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Research Article

Thermodynamic Analysis of Air Cooled Gas Turbine used in Marine Applications

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Abstract

Failure of turbine blades before the anticipated working hours is a common problem in gas turbine units, due to hot corrosion and creep developed because of exposure of blades to high temperature gases over a prolonged period of time. To prevent such a failure, several blade cooling techniques have been suggested. In this paper an attempt is made to estimate the performance of marine gas turbine with internal convection air cooling for both stator and rotor blades of a gas generator turbine. The model evaluates the efficiency and specific work of simple open cycle air cooled gas turbine where expansion is considered as a continuous process with simultaneous heat and work extractions and which is influenced by the mixing losses of the coolant with the combustion gases. The mass flow rate of coolant air as a percentage of gas is also determined. Pumping work of coolant and heat transfer from the combustion gases to the coolant significantly influences the performance of the plant. The sensitivity of the efficiency and specific work to each key input parameter is reported. The results are also compared with the previously published results of open loop air cooled gas turbine model.

Keywords: *Blade cooling – Open loop air cooling; Marine gas turbine; Efficiency; Specific work output; Coolant mass flow rate.*

1. Introduction

Because of simple, rigid and reliable operation, gas turbines have found wide applications in jet air crafts, standby power plants and in marine applications for driving ship vessels and auxiliary equipment (Saravanamuttoo *et al.*, 2013) For obtaining maximum performance from the gas turbine engine, it is always advisable to go for high pressure ratios along with increased turbine inlet temperatures (H. Kurt *et al.*, 2009)(M.M. Rahman *et al.*, 2011) Even though advanced gas turbine engines operate at high temperatures (1200 C – 2000 C)(Janusz Kotowicz *et al.*, 2015) , the blade failure occurs because they are exposed to high temperature gases over a definite period of time. The heat transfer to the blades increases considerably resulting in their thermal and mechanical failure (Nagabhushana Rao *et al.*, 2014). Taking into metallurgical constraints, we need to provide ceramic coatings to the blades or to provide sufficient cooling techniques to prevent the eventuality earlier, so that the temperatures can be maintained with in allowable limits. Increasing the TIT, CPR, component efficiencies, better cooling techniques, complex configurations with reheating, inter-cooling, regeneration along with combined power plants are the focus areas in developing gas turbine technologies which improves its performance. These developments require active cooling of the hot gas components of turbine in order to avoid a reduction in their operating life due to an unfavorable combined exposure of oxidation, creep and thermal stress on them (Sanjay *et al.*, 2008).

Most gas turbine components are cooled by air bleed off from compressor. Blades internally cooled by convection are treated as heat exchangers operating at constant metal temperature and coolant exit temperature is simply expressed as a function of heat exchanger effectiveness. In order to understand the performance of gas turbine cycle with high combustion temperatures, the impact of blade and vane cooling on cycle performance must be included. The model evaluates mixing losses of the coolant with the combustion gases for both stator and rotor blades of two stages of HP turbine. It also evaluates the pumping power of the coolant, heat transfer from the gases to the coolant and calculates the combined thermodynamic efficiency and specific power using state of the art knowledge. Optimum design conditions

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have been selected and sensitivity analysis can be conducted around these optimum conditions.

2. System Configuration

Figure 2.1 shows a flow diagram for the cycle of Marine gas Turbine consisting of gas generator and power turbine mounted on two separate shafts. It includes axial flow compressor consisting of 16 stages and gas generator (HP) turbine consisting of two stages and power (LP) turbine consisting of 6 stages.

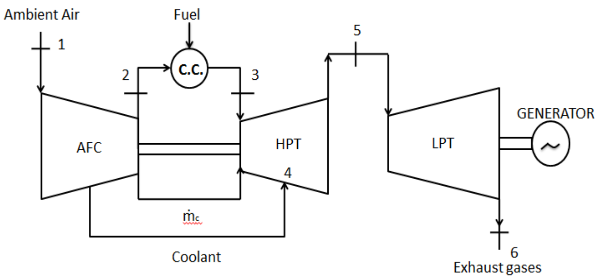

Fig. 2.1 Schematic of an air cooled model

The high temperature exhaust gases from the exit of HP turbine directly expand in the LP turbine. The air from compressor exit is bleed off and fed to the Gas generator turbine for cooling of stator and rotor blades of the two stages of HP Turbine. The thermodynamic state points are illustrated on Temperature –Entropy diagram in Figure.2.2.

