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Thermal Analysis of Power Plant Based on Gas Turbine

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***Abstract* - In recent development of gas turbine cycles, a new method for calculating the coolant flow requirements of a high-temperature gas turbine blade are necessary. It involved consideration of successive chordwise strips of blading; the coolant required in each strip was obtained by detailed study of the heat transfer processes across the wall of the blade and then setting limits on the maximum blade metal temperature. In the present work, a more sophisticated method is developed from the earlier work and is used to calculate the cooling flow required for a nozzle guide vane (the first and second blade row) of a high-temperature gas turbine, with given inlet gas temperature and coolant inlet temperature. The mass of coolant required for the cooling of the gas turbine blade is calculated as a fraction of the external gas flow. Analyses of gas turbine plant performance, including the effects of turbine cooling, are also calculated. If the value of blade temperature at the blade tip is assumed to be limited by material considerations to maximum blade temperature, then the elementary coolant flow rate may be obtained by iteration. Summation along the chord then gives the total coolant flow, for the whole blade. Results using the method are then compared to a simpler gas turbine cycle without the blade cooling. The thermal efficiencies are determined theoretically, assuming air standard cycles, and the reductions in efficiency due to cooling are established; it is shown that these are small, unless large cooling flows are required. The theoretical estimates of efficiency reduction are compared with calculations, assuming that real gases form the working fluid in the gas turbine cycles. It is shown from air standard analysis that there are diminishing returns on efficiency as combustion temperature is increased; for real gases there appears to be a limit on this maximum temperature for maximum thermal efficiency**.

1. INTRODUCTION

1.1.Elements of simple gas turbine power plants

The simple gas turbine power plant mainly consists of a gas turbine coupled to a rotary type air compressor and a combustor or combustion chamber which is placed between the compressor and turbine in the fuel circuit. Auxiliaries, such as cooling fan, water pumps, etc. and the generator itself are also driven by the turbine. Other auxiliaries are starting device, lubrication system, duct system, etc. A modified plant may have in addition to the above, an inter-cooler, a regenerator and a reheater. The arrangement of a simple gas turbine power plant is shown in Fig.1.The gas turbine can be classified into two categories, i.e. impulse gas turbine and reaction gas turbine. If the entire pressure drop of the turbine occurs across the fixed blades, the design is impulse type, while if the drop is taken place in the moving blades, the fixed blades serving only as deflectors, the design is called reaction type. The advantage of the impulse design is that there is no pressure force tending to move the wheel in the axial direction and no special thrust balancing arrangement is required.

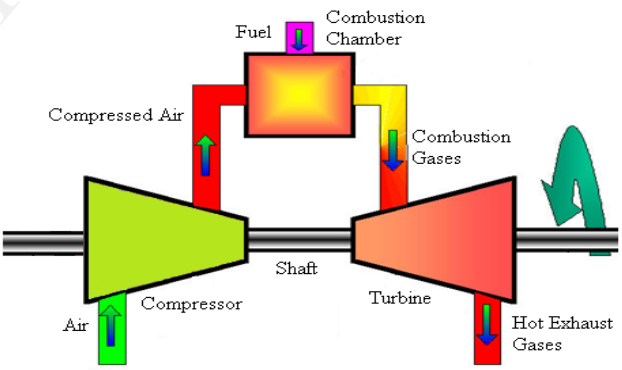

Fig.1. simple gas turbine power plant

Optimal combined cycle gas turbine power plants characterized by minimum specific annual cost values are here determined for wide ranges of market conditions as given by the relative weights of capital investment and operative costs, by means of a non-linear mathematical programming model , **E. Godoy et al., [49].,** have presented an analytical method for applicability evaluation of GTCCIAC with absorption chiller (inlet chilling) and saturated evaporative cooler (inlet fogging), **Cheng Yang et al., [51].**Have studied how to improve the capacity of the combined cycle (CC) power plant which has been operated for 8 years,**S. Boonnasaa et al., [54].,** have described a computational method of estimating the cooling flow requirements of blade rows in a high-temperature gas

turbine, for convective cooling alone and for convective plus film cooling,**Leonardo Torbidoni and J. H. Horlock, [55].**
Has presented the analyzed of gas turbine plant performance, including the effects of turbine cooling, are presented. The thermal efficiencies are determined theoretically, assuming air standard (a/s) cycles, and the reductions in efficiency due to cooling are established,**J. H. Horlock, [57].** Have investigated to identify and assess advanced improvements to the combined cycle such as gas turbine firing temperature, pressure ratio, combustion techniques, intercooling, enhanced blade cooling schemes and supercritical steam cycles that will lead to significant performance improvements in coal based power systems,. **Ashok D. Rao and David J. Francuz,[43].**

