ANALYSIS OF PRESSURE VESSEL

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ABSTRACT

Pressure vessel cylinders find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and fluid supply systems in industries. The failure of pressure vessel may result in loss of life, health hazards and damage of property. In addition to the pressure, the pressure vessels area also subjected to support loads that may be steady or variable, piping reactions, and thermal shocks which require an overall knowledge of the stresses imposed by these conditions on various vessel shapes and appropriate design means to ensure safe and long life. . In this paper, emphasis is on effect of stress concentration. It will be shown that an appropriate location and size of the opening in a pressure vessel results in minimizing the stresses induced due to the stress concentration resulting from the end flanges and other attachments. Also design and optimize the spherical and elliptical head profile with hole on the head, also Analysis the above profiles for various stress parameter.

Keyword: - *Pressure vessel , stress concentration, spherical, elliptical*

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1. INTRODUCTION

The pressure vessels are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it may combine with other reagents as in chemical plants. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and in water, steam, gas and air supply system in industries.In a appropriate location and size of the opening in a pressure vessel results in minimizing the stresses induced due to the stress concentration resulting from the end flanges and other attachments.

HIGH PRESSURE VESSELS

High Pressure vessels are used as reactors, separators and heat exchangers. They are vessel with an integral bottom and a removable top head, and are generally provided with an inlet, heating and cooling system and also an agitator system. High Pressure vessels are used for a pressure range of 15 N/mm^2 to a maximum of 300 N/mm^2. These are essentially thick walled cylindrical vessels, ranging in size from small tubes to several meters diameter. Both the size of the vessel and the pressure involved will dictate the type of construction used.

SOLID WALL PRESSURE VESSEL

A solid wall vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a ‘thick’ cylinder. In general, the physical criteria are governed by the ratio of diameter to wall thickness and the shell is designed as thick cylinder, if its wall thickness exceeds one-tenth of the inside diameter. A solid wall vessel is also termed as Mono Block pressure vessel.

PRESTRESSED PRESSURE VESSELS

With increase in internal pressures in mono block vessels, the problems of economic use of material as well as those of fabrication become critical. So a better utilization of the material is possible by setting upon initial stresses in the shell, known as prestressing.

The advantage of prestressing a vessel is either the reduction of the maximum stress existing under operating conditions i.e. to get uniform stress distribution throughout the shell or the reduction of the required shell thickness when a specified maximum allowable design stress is used. Regardless of the theory employed to calculate the maximum stress in the wall of a thick walled vessel, a non-uniform stress distribution under pressure will be found to exist in a mono block vessel that has not been pre-stressed. The principle of prestressing is to induce a permanent residual compressive stress existing under zero pressure at the point in the shell where the maximum tensile stress is induced under pressure.

2. LITERATURE SURVEY

M. Javed Hydera, M. Asifb (2007) main objective the research work is to optimize the location and size of opening (hole) in a pressure vessel cylinder using ANSYS. Analysis is performed for three thick-walled cylinders with internal diameters 20, 25 and 30 cm having 30 cm height and wall thickness of 20 mm. It is observed that as the internal diameter of cylinder increases the Von Misses stress increases. Optimization of hole size is carried out by making holes having diameter of 4, 8, 10, 12, 14, 16 and 20 mm located at center in each of the three cylinders, and it is observed that initially Von Misses stress decreases and then become constant with hole size. The optimum size of hole is found to be 8 mm for cylinder having internal diameter of 20 cm whereas a hole of size 10 mm for cylinder having internal diameter of 25 cm and 30 cm on the basis of lowest Von Mises stress value. Lastly, optimization of location of hole is carried out by making a 12 mm hole located at 1/16, 1/8, 2/8, 3/8 and 4/8 of cylinder height from top in all the three cylinders. The Von Misses stress is maximum at the center i.e., 4/8 location and decreases in the direction away from center and then stress increases as the location is changed from 1/8 to 1/16 from cylinder top due to the end effects. The optimum location of the hole is found to be at 1/8 of cylinder height.

