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Investigations of Gas Turbine Characteristics by Varying Tip Clearance and Axial Gap

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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 3.5 mm and 5% tip clearance is the optimum set value for the maximum performance.

Keywords **– Axial flow gas turbine, Axial gap, Tip clearance**

I. INTRODUCTION

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, an effort has been made in this work to investigate the effect of change in parameters such as axial gap and tip clearance which are source of inefficiency. 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 varying from 2.5 mm to 4.5 mm with the interval of 0.5 mm and a tip clearance out of values 3%, 5% and 7% (Table 1).

II. LITERATURE

Rashid [1] investigated curtis stage nozzle/rotor aerodynamic interaction and the effect on stage performance. Gaetani [2-3] in his first part experimentally analyzed the steady flow field in the stator-rotor axial gap by means of conventional five-hole probe and a temperature sensor. He also analyzed the unsteady flow measured downstream of a modern HP turbine. Denos [4] described some results of a large experimental programme on unsteady flow through the rotor of a transonic turbine stage in the large compression tube turbine facility. Pullan [5] investigated three-dimensional stator-rotor interaction in a turbine stage. Hodson [6] investigated that the interaction of wakes shed by a moving blade row with a downstream blade row causes unsteady flow. Prakash [7] explained that the tip leakage in high-pressure turbines contributes to aerodynamic losses and migration of hot gasses towards the tip resulting in increased thermal distress.

Table 1: Different Combinations of Axial Gap and Tip Clearance

		1	2	3
Setup	A	2.5 mm and 3%	2.5 mm and 5%	2.5 mm and 7%
	B	3.0 mm and 3%	3.0 mm and 5%	3.0 mm and 7%
	C	3.5 mm and 3%	3.5 mm and 5%	3.5 mm and 7%
	D	4.0 mm and 3%	4.0 mm and 5%	4.0 mm and 7%
	E	4.5 mm and 3%	4.5 mm and 5%	4.5 mm and 7%

III. MODELING AND ANALYSIS

CFX-BladeGen software is used to generate the 3-D profile of turbine blades. After importing geometry definitions from BladeGen, meshing is done by using ANSYS TurboaGrid software followed by analysis on ANSYS CFX software.

No boundary condition provides neat tip to check boundary profile, so it is to specify all the relevant parameters i.e. inlet and outlet pressure, temperature, properties of the gases which are coming from the combustion chamber (Table 2). Once all the critical parameters are set, the results are checked on the CFX.

The axial flow gas turbine used in this case is 2-stage. The gas turbine space is divided into five regions, viz. Inlet (I), Interface1 (IF1), Interface2 (IF2), Interface3 (IF3), and Outlet (O) (Fig. 1). For a setup, total pressure (P_T), static pressure (P_S), total temperature (T_T), static temperature (T_S) and Mach number (M) are calculated on these five regions (Table 3).

Table 2: Data for the Design and Analysis Softwares

Data			Value
Number of Blades (N_B)	Stator	First stage	47
		Second stage	73
	Rotor	First stage	72
		Second stage	69
Rotational Speed (N)			15000 rpm
Total Mass Flow Rate (m)			1.17 kg/sec
Boundary Conditions	Inlet Pressure (P_i)		5.4 bar
	Outlet Pressure (P_o)		2.66 bar
	Inlet Total Temperature (T_i)		640 K
Properties of Gases	Dynamic Viscosity (n)		1.849e-05 Pa/sec
	Thermal Conductivity (K)		0.25578 W/mK
	Specific Heat (Cp)		7918 J/kgK
	Fluid Gamma (γ)		1.378

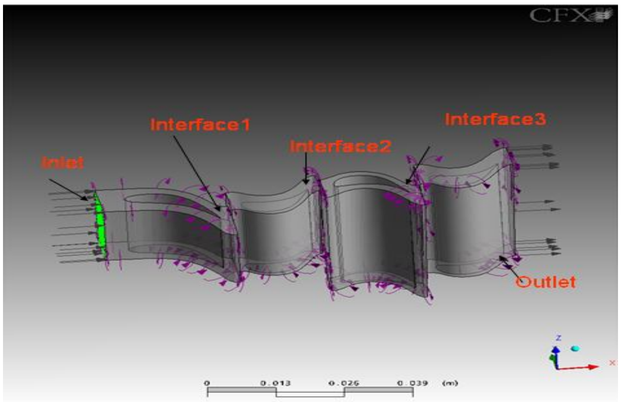


Fig. 1: Gas turbines regions

IV. PERFORMANCE

This section shows the performance in terms of efficiency for each set up. The efficiency is calculated on the basis of the torque generated by each blade of the two rotors. For an instance, efficiency calculation for a setup of 3.5 mm axial gap with 5% of tip clearance is shown. Similarly, efficiency for other combinations of axial gap and tip clearance has been calculated (Table 4).