1.2. Methods of Cooling

Cooling of components can be achieved by air or liquid cooling. Liquid cooling seems to be more attractive because of high specific heat capacity and chances of evaporative cooling but there can be problem of leakage, corrosion, choking, etc. which works against this method [1]. On the other hand air cooling allows to discharge air into main flow without any problem. Quantity of air required for this purpose is 1-3% of main flow and blade temperature can be reduced by 200-300°C [1]. There are many types of cooling used in gas turbine blades; convection, film, transpiration cooling, cooling effusion, pin fin cooling etc. which fall under the categories of internal and external cooling. While all methods have their differences, they all work by usingcooler air (often bled from the compressor) to remove heat from the turbine blades [1].

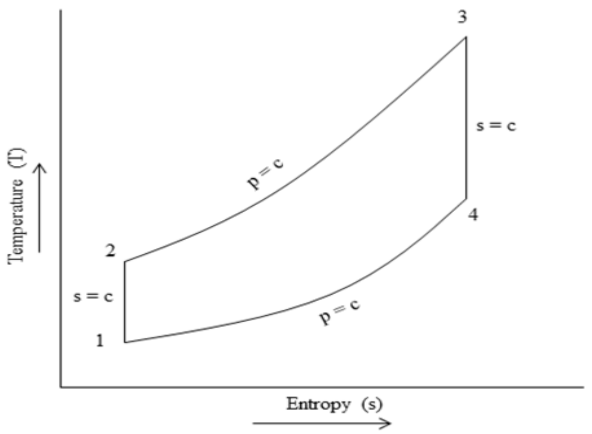

Fig.2.Temperature Entropy diagram of a closed cycle gas turbine

2. MATERIALS AND METHODS

2.1. Joule-Brayton cycle

The simple gas turbine power plant mainly consists of a gas turbine coupled to a rotary type air compressor and a combustor or combustion chamber which is placed between the compressor and turbine in the fuel circuit. Auxiliaries, such as cooling fan, water pumps, etc. and the generator itself are also driven by the turbine. Other auxiliaries are starting device, lubrication system, duct system, etc. A modified plant may have in addition to the above, an inter cooler, a regenerator and a reheater.The cooling of turbine blades also necessary when the blade undergoes at high temperatures. As Horlock, the limiting temperature above which cooling is required nearly 1123K. This temperature also depends upon the material of the blade. The combustion gases expand in the turbine before exhaust to the atmosphere.

