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PERFORMANCE PREDICTION OF GAS TURBINE

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ABSTRACT: *In this paper, the work is to demonstrate the performance of single stage gas turbine which is able to operate with hot air as a working fluid at specific boundary conditions. These boundary conditions were chosen to be compatible with a Brayton cycle. The evaluation is dependent on the turbine efficiency and output power. Firstly, preliminary design work was completed in order to figure out the turbines shapes and find initial information about the impact of various factors on their efficiency values and output powers. Factors considered were: inlet pressure, inlet temperature, rotational speed and the mass flow rate. The performance was predicted at a constant pressure of 3 bars by varying inlet temperature (400K & 600 K) at different rotational speeds. Subsequently, three-dimensional computational fluid dynamics (CFD) modelling was completed for a gas turbine to study the effect of other factors and have accurate results. The simulation results showed that an improvement in total turbine efficiency from 45 to 66% for a fixed cycle boundary conditions.*

Keywords: CFD, Gas Turbine, Rotational Speed, Turbine Efficiency.

1. INTRODUCTION

Medium-scale gas turbine (MSGT), which can operate at low mass flow rates, relatively low-pressure ratio and moderately high temperatures, was the driving force for investigating the Medium-scale gas Turbine (MSGT). A gas turbine function is to produce mechanical power to drive a pump, compressor or an electric generator etc. within the gas turbine, fuel chemical energy is converted into heat energy and is used to producing mechanical energy. Air is served as a working fluid for the engine which is compressed in the compressor, used in combustion in the combustor and resulting combustion gasses are fed into the expander for the production of mechanical energy. Gas turbines produce high-quality heat that can be used for industrial or district heating steam requirements. Gas turbine systems operate on the Brayton cycle. In a Brayton cycle, atmospheric air is compressed, heated, and then expanded, with the excess of power produced by the turbine over that consumed by the compressor used for power generation. The power produced by an expansion turbine and consumed by a compressor is proportional to the absolute temperature of the gas passing through the device. Consequently, it is advantageous to operate the expansion turbine at the highest practical temperature consistent with economic materials and internal blade cooling technology and to operate the compressor with inlet air flow at as low a temperature as possible. As technology advances permit higher turbine inlet temperature, the optimum pressure ratio also increases.

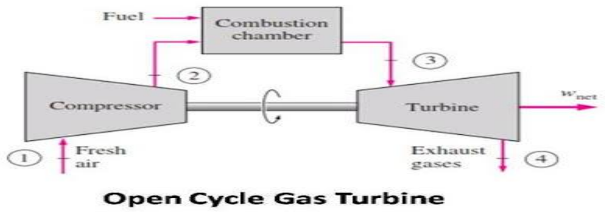

Figure 1 BRAYTON CYCLE

There are varying opinions about what characterizes a medium-scale gas turbine, however, the significance of the power output is commonly agreed upon. Several studies investigated separately different components of the cycle: such as the thermal cavity receiver of a medium-scale solar Brayton cycle [1]; the effect of some boundary conditions on the overall cycle efficiency [2]; and the optimum performance of the cycle [3]. However, they neglected the turbines' performance.

2. 3D GEOMETRY MODELING AND MESH GENERATION

The three-dimensional blade generations ability which ANSYS 17 has enables the users to generate three-dimensional Turbo Grid models for the rotor of the gas turbines. When the Preliminary Design was performed, the blade geometry and dimensions for rotor were exported to the detailed blade design module in ANSYS 17 called Blade-Gen to construct the blade geometry of turbine stage. CFX Turbo-Grid was used to mesh the fluid domain. As it is well known, the Discretization of the domain has a direct effect on the quality of the solution in terms of accuracy and computational costs. The structured 3D mesh generation for blade passage used in the simulations is shown in Fig.2&3. Also the Fig. 2shows a section of turbine stage consists of stator and rotor in which stator has single blade and rotor has two blades in a flow passage for a particular section.

