and tip gap geometry was investigated by Morphis and Bindon [2]. The second-stage nozzle efficiency was found to be significantly higher than for the first stage and even increased with tip clearance. Blade row interference and clocking effect in a one-and-a-half-stage turbine were studied by Billiard [3]. It consisted of investigating and analyzing the aerodynamic and heat transfer in turbine stage. Effect of axial gap on the aerodynamics of a single-stage turbine was studied by Subbarao and Govardhan [4]. Velocity and entropy distributions were used to describe the flow. Analysis found the varied performance of axial flow turbine with stator–rotor axial gap. Recently, Aziz et al. [5] conducted investigation in one-and-a-half-stage axial turbine aiming three-dimensional flow. Simulations were accomplished using steady state, profile scaling and time transformation approach in a single-passage arrangement with non-unity pitch ratios. Several constraints like pressure, flow angle and blade loading were deliberated. Němec et al. [6] investigated the flow field in one-and-half axial turbine stage. It was proved that the presence of the second stator and slightly changed rotor shroud outlet cavity configuration did not affect the stage characteristics. Since the performance situation with flow rate and axial gap has not been studied earlier, it is necessary to perform a study that would bring more facts about this limitation.

In the present work, the effect of varying axial gap and flow coefficient on the performance and flow field of an axial flow turbine for 1.5 stage is studied. Analysis is carried out for flow coefficients of 0.68, 0.78 and 0.96 with axial gaps of 15% and 50% of the average of the rotor and stator axial chords. Flow coefficient, ϕ is C_m/U, where C_m is meridional velocity (m/s), and U is mean blade speed (m/s).

2 Computational Methodology

Single-stage turbine consists of nozzle and rotor, respectively. One-and-a-half stage includes downstream stator as well. Particulars of the geometric configuration attained from Dring et al. [7] are presented in Table 1. Modeling and grid generation is done for all the axial gaps. Computational domain generated is as shown in Fig. 1 with all the three components sequentially placed. Periodic condition is applied. Hence, domain has three nozzles, four rotors and four stators. This keeps pitch ratio more or less equal to 1. For meshing, tetra method is used here. ANSYS® ICEM CFD 14.0 is used for modeling and meshing. Mesh distribution is shown in Fig. 2. Nozzle, rotor and stator contain about 1.8, 1.6 and 2.2 million elements correspondingly. CFX 14.0 is used for simulation. On the blade, hub and shroud surface, no-slip condition is assumed. Adiabatic condition is assumed for the walls. Standard k-ω based shear stress transport (SST) model is used as it gives highly accurate predictions both near and away from the walls of the turbine (Table 2).

Table 1 Blade configuration of CRT

Parameters	Nozzle	Rotor	Stator
Number of blades	22	28	28
Hub radius (mm)	610	610	610
Tip radius(mm)	762	762	762
Tip clearance (mm)	0	2.28	0