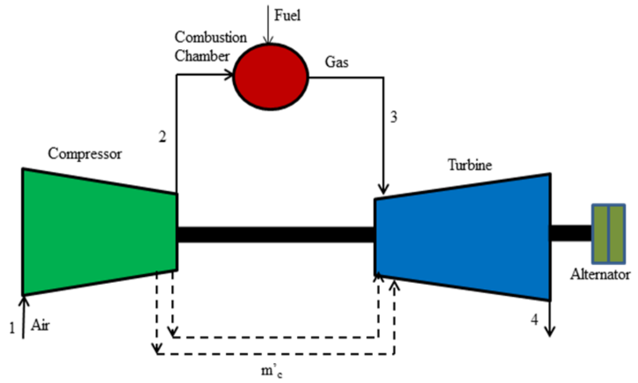

Fig.3.A schematic open cycle gas turbine with cooling of turbine blades

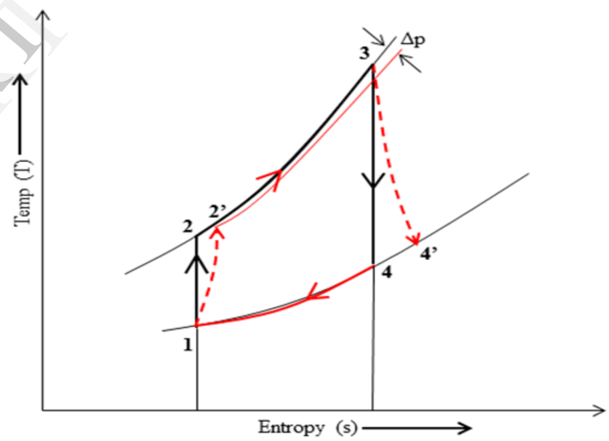

Fig.4.Actual gas turbine cycle representation on T-s chart

Fig.3. shows the schematic of an open cycle gas turbine with cooling of turbine blades with the compressed air and their T-s diagram for an actual Bryton cycle is shown in Fig.4. The Pressure loss in the combustion chamber is represented by Δp. In this cycle, the four processes occur to complete the cycle in the following sequence.
1-2 is isentropic compression.
1-2' is actual compression.
2-3 is the heat input in the combustor
3-4 is isentropic expansion.
3-4' is actual expansion.
4-1 is the exhaust process

2.2. Compressor Model

The compressor efficiency also known as isentropic compressor efficiency (η_c) is given by the following equation

The compressor efficiency

$$\eta_c = \frac{\text{isentropic compressor work}}{\text{actual compressor work}}$$

$$= \frac{W_c}{W_{ac}} \qquad (1)$$

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1} \qquad (2)$$

Or,

$$T_2' = T_1 + \frac{T_2 - T_1}{\eta_c} \qquad (3)$$

For isentropic compression process (1-2),

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma a - 1)}{\gamma a}} \qquad (4)$$

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma a - 1)}{\gamma a}} \qquad (5)$$

$$\gamma a = \frac{c_p}{c_v} \text{ For air} \qquad (6)$$

For actual compression process 1-2', the following equation is used to calculate thermal properties at the end of compression

$$\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma a - 1)}{\gamma a . \eta_c}} \qquad (7)$$

2.3. Gas Model

The thermodynamic properties of air and products of combustion are calculated by considering variation of specific heat and with no dissociation. Table containing the values of the specific heats against temperature variation have been published in many references such as Chappel and Cockshutt [11]. The curve fitting the data is used to calculate specific heats, specific heat ratio, and enthalpy of air and fuel separately from the given values of temperature. Mixture property is then obtained from properties of the individual component and fuel air ratio (FAR).

2.4. Combustion Chamber Model

For combustion process in combustion chamber process 2-3, the temperature in the combustor will be calculated by using following equations

$$T_3 = \frac{CV \times \eta_{comb} + (c_{pa} \times T_2') \times AFR}{c_{pg} \times AFR} \qquad (8)$$

3.5. Gas Turbine Model

The turbine efficiency is given by the following equation,

$$\eta_t = \frac{\text{Actual Turbine Work}}{\text{Isentropic Turbine Work}} = \frac{W_{at}}{W_t} \qquad (9)$$

$$\eta_t = \frac{T_3 - T_4'}{T_3 - T_4} (10)$$

For expansion process in turbine 3-4, the temperature and pressure at the end of expansion will be calculated by using the following equations:

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{(\gamma g - 1)}{\gamma g}} \qquad (11)$$

$$T_4' = T_3 - \eta_t (T_3 - T_4) (20)$$

Where $\gamma g = \frac{c_p}{c_v}$ for gas

For process 3-4', we have

$$\frac{T_3}{T_4'} = \left(\frac{p_3}{p_4}\right)^{\frac{(\gamma g - 1).\eta_t}{\gamma g}} \qquad (12)$$

Thermal efficiency of the cycle

$$\eta_{th} = \frac{c_{pg}[T_3 - T_4'] - c_{pa}[T_2' - T_3]}{(c_{pg} T_3 - c_{pa} T_2')} \qquad (13)$$

$$\text{Net power} = W_{net} \times \dot{m}_f \qquad (14)$$

2.5. Cooling Model

Horlock et al. cooling model have been used to calculate the mass of coolant required per kg of gas [55]

2.6. Transpiration air cooling

For an internally cooled turbine configuration, the ratio of coolant to main gas flow rates $(\dot{m}_{cl}/\dot{m}_g)$ required to cool the gas to T_b at the blade surface is proportional to the difference of enthalpy, which drives the heat transfer from gas to the blades to the ability of the coolant toabsorb heat, which is also termed as cooling factor R_c.

