THERMAL ANALYSIS OF COOLING EFFECT ON GAS TURBINE BLADE

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Abstract

Performance of a gas turbine is mainly depends on various parameters e.g. ambient temperature, compressor pressure ratio, turbine inlet temperature etc. The most important parameter to increase the life of the turbine blade is the cooling of the blade, which is necessary after reaching a certain temperature of the gases passing through the blades. Various types of cooling models are available for a turbine blade cooling. The power output of a gas turbine depends on the mass flow rate through it. This is precisely the reason why on hot days, when air is less dense, power output falls off. This paper is to analyze the film cooling technique that was developed to cool gases in the initial stages of the turbine blades, where temperature is very high (>1122 K). It is found that the thermal efficiency of a cooled gas turbine is less as compare to the uncooled gas turbine for the same input conditions. The reason is that the temperature at the inlet of the turbine is decreased due to cooling and the work produced by the turbine is slightly decreased. It is also found that the power consumption of the cool inlet air is of considerable concern since it decreases the net power output of gas turbine. In addition, net power decreases on increasing the overall pressure ratio. Furthermore, the reviewed works revealed that the efficiency of the cooled gas turbine largely depends on the inlet temperature of the turbine and previous research said that the temperature above 1123K, require cooling of the blade.

Keywords: *Gas turbine, Turbine blade cooling, film cooling technique, Thermal Efficiency*

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1. INTRODUCTION

In a bid to remain at the forefront of technological development as well as a technical expert to United States industry, NASA identified the need for an improved design process within the civilian aero engine industry, in hopes of improving their market share, reducing time to market, and minimizing research costs [2]. Areas of interest included, but were not limited to, high temperature materials, advancing turbine analysis techniques, and improving the overall engine design and analysis process. The latter interest called for the impact assessment of engine component technologies from the micro to system levels [2]. A good example of the need for this type of analysis comes from determining the required service life of a turbine blade, which is limited by the exit temperature from the combustor and the material properties that in turn, limits the performance of the gas turbine. Ideally, the engine would operate at a high enough temperature to achieve the highest possible thrust rating [3], while at the same time maintaining an economic service life. Currently, to address the issue of exit temperature, modern turbines utilize a cooled turbine blade to improve the possible rotor inlet temperature, and this necessary cooling flow has a strong impact on the turbine efficiency [4, 5]. By improving cooling technology for a gas turbine blade it is possible to increase the combustor exit temperature sufficiently, therefore achieving good improvement in turbine efficiency and thrust [6]. However, this improvement does not prove viable when considering the complete process. The increase in cooling flow to the turbine blades and vanes takes bleed air away from the compressor, reducing its on efficiency. This detrimentally affects the efficiency of the whole system, such that the improvements in the turbine are eclipsed. Being able to track all these whilst looking at a particular component within an aircraft gas turbine, is obviously very desirable, especially at the early stages of a design. A successful design process incorporates flexibility and freedom at the early conceptual stages and continuing as far into the design as possible, saving time and money in fixing problems that would have arisen had an investigation not taken place. Looking into technologies to improve an engine, one needs to provide useful benchmarks from which comparisons can be made. If a superior product is going to be produced, analysis of the areas affected by the new product needs to be considered. If this is not the case a lot of money could be invested in designs that in the end prove to be impractical. A new technique has been reviewed up to date that was developed to cool inlet air to gas turbine [9]. The techniques including the mechanical chillers, media type evaporative coolers and absorption chillers have been reviewed. It is found that the power consumption of the cool inlet air is of considerable concern since it decreases the net power output of gas turbine. Experimental tests to investigate the film cooling performance of converging slot hole (console) rows on the turbine blade have been performed [10]. Film cooling effectiveness of each single hole row is measured under three momentum flux ratios based on the wide-band liquid crystal technique. Measurements of the cooling effectiveness with all the hole rows open are also carried out under two coolant–mainstream flux ratios. Film cooling effectiveness of cylindrical hole rows on the same blade model is measured as a comparison. The thermodynamic performance of MS9001 gas turbine based cogeneration cycle having a two-pressure heat recovery steam generator (HRSG) for different blade cooling means have compared. Internal convection cooling technique, film cooling and transpiration cooling techniques, employing steam or air as coolants, are considered for the performance evaluation of the cycle[11]. The analysis is useful for power plant designers to select the optimum compressor pressure ratio, turbine inlet temperature, fuel utilization efficiency, power-to-heat ratio, and appropriate cooling means for a specified value of plant specific work and process heating requirement. The gas turbine plant performance and the effects of turbine cooling have been presented [8]. The thermal efficiencies are determined theoretically, assuming air standard (a/s) 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.

