ISSN:1687-1006

The Egyptian International Journal of Engineering Sciences and Technology

Vol. 20 (July 2016) 68-74

http://www.eijest.zu.edu.eg

Computed Effect of Varying Tip Clearance and Axial Gap on Gas Turbine Stage Performance Part (I): (Steady Flow)

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ARTICLE INFO

Article history:
Received: 23 March 2016
Accepted: 28 June 2016
Available online: 30 July 2016

Keywords:
Axial gap
Tip clearance
Efficiency
Axial-Turbine

ABSTRACT

It is desirable to further increase the efficiency of turbo-machines. In axial flow gas turbine, imprecise axial gap and tip clearance are two significant sources of inefficiency. This work investigates the effect of change in parameters such as axial gap and tip clearance on axial flow gas turbine used in power generation. Different combinations of axial gap and tip clearance have been tested to analyze the performance of the turbine. It is revealed that the axial gap of 18.5% of axial chord and 2.4% tip clearance of blade height is the optimum set value for the maximum performance

1 INTRODUCTION

Gas turbine engines are widely used in power plants, marine power and aircraft propulsion. Hence, attempts to improve the performance of gas turbines are encouraged through investigation of the effects of various parameters on their performance. Such parameter is the tip clearance percentage and axial gap. In axial flow gas turbine, imprecise axial gap (spacing between the stator and rotor blades) and tip clearance spacing between the (tips of blades and the stationary casing) are two significant sources of inefficiency. In order to increase the efficiency of axial flow gas turbine used in power generation, effort has been made in this work to investigate the effect of change in such parameters .In particular, the effect of different sets of axial gap and tip clearance have been studied to find out the optimum set value for maximum performance of the turbine. Five (A to E) sets are defined from different values of axial gap out of values 10.5%, 13.5%, 16%, 18.5%, and 21% of rotor blade chord and a tip clearance out of values 2.4%, 5% and 6.9% of rotor blade height as shown in (Table1).

Hass et al.[4] (1984) studied experimentally as well as numerically the effect of stator end wall contouring on turbine stage performance. In his investigation three stator end wall configurations were evaluated with the same rotor. One configuration was a cylindrical end wall and the other two were contoured end walls, one of S-shaped profile and the other of conical-shaped. The results showed total efficiencies of 0.845, 0.851, and 0.853, respectively.

Dey1 (2001) indicates that the leakage flow near the tip of an unshrouded rotor blade in an axial turbine imposes significant thermal load on such blade. This study used several techniques to reduce the severity of losses caused by the leakage vortex. Three desensitization techniques, both active and passive, were examined.. (N.B. : the fact that the leakage vortex is weakened by closing the tip gap has been well established in literature).

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Pfau7 (2003) studied the interaction flows associated with open cavities in shrouded high pressure turbines. The measurements focused on the rotor tip labyrinth seal, comprising two seal gaps, 0.3% and 0.8% of blade height.

QingJun et al.8 (2009) studied experimentally the unsteady pressure fluctuation of rotor tip region in high pressure stage turbine. The experiment was carried out on a blow-down short duration turbine facility.

Gao, et al.3 (2011) studied the effect of axially non-uniform tip clearance on the aerodynamic performance of an unshrouded axial turbine at design and off-design conditions. Different types of axially uniform and non-uniform rotor tip clearances were used in this investigation, which include uniform, expanding, shrinking, and back- and front-step tip clearances.

Lakshminarayana et al.6 (1998) Studied the experimental and computational effects of the nozzle wake-rotor interaction and effects of the unsteady flow in turbine rotors.

Li et al.5 (2007) studied the percentage of tip leakage flow losses of the shrouded rotor blade contribute significantly to overall losses of the turbine stage.

Yadav et al.9 (2008) studied the effect of change in parameters such as axial gap and tip clearance on axial -flow gas turbine used in power generation. Different combinations of axial gap and tip clearance have been tested to analyze the performance of the turbine. The results show that the axial gap of 3.5 mm and 5% tip clearance is the optimum set value for the maximum performance.