Thus

$$\frac{\dot{m}_{cl}}{\dot{m}_g} = \frac{\text{Heat transfer to blades}}{\text{Ability of coolant to absorb heat}} \propto \frac{h_{g,i} - h_{bl}}{h_{cl,e} - h_{cl,i}} \qquad (15)$$

The concept of heat exchanger effectiveness (ε) is introduced to account for the exit temperature of coolant

$$\varepsilon = \frac{T_{cl,e} - T_{cl,i}}{T_{bl} - T_{cl,i}} \qquad (16)$$

Thus, the cooling factor 'R_c' is expressed as

$$R_c = \frac{(T_{g,i} - T_{bl}).c_{p,g}}{\varepsilon.(T_{bl} - T_{cl,i}).c_{p,cl}} \qquad (17)$$

$$\text{or} \frac{\dot{m}_{cl}}{\dot{m}_g} = [\bar{S}_{tn}].\left[\frac{S_g}{t\cos\alpha}.F_{sa}\right].[R_c] \qquad (18)$$

As given by Sanjay et al. [63], eq. (33) shows that the cooling requirement in a blade row depends upon average Stanton number ($\bar{S}_{tn}$), turbine blade geometry $\left(\frac{S_g . F_{sa}}{t\cos\alpha}\right)$and cooling factor ($R_c$). In general $\bar{S}t_m$=0.005, $\frac{S_g}{t\cos\alpha}$=3.0 and if F_{sa}=1.04, so equation takes the form as

$$\frac{\dot{m}_{cl}}{\dot{m}_g} = 0.0156 R_c \qquad (19)$$

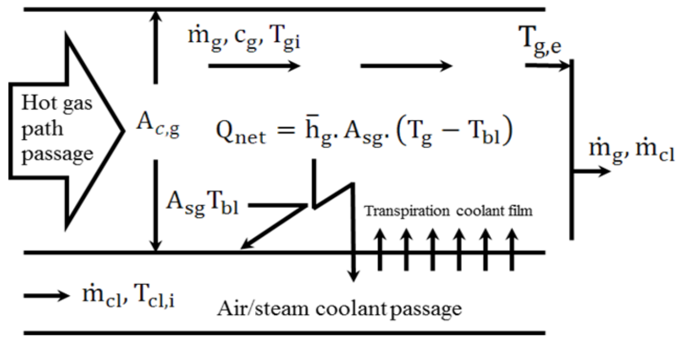

Fig.5. Model for transpiration cooling of turbine blade

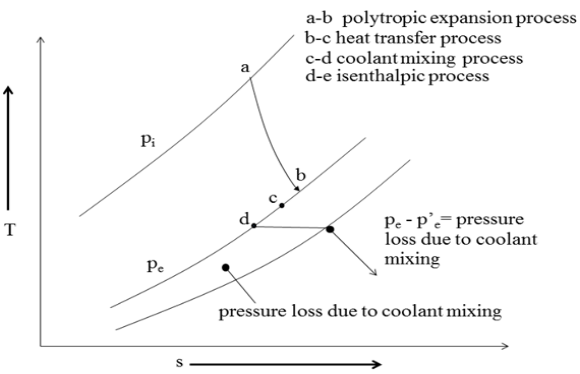

Fig.6. T-s representation of the expansion path in a single cooled

In the present work transpiration cooling includes air and steam as the cooling media has been taken. So in calculating the cooling requirement of air and steam, the value of specific heat of coolant ($c_{p,cl}$) will be taken accordingly in the expression of R_c equation (26) i.e. for air $c_{p,cl}$ will be $c_{p,a}$ and for steam $c_{p,cl}$ will be $c_{p,s}$. For air and steam transpiration cooling the value of $c_{p,cl}$ will be taken according to the coolant used. Thus the cooling requirement for transpiration cooling is

$$\frac{\dot{m}_{cl}}{\dot{m}_g} = \bar{S}t_{in} . \left[\frac{S_g}{t\cos\alpha} . F_{sa}\right] . [R_c]_{trans} = 0.0156[R_c]_{trans} \qquad (20)$$

The total pressure losses in mixing of coolant and mainstream are expressed as

$$\frac{\Delta p}{p} = 0.07 \frac{\dot{m}_{cl,i}}{\dot{m}_{g,i}} \qquad (21)$$

2.7. Analysis of the present work

The software developed in C++ has been used for the calculations of various parameters. Further a menu driven Origin 6.1 software used to plot the various graphs. The program developed on the basis of the modelling of different components of a gas turbine cycle..