$P_{\text{First Rotor}} = 2PNT_1N_{B1}/60000 = 295.008$ KW (1)

$P_{\text{Second Rotor}} = 2PNT_2N_{B2}/60000 = 193.7$ KW (2)

$P_A = P_{\text{First Rotor}} + P_{\text{Second Rotor}} = 488.708$ KW (3)

$P_T = m\, Cp\, T_i\, (1- 1/ (P_i/P_o))\, e^{(\gamma-1)/\gamma)} = 1046.66$ KW (4)

$\eta = P_A/P_T = 46.69$ % (5)

Table 3: Parameters at Different Regions of Gas Turbine

Set up	P	I	IF1	IF2	IF3	O
A1	P_T	5.39	4.39	3.50	3.08	2.96
	P_S	5.26	3.65	3.16	2.84	2.65
	T_T	640.00	622.75	615.17	601.97	601.61
	T_S	636.30	598.81	603.06	592.09	588.53
	M	0.17	0.45	0.29	0.31	0.36
B1	P_T	5.39	4.38	3.49	3.08	2.97
	P_S	5.26	3.60	3.14	2.83	2.65
	T_T	640.00	622.41	614.31	601.17	600.76
	T_S	636.25	597.16	601.79	591.15	587.60
	M	0.17	0.47	0.30	0.31	0.36
C1	P_T	5.39	4.38	3.45	3.15	3.03
	P_S	5.27	3.53	3.06	2.84	2.66
	T_T	640.00	620.10	610.86	594.98	594.07
	T_S	635.81	585.72	592.74	579.03	573.11
	M	0.18	0.54	0.37	0.38	0.45
D1	P_T	5.39	4.37	3.47	3.08	2.97
	P_S	5.26	3.54	3.11	2.83	2.65
	T_T	640.00	621.98	613.20	600.31	599.90
	T_S	636.18	594.97	600.06	590.40	586.65
	M	0.17	0.48	0.31	0.31	0.36
E1	P_T	5.39	4.42	3.53	3.14	3.02
	P_S	5.27	3.57	3.17	2.83	2.66
	T_T	640.00	620.19	610.32	594.91	594.36
	T_S	635.86	585.90	593.76	578.08	574.30
	M	0.18	0.54	0.35	0.40	0.44
A2	P_T	5.39	4.40	3.52	3.07	2.97
	P_S	5.26	3.64	3.21	2.81	2.66
	T_T	640.00	622.73	616.26	602.16	601.52
	T_S	636.28	598.18	604.99	591.36	588.39
	M	0.17	0.46	0.28	0.32	0.36
B2	P_T	5.39	4.41	3.51	3.13	3.00
	P_S	5.27	3.61	3.18	2.81	2.66
	T_T	640.00	620.89	613.12	596.65	595.64
	T_S	635.92	589.10	597.75	579.95	579.47
	M	0.18	0.52	0.33	0.39	0.43
C2	P_T	5.39	4.38	3.45	3.14	3.04
	P_S	5.27	3.52	3.08	2.84	2.66
	T_T	640.00	620.08	612.14	595.66	594.49
	T_S	635.81	585.6	594.37	579.89	573.42
	M	0.18	0.54	0.36	0.38	0.45
D2	P_T	5.39	4.43	3.55	3.14	3.02
	P_S	5.27	3.59	3.19	2.83	2.66
	T_T	640.00	620.43	611.97	596.04	595.25
	T_S	635.86	578.84	595.68	579.41	574.81
	M	0.18	0.54	0.35	0.40	0.44
E2	P_T	5.39	4.43	3.35	3.15	3.03
	P_S	5.27	3.60	3.19	2.83	2.66
	T_T	640.00	620.56	612.01	595.92	595.06
	T_S	635.87	587.71	595.36	578.83	574.45
	M	0.18	0.53	0.35	0.40	0.44
A3	P_T	5.39	4.40	3.50	3.06	2.94
	P_S	5.26	3.62	3.18	2.79	2.65
	T_T	640.00	622.53	616.65	602.47	601.76
	T_S	636.30	597.17	604.99	591.57	589.41
	M	0.17	0.47	0.29	0.32	0.35