2. METHODOLOGY

2.1 Open Gas Turbine Cycle

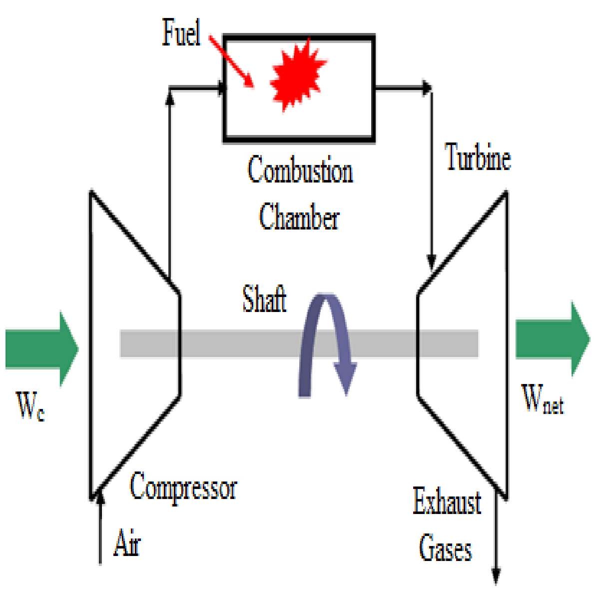

Fig. 1: Schematic for an open gas-turbine cycle

Fresh air enters the compressor at ambient temperature where its pressure and temperature are increased. The high pressure air enters the combustion chamber where the fuel is burned at constant pressure. The high temperature (and pressure) gas enters the turbine where it expands to ambient pressure and produces work.

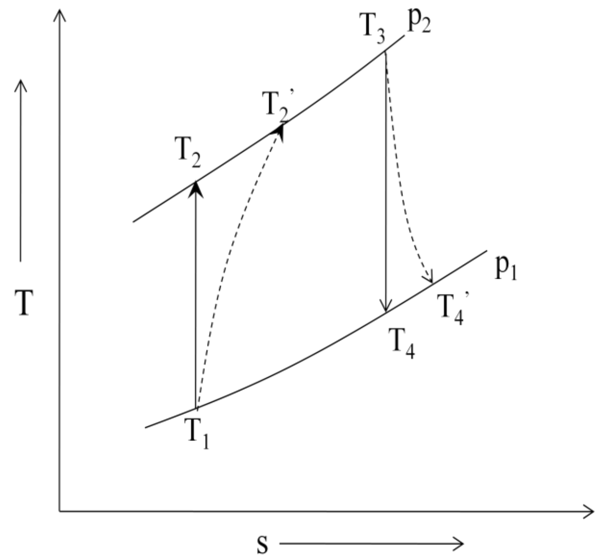

Fig 2 T-s representation of a gas turbine engine

2.2 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. Following equations are used to calculate the specific heats of air and gas as a function of temperatures.

For $T_a \leq 800$ the value of cpa will be calculated from the following equation.

$$c_{pa} = (1.0189 \times 10^3) - (0.13784 \times T_a) + (1.9843 \times$$
$$10-4\times Ta2+4.2399\times10-7\times Ta3-3.7632\times10-10\times Ta4 \quad (1)$$

In the above equations, T stands for gas or air temperature in deg K and $t = \frac{T}{100}$

Compressors are subjected to various aerodynamic losses which are accounted here by introducing the concept of polytropic efficiency. For a given polytropic efficiency, pressure ratio and bleed point pressure, the temperature and other variables for each stream are calculated. The mass and energy balances yield the compressor work, as given below

Mass balance:

$$\dot{m}_{a,i} = \dot{m}_{a,e} + \sum \dot{m}_{cl} \quad (2)$$

Energy balance:

$$W_c = \dot{m}_{a,e} \,.\, h_{a,e} + \sum \dot{m}_{cl} \,.\, h_{cl} - \dot{m}_{a,i} . h_{a,i} \qquad (3)$$

Where determination of $\dot{m}_{cl}$ (mass of the cooling air) extracted from the compressor, is given after turbine cooling flow calculation.

$$p_2 = p_1 \left(\frac{p_2}{p_1}\right) \qquad (4)$$

$$T_2 - T_1 = \frac{T_1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right] \qquad (5)$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma_a - 1)}{\eta_{pc} \cdot \gamma_a}} \qquad (6)$$

Where $\frac{p_2}{p_1}$ is the overall pressure ratio of compressor and η_c and η_{pc} are isentropic and polytropic efficiencies of the compressor, respectively.