Rashmi Ranjan Nath (2011) It is proposed to conduct stress analysis of a thick walled cylinder near the radial hole on the surface. The literature indicated that there will be a ductile fracture occurring in such cases. The radial holes cannot be avoided due to various piping attachments . Hence the stress analysis of cylinder and its ultimate failure under internal pressure beyond elastic limit is an appropriate scenario. The plastic zone appearing in vicinity of internal surface of cylinder propagates more fastly along hole side . When cylinder is unloaded it will cause reverse plasticity. Therefore it is proposed to obtain numerical solution using Finite Element analysis of cylindrical segment to obtain the radial & hoop stress distribution by including elastoplastic conditions. The stress analysis of thick walled cylinders with variable internal pressure states is conducted Elastic analysis of uniform cylinder & cylinder with holes is predicted both from theory (lame’s formulae) under & Finite element method. Also elasto plastic analysis with bilinear kinematic hardening material is performed to know the effect of hole sizes. It is observed that there are several factors which influence stress intensity factors. The Finite element analysis is conducted using commercial solvers ANSYS & CATIA. Theoretical formulae based results are obtained from MATLAB programs. The results are presented in form of graphs and tables.

Peng-fei LIU, Ping XU, Shu-xin HAN, Jin-yang ZHENG (2008) As the idea of simulated annealing (SA) is introduced into the fitness function, an improved genetic algorithm (GA) is proposed to perform the optimal design of a pressure vessel which aims to attain the minimum weight under burst pressure constraint. The actual burst pressure is calculated using the arc-length and restart analysis in finite element analysis (FEA). A penalty function in the fitness function is proposed to deal with the constrained problem. The effects of the population size and the number of generations in the GA on the weight and burst pressure of the vessel are explored. The optimization results using the proposed GA are also compared with those using the simple GA and the conventional Monte Carlo method.

3. MATERIAL SELECTION

Table 3.1

Description	Material	UTS MPa (Min)	YS MPa (Min)
Vessel	SA 515 GR 70	492.9	267.6
Dished Ends	SA 515 GR 70	492.9	267.6