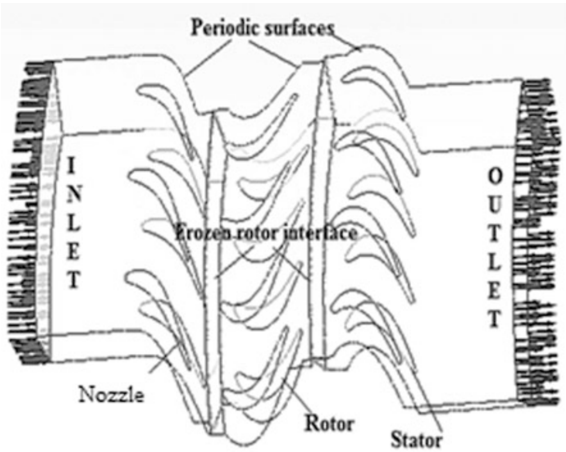


Fig. 1 Computational domain of CRT

Fig. 2 Mesh pattern for the blades

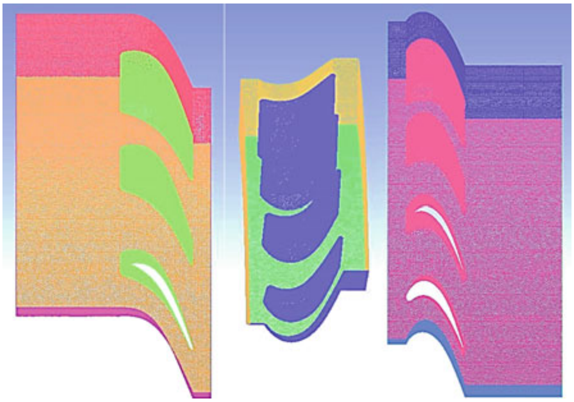


Table 2 Flow parameters used in simulation

Parameters	Values
Inlet temperature (K)	288.97
Meridional velocity, C_m (m/s) for ϕ = 0.78	22.86
Rotational speed of rotor (RPM)	410
Rotor–stator interface	Frozen rotor
Working fluid	Air ideal gas
Axial gaps (x/ch)	0.15, 0.5
Flow coefficients (ϕ)	0.68, 0.78, 0.96

3 Results and Discussion

3.1 Validation

Pressure coefficient on the blade is calculated by obtaining the pressure at all the locations of pressure and suction sides of the blade. Simulation results are validated with the experiments conducted in a large-scale rotating turbine rig at the United Technologies Research Center (UTRC), USA, by Dring et al. [7]. Mid-span C_p distribution for the stator is shown in Fig. 3 for ϕ = 0.78 and x/ch = 0.5 in case of one-and-one-half stage. The match between the simulation and measured experimental results is good, except in the region rear of the throat on the suction surface. The slight deviation near the leading edge is due to over-prediction of simulation results. Capturing of edges by the computational domain may be more fine than in case of experiments.

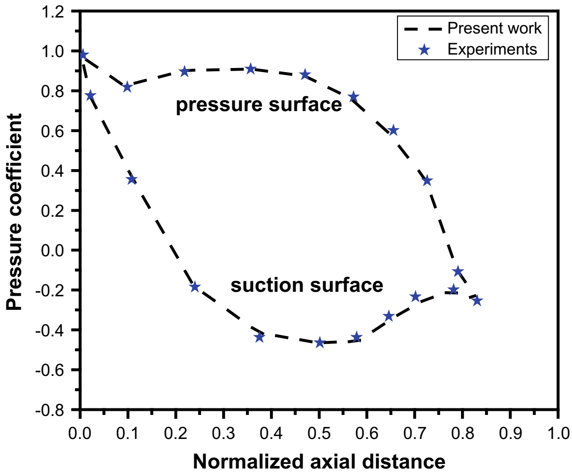


Fig. 3 Pressure coefficient of stator

3.2 *Velocity Distribution in One-and-a-Half Stage*

Figure 4a describes the velocity pattern throughout the one-and-one-half stage in case of x/ch = 0.5. This would help in analyzing the features of the flow field in the turbine passage near all the locations of the blade. Flow at the mid-span of the nozzle is well behaved. On the pressure side, velocity increases steadily from the leading edge to the trailing edge. In case of rotor, flow on the suction side undergoes speed increment before decelerating to the trailing edge. There is a slight speed increment near the leading edge. In the following rows of stator, velocity increases after the flow passes the leading edge. At the end of the stage, higher velocities are observed. Similar pattern is detected in all the flow coefficients. Closer velocity distribution between nozzle and rotor for x/ch = 0.15 and 0. 5 (ϕ = 0.78) is shown in Fig. 4b. In lesser gap case, wake region is impinging on rotor, effecting the flow pattern. In x/ch = 0.5, it is getting mixed with the flow and velocities are varying. Both Figs. 4a, b reveal that the effect of gap is distinct. The performance of rotor is clearly dependent on how flow impinges on blade rows.