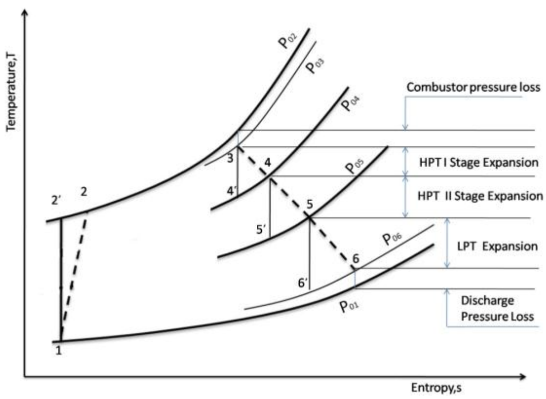

Fig. 2.2 Temperature-Entropy diagram of marine gas turbine model

3. Modeling and governing equations

Modeling of cooled marine gas turbine has been done in the following steps:

3.1. Gas model

Ambient air taken at 1.013 bar and 288 K with 50 % relative humidity is fed to the compressor. The fuel is assumed to be natural gas. The specific heat of gas at constant pressure is assumed to be a function of temperature given by the polynomial

$$C_p(T) = a + bT + cT^2 + dT^3 + \ldots \quad (1)$$

where a, b, c, d etc., are coefficients of the polynomial, their values taken from the work of Toulounkain and Tadash. Thus the enthalpy of gas can be calculated using the above polynomial in the following equation.

$$h = \int_{Ta}^{T} C_p(T)\,dT \quad (2)$$

3.2. Compressor

Due to irreversibilities, the compression in axial flow compressor is assumed to be polytropic. At any section of the compressor, temperature of air can be expressed in term s of polytropic efficiency and local pressure, by the equation given as under:

$$\frac{dT}{T} = \left[\frac{R}{\eta_{pc} C_{pa}}\right] \frac{dP}{P} \quad (3)$$

Mass balance and energy balance across the control volume of compressor,

$$\dot{m}_{c,i} = \dot{m}_{c,e} + \sum \dot{m}_{c,j} \quad (4)$$

$$\dot{W}_C = \dot{m}_{c,e} C_{P_{c,e}} T_{c,e} + \sum \dot{m}_{c,j} C_{P_{c,j}} T_{c,j} - \dot{m}_{c,i} C_{P_{c,i}} T_{c,i} \quad (5)$$

Compressor temperature rise can be obtained by

$$(\Delta T)_c = [T_{01}/\eta_C][r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1] \quad (6)$$

3.3 Combustor

The combustion process in a combustor is irreversible and it suffers from pressure losses and incomplete combustion. This can be taken care by considering pressure drop as a percentage of air pressure at compressor exit and by introducing combustion efficiency. For a given combustor discharge temperature i.e., turbine inlet temperature (TIT), pressure drop and combustion efficiency, the fuel flow is found from the mass and energy balance.

$$(\dot{m}_e)_{comb} = (\dot{m}_i + \dot{m}_f)_{comb} \quad (7)$$

$$\dot{m}_f . LHV . \eta_{comb} = \dot{m}_e h_e - \dot{m}_i h_i \quad (8)$$

3.4 Cooled gas turbine

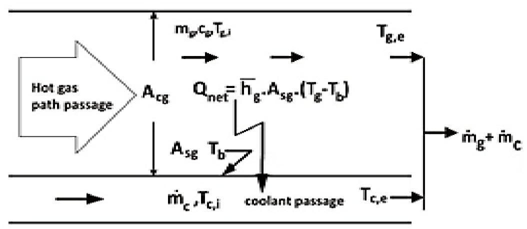

Fig. 3.1 Internal convection open loop air cooled model

In the present gas turbine blade cooled model, the coolant bleed is taken from compressor discharge and it is used for cooling both stages of gas generator turbine including stator and rotor blades. The simple model for internal convection cooling of blades is shown in Figure 3.1 .The blade coolant requirement mainly depends upon the cooling technique employed. The losses in the open loop cooling scheme are reduction in enthalpy and loss of pressure. The mixing of the spent coolant with the main gas stream causes reduction in temperature and pressure loss. Several approaches for the determination of cooling flow rate are given in literature by (Louis, *et al* 1983), El- Masri 1988), Bolland and Stadaas,1991), (Horlock, *et al* 2001), (Sanjay, *et al* 2008), (Kristin Jordal, *et al* 2004). The model used in this investigation for cooled turbine is the refined version of those of (Louis, *et al* 1983), Horlock, *et al* 2001) and Sanjay, *et al* (2008). The expansion path in two stage generator turbine with cooling provided to both stator and rotor blades is shown in Figure 3.2