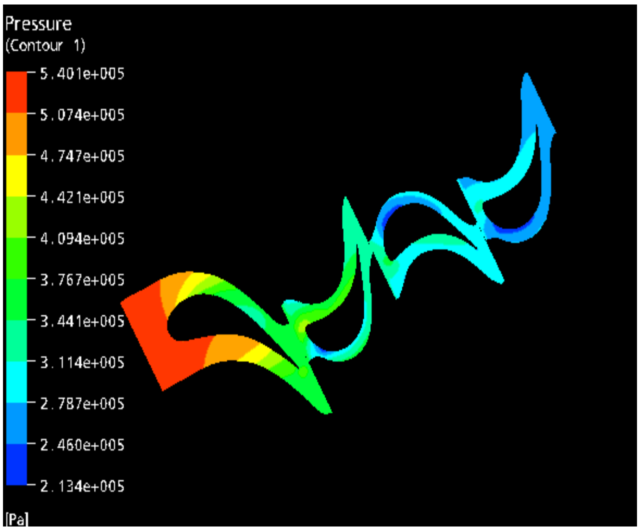
B3	P_T	5.39	4.38	3.50	3.06	2.95
	P_S	5.26	3.56	3.15	2.81	2.65
	T_T	640.00	622.16	615.81	601.81	601.15
	T_S	636.19	595.57	603.17	591.44	588.43
	M	0.17	0.48	0.30	0.32	0.35
C3	P_T	5.39	4.37	3.48	3.06	2.96
	P_S	5.25	3.53	3.13	2.81	2.65
	T_T	640.00	621.92	615.21	601.47	600.76
	T_S	636.14	594.33	602.49	591.05	587.94
	M	0.17	0.49	0.30	0.32	0.35
D3	P_T	5.39	4.38	3.46	3.06	2.96
	P_S	5.25	3.53	3.13	2.81	2.65
	T_T	640.00	621.92	614.99	601.29	600.46
	T_S	636.10	592.50	602.00	591.01	587.41
	M	0.17	0.51	0.30	0.31	0.36
E3	P_T	5.39	4.39	3.49	3.06	2.96
	P_S	5.25	3.49	3.13	2.80	2.65
	T_T	640.00	621.64	615.42	600.99	600.17
	T_S	636.09	592.13	602.41	590.26	587.47
	M	0.17	0.51	0.30	0.32	0.35

Table 4: Performance for Different Setups

Sets	**T_1 (Nm)**	**T_2 (Nm)**	**P_{Actual} (KW)**	**m Kg/sec**	**η (%)**
A1	2.45	1.67	463.33	1.33	44.26
B1	2.53	1.69	474.22	1.34	45.30
C1	***2.62***	1.84	500.26	1.19	47.79
D1	2.53	1.70	486.51	1.35	46.48
E1	2.55	***1.85***	494.50	1.18	47.24
A1	2.41	1.64	454.33	1.33	43.40
B1	2.42	1.74	467.46	1.17	44.64
C1	***2.57***	1.78	488.70	1.19	46.69
D1	2.47	1.80	479.20	1.18	45.78
E1	2.50	***1.83***	485.59	1.19	46.39
A1	2.37	1.58	443.66	1.33	42.38
B1	2.50	1.60	461.73	1.35	44.11
C1	2.55	1.61	468.39	1.36	44.75
D1	2.58	1.62	472.72	1.36	45.16
E1	***2.58***	***1.64***	474.93	1.36	45.37

V. PRESSURE AND TEMPERATURE DISTRIBUTION

Pressure and temperature distribution for the set up of 3.5 mm axial gap and 5% tip clearance is shown in Fig. 2. It can be seen from the distribution that maximum pressure at the inlet and outlet is 5.4 bar and 2.66 bar respectively. In the first stator, pressure loss is more as compared to second stator. Velocity at the first rotor inlet is maximum and again pressure builds up between the first rotor and second stator and velocity increases slightly in the stator. Temperature at the inlet is maximum and it reduces downstream of the stage. The gas expands in the rotor row and the temperature reduces in the stages. The turbine extracts kinetic energy from the expanding gases, as the gases come from the burner, converting this kinetic energy in to shaft power to drive the compressor and the engine accessories.