3.3 *Pressure Distribution in the Stage*

Figure 5 shows the static pressure distributions in the blade passage with and without legend for clear visibility of flow. As shown in Fig. 5a, pressures are varying in all the blade rows. In the nozzle, values are changing more on suction side than pressure side. Same is the case with rotor and stator. Overall, pressures are

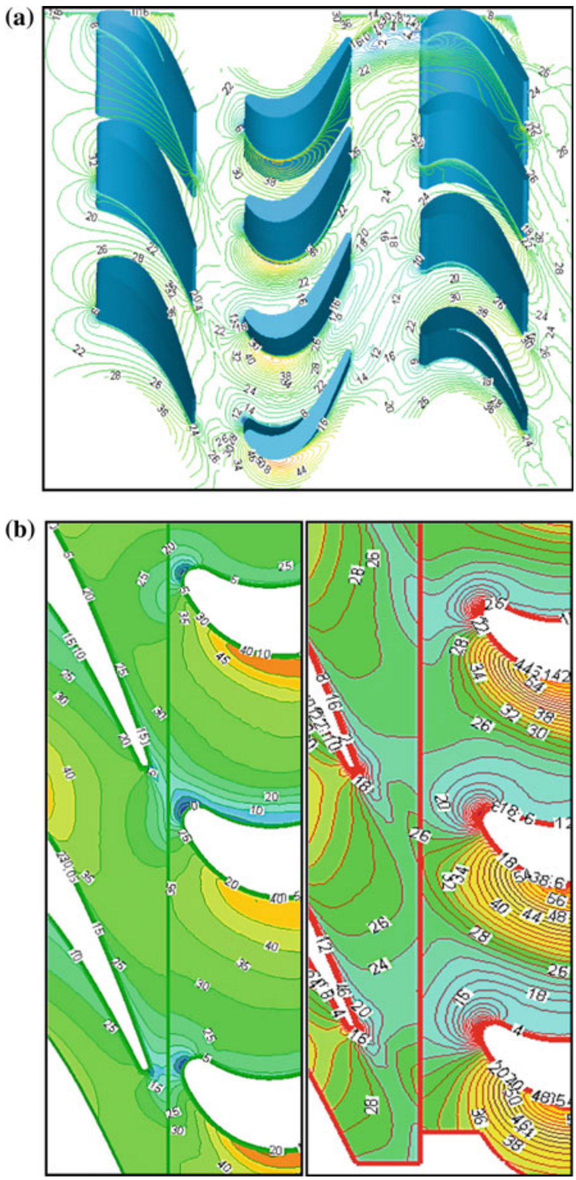


Fig. 4 **a** Velocity (m/s) distribution from inlet to outlet for x/ch = 0.5 (ϕ = 0.78), **b** Velocity (m/s) distribution from inlet to outlet for x/ch = 0.15 and 0. 5 (ϕ = 0.78)

reducing from inlet to outlet of the blade rows, however, with varied magnitude. As the incidence is zero, the streamline splits at the stagnation point (s) corresponding to the blade leading edge with one part moving along the pressure side and the other moving along the suction side of the blade as shown in Fig. 5b. However, the positions of stagnation point vary in rotor and the following stator. The pressure gradient from the pressure side to the suction side leads to the development of