2.3 Combustion Chamber

The purpose of the combustion system of a gas turbine engine is to increase the thermal energy of a flowing gas stream by combustion which is an exothermic chemical reaction between the hydrocarbon fuel and oxygen in the air-stream. Inflowing air is diffused at the entrance of the burner. Kerosene is the standard aviation fuel but the highly distilled forms of diesel used make little difference in performance compared to kerosene and hence diesel is used here for the analysis. The mass and energy balances for combustor, with suffixes i and e denoting its inlet and outlet sections, are given by

Mass balance:

$$\dot{m}_g = \dot{m}_{a,i} + \dot{m}_{f,cc} \qquad (7)$$

Energy balance:

$$\eta_{cc} . \dot{m}_{f,cc} . LCV_f = \dot{m}_{g,e} . h_{g,e} - \dot{m}_{a,i} . h_{a,i} \qquad (8)$$

The pressure at combustor exit is given by (Δp_{cc} =2% pressure drop has been assumed in the combustor)

$$p_{cc,e} = p_{\,cc,i} - \Delta p_{cc} \qquad (9)$$

The fuel to air ratio (FAR) is calculated as,

$$\frac{\dot{m}_{f,cc}}{\dot{m}_a} = \frac{c_{pg} . T_e - c_{pa} . T_i}{\eta_{cc} . LCV_f - c_{pg} . T_e} \qquad (10)$$

In this equation T_e is the turbine inlet temperature, T_i is exit temperature of compressor, η_{cc} is the combustion efficiency of the main combustion chamber, normally taken between 0.98 to 0.99 and LCV_f is the lower calorific value of the fuel taken as 42000 kJ/kgK assuming fuel as diesel. Values of specific heat of air and gases are to be calculated from the gas model.

2.4 Cooled Gas Turbine

Turbine produced power to drive engine compressor due to expansion of gas stream apart from the excess work developed at the shaft of the turbine. A general model that represents the gas turbine with turbine blade cooling has been developed. The model is intended for use in cycle analysis applications. Aerodynamic losses and hence the inefficiency of gas turbine is taken care by considering polytropic efficiency of turbine.

$$\frac{T_i}{T_e} = \left(\frac{p_i}{p_e}\right)^{\frac{\eta_{pt} \cdot (\gamma_t - 1)}{\gamma_t}} \qquad (11)$$

Pressure ratio, the temperature and other variables for each stream are calculated for a given polytropic efficiency. W_t is the work developed by turbine, and η_m is the mechanical efficiency which is used to account for the losses due to windage, bearing friction, and seal drag and is defined as

$$\eta_m = \frac{\text{mechanical power output}}{\text{mechanical power input}}$$

Mass and energy balances yield the turbine work, as given below.

Mass balance:

$$\dot{m}_{g,i} = \dot{m}_{g,e} + \sum \dot{m}_{cl} \qquad (12)$$

Where

$$\dot{m}_{g,i} = \dot{m}_{a,i} + \dot{m}_{f,cc} \qquad (13)$$

and $\sum \dot{m}_{cl}$ represents the sum of air coolant flow rates to the stage (stator and rotor).