Da Silva et al.2 (2011) studied the influence of the rotor tip clearance on the performance of a multistage axial flow turbine, the performance parameters such as turbine efficiency, mass flow and pressure ratios, for several tip clearances. The influence of the rotor tip clearance on the performance of a multistage axial flow turbine is evaluated by means of turbulent, viscous, 3D flow calculations. The results from CFD calculations show clearly the influence of the tip clearance and its gap values on the efficiency and pressure ratio of a low pressure multi-stage axial flow turbine.

2 NUMERICAL ANALYSIS

The numerical study is conducted to simulate a single stage axial flow turbine considering rotor tip clearance effect. The steady, viscous, and 3D governing equations (continuity, momentum and energy) are solved together with standard $k-\varepsilon$ turbulence model. FLUENT 6.3.26 software provided by FLUENT Inc. is used to simulate the problem under consideration.

Table 1: Different combinations of axial gap and tip clearance

Setup	1	2	3
A	10.5% 2.4%	10.5% 5.0%	10.5% 6.9%
B	13.5% 2.4%	13.5% 5.0%	13.5% 6.9%
C	16% 2.4%	16% 5.0%	16% 6.9%
D	18.5% 2.4%	18.5% 5.0%	18.5% 6.9%
E	21% 2.4%	21% 5.0%	21% 6.9%

2.1 Governing equations

The mass conservation equation for unsteady flow is given by

$$\frac{\partial}{\partial x_i}(\rho V_i) = 0 \qquad (1)$$

where

V_i : the velocity in the i^{th} direction

x_i : the coordinate in the i^{th} direction

ρ : is the air density.

i : a tensor indicating 1, 2, 3.

The relative velocity $V_{r,i}$ in the rotating frame can be obtained by:

$$V_{r,i} = V_i - e_{jki}\,\omega_j\,x_k$$

where;

ω_j : the angular velocity for the rotating frame in the j direction

x_k : The coordinates in the rotating frame in the k direction.

j, k, i: Tensors indicating 1, 2, 3.

e_{jki} the permutation symbol given by:

$$e_{jki} = \begin{cases} 1 & \text{If } j,k,i \text{ are in a repeating order as 1, 2, 3.} \\ -1 & \text{If } j,k,i \text{ are in different repeating order.} \\ 0 & \text{If any two of } j,k,i \text{ are equal.} \end{cases}$$

The conservation of momentum equation in the i_{th} direction for unsteady flow can be written as follows:

$$\frac{\partial}{\partial x_j}(\rho V_i V_j) = \rho \bar{g}_j - \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + F_j \quad (2)$$

where p is the static pressure, and τ_{ij} is the viscous stress tensor given by

$$\tau_{ij} = \mu\left[\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial V_l}{\partial x_l}\right] \quad (3)$$

where

μ: is the absolute viscosity.

i, j, l: are tensor indices indicating 1, 2, 3.

$$\delta_{ij} = \begin{cases} 1 & if \quad i = j \\ 0 & if \quad i \neq j \end{cases}$$

$\rho\bar{g}$ And F are the gravitational body force and external body forces (that arise from interaction with the dispersed phase), respectively.

The unsteady equation of conservation of energy is given by

$$\frac{\partial}{\partial x_i}\left[V_i(\rho E + p)\right] = \frac{\partial}{\partial x_i}$$

$$\left(K\frac{\partial T}{\partial x_i} - \sum_j h_j J_i + (V_i \tau_{ij})\right) + S_h$$

where

E : is the total energy of the air.

K : is the air thermal conductivity.