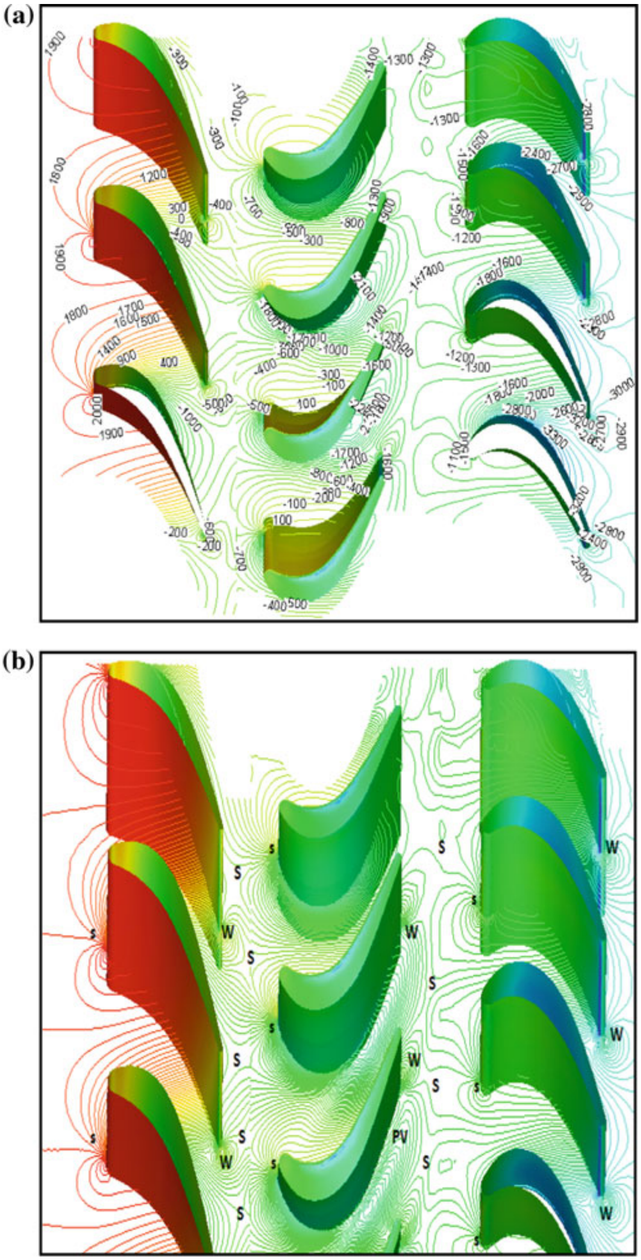


Fig. 5 **a** Pressure distribution for x/ch = 0.5 (ϕ = 0.78), **b** Pressure contours for x/ch = 0.5 (ϕ = 0.78)

losses. The area near the passage throat, where velocity is high, corresponds to the location, where *Cp* is low. It is also clear from the flow lines is that wake regions (*W*) after nozzle and stator blades are clear, with varied magnitude. Saddle points (*S*) are also observed in the interface region, where there is a chance of flow separation depending on flow rate. There is also the effect of the passage vortex (PV) on the suction side in the rotor flow. From the two figures of velocity and pressure, it is clear that flow on the blade suction surface becomes more three dimensional and distorted.

3.4 Mach Number and TKE Distributions

Figure 6a shows the local Mach number distribution for all the three components in the case of x/ch = 0.5. Flow over most of the pressure side rushes speedily. On the suction side, it accelerates up to the throat only. Near to the trailing edge, there is sudden flow deceleration after throat. Mach number contours are also drawn without values in order to display the exact pattern of flow, as shown in Fig. 5b. Nozzle wake is more stronger than the stator one. In rotor, it is dissimilar in magnitude and pattern.

Turbulent kinetic energy (TKE) values are negligible in stationary nozzle for all the flow rates as comprehended in Fig. 7. In case of rotor, high TKE values are observed on the suction side region than on the pressure side. From mid-axial chord section of the rotor, TKE values rise, which get transmitted along the turbine stage further. In stator, less TKE is witnessed near the leading edge, and further, it is more as the flow passes through the trailing edge region. High TKE variation is observed from leading edge to trailing edge on pressure and suction side regions. But, on suction side, TKE values are more for all the blade rows. These values may vary with change in flow rate. Seeing the trend, it can be concluded that if the flow rate is low, TKE values will be low. However, there may be an optimum flow rate, at which the turbulent flow is useful for better energy conversion in rotor. Both the Mach number and TKE distributions show the flow pattern of the turbine stage that may vary with flow rate.

3.5 Total Pressure Variation with Flow Coefficient and Axial Gap

Total pressure variation for the stage with flow coefficients and axial gaps is shown in Fig. 8. From nozzle (S_1) to rotor (R_1) and then to stator (S_2), total pressure values are found to be decreasing, while moving from inlet to outlet of the turbine stage. As the flow coefficient is increased, the total pressure difference is changing. This gives the measure of useful energy. Plot suggests that as the flow coefficient is

