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Thermal and Structural Analysis of a Gas Turbine Casing Using Finite Element Method

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Abstract:- Gas turbines are becoming increasingly used as power generators for a wide variety of applications around the world. With such wide range of applications, it is necessary to improve the designs on continual basis for increased efficiency, reliability, availability and cost reduction. Gas turbine casings are generally of thin wall design to reduce thermal inertia to enable quick start up and shut downs. The main objective of the present investigation is to analyze the temperature distribution, stresses developed throughout the turbine casing using finite element method concept. In this paper, steady state thermal and structural analysis on a gas turbine casing of 26.82 MW capacity is carried out using ANSYS software with increased working gas temperatures and reduced outside casing thickness than the existing operating conditions. The outcome of the present work can be used for changing the operating conditions of the gas turbine to higher parameters or for resolving any machining deviations.

Keywords:- Gas Turbine Casing, Finite Element Method, ANSYS, Thermal Analysis, Structural Analysis.

I. INTRODUCTION

A gas turbine also called a combustion turbine, is a rotary engine that extracts energy from a flow of hot gas produced by combustion of gas or fuel oil in a stream of compressed air. It has an upstream air compressor with radial or axial flow mechanically coupled to a downstream turbine and a combustion chamber in-between. The thermodynamic cycle upon which a gas turbine works is called the Brayton Cycle. A schematic diagram of a single shaft, simple cycle gas turbine is shown in Fig. 1. Gas turbine may also refer to just the turbine element. Compressed air from the compressor flows into the annular space surrounding the combustion chambers from which it flows into the combustion liners and enters the combustion zone through metering holes in each of the combustion liners for proper fuel combustion. Fuel from an off base source is provided to equal flow lines, each terminating at a fuel nozzle centered in the end plate of a separate combustion chamber prior to distributed to the nozzles, the fuel is accurately controlled to provide an equal flow into the nozzle feed lines at a rate consistent with the speed and load requirements of the gas turbine. The nozzles introduce the fuel into the combustion chambers where it mixes with the combustion air and is ignited by one or more spark plugs. At the instant when fuel is ignited in one combustion chamber flame is propagated through connecting crossfire tubes to all other combustion chambers. After turbine rotor approximates operating speed, combustion chamber pressure causes the spark plugs to refract to remove their electrode from the hot flame zone. The hot gases from the combustion chamber expands into the separate transition pieces attached to the aft end of the combustion chambers liners and flow from there to the stages of the turbine part of the machine. Each stage consists of a row of fixed nozzles followed by the arrow of rotatable buckets. In each nozzle row, the kinetic energy of the jet is increased, with an associated pressure drop, and in each following row of moving blades, a portion of the kinetic energy of the jets absorbed as useful work on the turbine rotor. After passing through the last stage buckets, the gases are directed into the exhaust hood and diffuser which contains a series of turbine vanes to turn the gases from axial direction to a radial direction, thereby minimizing exhaust hood losses. The gases then pass into the plenum and are introduced to atmosphere through the exhaust stack. Resultant shaft rotation is used to turn the generator rotor and generate electrical power. Gas turbines are always used if high power density, low weight and quick starting are required. As the moving parts of a gas turbine only perform rotary motion, almost vibration free running can be achieved if the turbine is well balanced. Fig. 2 shows Temperature-Entropy diagram for the Brayton Cycle.

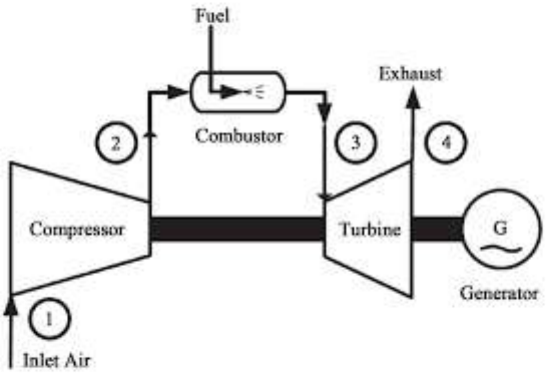

Fig 1: Simple Gas Turbine Power Plant

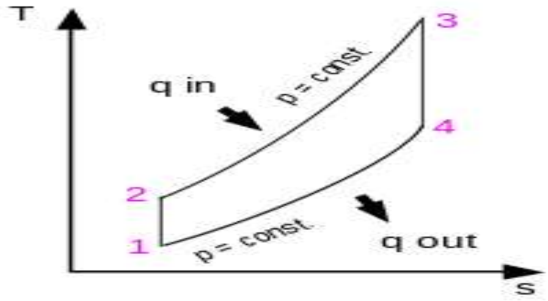

Fig 2: T-S Diagram of Brayton Cycle

II. FINITE ELEMENT METHOD CONCEPT

The finite element method is a numerical technique well suited to digital computers, which can be applied to solve problems in solid mechanics, heat transfer and vibrations. The procedures to solve problems in each of these fields are similar. However, this discussion will address the application of finite element methods to solid mechanics problem. In all finite element models, the domain (the solid in solid mechanics problem) is divided into a finite number of elements. These elements are connected at all points called nodes. In solid models, displacements are directly related to the nodal displacements. The nodal displacements are then related to the strains and the stresses in the elements. The finite element method tries to choose the nodal displacements so that the stresses are in equilibrium with the applied loads. The nodal displacements must also be consistent with any constraint on the motion of the structure. The finite element method converts the conditions of equilibrium into a set of algebraic equations for the nodal displacements. Once the equations are solved, one can find the actual strains and stresses in all the elements. By breaking the structure into a larger number of smaller elements, the stresses become closer to achieving equilibrium with the applied loads. Therefore, an important concept in the use of finite element methods is that, in general, a finite element model approaches the true solution to the problem only as the element density is increased.

III. METHODOLOGY

The basic steps involved in solving CFD problems are as follows:

- Specifying the Geometry
- Specifying Element Type and Material Properties
- Meshing the Object
- Applying Boundary Conditions and External Loads
- Generate a Solution
- Post Processing
- Refining the Mesh
- Interpreting the Results

In the present study, the thermal and structural analysis is carried out on the actual design of the turbine casing with the existing operating conditions and at increased temperatures beyond the operating conditions. Furthermore, investigations were carried out on the actual casing design by reducing its outside thickness up to the resultant stress value reaching the yield stress of the material. The following boundary conditions are applied on the casing for obtaining the solution.

Boundary Conditions Thermal Analysis:

There is hot ambient inside the casing and relatively cold ambient outside the casing, because of which there is a temperature gradient. This condition is taken care in the analysis by defining the boundary conditions both on inside and outside of the casing.