S_h : is heat source term form the dispersed phase

J_i : is the diffusion flux of j^{th} species in the i^{th} direction

After importing geometry definitions from Gambit software followed by analysis on Fluent software. GAMBIT is the software, which combines the features of both modelling and meshing. GAMBIT consists of four stages: Modelling, Meshing, Boundary conditions, and Pre-processing. The axial flow gas turbine used in this case is 1-stage. The gas turbine space is divided into three regions, Inlet, Mixing plane, and Outlet. For a setup, total pressure, static pressure, total temperature, static temperature and Mach number are calculated on these three regions. Data analysis is given in Table 2.

Table 2: Data for design and analysis software

Data		Value
Number of Blades	Stator	28
	Rotor	30
Rotational Speed (N)		70000rpm
Boundary Conditions	Inlet Pressure	9.1 bar
	Inlet Temperature	1478 K
Mass Flow Rate		0.035 g/s

2.2 Computational Domain

The section of computational domain is shown in "Figure 1" for stator and rotor. The code supports different mesh types that include 2D triangular / Quadrilateral, 3D tetrahedral/ hexahedral / pyramid / wedge, and mixed (hybrid) meshes. The hexahedral mesh type was used in the present work. The mesh is very fine next to the solid boundary of the stator and rotor. The size of the element increases towards the far field away from the solid boundaries. Careful consideration was paid to minimize the dependence of solution on the mesh by improving the clustering of cells near solid walls until results are almost constant. The meshes for domain section and rotor are shown in "Figure 2", "Figure 3", respectively.

Mesh Sensitivity Analysis: The effect of the total number of cells on the stage characteristic is studied, to select the suitable one. Three mesh densities (1,106,267, 1,181,757, and 1,312,825) are examined and their results are compared with experimental results. The results of 1,181,757 and 1,312,825 cells are identical and close to the experimental results. So, the suitable number of cells is 1,181,757 nodes and beyond that the solution is independent on the number of cells.

3 RESULTS

3.1 Pressure Variation

The pressure variation for different axial gaps and tip clearances are plotted from inlet to outlet.

"Figure 4-a" shows the contour of total pressure at mean blade–to–blade section through the turbine stage. Through stator blade passage its clear that the total pressure decrease slowly due to friction till it reaches the stator exit. The total pressure decrease through the rotor till it reaches its outlet. "Figure4-b" and "Figure 4-c" shows the contour of total pressure on rotor pressure and suction sides, respectively. In suction side, the total pressure has a lower value than in rotor pressure side and at the corner between the rotor blade tip and the trailing edge has a maximum value. The distribution of total pressure at different tip clearance ratio is shown in "Figure 5"**.**