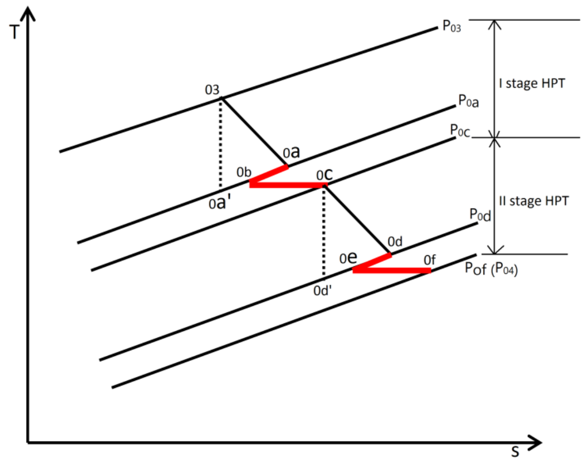

Fig. 3.2 Model for expansion in a two stage HP turbine

3.5 Open Loop Cooling Scheme

The following assumptions are made for the model.

(1). Gas turbine blades cooled by internal convection are treated as heat exchangers operating at uniform temperature and the coolant exit temperature is expressed as a function of heat exchanger effectiveness.

(2). After removing heat from the hot components, the coolant is mixed with the main flow of gas. Mixing and pumping losses are proportional to the coolant flow in this cooling process and it is always suggestible to minimize the ratio of coolant to main gas flow.

(3). Cooling of stator and rotor of I stage of HP turbine results in a drop of stagnation temperature from stage outlet conditions 0a to 0b which is assumed to occur at constant stagnation pressure. Enthalpy at 0c is determined by the enthalpy balance equation.

$$\dot{m}_g h_g + \dot{m}_c h_c = (\dot{m}_g + \dot{m}_c)\, h_b \tag{9}$$

4). Mixing of coolant with primary flow results in stagnation pressure drop which considered at exit of stator cascade results in a change of stage from 0b to 0c at constant stagnation temperature which can be calculated by model of (Shapiro 1953) assuming that the flow is one dimensional and semi perfect. The incremental change of stagnation pressure can be expressed by momentum balance as

$$\frac{dP_{loss}}{P_0} = -\gamma M^2\, Y \frac{\dot{m}_c}{\dot{m}_g} \tag{10}$$

(5).The expansion is assumed to occur at polytropically with isentropic efficiency of η_{pHPT}at constant mass flow rate equal to that at state03. Hence 03 -0a' represents isentropic expansion and 03-0a represents actual expansion such that

$$\eta_{pHPT} = \frac{h_{03}-h_{0a}}{h_{03}-h_{0a'}} \tag{11}$$

This enables temperature at point 0a to be determined for a given stage pressure ratio and a given value of η_{pHPT}. The power developed in this stage is

$$W_{HPT1} = \dot{m}_g(h_{03} - h_{0a}) \tag{12}$$

(6).The stagnation pressure and temperature drop at the exit of II stage of HP turbine can also be determined by a procedure similar to that described step (3) and step (4) mentioned above.

(7) The second stage of HP turbine can also be analyzed in the similar way i.e., repeat the calculations shown from points (3) to (5)

The power developed in the second stage is

$$W_{HPT2} = \dot{m}_g(h_{0c} - h_{0d}) \tag{13}$$

The total power developed in the HP turbine is

$$W_{HPT} = W_{HPT1} + W_{HPT2} \tag{14}$$

3.6 Calculation of Coolant flow rate

To simplify the calculation, internally cooled blades by convection, coolant temperatures are expressed as a function of blade heat exchanger effectiveness,

$$\varepsilon = \frac{T_{ce}-T_{ci}}{T_b-T_{ci}} \tag{15}$$

The turbine is considered as an expander whose walls continuously extract work. The stagnation temperature of the gas relative to those walls is a step like variation in real machine, but it is approximated by a very close to continuously varying function which is depicted in Figure 3.3.

It shows that the assumed temperature profile underestimates the relative gas to surface temperature difference for a stator, while overestimating it for the rotor. The stage average is essentially correct. An element of the expansion is shown in Figure 3.4.