Ambient Temperature on the Inner Surface - 963^{o} C
Ambient Temperature on the Outer Surface - 485^{o} C

Structural Analysis:

Symmetric boundary conditions are applied in the structural analysis. Since any point on the vertical cutting plane cannot move in a horizontal direction, due to symmetric conditions, this plane is defined with Ux=0. Similarly, the horizontal parting plane is defined with Uy=0. A pressure load of 10 bar is specified on the inner surface of the casing, which is the maximum air pressure inside the turbine section at the base load of the unit.

Material Properties:

The material of the turbine casing is Cast Iron. Following are the material properties.

Thermal:

Thermal Conductivity	- 34.6 W/mK
Coefficient of Thermal Expansion	- 2.12 E-06/^{O}C
Density	- 7753 Kg/m^3

Structural:

Modulus of Elasticity	- 2.10 E 5 N/m^2
Poisson's Ratio	- 0.29
Yield Strength	- 276 E 6 N/m^2
Tensile Strength	- 414 E 6 N/m^2

Chemical Composition

Carbon	- 3%
Silicon	- 2.5%
Phosphorus	- 0.008%
Cast Iron	- 94.492%

IV. SOLUTIONS

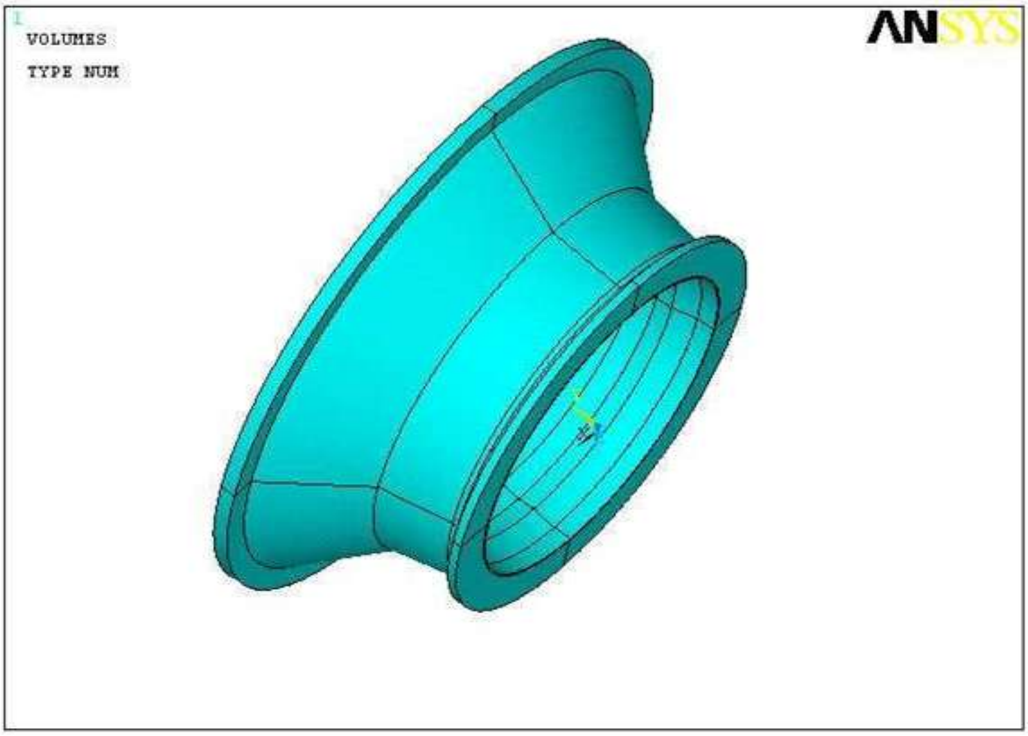

Fig. 3: 3-D Model of the Gas Turbine Casing in ANSYS

Fig. 3 represents the three-dimensional (3-D) view of the actual design of the gas turbine casing, which is analysed for thermal and structural analysis by using ANSYS software. This is obtained from the model created by feeding the key points to the computer from the original drawing.