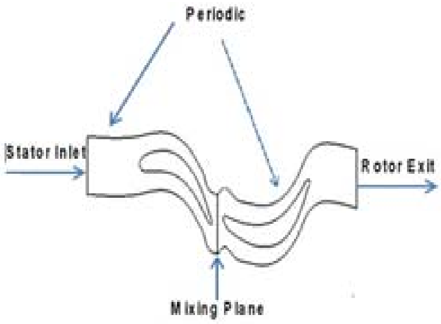


Figure 1: Section of Computational Domain

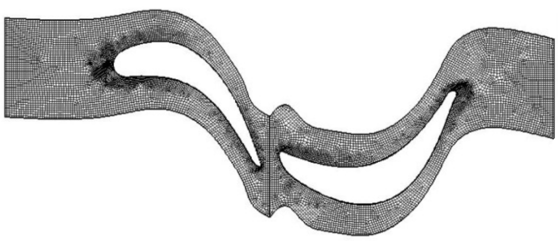


Figure 2: Section of stage mesh

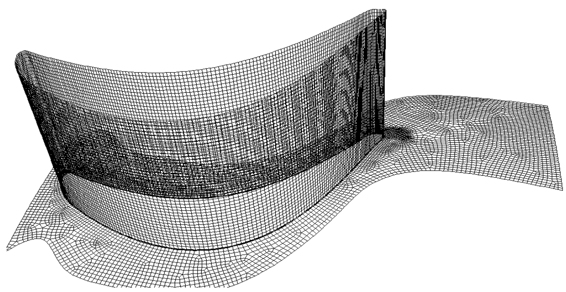


Figure 3: Rotor mesh

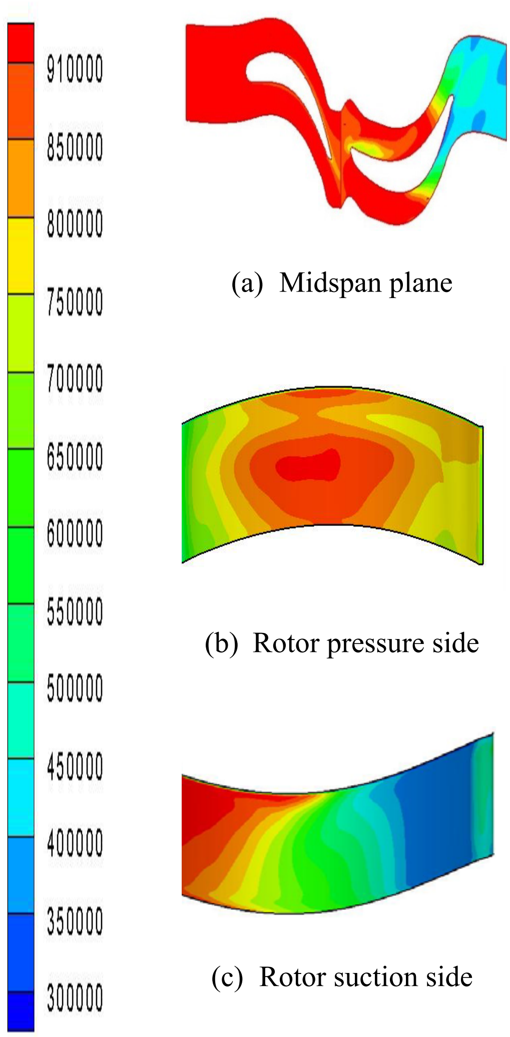
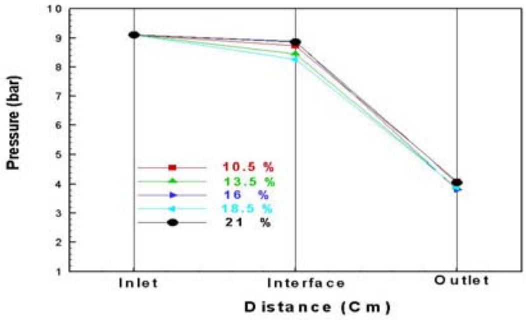
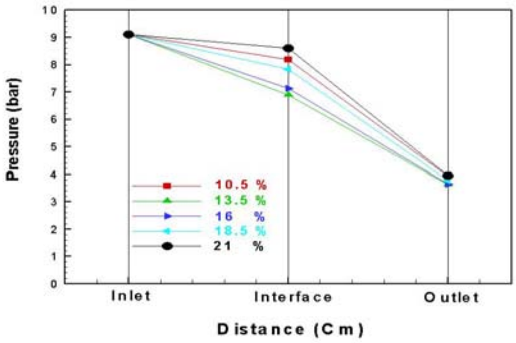


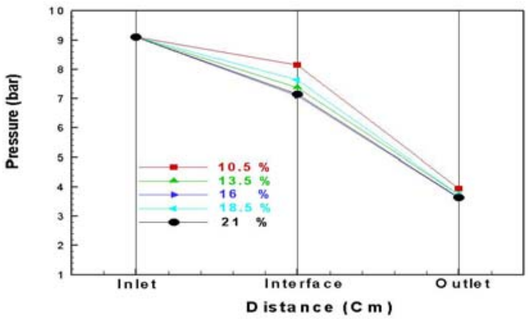
Figure 4: Contour of total pressure (Pa)