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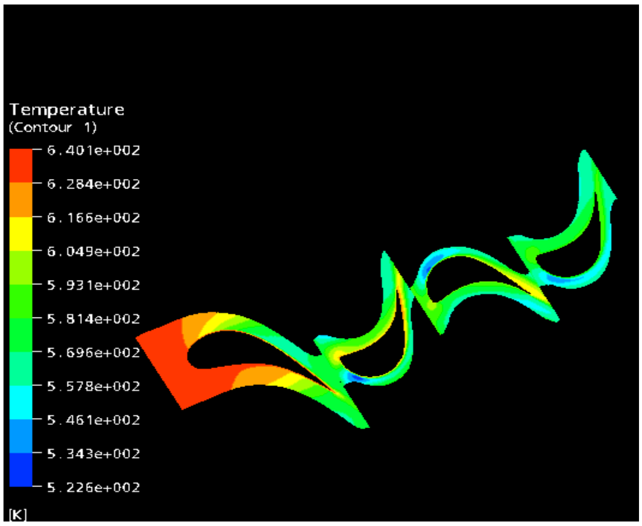

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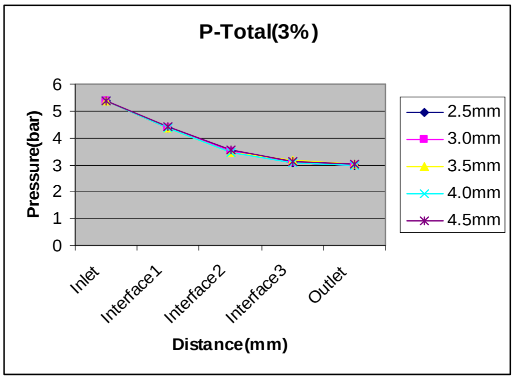
Fig. 2: (a) Pressure distribution contour (b) Temperature distribution contour

VI. RESULTS

Pressure and Temperature Variations

The pressure and temperature variations for different axial gaps and tip clearances are (Fig. 3 and 4) plotted from inlet to outlet along with various interfaces. It can be seen that the pressure and temperature of the gas decreases as it passes through various stages of turbine.



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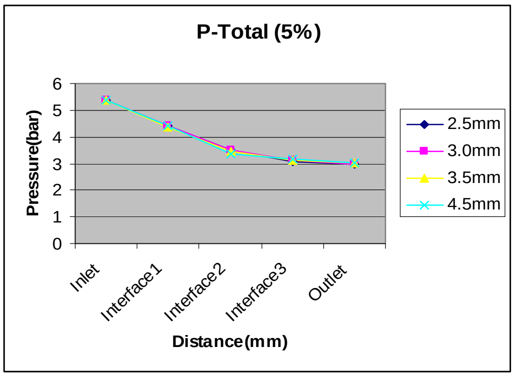

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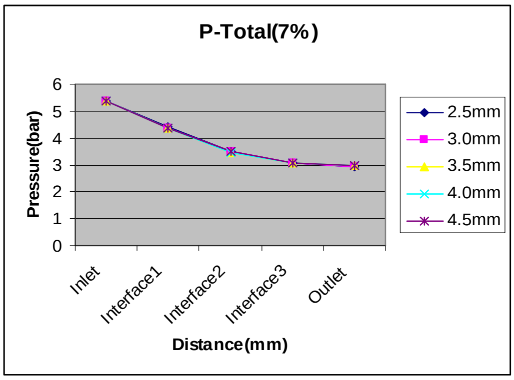
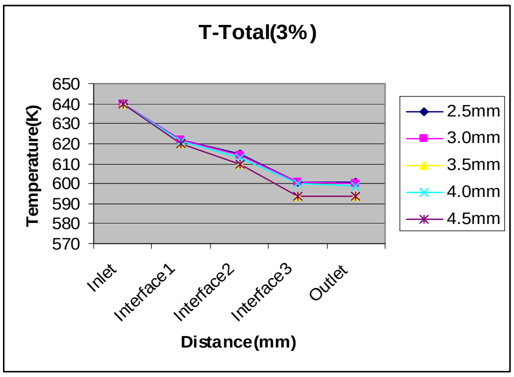