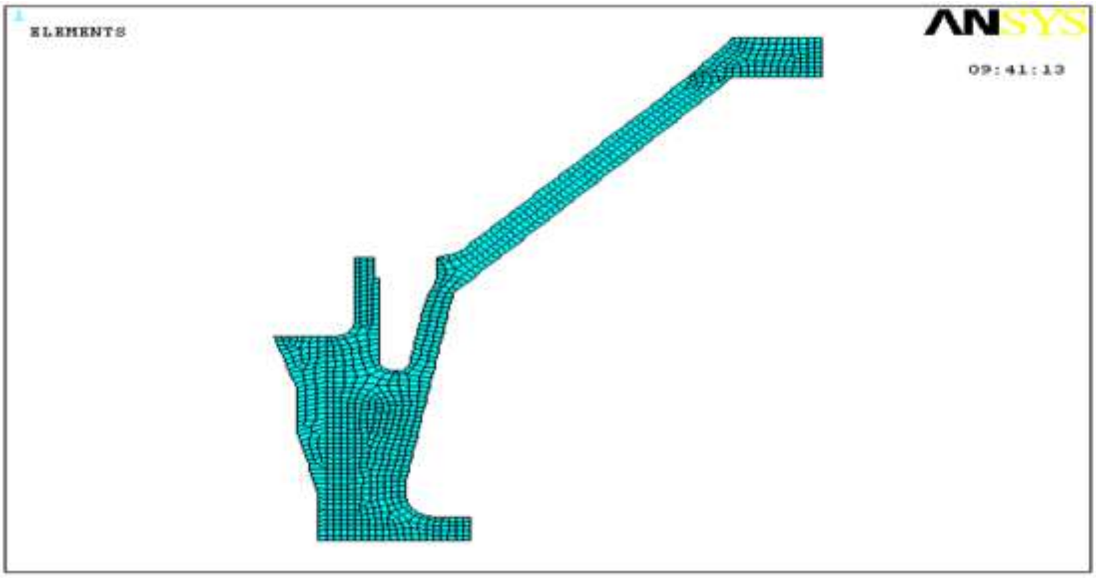

Fig. 4: Meshed View of the Gas Turbine Casing

Case 1: Temperature and Von Mises Stress Distribution of Actual Casing Design

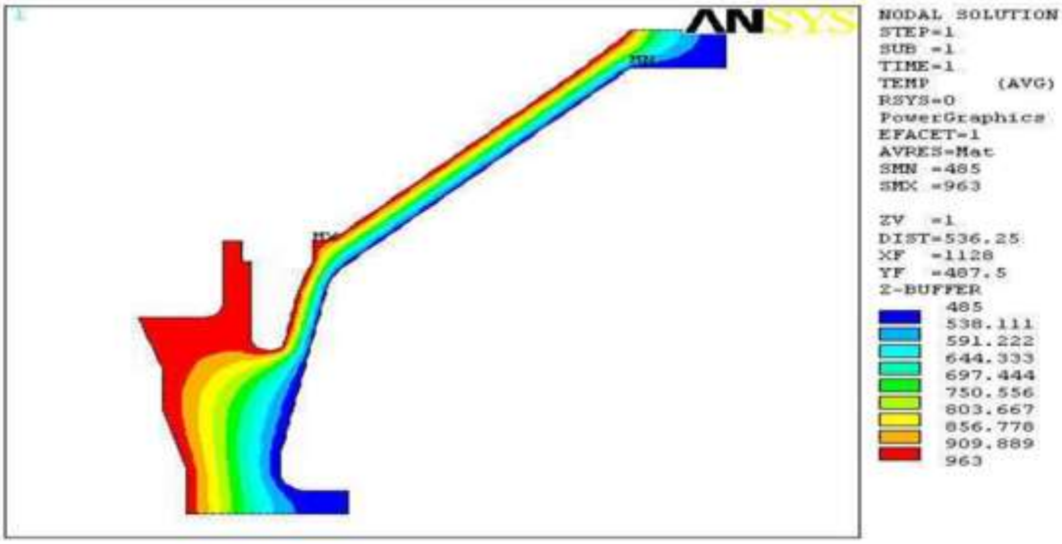

Fig. 5: Temperature Distribution Contours in ANSYS

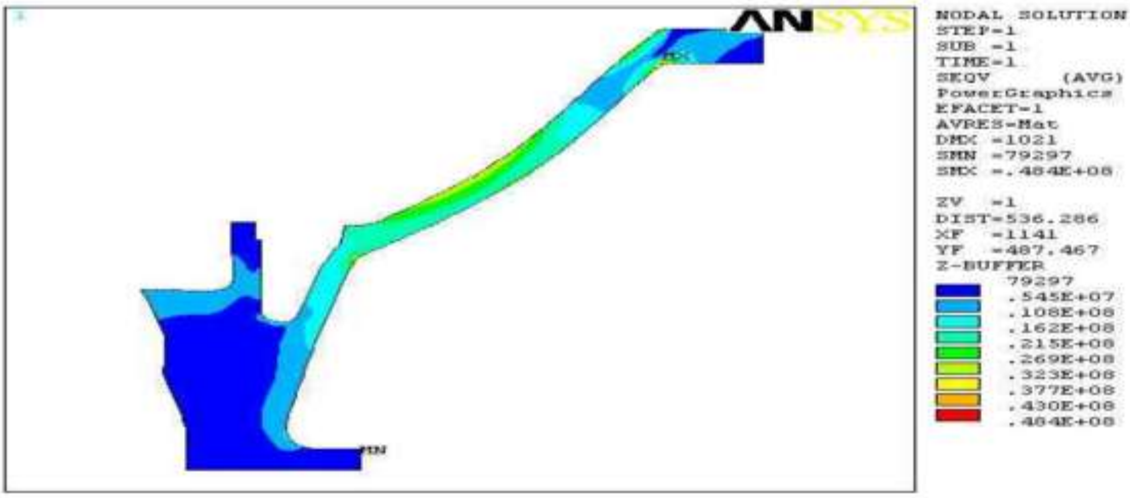

Fig. 6: Von Mises Stress (Pressure) Contours in ANSYS

Fig. 5 and fig. 6 represents the thermal analysis and structural analysis of the turbine casing. In this analysis, inside heat transfer coefficient and bulk temperatures are applied to the inner surface of the gas turbine casing, outside heat transfer coefficient and bulk temperatures of air side are applied to the outer side of the casing master model. Inside temperature of 963^{o} C and outside temperature of 485^{o} C are applied along the inner side of the casing. For structural analysis, inside and outside pressure of 10 bar and 1.013 bar are applied respectively to the casing. From fig. 6, the maximum stress obtained is 48.4 Mpa, with a factor of safety 5, the maximum stress is 242 Mpa. This value is found within the limit of yield stress value of 276 Mpa. Therefore, it is concluded that the existing design is safe.

Case 2: Temperature and Von Mises Stress Distribution at Increased Temperatures of 1100^{o}C and 600^{o}C

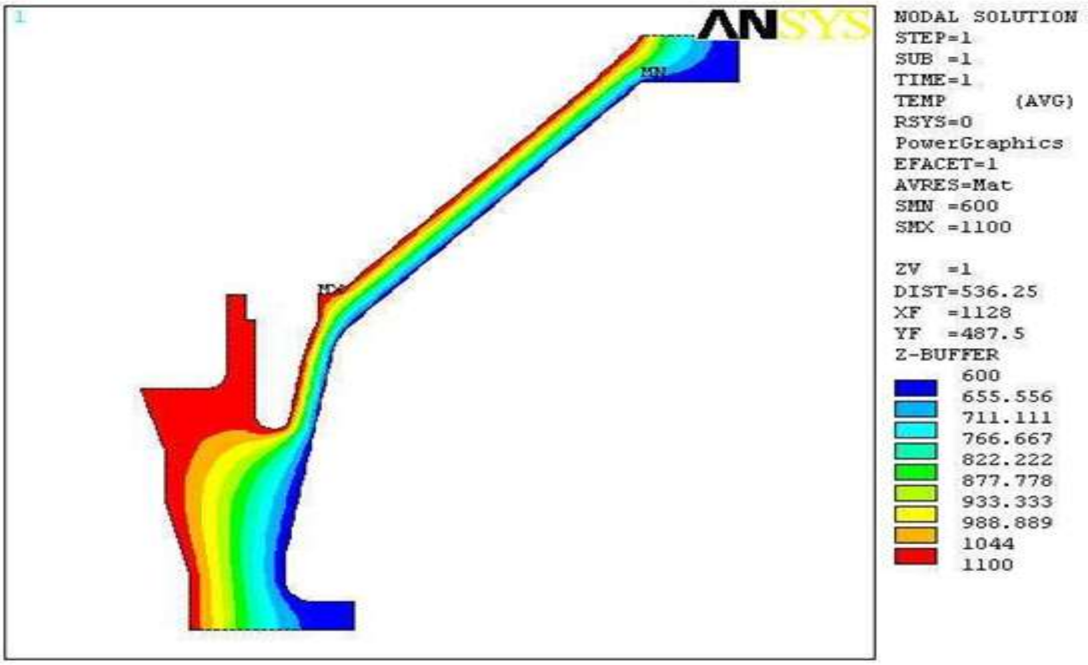

Fig. 7: Temperature Distribution Contours (Increased Temperature)