(a) 2.4 %



(b) 5.0 %

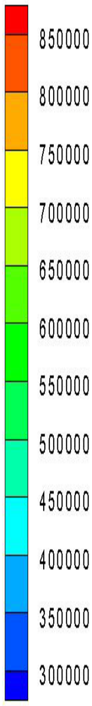


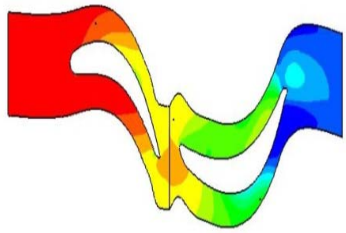
(c) 6.9 %

Figure 5: Total Pressure Distribustion at differents tip clearance Ratio

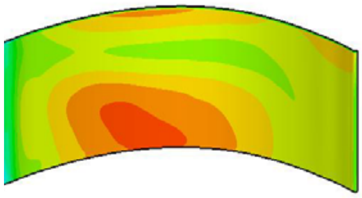
Figure 6-a" shows the contour of static pressure at mean blade–to–blade section through the turbine stage. The static pressure decreases through the turbine passage till it reaches turbine exit.

"Figures 6-b" and "Figure 6-c" shows the contour of static pressure on rotor suction and pressure side. In suction side, the static pressure has a value less than in rotor pressure side.

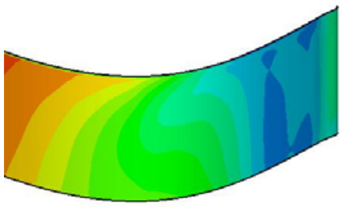




(a) Midspan plane



(b) Rotor suction side

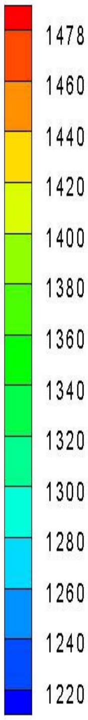


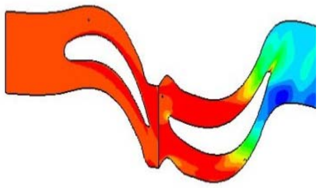
(c) Rotor pressure side

Figure 6: Contour of static pressure (Pa)

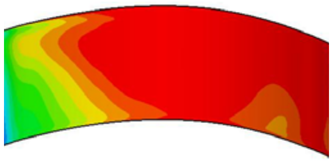
3.2 Temperature Variation

Figure 7-a shows the contour of total temperature at mean blade–to–blade section through the turbine stage. Through stator blade passage it is clear that the total temperature is almost constant till it reaches the stator exit. The total temperature decrease through the rotor passage till it reaches the rotor outlet. "Figure 7-b" and "Figure 7-c" shows the contour of total temperature on rotor pressure and suction side. In suction side, the total temperature is axially decreased and has a lower value than in rotor pressure side. In pressure side, the total temperature is increased axial and at the corner between the rotor blade tip and the trailing edge has a maximum value. The temperature variation for different axial gaps and tip clearance ratio are plotted from inlet to outlet as shown in "Figure 8".

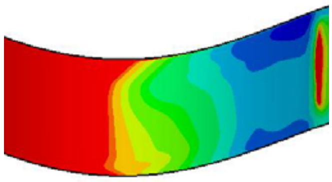




(a) Midspan plane



(b) Rotor pressure side



(c) Rotor suction side

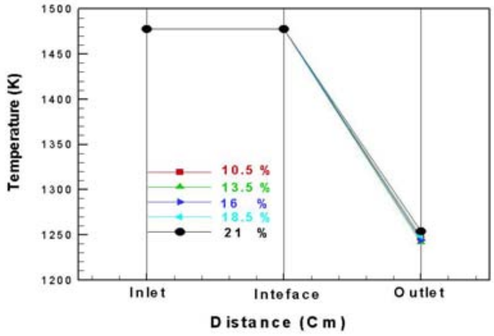
Figure 7: Contour of total temperature (K)