(c)

Fig. 3: Total Pressure (a) 3% tip clearance (b) 5% tip clearance (c) 7% tip clearance



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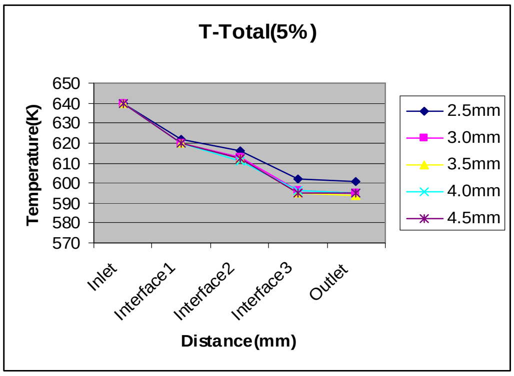

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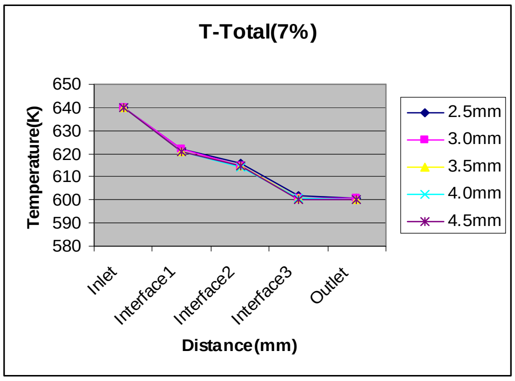


(c)

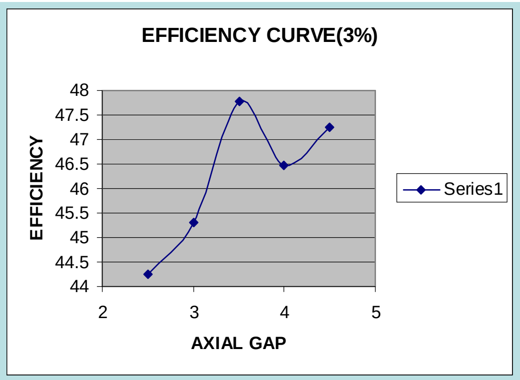
Fig. 4: Total Temperature (a) 3% tip clearance (b) 5% tip clearance (c) 7% tip clearance

Efficiency Curves

For 3%, 5% and 7% tip clearances, efficiency curves are plotted between efficiency and axial gap (Fig. 5).

For 3% and 5% tip clearance, efficiency is maximum at 3.5 mm axial gap because the leakage flows finds space for redistribution with sufficient velocity. The leakage flow mixing with the rotor passage flow reduces the efficiency. The leakage flows for 3% is less and hence efficiency is more. The leakage flows for 5% is more than as compared to the tip of 3% and less for than for 7% of tip clearance. Efficiency is increasing again for the axial gap of 4.5 mm because the rotor dimension and flow has sufficient space for mixing and the flow become streamline. For the 4.5 mm axial gap torque on first rotor is minimum but increases for the second rotor because flow mixes out to form streamline in the first stage and velocity reduces and in the second stator pressure drop has taken place and velocity increases.

For the case of 7%, efficiency is maximum for the axial gap of 4.5 mm because losses is more through the tip and for less axial gap the flow becomes complicated because of the mixing of tip leakage flow with the rotor exit flow and this reduces the efficiency for the small gap and slowly increases with increase in axial gap. Velocity reduces with the increase in axial gap.