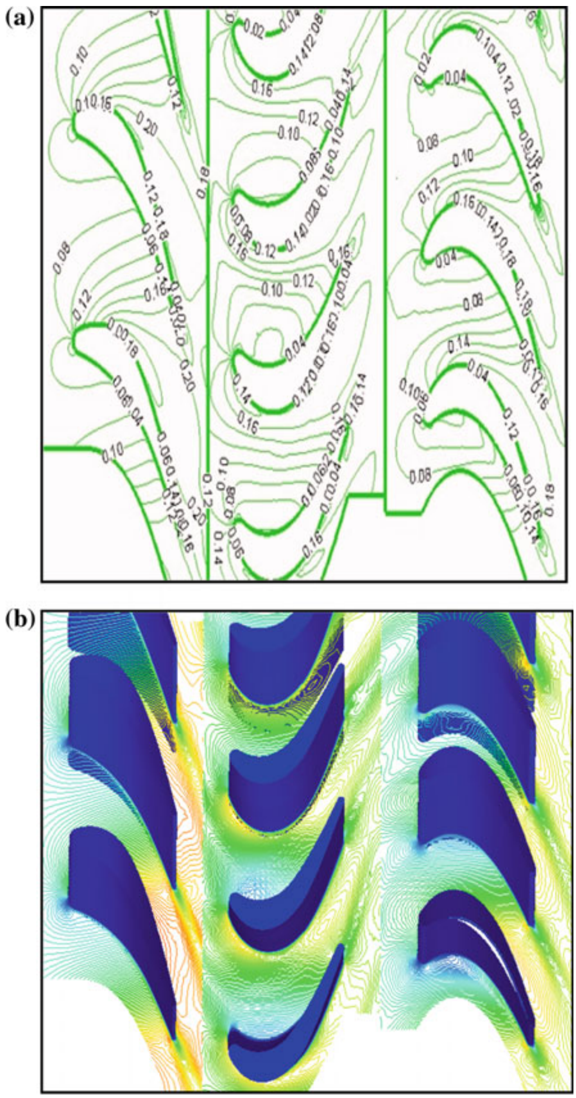


Fig. 6 **a** Mach number distribution for x/ch = 0.5 (ϕ = 0.78), **b** Mach number contours for x/ch = 0.5 (ϕ = 0.78)

increased, energy values increase. Magnitudes of total pressures reduced as the axial gap is increased, even though the pattern remained same from the turbine inlet to outlet. Thus, total pressure changes in rotor and the stage varied, associated with the performance of the turbine. It is clear that moderate loss is there in case of higher gaps. As the flow coefficient is increased, the losses increase almost linearly. Total pressure variation for the intermediate flow rate in case of both the axial gaps is shown in Fig. 9. For higher gap, total pressure change or energy converted in the rotor is only slightly less. The pattern of change is exactly similar for both the gaps,

Fig. 7 TKE distribution for x/ch = 0.5 (ϕ = 0.78)