4. PRO-E MODELING OF PRESSURE VESSEL

4.1 Elliptical Head Without And With Hole

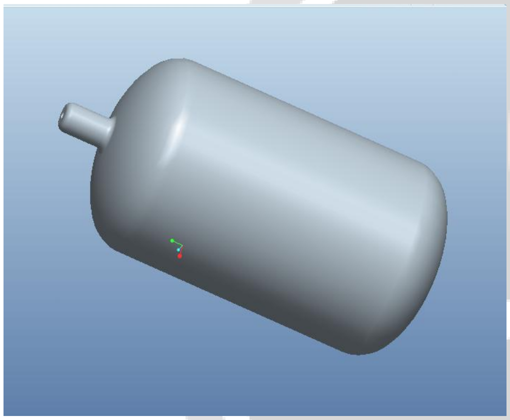


Fig 4.1 Elliptical Head Without Hole

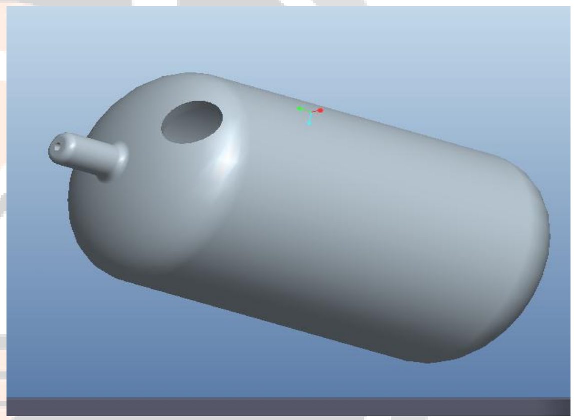


Fig 4.2 Elliptical Head With Hole

4.2 Spherical Head Without And With Hole

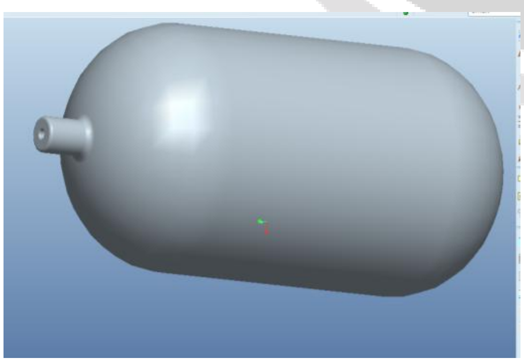


Fig 4.3Spherical Head Without

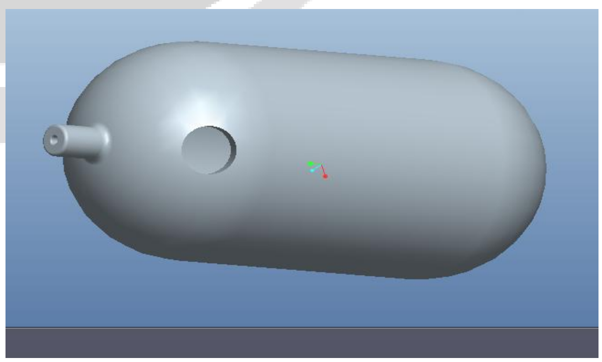
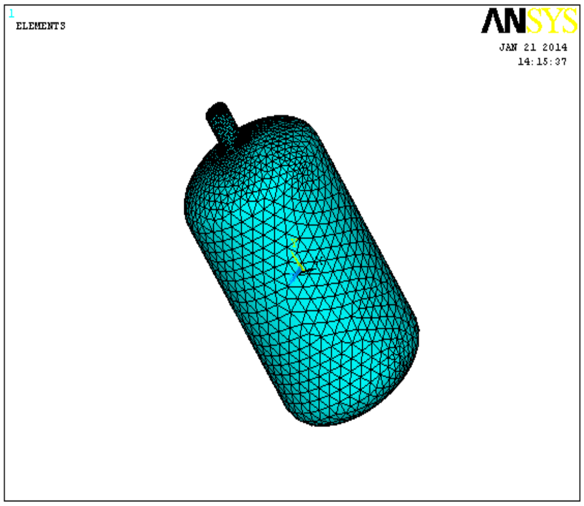