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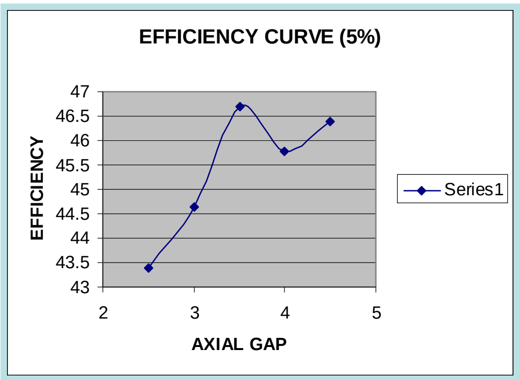

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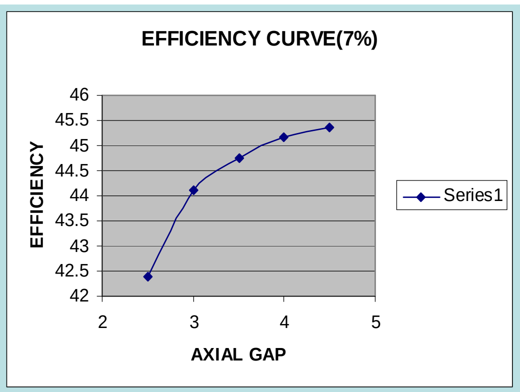


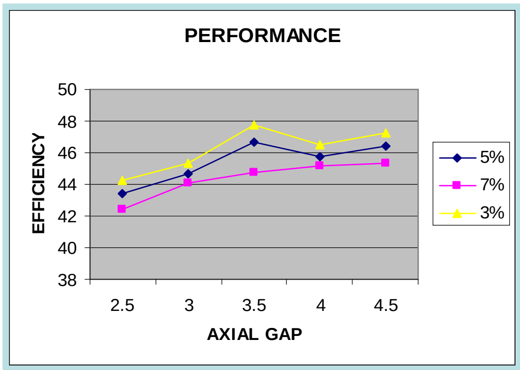
(c)

Fig. 5: Efficiency curves (a) for 3% (b) for 5% (c) for 7%

Performance Curves

A comparison has been made for different tip clearances with different axial gaps. The plot (Fig. 6a) shows that the efficiency is maximum for the axial gap of 3.5 mm with tip clearance of 3%. For the optimum tip clearance (5%), the efficiency is maximum for 3.5 % and this is the design efficiency. Efficiency is minimum for the axial gap of 2.5 mm with tip clearance of 7%.

For different axial gap, the performance has been shown in the Fig. 6b. For 3% tip clearance having axial gap of 3.5 mm, efficiency is maximum and minimum for the case of 2.5 mm. Similarly, for 5% tip clearance, efficiency is maximum for 3.5 mm axial gap and minimum for 2.5 mm axial gap. For 7% tip clearance efficiency is maximum for 4.5 mm axial gap and minimum for 2.5 mm axial gap.



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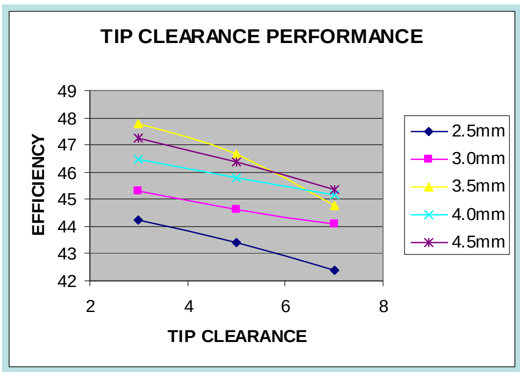

(b)

Fig. 6:- Performance curves

VII. 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 3.5 mm axial gap and tip clearance 3%. But for the actual case, the tip clearance of 3% is not sufficient because of thermal expansion of rotor blade working under high temperature and rotating with high speed. Losses are less in case of minimum tip as compared to maximum tip clearance. The second highest efficiency is for the set of axial gap of 3.5 mm and 5% tip of i.e. the optimum set value axial gap and tip clearance for the maximum performance. The losses are maximum for the case of 7% tip clearance. The leakage flow mixing with the rotor passage flow also reduces turbine stage efficiency; hence for the maximum tip efficiency is less. From the result for different axial gap, efficiency increases with increase in axial gap and become maximum and after that it starts reducing with further increase in axial gap. The torque on the first rotor is maximum as compared to second rotor because of large pressure drop. Mach number is maximum for the first rotor and it decreases for the second rotor.

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