3.3 Mach number Variation

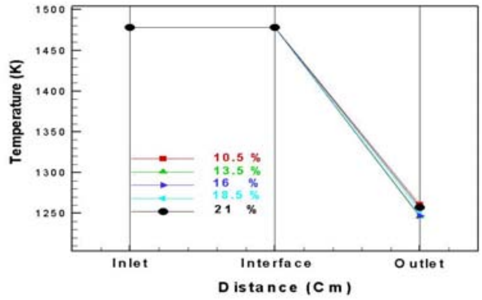
"Figure 9" shows the contour of absolute Mach number at mean blade–to–blade section through the turbine stage. In the stator Mach number is axially increasing due to change of area, but it decreases through the rotor.

4 EFFICIENCY CURVES

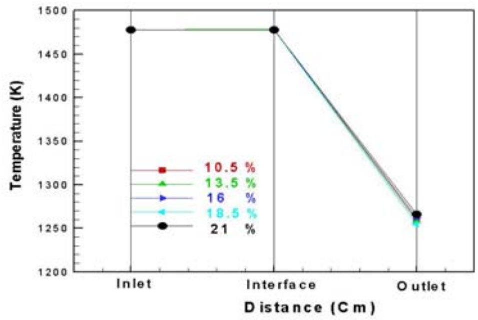
The efficiency for combinations of axial gap and tip clearance has been calculated (Table 3). "Figure 10-a" shows the turbine efficiency at different rotor tip clearance ratio with different axial gaps, which indicates that the turbine efficiency decreases by increasing rotor tip clearance ratio. "Figure 10-b" shows the turbine efficiency at different axial gaps which indicates that the turbine efficiency increase by increasing axial gaps to the value of 18.5% and then decrease with increasing axial gaps.



(a) 2.4 %



(b) 5.0 %



(c) 6.9 %

Figure 8: Total Temperature Distribustion at differents tip clearance Ratio

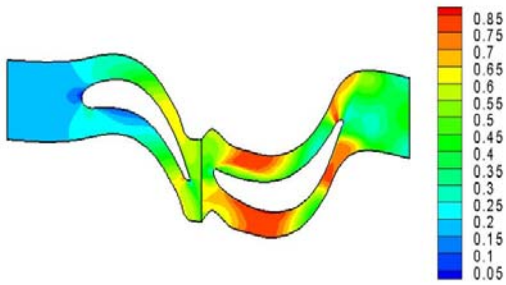
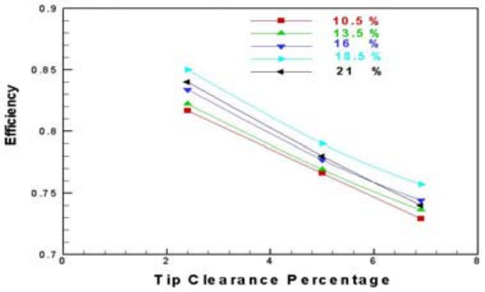
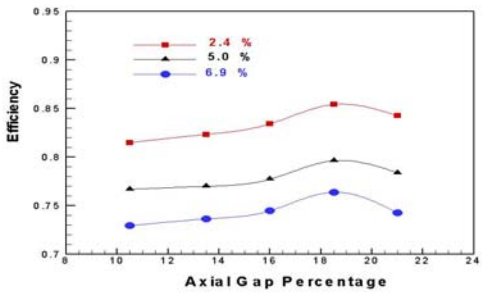


Figure 9: Contour of absolute Mach number at midspan plane



(a) At different tip clearance



(b) At different axial gap

Figure 10: Performance curves

5 CONCLUDING REMARKS

In this study , turbine efficiency has been calculated for the different combinations of axial gap and tip clearance. The results show that the efficiency is maximum for the combination of 18.5% axial gap and tip clearance 2.4%.. From the result investigating different axial gap, efficiency increases with increase in axial gap and become maximum and then starts reducing with further increase in axial gap. The results are compared with Hass [1] and agreement with its experiment is achieved as shown in "Figure 11".