The Cooling factor R_{conv} in this case can be written as

$$R_{conv} = \frac{(T_{gi}-T_b)\, c_{pg}}{\varepsilon(T_b-T_{ci})c_{pc}} \quad (16)$$

The required cooling air mass flow for the corresponding gas flow, for an element of the expansion path can be derived and can be expressed in the following way.

$$\frac{\dot{M}_c C_{pc}}{\dot{M}_g C_{pg}} = \sigma \frac{dT_g}{T_a} \frac{(T_g-T_b)}{(T_b-T_c)} \quad (17)$$

where, $$\sigma = \frac{St\frac{A_{w,stage}}{A_g}}{C(k-1){M_a}^2\varepsilon} \quad (18)$$

and Mach number M_a is

$$M_a = \frac{u}{\sqrt{T_a k R_a}} \quad (19)$$

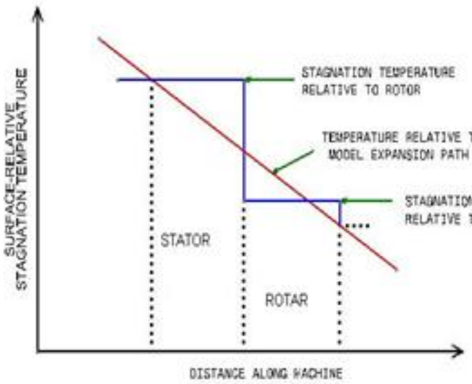

Fig. 3.3 Schematic representation of the temperature profile in the expansion path with continuous work extraction

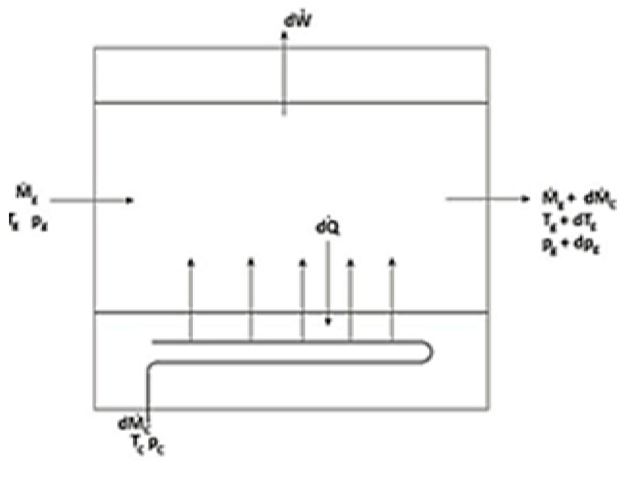

Fig.3.4 Element of the cooled expansion path

For a cascade, the ratio of wall area to gas flow path inlet area is

$$\frac{A_b}{A_g} = \frac{AP}{BCS} + \frac{2AB}{BH} \quad (20)$$

where, AP= airfoil perimeter, BCS = blade circumferential spacing, AB = axial breadth, and BH = blade height. Typical value of $\frac{A_{bs}}{A_g}$is around 4 for a row and 8 for a stage. Stanton numbers based on the cascade inlet area, according to (EI Masri 1986), are typically about 0.005. The Mach number M_a, based on the pitch line velocity, is typically in the range of 0.6 to 0.9 for current gas turbines. With k = 1.4 and C = 1.2, Heat exchanger effectiveness ε depends upon the cooling air delivery temperature and the blade internal heat transfer area, the heat transfer coefficient and the temperature difference. The value of ε by (Louis *et al* 1983) is assumed to be in the range of 0.3-0.5 where the lower value is typical for film cooled blades with multiple compressor extractions, and the upper value is typical for internally cooled blade with cooling air taken from the compressor discharge. The parameter σ characterizes the relative heat to work loadings on the machine surfaces, and can be regarded as a descriptor of the level of technology for a cooled machine.

Combining equations (15-17) the final expression for coolant mass flow rate can be obtained from

$$\frac{\dot{m}_c}{\dot{m}_g} = 0.0158\, R_{conv} \quad (21)$$

4. Results and Discussion

In this model, cycle efficiency and specific power depends on four primary variables which are (1) The cycle pressure ratio (2) Turbine inlet temperature (3) heat exchanger effectiveness (4) Turbine blade temperature.