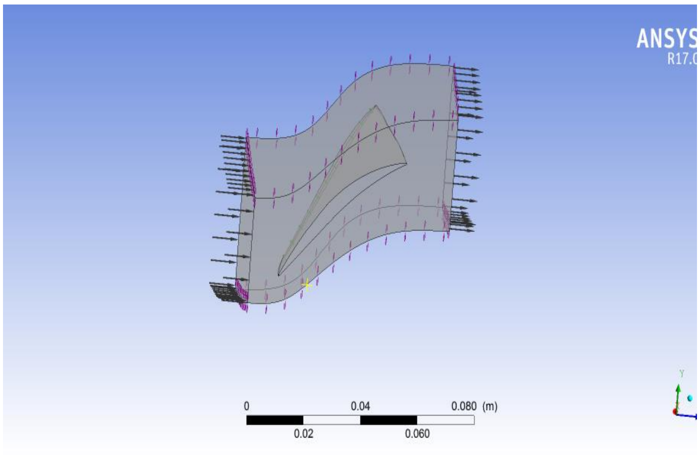

Figure 2 DESIGN OF TURBINE STAGE

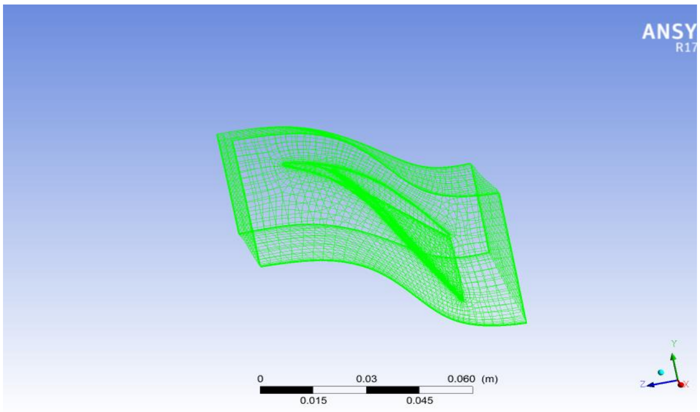

Figure 3 TURBOGRID MESH

Table 1. Mesh Information for AXIAL TURBINE

Domain	Nodes	Elements
R1	58095	52784

In a mesh, grid size for the gas turbine section of 58095 nodes was used with a refined mesh near the blade wall. It is worth noting that in the zone near to the blade surface and walls, the grid was refined to maintain a good compromise between computational costs and solution accuracy. The k-ɷ based on SST turbulence model was implemented to produce a highly accurate prediction by the inclusion of transport effects in terms of flow separation prediction into the formulation of the turbulent viscosity (eddy-viscosity). To account the wall effects in the simulation, an automatic wall treatment was applied, which allows smooth shift between wall functions formulation and low-Reynolds number through computational grids without losing accuracy [6]. Y+ is the dimensionless distance from the wall which is used to check the distance from the wall to the first node. The k-ω based SST model accounts for the transport of the turbulent shear stress and gives highly accurate predictions of the onset and the amount of flow separation under adverse pressure gradients.

3. 3D NUMERICAL SIMULATION

The simulation of 3D turbulent viscous flow in the gas turbine of SST geometry was performed using ANSYS 17-CFX solver. Steady state 3D viscous, single phase, compressible flow was used. Topology with a first order upwind advection scheme was chosen because it is numerically stable. These assumptions were suggested by [7]. A stage interface was applied to the rotor. The Generalized Grid Interface feature of CFX was chosen for stage analysis and the steady state flow. The periodic boundary conditions were applied for blade passages for the rotor. The shear stress turbulence model (SST) was chosen and combined with Navier-Stokes equations. Simulations are carried out on the gas turbine with different operating temperature 400K and 600K at a constant pressure 3bar.The applied Boundary conditions are the total temperature, total pressure, flow direction, and the rotational speed as inlet conditions. A rotational, adiabatic wall was chosen for the blade and hub surface. The static pressure was chosen to be an output Boundary Condition. The convergence criteria for the residuals of both velocity and the continuity equations were of the order of 10-4 while for the energy equation 10-6.