3. Results and Discussion

The results are presented in the form of graphs as computed by the software developed in "C++" using various equations for individual components of a gas turbine cycle. The graphs plotted using menu driven software "Origin6.1". The data has been taken from the literature published and the cooling model has been taken from Horlock work. In the present analysis, the transpiration cooling technique has been taken for the cooling of turbine blade at high temperatures.

Table.1. Input values taken for calculation

S. No.	Symbol	Values
1	T1	288.15K
2	PA	1.01325 bar
3	R	287 kJ/kg
4	ETAT	0.93
5	ETAF	0.9
6	ETAC	0.9
7	ETAP	0.9
8	OPR	10-50
9	TIT	1400-2000 K
10	RG	0.296 kJ/kg
11	ETAM	0.99
12	TRS	25
13	DOR	0.5
14	C	0.03
15	ETACL	0.8
16	TBL	1122 K
17	CCL	2
18	MA	1 kg
19	ETAB	0.98
20	CV	42000 kJ/kgK

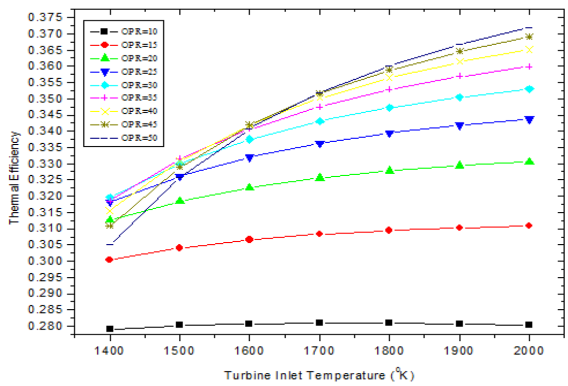

Fig.7. Variation of thermal efficiency with TIT for various OPR

Fig.7. shows the variation of thermal efficiency with turbine inlet temperature for various overall pressure ratios of the compressor when the turbine blade cooling is not been considered. The figure represents that the thermal efficiency is low at low overall pressure ratio for a given turbine inlet temperature. On the other hand, the efficiency increases on increasing the OPR. For high range of TIT (1800K and above), the increase in thermal efficiency as

OPR increase but in the low range of TIT, the result is not uniform for higher OPR.

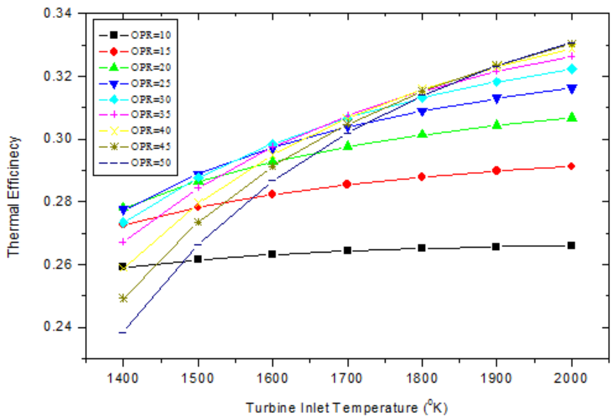

Fig.8. Variation of Thermal Efficiency with TIT (with blade cooling)