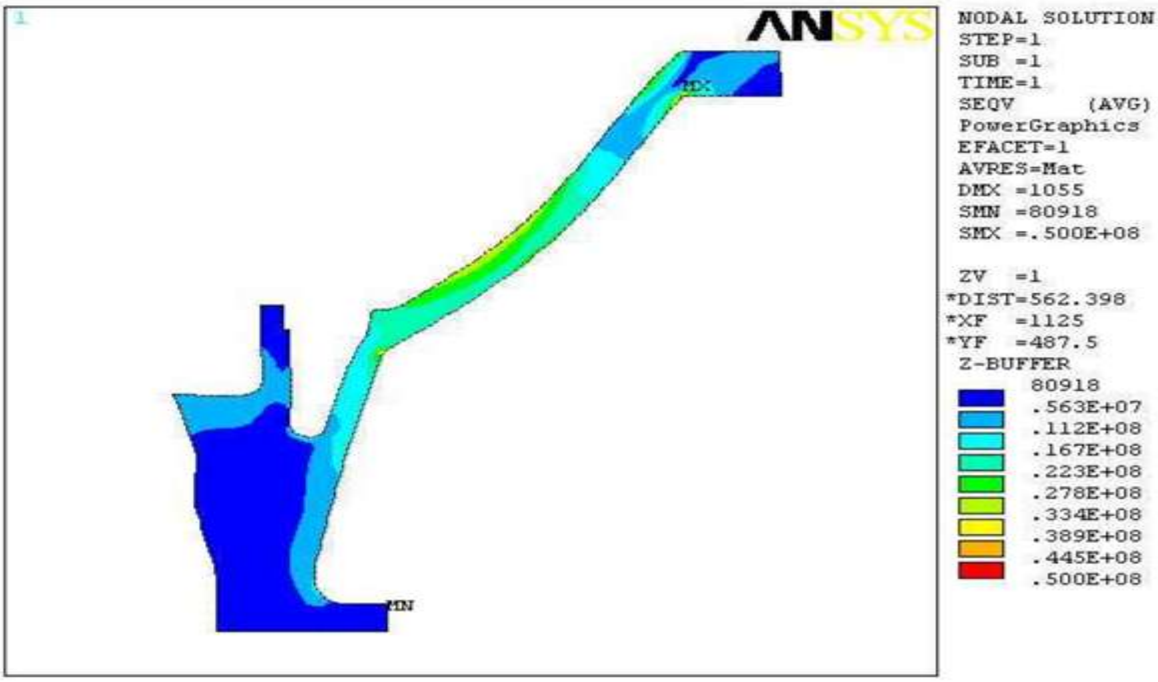

Fig. 8: Von Mises Stress (Pressure) Contours at Increased Temperatures

Fig. 7 and fig. 8 represents the thermal and structural analysis of the turbine casing with increased working gas temperature i.e. 1100^o C at inside and 600^o C at outside of the casing. The temperature distribution and stress distribution at these conditions are shown in the above figures. It is observed that the stress in the casing is increased with the increased gas temperature and pressure. Maximum stress developed is 50 Mpa. The stress value with a factor of safety reaches to 250 Mpa. It is observed that this value is also within the limit.

Case 3: Temperature and Von Mises Stress Distribution at Increased Temperatures of 1200^{o}C and 700^{o}C

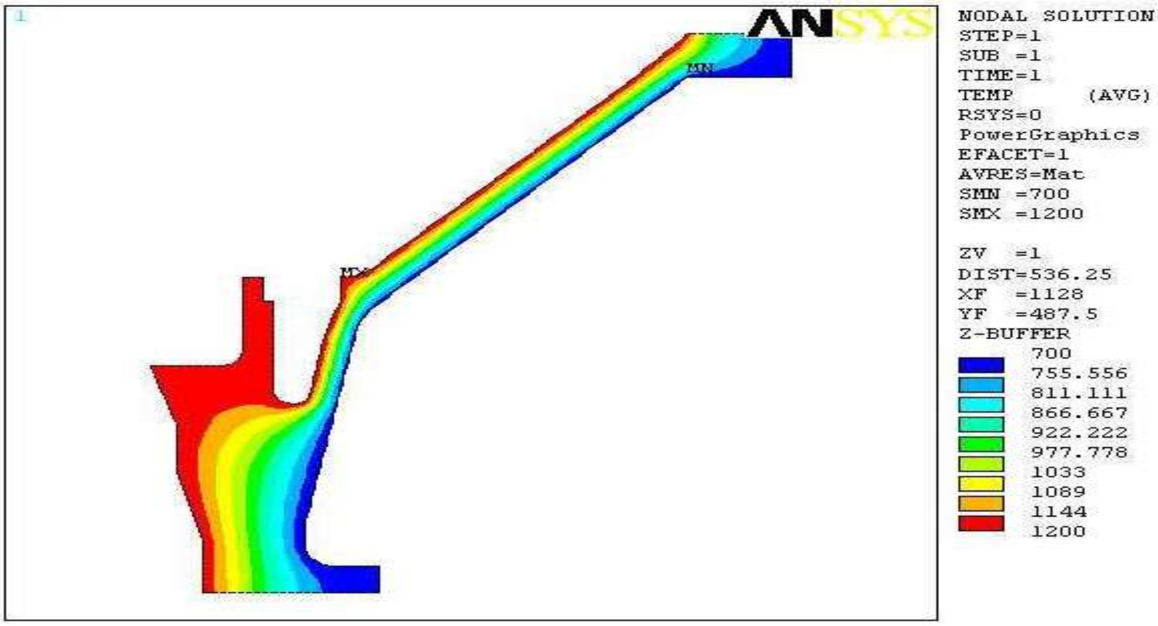

Fig. 9: Temperature Distribution Contours (Increased Temperature)