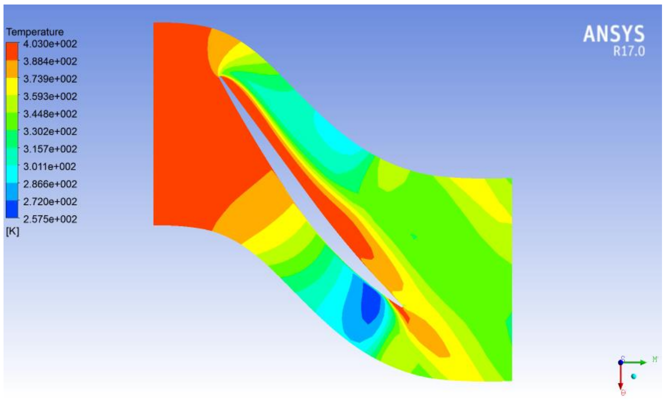

Figure 4 TEMPERATURE CONTOUR AT 400K, 1800 RPM

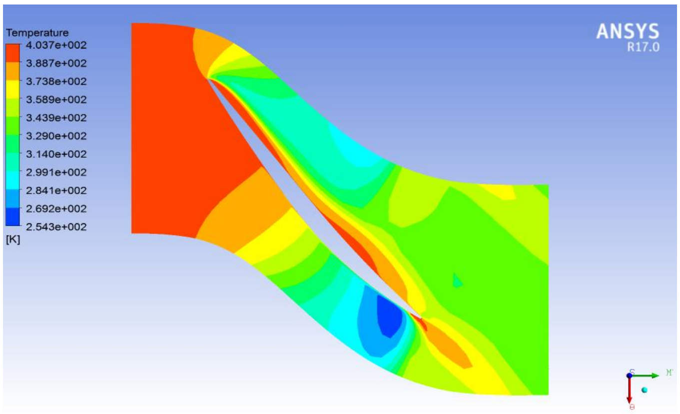

Figure 5 TEMPERATURE CONTOUR AT 400K, 3000 RPM

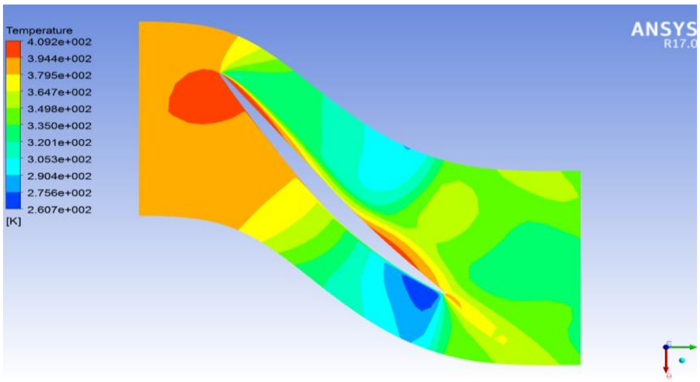

Figure 6 TEMPERATURE CONTOUR AT 400K, 5000 RPM

It shows that from figures 4, 5,6 temperature distribution of turbine blade at different speeds at constant pressure 3 bars and temperature 400K, the maximum temperature is at tip of the blade i.e., 4e+002 K and minimum at root of the blade 3.35e+002 K. It shows that temperature is decreasing from tip to root of the blade.