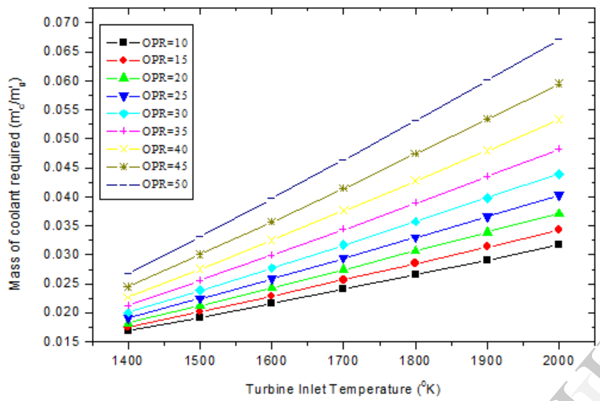

Fig.9.Variation of mass of coolant required with TIT for various OPR

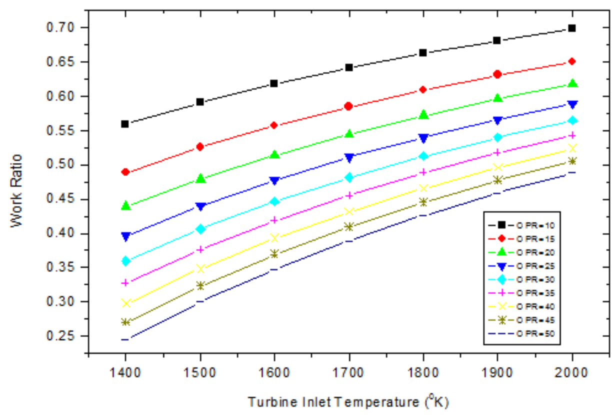

Fig.10.Variation of Work ratios with TIT for various OPR

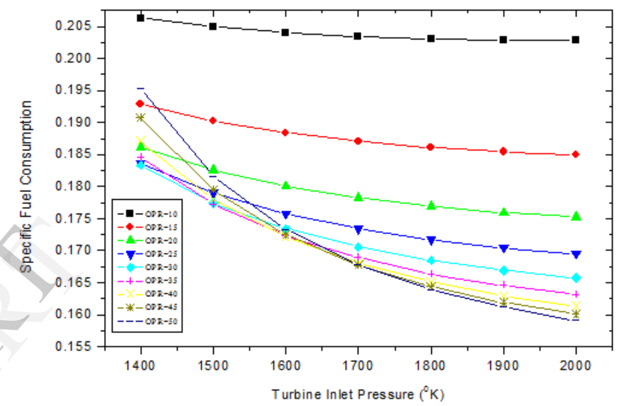

Fig.11. Variation of Specific Fuel Consumption with TIT for various OPR

Fig.8. represents the variations of thermal efficiency of the cycle with turbine inlet temperature when the turbine cooling has been taken into account. This figure clearly shows that the efficiency increase on increasing the overall pressure ratio for a given turbine inlet temperature. While the efficiency increases with the turbine inlet temperature but this increase is low at lower overall pressure ratio but at higher OPR, it increases abruptly. Also it is clear from the graph that the efficiency is quite low at a lower TIT even at higher OPR.

Fig.9. shows the variation of mass of coolant required with TIT for various OPR. It is clear from the graph that the mass of coolant required per kg of gas is low at lower values of TIT for a given overall pressure ratio. The mass of coolant per kg of gas is increased on increasing the OPR for a given value of turbine inlet temperature. The mass of coolant per kg of gas is mostly depends upon the inlet temperature of the turbine. If it works at high TIT, the mass of coolant required is more compared to low TIT

Fig.10. shows the variation of work ratios with turbine inlet temperature (TIT) for various overall pressure ratios (OPR). Work ratio increases on increasing the TIT for a given value of OPR. But the values of work ratio are higher for lower values of OPR for a given TIT.Fig.11. represents the variation of specific fuel consumption with turbine inlet temperature for various overall pressure ratios (OPR). At low pressure ratio, the fuel consumption is very high as compared to the higher values of overall pressure ratio. This value is decreases as turbine inlet temperature increases.