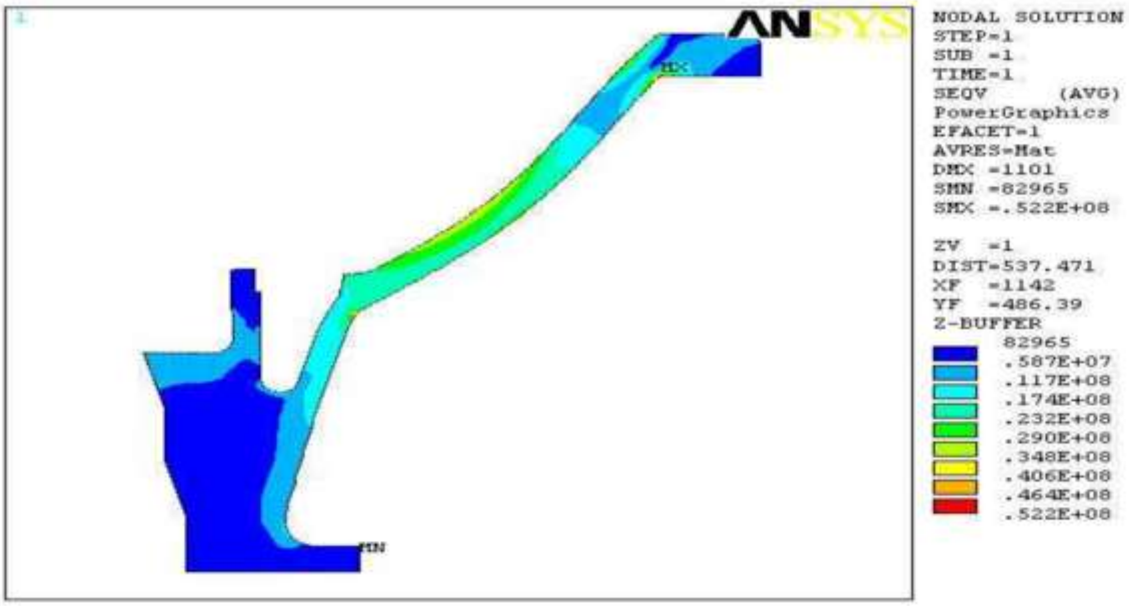

Fig. 10: Von Mises Stress (Pressure) Contours at Increased Temperatures

Fig. 9 and fig. 10 represents the temperature distribution and stress distribution of the turbine casing with increased temperature of working gas i.e. 1200^o C at inside and 700^o C at outside of the casing. From fig. 10, the maximum stress value observed is 52.2 Mpa which is far lower to the yield stress value of the casing material. With a factor of safety, the stress value is 262.5 Mpa. This value is also less than the yield stress value of the material. Therefore, it is concluded that one more attempt can be made with further increasing working gas conditions.

Case 4: Temperature and Von Mises Stress Distribution at Increased Temperatures of 1300°C and 800°C

Fig. 11: Temperature Distribution Contours (Increased Temperature)

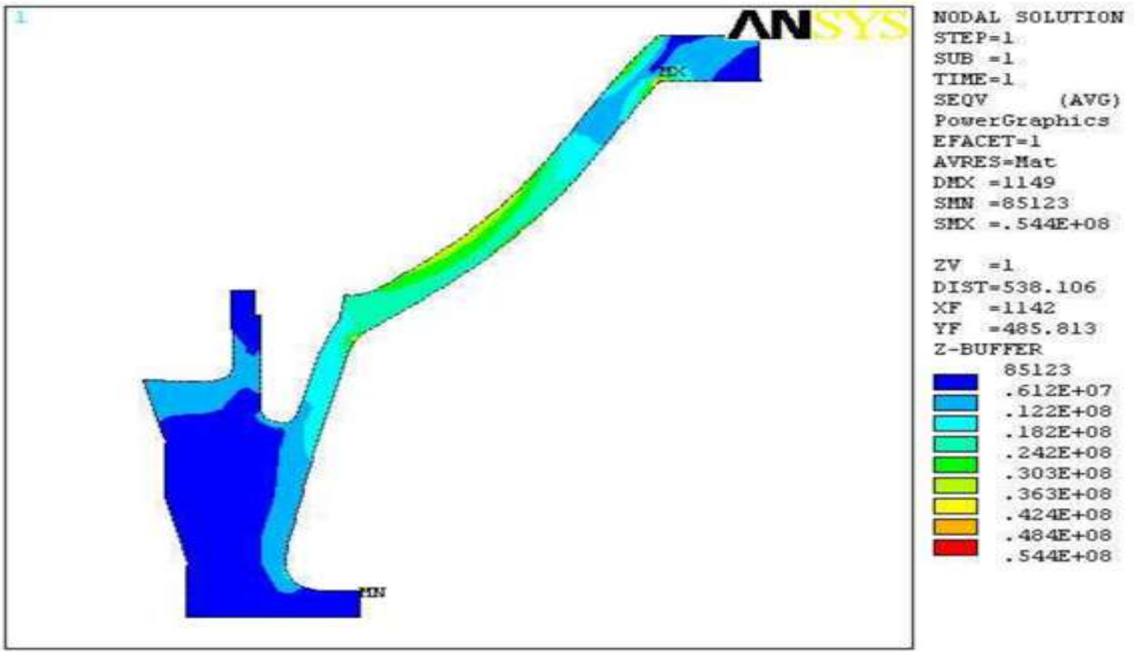

Fig. 12: Von Mises Stress (Pressure) Contours at Increased Temperatures

Fig. 11 and fig. 12 represents the thermal and structural analysis of the turbine casing with increased working gas temperature i.e. 1300° C at inside and 800° C at outside of the casing. From fig. 12, the maximum stress observed is 54.4 Mpa. By considering the factor of safety as 5, the stress value is 272 Mpa which is equal to the yield stress value of the casing material.

Case 5: Temperature and Von Mises Stress Distribution at Reduced Actual Casing Thickness of 5 mm