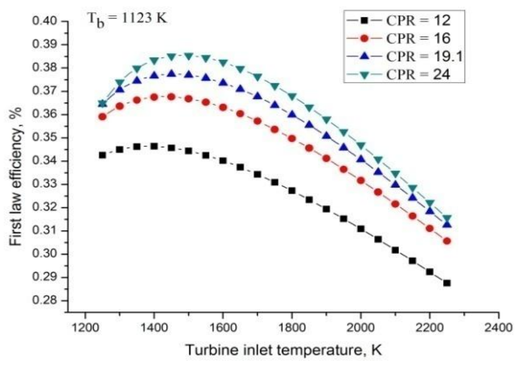

Fig. 4 Effect of TIT on First law efficiency for various CPRs

Figure 4 Shows the variation of the First law efficiency with turbine inlet temperature (TIT) at a given blade temperature T_b 1123 K for different pressure ratios ranging from 12 to 24. It shows that the efficiency increases with increase in TIT up to a certain value of TIT and thereafter decreases for each CPR variation. There exists an optimum value of TIT for a given CPR. This is because of the fact that as the TIT increases coolant requirement increases. In the case of air cooling, since the coolant is taken from compressor bleed points the compressor work decreases and turbine work out put increases due to more energy available with gases, so net work output increases and hence efficiency also increases up to a certain limit of TIT. But after some TIT value reaches, the increase in cooling requirement with increasing TIT, reduces the turbine output drastically owing to the mixing and cooling losses which offsets the advantage gained due to the reduction in compressor work. Hence efficiency fall down more rapidly at higher TITs It also evident from the plots that as pressure ratio increases turbine efficiency also increases but the increase in value decreases as CPR increases to higher values.

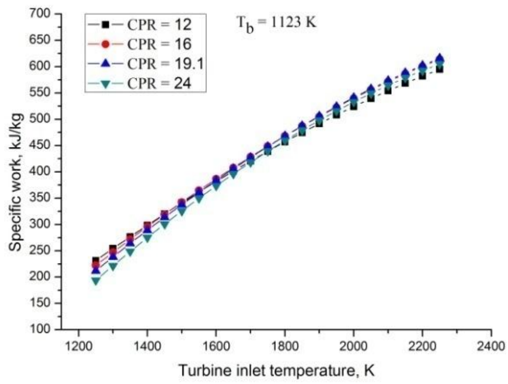

Fig. 5 Effect of TIT on Specific work output for various CPRs

Figure 5 Shows the variation of the Specific work output with turbine inlet temperature (TIT) at a given blade temperature T_b 1123 K for different pressure ratios ranging from 12 to 24. . It is found that specific work output vary linearly with turbine inlet temperature for all CPR variations, as it could be expected from the variation of R which indicated that pumping and mixing losses would vary linearly with turbine inlet temperature. Up to TIT 1700 K specific work decreases as CPR increases because of fact that as CPR increases compressor work increases and hence less specific work output, but after 1700 K the specific work increases as CPR increases because of the fact that increase in turbine work due to increase in TIT is more predominant than increase in compressor work and hence specific work output increases with respect to both increase in TIT and increase in CPR. At TIT =1700K Specific work output have 425 kJ/kg for all CPR variation from 12 to 24.

Figure 6 shows the effect of TIT on blade cooling requirement at a given blade temperature T_b = 1123 K for various pressure ratios ranging from 12 to 24. The coolant requirement increases with increase in TIT for open loop air cooling scheme because since air as the coolant it has a low specific heat as compared to steam or water.

From the figure it also evident that as the pressure ratio increases, the coolant flow rate requirement also increases, because as the pressure ratio increases, the compressor outlet temperature and hence combustor temperature i.e., TIT increases which necessitates more supply of coolant which is air in this system