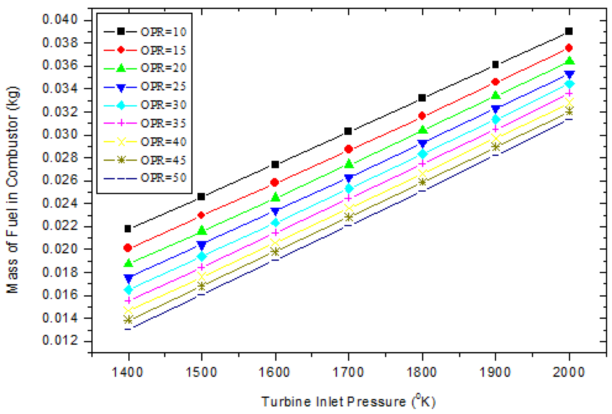

Fig.12. Variation of Mass of fuel in combustor with TIT for various OPR

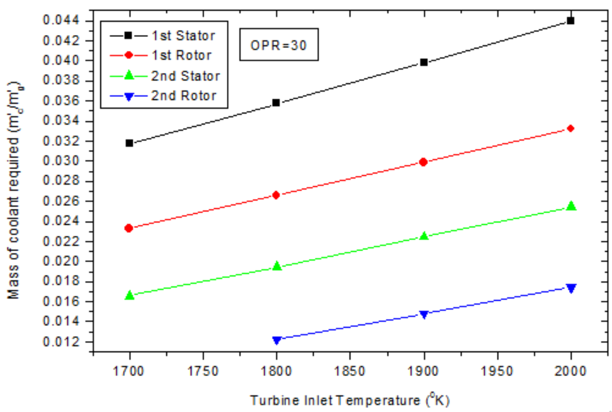

Fig.13. Variation of Mass of coolant required with TIT for first 2 stages

Fig.12.shows the variation of mass of fuel required in combustor with turbine inlet temperature (TIT) for different overall pressure ratios (OPR). As the TIT increases, the mass of fuel required in the combustion chamber is increases for a given value of OPR. While on increasing the OPR, the mass of fuel required in the combustion chamber is decreases for a fixed value of turbine inlet temperature. This is due to fact that, on increasing the OPR, the temperature of air at the exit of compressor increases therefore it requires less fuel to reach the desired temperature of turbine at inlet.

Fig.13. represents the variation of Mass of coolant required with turbine inlet temperature for a fixed value of overall pressure ratio (30 bar). The first 2 stages of rotor and stator have been considered. It is clearly reflects from the graph that the first stage required more mass of coolant due to high temperature of the blade while for the later stages the coolant required is low.

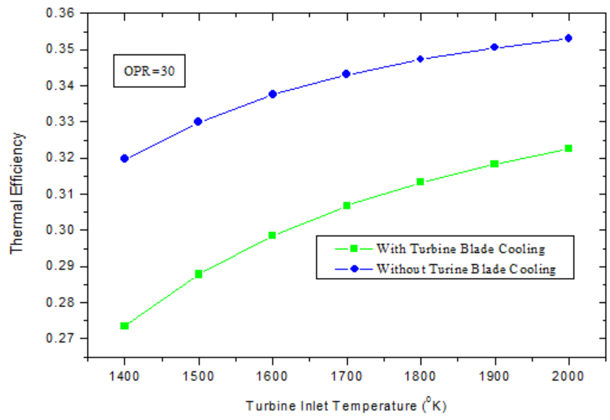

Fig.14. Variation of Thermal Efficiency with TIT

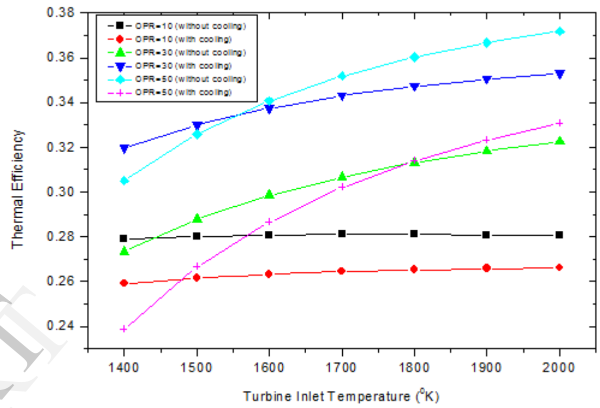

Fig.15. Variation of Thermal Efficiency with TIT for various OPR

Fig.14. represents the variation of thermal efficiency with turbine inlet temperature for a fixed vale of overall pressure ratio. The thermal efficiency increases on increasing the values of turbine inlet temperature. This graph has been plotted for two different cases. One, for the thermal efficiency calculation, when there is no cooling of the turbine blades while the other one is with the turbine blade cooling. It is obvious that the efficiency is high for the cycle with turbine blade cooling. This is due to fact that the air bleed from the compressor for the turbine blade cooling decreases the exit temperature of the compressor and at the same time the turbine inlet temperature decreases as soon as the cooling came into effect. Therefore the overall efficiency of the cycle has been decrease for the cycle with turbine blade cooling.