Energy balance:

$$\dot{m}_{g,i} . h_{g,i} + \dot{m}_{cl,i} . h_{cl,i} = \dot{m}_{g,e} . h_{g,e} + W_T \qquad (14)$$

Mass of coolant required per kg of gas flow is given by following equations as explained in [8].

$$\frac{\dot{m}_{cl}}{\dot{m}_g} = 0.0156 . [R_c]_{trans} \qquad (15)$$

$$(\eta_{iso})_{film} = 0.4$$

Where

$$(R_c)_{trans} = \frac{(T_{g,i}-T_{bl}).\, c_{p,g}.\, \left(1-(\eta_{iso})_{film}\right)}{\varepsilon(T_{bl}-T_{cl,i}).\, c_{p,cl}} \qquad (16)$$

Work Ratio: Work ratio is also very important parameter for the selection of a gas turbine cycle. The work ratio is given by the following equation:

$$WR = \frac{W_{net}}{W_t} \qquad (17)$$

Specific Fuel Consumption: Specific fuel consumption is defined as the fuel consumed by the combustor per unit net work of the cycle; If the mass of fuel consumed is given in kg/s and the net work developed is in kW then the specific fuel consumption will be calculated in kg/kWhr.

$$SFC = \frac{3600 \times m_f}{W_{net}} \text{ (kg/kWhr)} \qquad (18)$$

Net Power: Net power available is calculated by the following equation.

$$\text{Net Power} = m_a \times w_{net} \times \eta_{gen} \qquad (19)$$

If mass of air flow is given in kg/s and net power is given in kJ/kg then the net power will be calculated in kW.

3. RESULTS

The results are based on the software developed in C++ and afterwards graphs have plotted with the help of menu driven software "Origin 50".