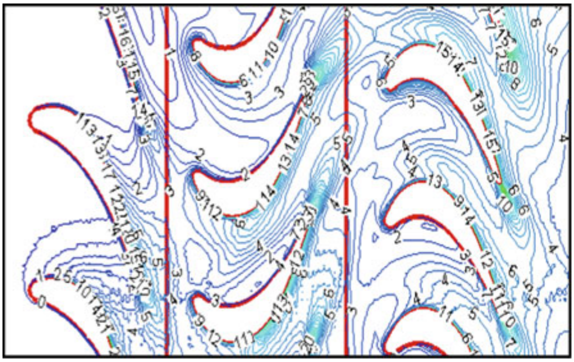


Fig. 8 Total pressure variation from inlet to outlet

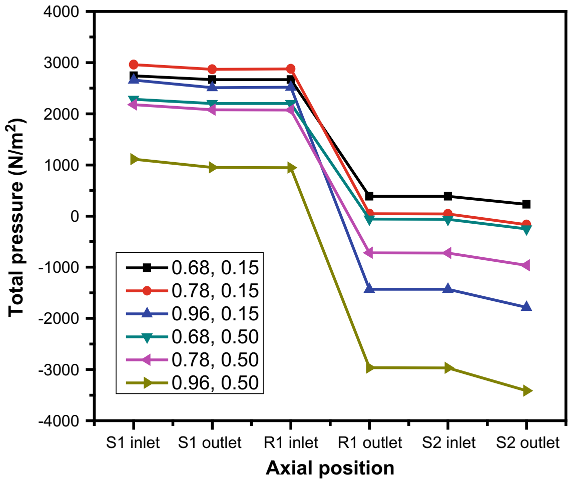
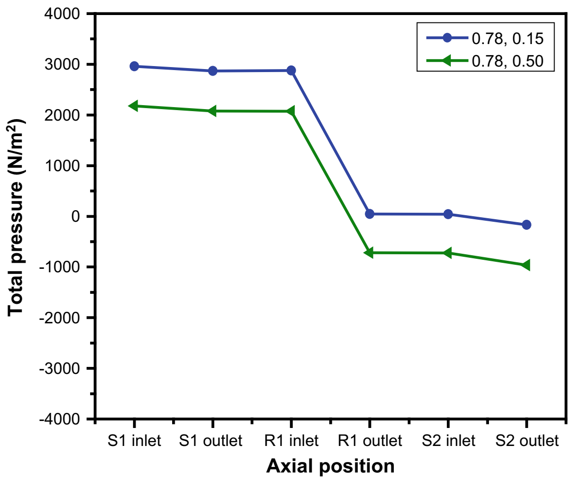


Fig. 9 Total pressure variation from inlet to outlet for ϕ = 0.78



even though the gap sizes are differed by about 35% of *x*/ch. In order to obtain better understanding, less variation in gap may be the right choice with more range in the permissible x/ch. This in turn may depend on profile of blades and the configuration of the turbine.

3.6 Entropy Variation with Flow Coefficient and Axial Gap

Figure 10 describes the variation of static entropy over the whole span for the entire stage with 15 and 50% axial gaps. From nozzle (S_1) to rotor (R_1) and then to stator (S_2), entropy is found to be increasing as the gap is increased. It is evident that entropy values are more for higher gap, when compared to the lesser axial gap. This confirms the decrement of total pressure part in the axial flow turbine as the gap is increased, which is a measure of useful energy.

Figure 11 labels the entropy drop across the stage. It is seen that this drop is less in case of the *x*/ch = 0.15. This may be due to the fact that there can be better energy conversion in rotor if the gap is less. Also, losses might have increased with increase in gap. As the flow coefficient is increased, the entropy drop is more and it will reflect in the performance of the turbine. It is interesting to note that the pattern of entropy drop variation with flow rate is slightly different for higher gap of *x*/ch = 0.5.