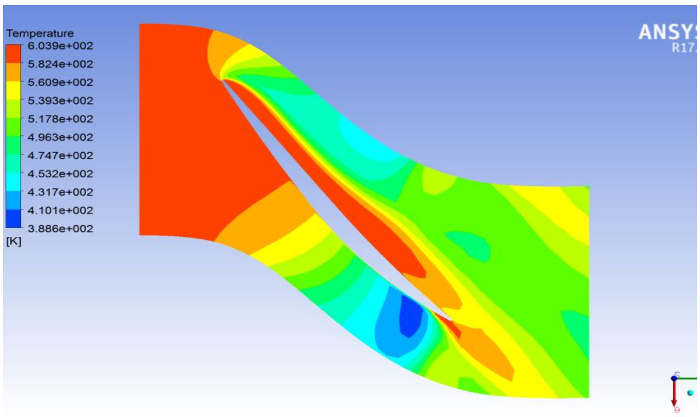

Figure 7 TEMPERATURE CONTOUR AT 600K, 1800 RPM

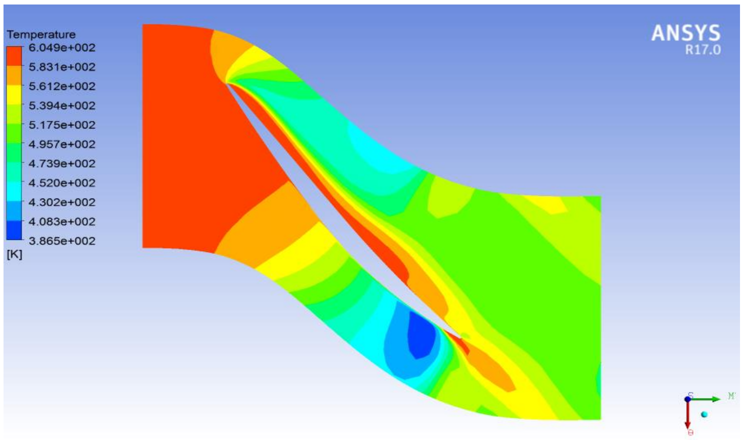

Figure 8 TEMPERATURE CONTOUR AT 600K, 3000 RPM

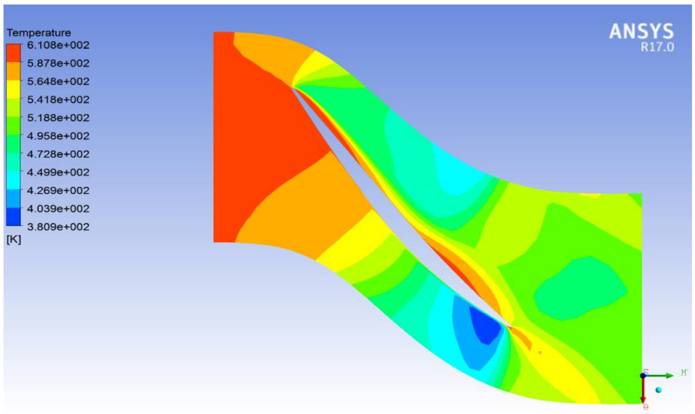

Figure 9 TEMPERATURE CONTOUR AT 600K, 5000 RPM

It shows that from figures 4.10, 4.11, 4.12 temperature distribution of turbine blade at different speeds at constant pressure 3 bars and temperature 600K, the maximum temperature is at tip of the blade i.e., 6e+002 K and minimum at root of the blade 4.95e+002 K. It shows that temperature is decreasing from tip to root of the blade.

4. RESULTS AND DISCUSSIONS

In this paper, the simulation of 3D turbulent viscous flow of SST geometry has been performed by using ANSYS 17-CFX solver. As a result, the effect of each: pressure ratio, rotational speed and temperature on the total to static turbines efficiency and output power have been figured out. Some of these results are shown in Figs. 10, 11, 12. Since the simulations are performed on a gas turbine with two different inlet temperatures at a constant pressure. The analysis is progressed at various rotational speeds to estimate the behavior of performance parameters. From the fig. 10 as the rotational speed is increased, total isentropic efficiency is simultaneously increased. The comparison is made between the 400K and 600K two inlet temperature conditions. It was seen that the former gives slightly better efficiency as compared to the latter. At 400K inlet temperature condition due to low enthalpy drop the flow

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velocity is decreased because the fluid having low Mach number. As the rotational speed increased the two efficiency lines of 400K and 600K temperature conditions shows that 600K temperature condition the total isentropic efficiency is slightly lower than 400K temperature condition

This is because increasing the Rotational speed allows a reduction in both Secondary and leakage losses. And also from the results obtained from the simulation, the entropy variation is more in 600K condition and hence more irreversibility is the major cause of slight variation of efficiency compared to the 400K condition.