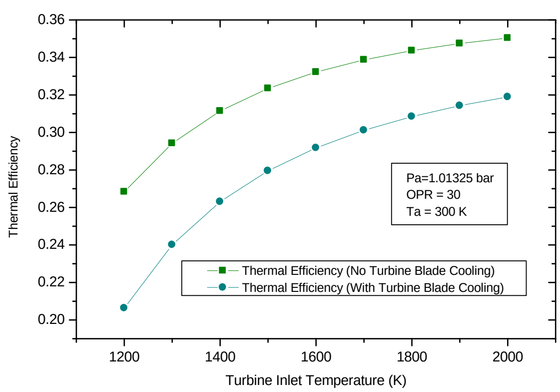

Fig 3 Variations of Thermal Efficiency with TIT

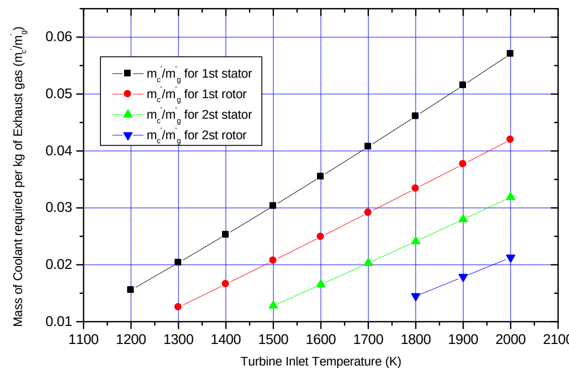

Fig 4 Variations of m'c / m'g with TIT

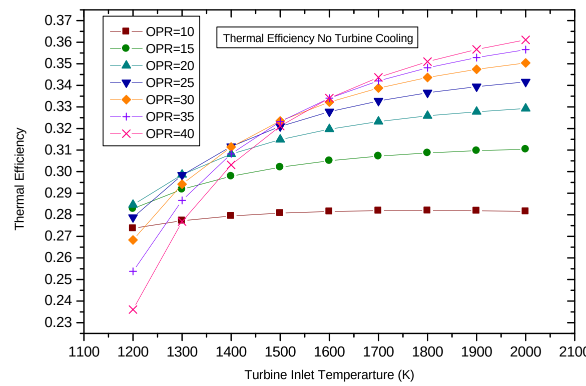

Fig 5 Variations of Thermal Efficiency with TIT

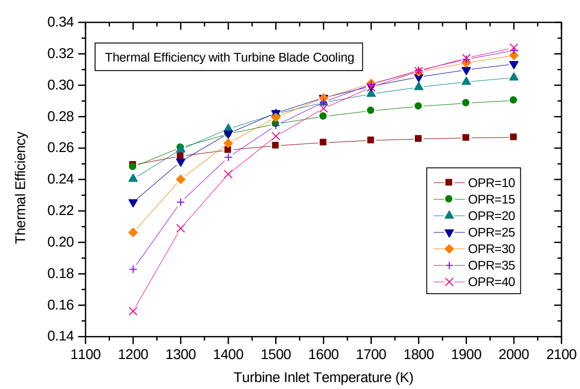

Fig 6 Variations of Thermal Efficiency with TIT

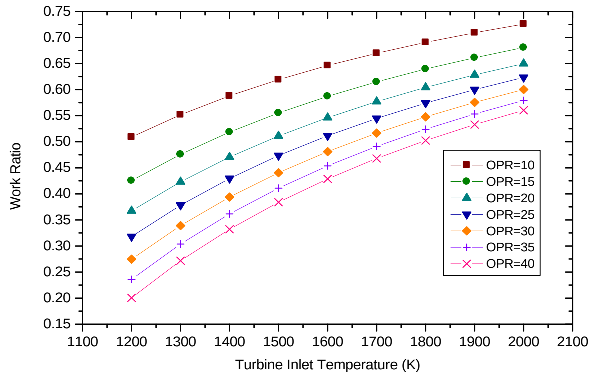

Fig 7 Variations of Work Ratio with TIT

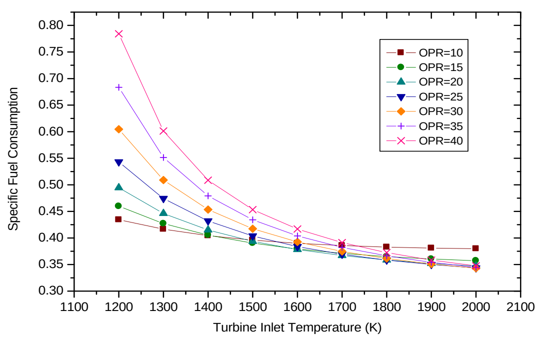

Fig 8 Variations of Specific Fuel Consumption with TIT

The variations of the graphs are based on the input values taken and the thermodynamic laws governed those equations. The parametric analysis of the gas turbine cycle of the present work, film cooling has been taken for the cooling of the turbine blade to increase the life and better performance.

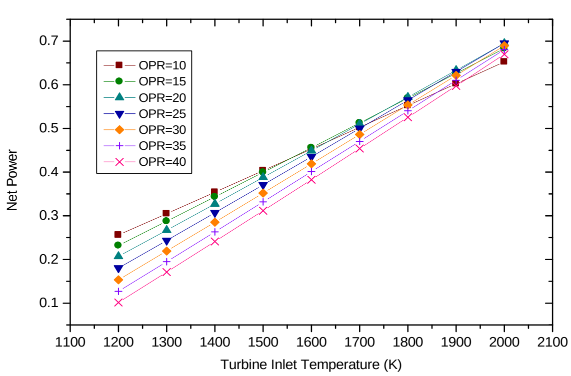

Fig 9 Variations of Net Power with TIT

Figure 3 shows the variations of the thermal efficiency for various turbine inlet temperature (TIT) at a given values of atmospheric pressure, atmospheric temperature and overall pressure ratio (OPR). It seems from the graph that the efficiency increases with the increase in TIT for both type of cycles i.e. for cooled turbine blade and uncooled turbine blade. It has been observed that the thermal efficiency of the uncooled turbine blade is slightly better that the one with cooled turbine blade. It is due to fact that the temperature at the inlet of the turbine is decreases. Figure 4 shows the variations of m'c / m'g with TIT. The mass of coolant required is calculated as the fraction of the exhaust gases. It has been observed from the figure that the mass of coolant required is limited up to first stator only at low turbine inlet temperature but as long as the temperature increases, the cooling required is in later stages as well. As shown in the figure that for TIT 1800 and above, the cooling is must for at least two stages. That is two stators and two rotors.

Figure 5 represents the variations of Thermal Efficiency with TIT for an uncooled turbine blade for various overall pressure ratios. The thermal efficiency increases on increasing the OPR for higher range of TIT. At lower TIT, the thermal efficiency is high for low OPR and it has little variation for the entire range of TIT. Figure 6 represents the variations of Thermal Efficiency with TIT for a cooled turbine blade for various overall pressure ratios. Almost same variations have been found in this graph also. Figure 7 shows the variations of Work Ratio with TIT for various overall pressure ratios. For a given value of OPR, the work ratio increases on increasing the TIT. At a given TIT, the work ratio decreases on increasing the OPR. Figure 8 shows the variations of Specific Fuel Consumption (SFC) with TIT for various Overall pressure ratios. SFC decreases on increasing the TIT for a given OPR. At low OPR, this change is in a limited range but at higher OPR it varies in wide range. Figure 9 shows the variations of Net Power with TIT. Net power increases on increasing the value of TIT. It has been observed that the net power decreases on increasing the OPR for a given value of TIT. Figure 10 represents the variations of Thermal Efficiency for an uncooled turbine with TIT for various ambient temperatures. Thermal efficiency increases on increasing the TIT for a given value of ambient temperature. At low ambient temperature the thermal efficiency is higher compared to high ambient temperature. Figure 11 shows the variations of Thermal Efficiency of a cooled gas turbine with TIT for various ambient temperatures. It also has the same effect as the uncooled turbine blade does. The difference in that, the uncooled gas turbine has more efficiency that a cooled gas turbine blade for a given ambient temperature.