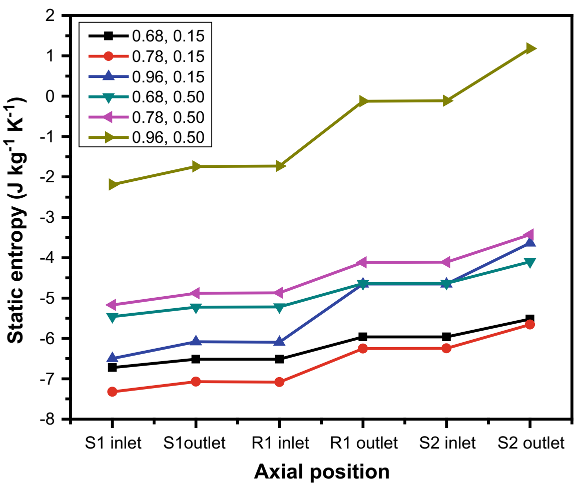


Fig. 10 Entropy variation from inlet to outlet

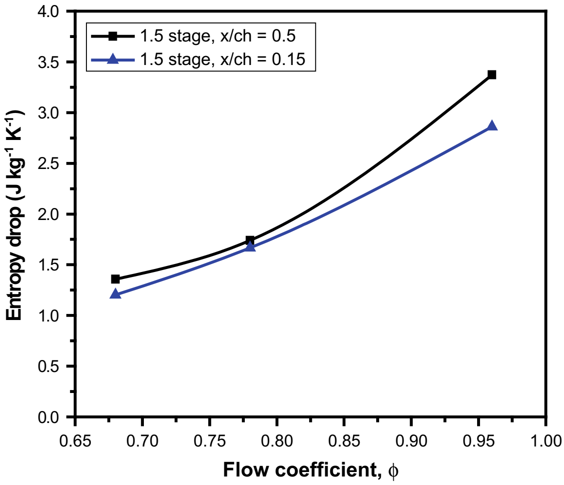


Fig. 11 Entropy drop across the turbine stage

3.7 *Torque and Efficiency with Flow Coefficient and Axial Gap*

Figure 12 describes the behavior of torque obtained from rotor for both the gaps, as the flow coefficient is changed. It is seen that the torque values are slightly changing with respect to the axial gap. Plot suggests that the decrement of axial gap is advantageous. Also, the effect of flow coefficient on the torque obtained from the rotor is shown. As the flow coefficient is increased torque obtained is increasing highly. The rate of increase is more beyond $\phi = 0.78$ in case of both the gaps. This confirms a relationship between the efficiency decrement and total pressure losses

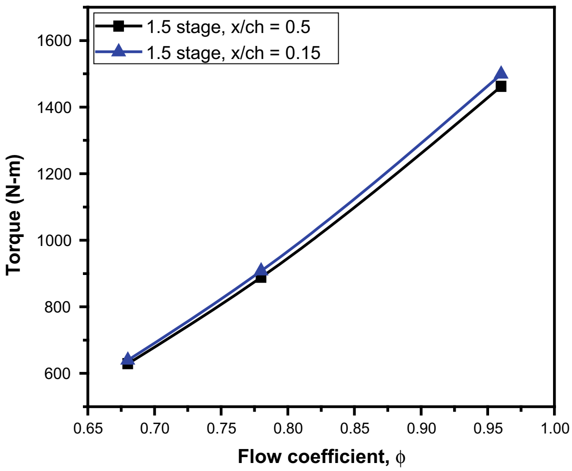
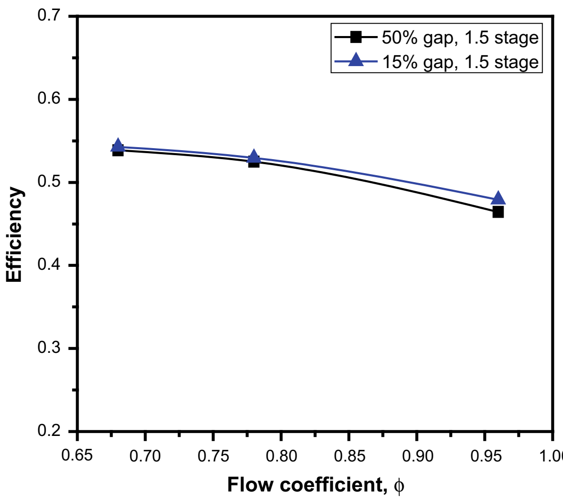


Fig. 12 Rotor torque with axial gap and flow coefficient

Fig. 13 Efficiency with axial gap and flow coefficient



with the increase in axial gap. Torque obtained from rotors is slightly decreasing with the gap. Evidently, the efficiency of the stage is slightly more for 15% gap as shown in Fig. 13. This may be due to the presence of downstream stator. Thus, it is recommended that the axial gap between the stator and the rotor should be as small as possible, especially if the turbine is operated at lower than designed mass flows and rotating speeds. This kind of performance was reported for a single-stage turbine earlier [4]. In addition, the loss of flow velocity at the rotor inlet when axial gap is increased encourages the use of smallest possible axial gap.

4 Conclusions

Computational study of a one-and-a-half-stage axial flow turbine with axial gaps of 15% and 50% of the average chords is conducted for three flow coefficients. The turbine components, nozzle, rotor and stator are modeled for both the axial gaps. Mid-span pressure distribution of the stator is compared with the experimental results and found to be in good agreement. From inlet to out of the stage, parametric variation is studied. Flow impingement, presence of wakes, stagnation and saddle points are observed for all configurations. Total pressure and entropy plots in the stage depict the loss pattern, which varies with flow coefficient and axial gap. Entropy drop across the stage is higher for 50% gap. Torque obtained from rotors is slightly increasing flow coefficient and decreasing with axial gap. Clearly, the efficiency of the stage is slightly more for 15% gap. This aspect can be clearly attributed to the presence of downstream stator. Thus, the study shows that the flow and performance aspects of a one-and-a-half-stage axial flow turbine are clearly dependent on flow coefficient and the stator–rotor axial gap.

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