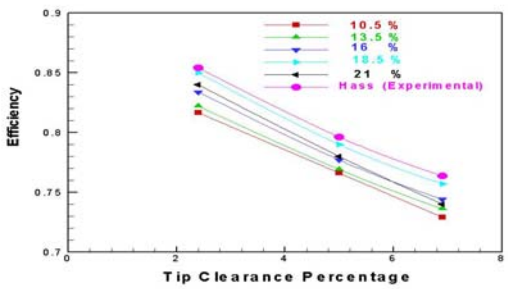


Figure 11: Comparison between present work and Hass[1]

Table 3: Performance for Different Setups

Sets	P_T(bar) (Inlet)	P_T(bar) (Outlet)	T_T K) (Inlet)	T_T (K) (Outlet)	η %
A1	9.1	3.863	1478	1247	81.66
B1	9.1	3.803	1478	1241.5	82.46
C1	9.1	3.894	1478	1243.97	83.414
D1	9.1	4.049	1478	1248.34	85.42
E1	9.1	4.09	1478	1254	84.26
A2	9.1	3.61	1478	1245.8	76.69
B2	9.1	3.641	1478	1246.8	76.98
C2	9.1	3.779	1478	1253	77.71
D2	9.1	3.938	1478	1257.3	79.62
E2	9.1	3.933	1478	1260.4	78.39
A3	9.1	3.623	1478	1258	72.91
B3	9.1	3.727	1478	1261.9	73.625
C3	9.1	3.651	1478	1255	74.45
D3	9.1	3.937	1478	1266.22	76.37
E3	9.1	3.733	1478	1260.4	74.26

REFERENCES

[1] Dey, D., "Aerodynamic Tip Desensitization In Axial Flow Turbines", Phd, December, 2001 The Pennsylvania State University.

[2] Da Silva, L. M., Tomita, J. T., and Barbosa, J. R., " A study Of The Influence Of The Tip-Clearance Of An Axial Turbine On The Tip-Leakage Flow Using CFDTechniques", 21st Brazilian Congress of Mechanical Engineering 2011, October ,24-28, 2011, Natal, RN, Brazil. Natal, RN, Brazil.

[3] Gao, J., Zheng, Q., Li, Y., and Yue, G., " Effect of axially non-uniform rotor tip clearance on erodynamic performance of an unshrouded axial turbine", Journal of Power and Energy, November, 2011.

[4] Hass. J.E, Boyle, R.J, "Analytical and Experimental Investigation Of Stator Endwall Contouring In A Small Axial-Flow Turbine", National Aeronautics and space Administration, 1984.

[5] Li, J., Xin, Y., Qiang L., Yonghui, X., Zhenping, F., "The Effect of the Shrouded Rotor Blade Tip Leakage Flow on the Aerodynamic Performance of a One and Half Turbine Stage", International Conference on Power Engineering October, 23-27, 2007, Hangzhou, China.

[6] Lakshminarayana, B., Chernobrovkin, A., and Ristic, D., "Experimental and Numerical Study Of Unsteady Viscous Flow Duo To Rotor-Stator Interaction In Turbines", 34th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit Jul y13-15,1998/Cleveland, OH.

[7] PFAU, A., "Loss Mechanisms in Labyrinth Seals of Shrouded Axial Turbines", Phd, Zurich, Germany,2003.

[8] QingJun, Z., XiYang, L., HuiShe, W., XiaoLu, Z. and JianZhong, X., " Experimental investigation on unsteady pressure fluctuation of rotor tip region in high pressure stage of a vaneless counter-rotating turbine", Science in China Series E: Technological Science, 2009, vol. 52,1478-1483.

[9] Yadav, R., Gulati, V., and Katyal, P., "Investigations of Gas Turbine Characteristics by Varying Tip Clearance and Axial Gap", International Journal of Engineering Research and Applications (IJERA) , Vol. 1, Issue 3, pp.1058-1064., 2008.