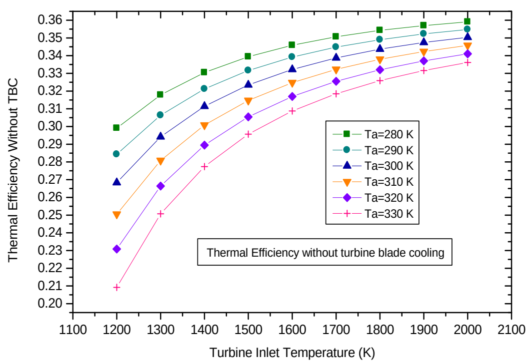

Fig 10 Variations of Thermal Efficiency uncooled turbine with TIT

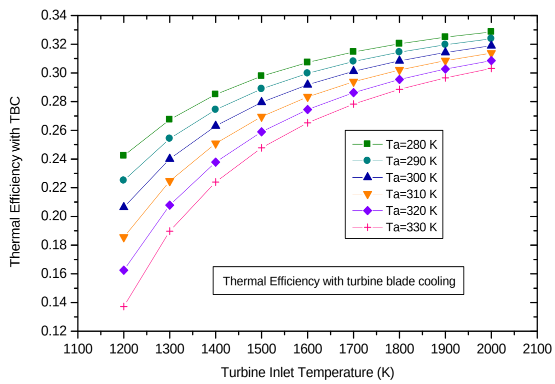

Fig 11 Variations of Thermal Efficiency of cooled Turbine with TIT

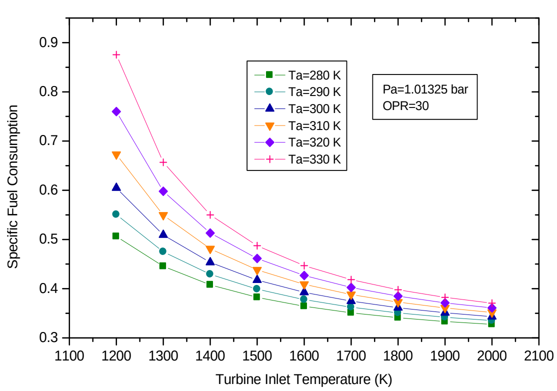

Fig 12 Variations of SFC with TIT

Figure 12 shows the variations of SFC with TIT for various ambient temperatures. SFC decrease continuously on increasing the TIT. It decreases steeply in the low range of TIT but at the higher range of TIT, this variation is slightly low. It also reflects that the maximum value of SFC is for high ambient temperature. Figure 13 represents the variations of mass of coolant required per kg of gas (m'c/m'g) with ambient temperatures. The mass of cooling air required is increases on increasing the ambient temperature. In the first stage of turbine, the mass of coolant required is increases continuously with the ambient temperature but in the second stage of turbine the variation of coolant required is almost constant with the ambient temperatures.