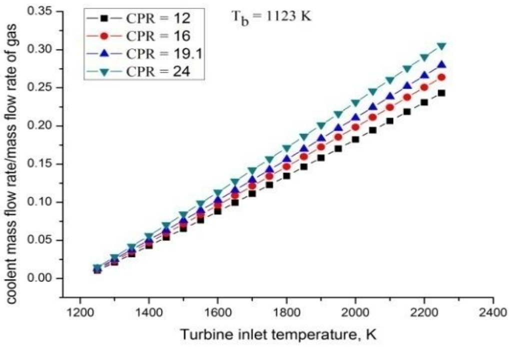

Fig. 6 Effect of TIT on Coolant flow rate for different CPRs

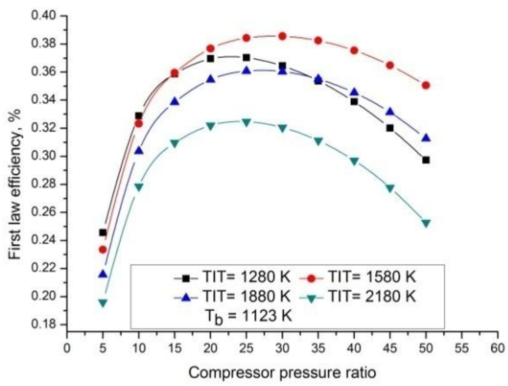

Fig. 7 Effect of CPR on First law efficiency for different TITs

Figure 7 shows variation of First law efficiency with compression ratio for different turbine inlet temperatures varying from 1280 K to 2180 K for a given blade temperature T_b 1123K. It is seen that efficiency increases with increase in pressure ratio up to a certain value of CPR thereafter it decreases, this is due to the less heat supply requirement in combustion chamber. It is also evident from the figure that as the TIT increases the efficiency decreases even the CPR increases for a cooled gas turbine.. This is due to the fact that at high TITs more heat is lost to coolant so turbine work likely to be decreased which causes reduction in first law efficiency even though CPR increases.

Figure 8 Shows the variation of the Specific work output with compressor pressure ratio (CPR) at a given blade temperature T_b 1123 K for different TITs ranging from 1280K to 2180K. It is found that specific work output increases with increase in CPR up to a certain CPR and thereafter it decreases. This is due to the fact that as CPR increases compressor work increases and hence network output decreases. Similarly as TIT increases specific work also increases because of more energy available with gases can be extracted during the expansion by the turbine and hence specific work increases as TIT increases.

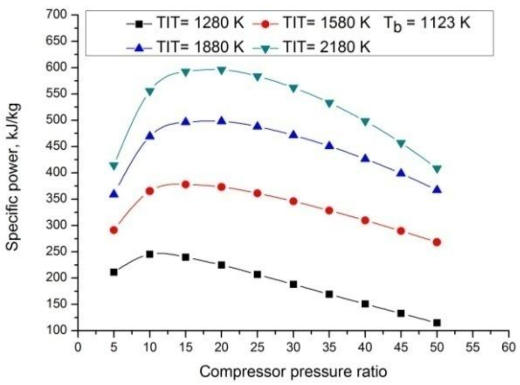

Fig.8 Effect of CPR on Specific Work output for different TITs

Figure 9 Shows the variation of Coolant mass flow rate with compressor pressure ratio for a given blade temperature of 1123 K for different TITs ranging from 1280 K to 2180 K. It is observed from the plots that variation of CPR doesn't have much effect on the coolant requirement at low TIT value of 1280K. But it have significant variation at high TIT values of 1880K and 2180 K. The reason for this is as CPR increases compressor outlet temperature and hence TIT also increases for a given fuel flow rate and hence needs more coolant. Similarly as TIT increases also the requirement of coolant increases along with increase in CPR.

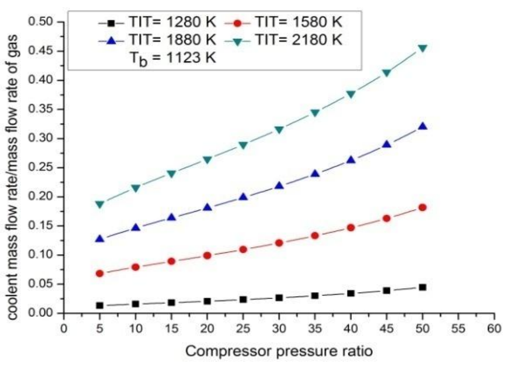

Fig. 9 Effect of compression ratio on coolant flow requirement for different TITs

Conclusions

The gas turbine used in Marine applications has been analyzed for various parameters. There are various types of cooling methods which have been published in the literature. In the present work, open loop internal convection air cooling method is applied to both stator and rotor blades of two stages of the gas generator (HP) turbine and thermodynamic analysis has been carried out. The results are obtained using the suitable input parameters from the published work, for the simple open cycle configuration. The cycle efficiency and specific work are found to be sensitive to the TIT and CPR. The results from the analysis are as follows

(i) The First law efficiency has a maximum value around TIT value of 1400 K to 1450 K even the compressor pressure ratio varies from 12 to 24.
(ii) The Specific work has maximum value around TIT 1700 K even the compressor pressure ratio varies from 12 to 24.
(iii) The coolant mass requirement is found to be maximum when TIT is 2200 K and CPR 24.
(iv) The First law efficiency is maximum at TIT of 1580 K and at a CPR value of 30.
(v) The Specific work reaches maximum value at about CPR value 15 even the TIT increases from 1580 K to 2180 K.
(vi) The coolant mass requirement does not depend much on CPR and mainly depends on TIT.