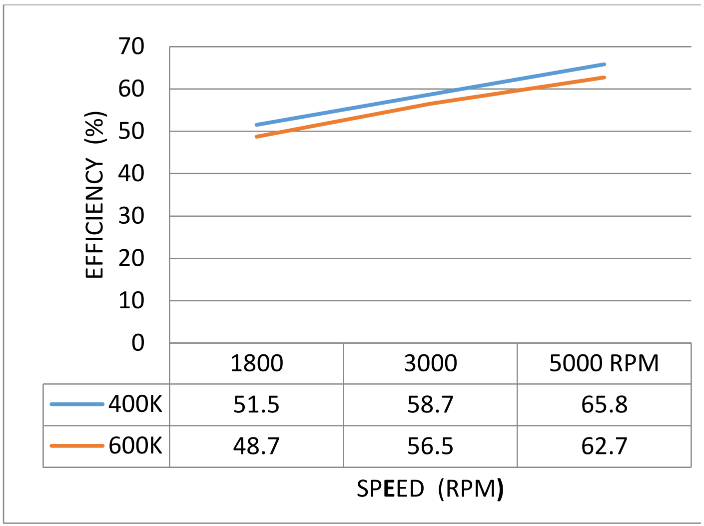

Figure 10 speed vs. efficiency

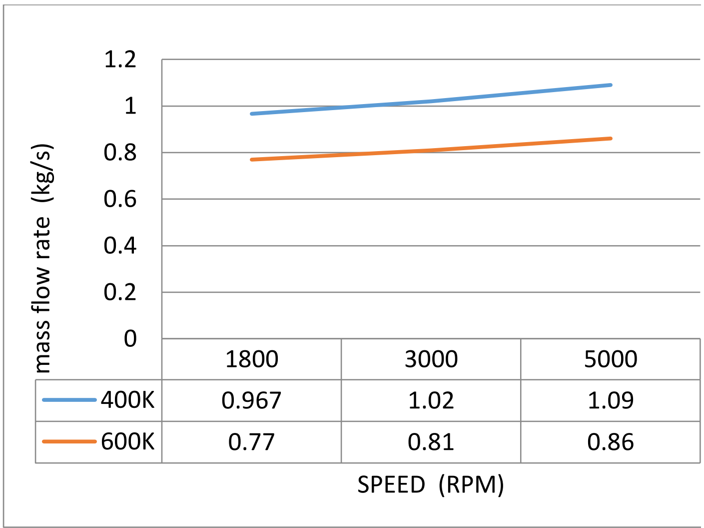

Figure 11 speed vs. mass flow rate

Figure 12 speed vs. outlet temperature

From the above graphs, it is shown that if temperature increases then efficiency and mass flow rate decreases but as speed increases, it gives maximum efficiency and maximum power output because as temperature increases then the density of the air decreases and then the mass flow rate of air decreases in ideal gas condition.

PV= MRT

ρ = m/v

Where P = pressure

V = volume

m = mass flow rate

ρ = density

If mass flow rate decreases, then the efficiency of the gas turbine also decreases. Turbine outlet temperature is also a major reason for efficiency difference between 400K and 600K conditions. Since the pressure is constant (600K) more outlet temperature results in higher efficiency of a turbine stage. Maximum efficiency and mass flow rate are at 400K and 5000 rpm which are 65.8% and 1.09 kg/sec.

5. CONCLUSION

In this paper, a series of computational simulations have been conducted to provide pre-test performance for the single stage gas turbine. The performance of single stage gas turbine with compressed air as a working fluid has been investigated at different boundary conditions.

The maximum temperature is observed at the tip of blade and minimum at the root of the blade. The temperature distribution is almost uniform and is linearly decreasing from the tip of the blade to root of the blade.

The performance was predicted at a constant pressure at two different inlet temperatures by varying rotational speeds of the gas turbine. At 400K constant pressure boundary condition the total isentropic 51%, 59%, 66% which are maximum as compared to 600K temperature. The maximum power output of a gas turbine is 9KW,

12.4KW, 17.6KW which is obtained at 600K constant pressure boundary condition.
From this project, we conclude that as the temperature increases then efficiency decreases but as speed increases then efficiency increases because it depends on mass flow rate. Mass flow rate of air depends on density which varies with temperature.

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