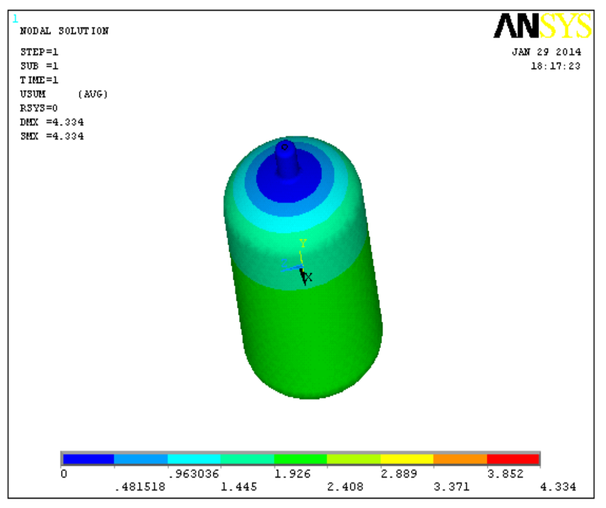
Fig 4.4Spherical Head With Hole

5. FINITE ELEMENT ANALYSIS OF PRESSURE VESSEL

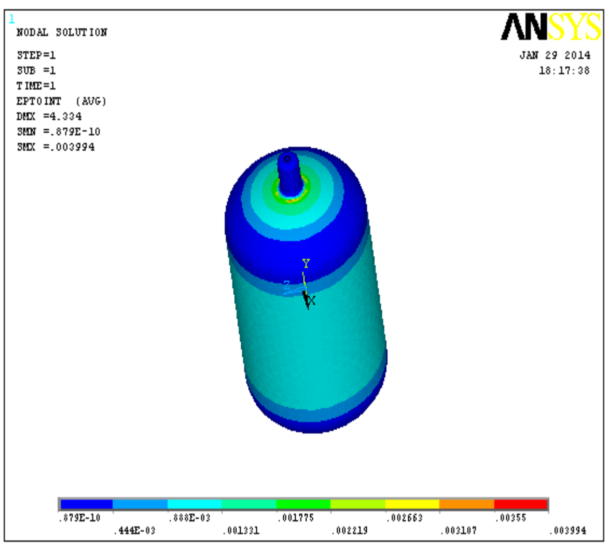
5.1 Elliptical Head Without Hole



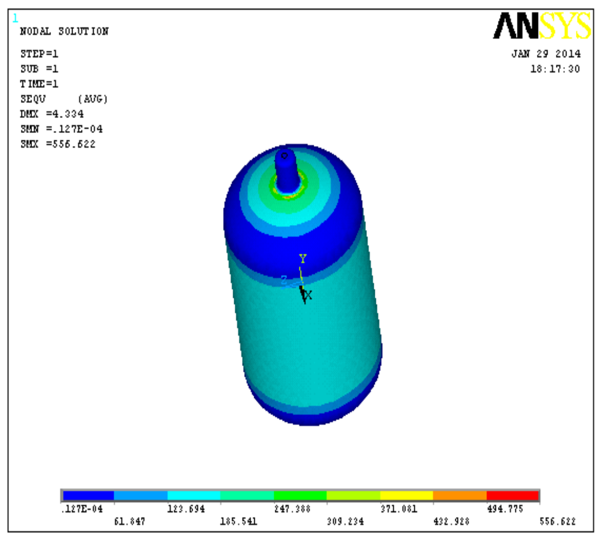
Meshing



Displacement

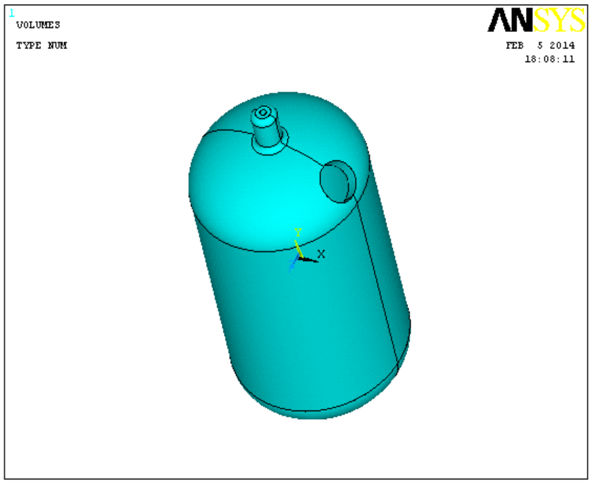


Stress

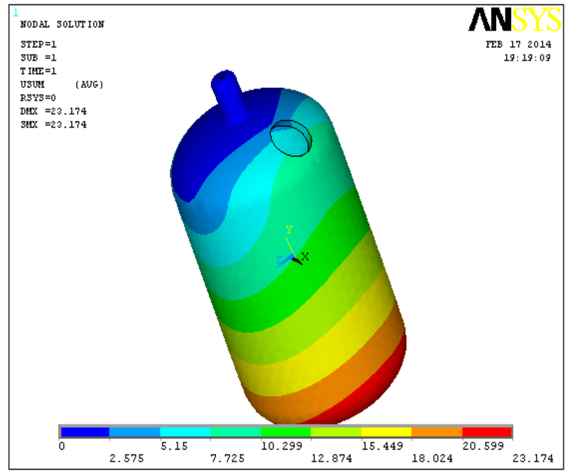


Strain

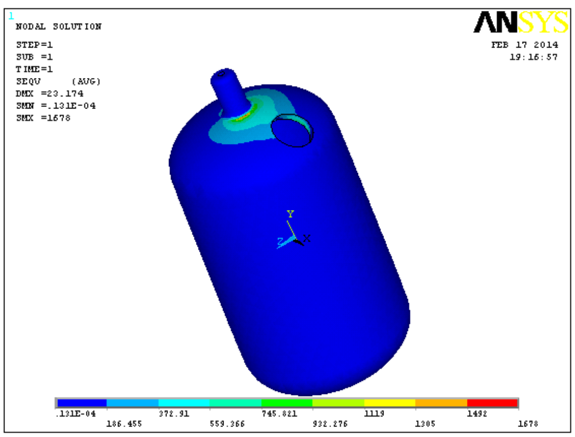
5.2 Elliptical Head With Hole



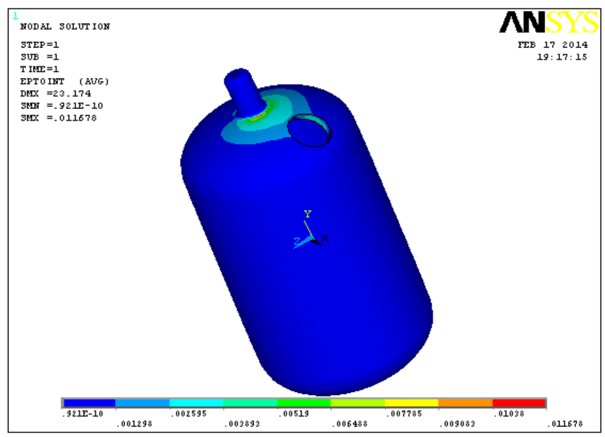
Imported Model



Displacement

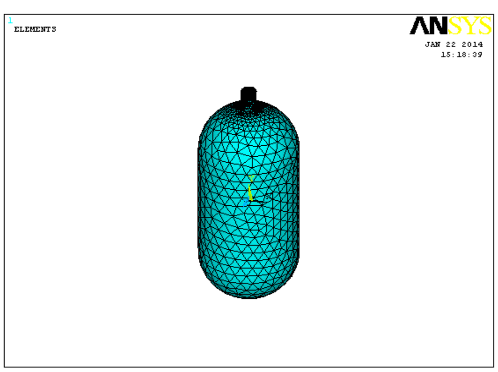


Stress

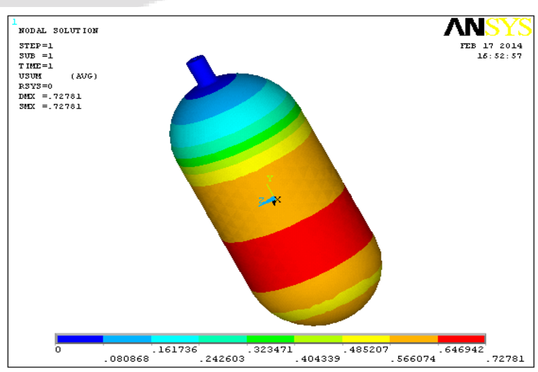


Strain

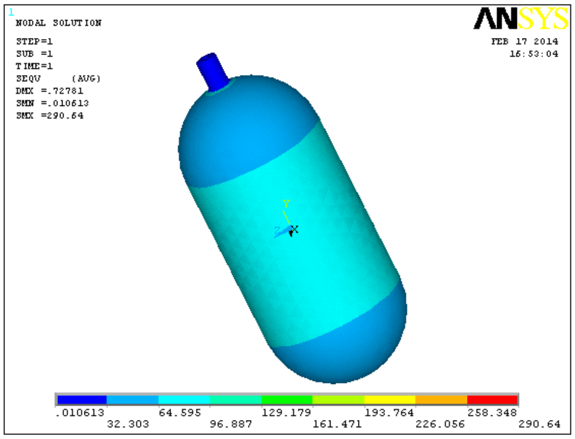
5.3 SPHERICAL HEAD WITHOUT HOLE