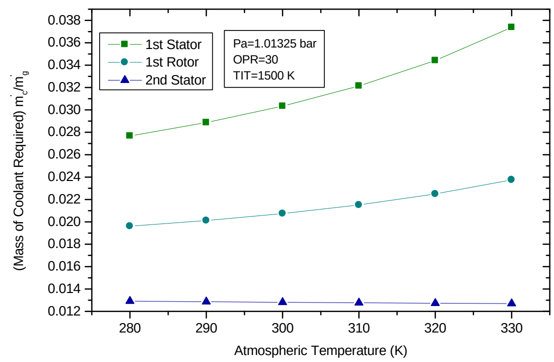

Fig 13 Variations of (m'c/m'g) with Ambient Temperatures

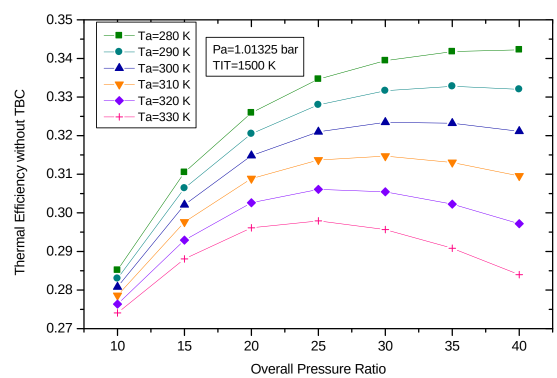

Fig 14 Variations of Thermal Efficiency of uncooled turbine with OPR

Figure 14 represents the variations of thermal efficiency of an uncooled turbine with OPR for various ambient temperatures. It has been observed that, the thermal efficiency first increases and then decreases on increasing the OPR for a given value of ambient temperature. The thermal efficiency is higher for low ambient temperature and also the thermal efficiency for this range of ambient temperature is continuously increases on increasing the OPR.

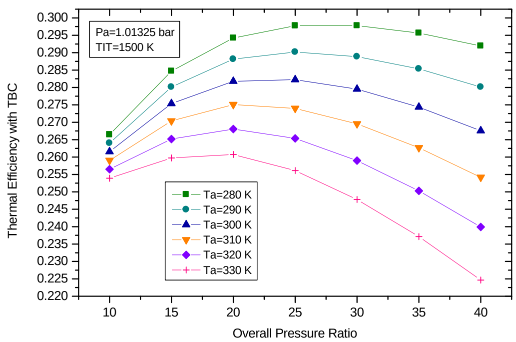

Fig 15 Variations of Thermal Efficiency of cooled blade with OPR

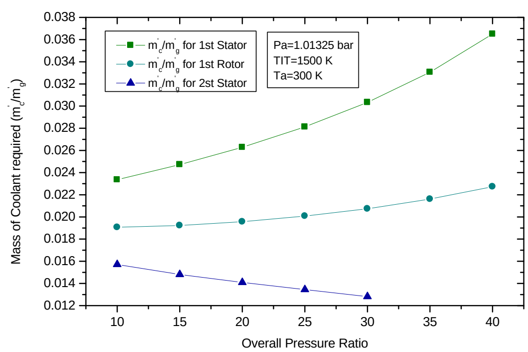

Fig 16 Variations of (m’c/m’g) with OPR

Figure 15 shows the variations of thermal efficiency of a cooled blade gas turbine with OPR for various ambient temperatures. In this case, the natures of the curves are almost similar to the previous graph. The difference is that, efficiency is decreases for the entire range of the OPR after a slight increase at the low range of OPR. Figure 16 shows the variations of mass of coolant required (m’c/m’g) with OPR for a given value of ambient temperature and turbine inlet temperature. In the first stage of turbine, the mass of coolant required is increases on increasing the OPR but for the second stage the mass of coolant is required are decreases.

4. CONCLUSIONS

The gas turbine power plant has been analyzed for various parameters. The most important parameter which has been covered in this work is the new method of calculating the coolant flow requirements for gas turbine blades especially at higher temperatures. There are various types of cooling method approach have already been published in various literatures. In the present wok the film cooling has been taken for the calculation of the blade cooling. It describes how the cooling flow in each chordwise strip may be calculated, assuming a particular internal cooling geometry. Comparisons with a simplified method of calculation, assuming constant blade temperature along the span, indicate broad agreement. The new method presented should then give more reliable evaluations than the semi-empirical methods, particularly for the study of innovative cycles, where conventional fluids and operating conditions cannot be assumed in the performance calculations. It is hoped that such full development of the code will provide a useful design tool for the turbine designer faced with the necessity of blade cooling.

Table 1: Input parameters for calculation

S. No.	Parameter	Value
1	Atm. Press.	1.01325 bar
2	Turbine Efficiency	90%
3	Compressor Efficiency	90%
4	Mechanical Efficiency	99%
5	Mass of air flow	1 kg/s
6	Turbine Inlet Temp	1200-2000K
7	Overall Pressure Ratio	10-40

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BIOGRAPHIES

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