Table 1 Nomenclature

A	Area , m^2
C_{pa}	Specific heat at constant pressure for air, kJ/kg K
C_{pg}	Specific heat at constant pressure for gas, kJ/kg K
Cp	Specific at constant pressure, kJ/kg
T	Temperature, K
P	Stagnation pressure, bar
M	Mach number
γ, k	Adiabatic index
η_C	isentropic efficiency of compressor
η_t	isentropic efficiency of turbine
η_{PC}	Polytropic efficiency of compressor
$\eta_{p\,HPT}$	Polytropic efficiency of HP turbine
ηp_{LPT}	Polytropic efficiency of LP turbine
h	Specific enthalpy, kJ/kg
$\dot{m}, \dot{M}$	mass flow rate, kg/s
$\dot{m}_c$	mass flow rate of coolant, kg/s
r_p	Pressure ratio
$\dot{m}_g, \dot{M}_g$	mass flow rate of gas, kg/s
W	Power, kW
ϵ	Heat exchanger effectiveness
R_a	Gas constant of air, kJ/kg K
R_g	Gas constant of gas, kJ/kg K
St	Stanton number
LHV	Lower calorific value, kJ/kg
r	recovery factor
Pr	Prandtl number
u	pitch line velocity, m/s
C	proportionality constant
ΔT, dT	temperature difference, C/K
dp	pressure difference, bar
dA	differential element area, m^2
dW	work extraction for blade element , kJ/kg
dQ	heat flow for element during expansion, kJ/kg
Y	cooling air mixing loss factor

ρ	density, kg/m^3
σ	Technology level descriptor
R	Gas constant, J/kg K
$\dot{m}_s$	Mass flow rate of fuel. kg/s
dP_{loss}	Pressure loss, bar
$A_{w,Stage}$	Stage wall (blade) area

Subscripts

a	ambient, air
i	inlet
o, e	Exit
comb	combustor
ci	compressor inlet, coolant inlet
ce	compressor exit, coolant exit
g	Gas
c	compressor, coolant, cooling air loss
HPT	high pressure turbine
LPT	low pressure turbine
S	stage
Ab	adiabatic blade
∞	free stream static condition
TIT	turbine inlet temperature, K or combustor exit temperature
b	blade
w	Wall

Appendix

Table 3 Input Data for The Analysis

Power of HP Turbine, P_{HPT}	= 29 MW
Power of LP Turbine, P_{LPT}	= 23.2 MW
HP Turbine Inlet Temperature, TIT	= 1580 K
Pressure ratio, r_p	= 19.1
Air inlet pressure to compressor, $P_1 = P_{01}$	= 1.013 bar
Air inlet temperature to compressor , $T_1 = T_{01}$	= 288 K
Combustion efficiency ,η_{comb}	= 99.5 %
Fuel used	= Natural gas
Lower Heating Value	=42000 kJ/kg
Isentropic efficiency of Compressor ,η_c	= 0.85
Isentropic efficiency of turbine, η_t	= 0.90
Polytropic efficiency of compressor ,η_{PC}	= 0.92
Polytropic efficiency of HP turbine, η_{HPT}	= 0.92

Table 2 Different states on T-s diagram

03	Gas entry condition to the I stage of HP turbine at pressure P_{03}
03–0a'	Isentropic (ideal) expansion through I stage of a HP turbine
03-0a	Polytropic (actual) expansion through I stage of a HP turbine
0a-0b	I stage internal cooling results in temperature drop
0b-0c	I stage pressure drop due to mixing of gases with coolant
0c	Gas Entry temperature for the II stage of HP turbine
0c-0d'	Isentropic (ideal) expansion through II stage of a HP turbine
0c-0d	Polytropic (actual) expansion through II stage of a HP turbine
0d-0e	II stage internal cooling results in temperature drop
0e-0f	II stage pressure drop due to mixing of gases with coolant
0f(04)	Gas entry temperature for the LP turbine

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