Fig.15. shows the variation of thermal efficiency with turbine inlet temperature for various overall pressure ratios (OPR). This graph has been plotted for three different pressure ratios to calculate the thermal efficiency of the cycle with and without the turbine blade cooling.

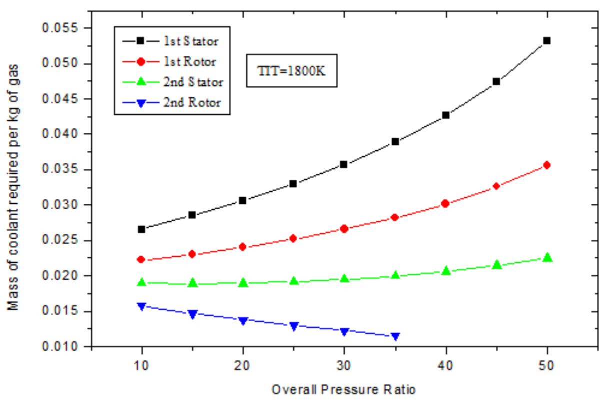

Fig.16. Variation of Mass of coolant required with OPR for first 2 stages

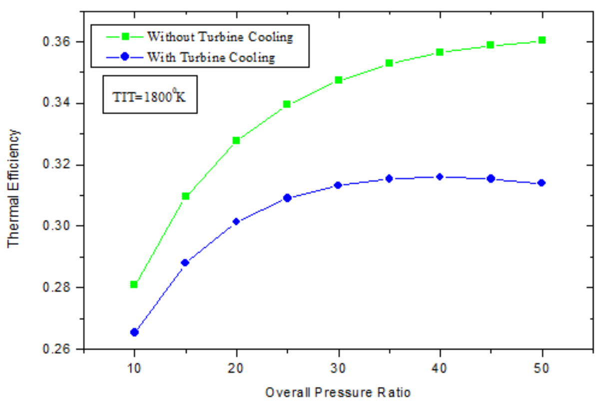

Fig.17. Variation of Thermal Efficiency with OPR

Fig.16. shows the variation of mass of coolant required for the cooling of turbine blade with overall pressure ratio (OPR) for first 2 stages of turbine at a given turbine inlet temperature (1800 K). The graph shows that, for the first stage of turbine, the mass of coolant required is increase continuously on increasing the OPR while in the later stages of turbine, the mass of coolant required is decreases on increasing the OPR. Fig.17. shows the variation of thermal efficiency with overall pressure ratios of the compressor (OPR). The variations of efficiencies are plotted for the cycle with cooling and without cooling of turbine blades. It is found that the cycle having turbine blade cooling is less efficient than the cycle having no turbine blade cooling. It is also found that the efficiency decreases after a certain pressure rise in the compressor. That means the efficiency increases upto a certain extent for a cycle with turbine blade cooling. An optimum value of thermal efficiency hasbeen obtained at an overall pressure ratio 30 and beyond that it falls gradually

4. CONCLUSIONS

It has been concluded from air standard Braytoncycle analysis that

- Plant thermal efficiency drops relatively little due to turbine cooling.
- The thermal efficiency of the cooled turbine plant, when expressed as a function of the rotor inlet temperature**,** is virtually identical to the efficiency of the uncooled plant when expressed as a function of combustion temperature.
- The difference between uncooled and cooled thermal efficiency decreases at high turbine inlet temperature.
- The rate of increase of thermal efficiency of the cooled cycle falls with increasing turbine inlet temperature,but there is no prediction of a maximum efficiency being attained at high TIT.

Nomenclature

1, 2, 3… are the state points
T_1 = ambient temperature
P_a = ambient pressure
R = gas constant
ETAT = turbine efficiency
ETAF = compressor efficiency
OPR = over all pressure ratio
TIT = turbine inlet temperature
ETAM = mechanical efficiency
TRS = no. of stages
CV = calorific value

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