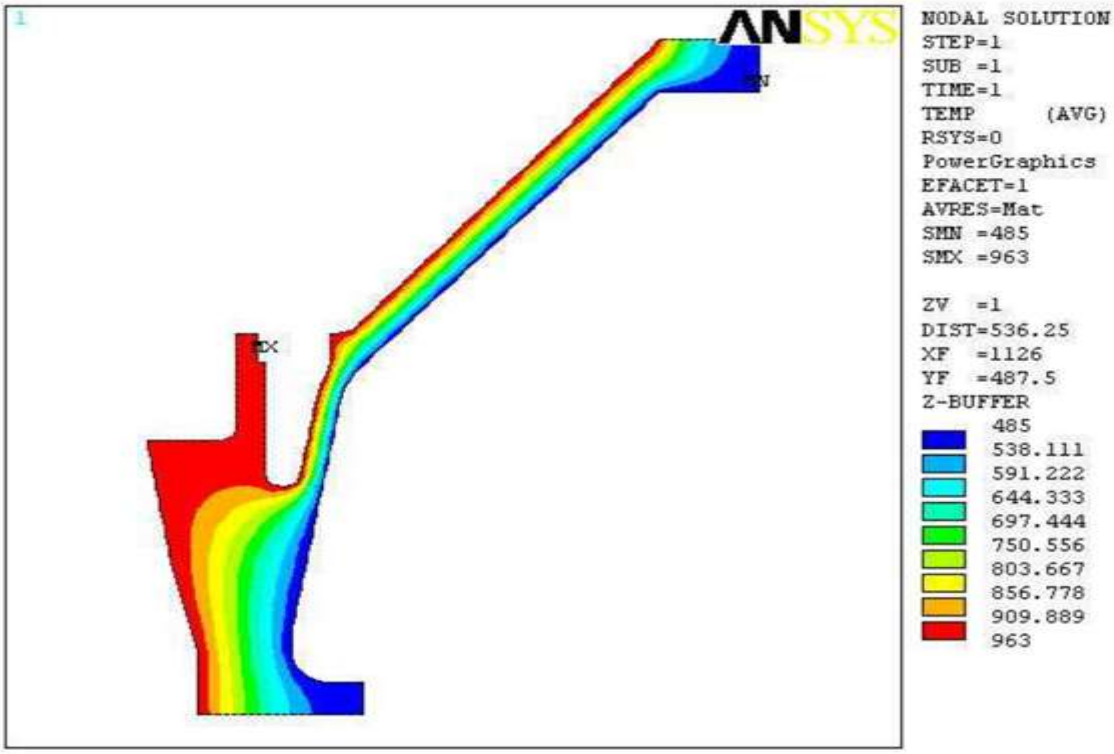

Fig. 13: Temperature Distribution Contours (Reduced Thickness of 5 mm)

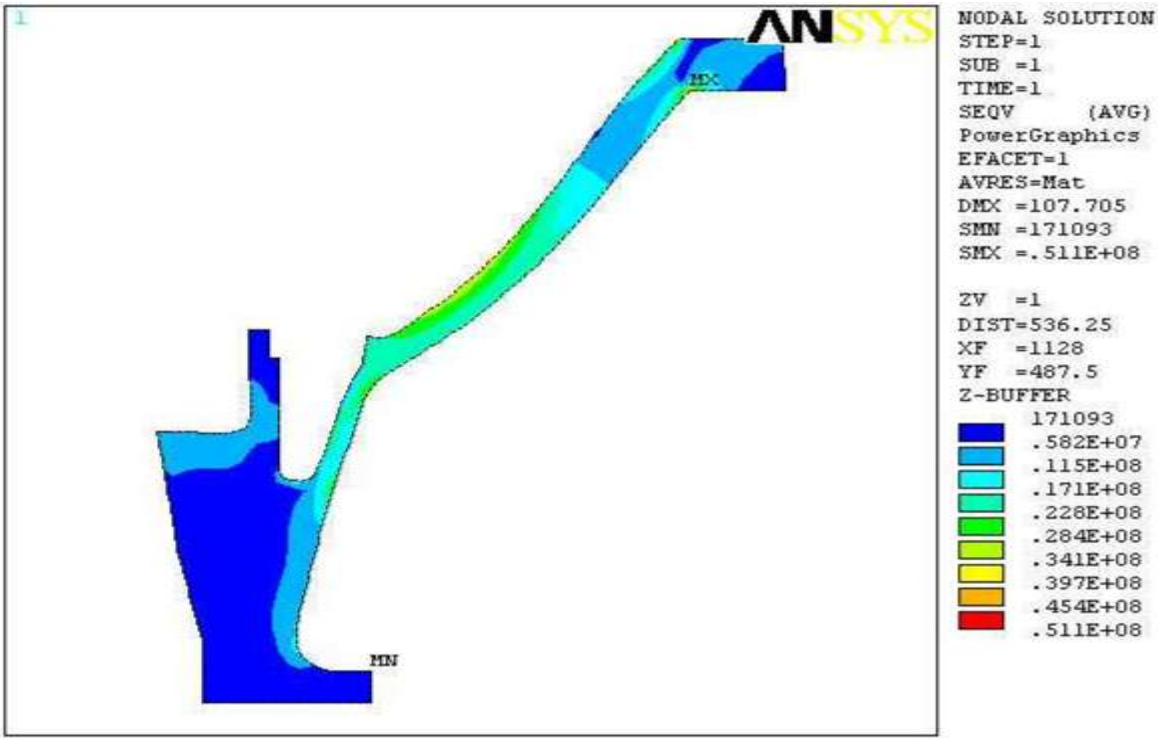

Fig. 14: Von Mises Stress Contours at Reduced Casing Thickness of 5 mm

Fig.13 and fig. 14 represent the thermal and structural analysis of the turbine casing at a reduced thickness of 5 mm at the outside of the casing uniformly to the master model. From the structural analysis, the maximum stress value observed is 51.1 Mpa. With a factor of safety, the stress value is 255.5 Mpa. It is observed that this value is far less than the yield stress value of the material.

Case 6: Temperature and Von Mises Stress Distribution at Reduced Actual Casing Thickness of 10 mm

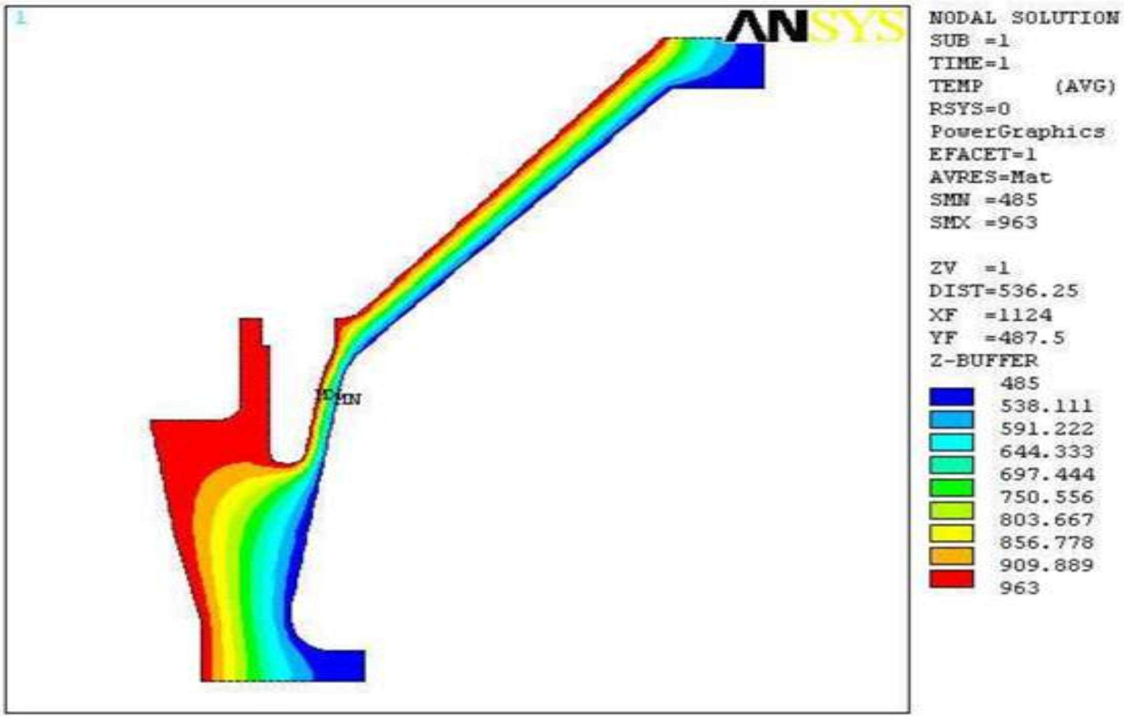

Fig. 15: Temperature Distribution Contours (Reduced Thickness of 10 mm)