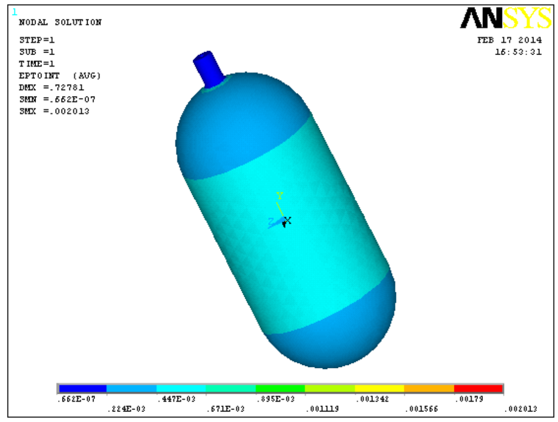
Meshing



Displacement

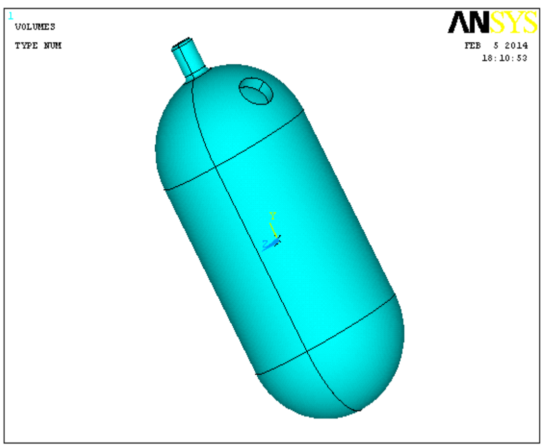


Stress

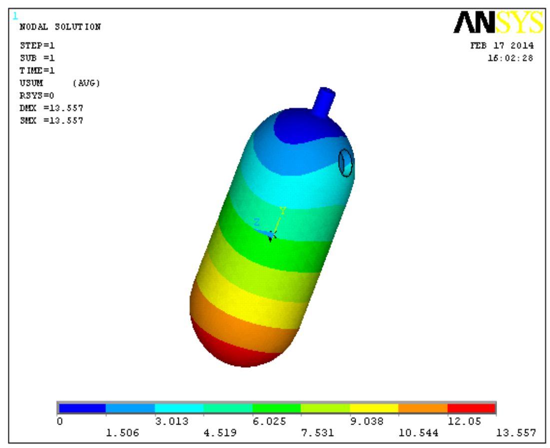


Strain

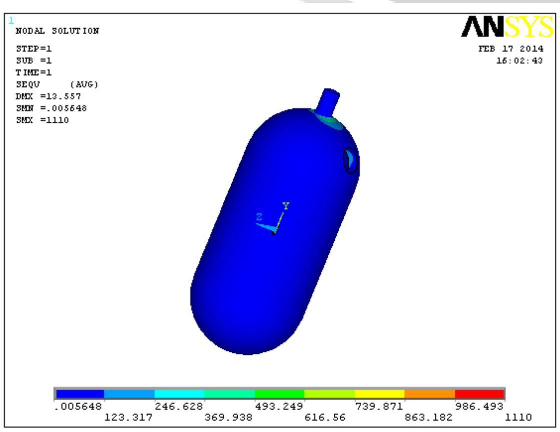
5.3 SPHERICAL HEAD WITH HOLE

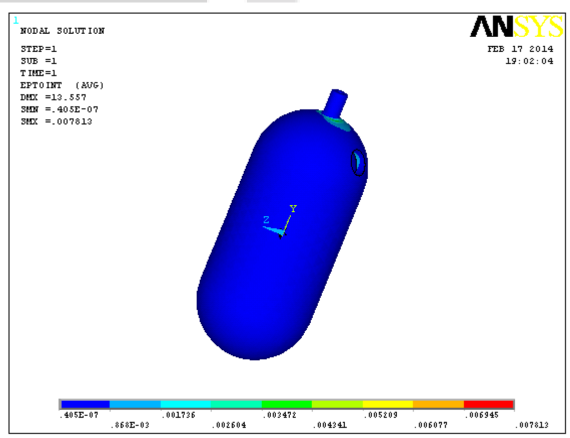


Imported Model



Displacement





Stress Strain

WITHOUT HOLE AND WITH HOLE

	Displacement (mm)	Stress (N/mm^2)	Strain
Elliptical without hole	4.334	556.6622	0.003994
Elliptical with hole	23.124	1628	0.011678

	Displacement (mm)	Stress (N/mm^2)	Strain
Spherical without hole	0.72791	290.64	0.002013
Spherical with hole	13.557	1110	0.007813

CHANGING DISTANCES AND DIAMETERS

Spherical

	500mm DIA, DISTANCE- 500mm	DIA 400mm, DISTANCE-400mm	450mm DIA, DISTANCE- 350mm
Displacement (mm)	11.682	6.146	7.367
Stress (N/mm$^{2)}$	499.219	619.242	661.739
Strain	0.003401	0.004317	0.004583

Elliptical

	550mm DIA, DISTANCE- 700mm	500mm DIA, DISTANCE- 750mm	DIAMETER-450mm, DISTANCE- 350mm
Displacement (mm)	17.494	15.246	14.411
Stress (N/mm^2)	1349	1208	907.861
Strain	0.009381	0.008418	0.006356

THERMAL ANALYSIS

	Temperature(K)	Thermal Gradient(K/mm)	Heat Flux(W/mm^2)
Steel	673	49.016	9.313

4. CONCLUSIONS

Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work Conclusion related your research work

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