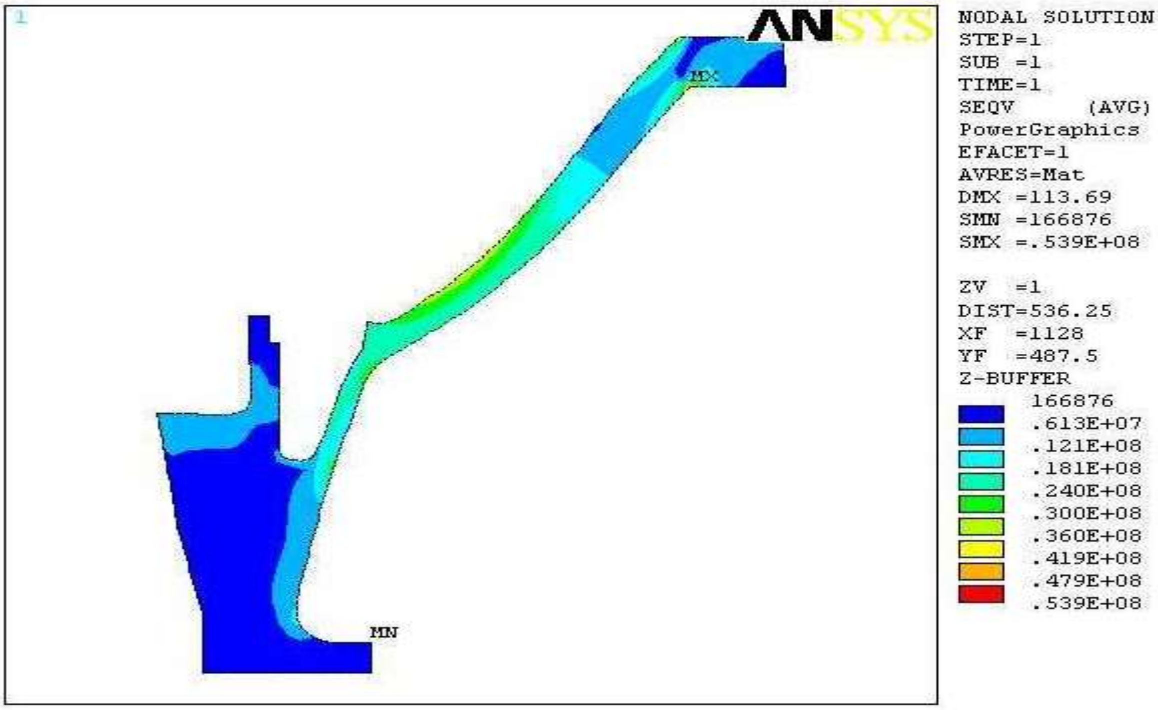

Fig. 16: Von Mises Stress Contours at Reduced Casing Thickness of 10 mm

Fig. 15 and fig. 16 represents the temperature and stress distribution at reduced outside casing thickness of 10 mm. the maximum stress value observed is 53.9 Mpa. By considering the factor of safety, the stress value is 269.5 Mpa. This value is within the yield stress value of the material. So it is concluded that one more attempt can be made with a further reduction of outside casing thickness.

Case 7: Temperature and Von Mises Stress Distribution at Reduced Actual Casing Thickness of 15 mm

Fig. 17: Temperature Distribution Contours (Reduced Thickness of 15 mm)

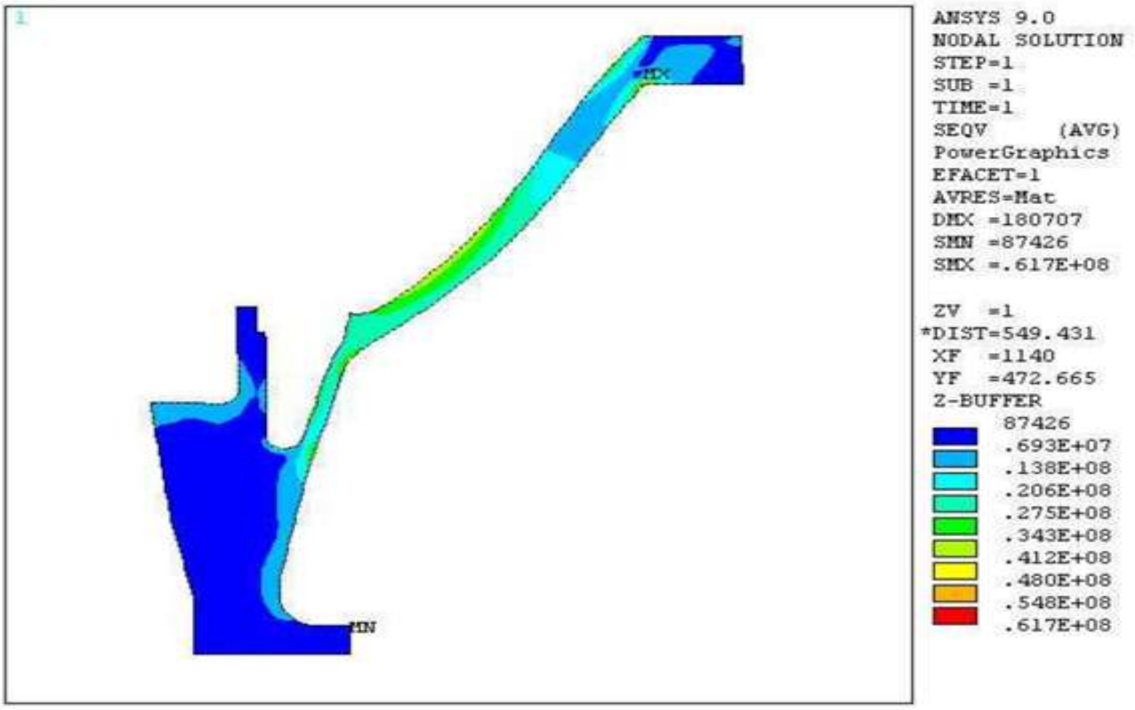

Fig. 18: Von Mises Stress Contours at Reduced Casing Thickness of 15 mm

Fig. 17 and fig. 18 represents the temperature and stress distribution at reduced outside casing thickness of 15 mm. the maximum stress value observed is 61.7 Mpa. By considering the factor of safety, the value reaches to 308.5 Mpa. This value is just crossing the yield stress value of the material. Hence, it is observed that the outside thickness of casing cannot be reduced further at existing design conditions of temperature and pressures.

V. RESULTS

Table 1: Von Mises Stresses at Different Increased Temperatures

Temperature and Stress	Case 1	Case 2	Case 3	Case 4
Inside Temperature (^{o}C)	963	1100	1200	1300
Outside Temperature (^{o}C)	485	600	700	800
Maximum Stress (Mpa)	48.4	50	52.2	54.4
Maximum Stress with Factor of Safety (Mpa)	242	250	262.5	272

Table 2: Von Mises Stresses at Reduced Thickness

Thickness and Stress	Case 1	Case 5	Case 6	Case 7
Reduced Thickness (mm)	**0**	**5**	**10**	**15**
Maximum Stress (Mpa)	**48.4**	**51.1**	**53.9**	**61.7**
Maximum Stress with Factor of Safety (Mpa)	**242**	**255.5**	**269.5**	**308.5**

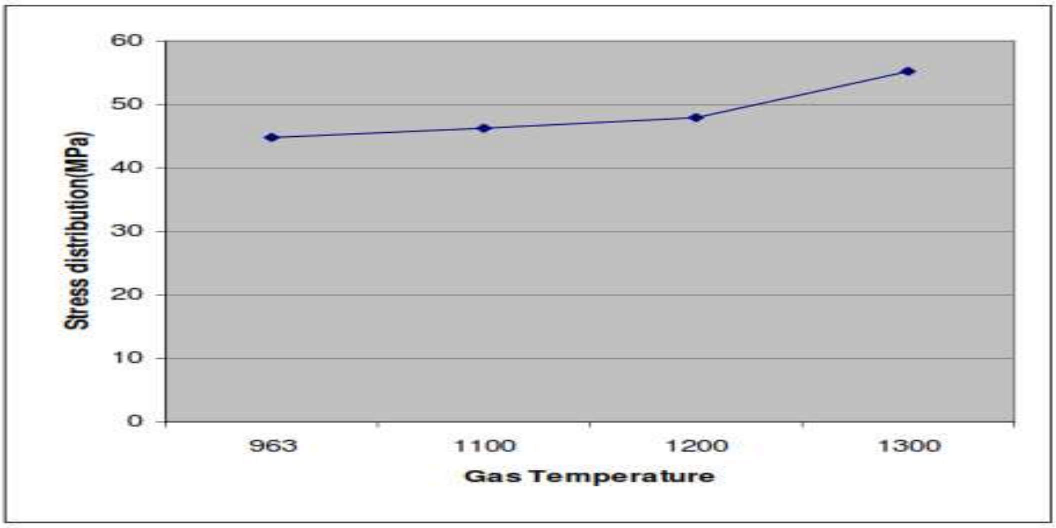

Fig. 19: Stress Distribution vs Gas Temperature

Fig. 19 represents the stress distribution of the gas turbine casing against various temperatures of working gas. From the figure, as the temperature of the working gas increases, the stress values in the casing material also increases.

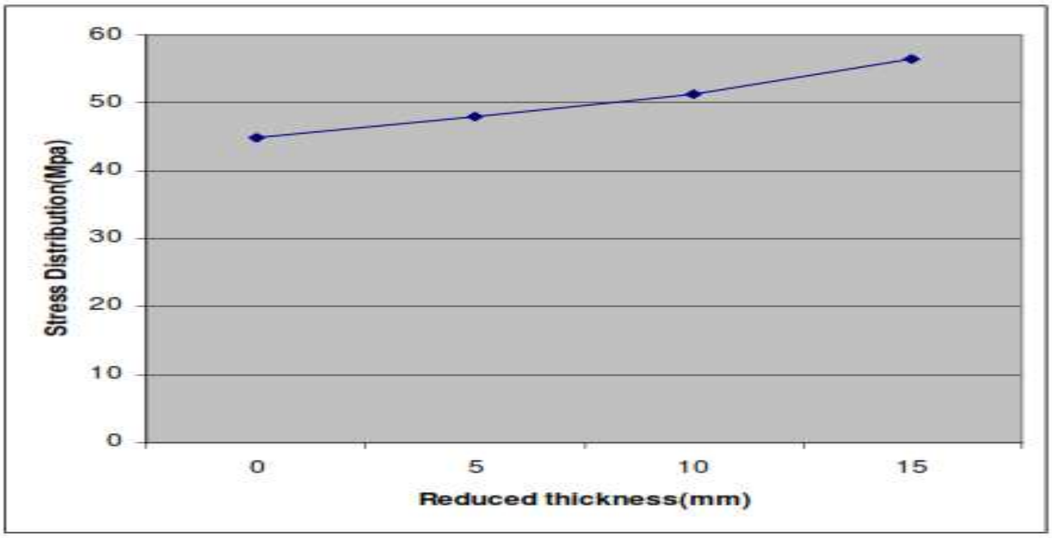

Fig. 20: Stress Distribution vs Reduced Casing Thickness

Fig. 20 represents the stress distribution of the gas turbine casing against reduced thickness at the outside surface of the turbine casing in equal decrements. From the figure, as the outside thickness of the casing decreases, the stress values increases.

VI. CONCLUSIONS

The structural and thermal analysis is carried out on the gas turbine casing using finite element method concept. The following conclusions are drawn from the present investigation.

The temperature has a significant effect on the overall stresses in the turbine casing and maximum stresses induced are within safe limit. As the temperature of the working gas increases in equal range, the stress values induced in the casing material also increases.

Corresponding to the inside and outside temperature of 963^oC and 485^oC, the stress developed in the turbine casing is 48.4 Mpa. With a factor of safety 5, the stress is 242 Mpa. Both these values are within the yield stress value of the casing material. Similarly, by increasing the working gas temperatures at inside and outside of the casing up to 1300^oC and 800^oC respectively, maximum stress value observed is 54.4 Mpa. With a factor of safety 5, the stress value is 272 Mpa. This value is nearly equal to yield stress value of the casing material. Hence, it is concluded that the temperatures of the working gas cannot be increased beyond this value.

Additionally, it is concluded that the outside thickness of the casing cannot be decreased beyond 15 mm of the actual design since the induced stresses in the modified design are more than the yield stress value of the casing material which leads to turbine failure. It is also concluded that as the outside thickness of the casing decreases, the stress values simultaneously increases.

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