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INVESTIGATIONS ON FAILURE OF CYLINDRICAL JET DYING MACHINE

C.K. Desai[1], D.P. Vakharia[2] and S.A. Channiwala[3]

[1] Lecturer, C.K.Pithawala College of Engineering & Technology, Surat, India.
[2] Lecturer, Sardar Vallabhbhai National Institute of Technology, Surat, India.
[3] Professor, Sardar Vallabhbhai National Institute of Technology, Surat, India.

ABSTRACT
The present paper discusses the failure analysis of an industrial cylindrical jet dying machine, which is also one type of pressure vessel used in textile industries. Failure of pressure vessel largely depends upon various parameters like shell thickness, head thickness, pressure, material properties, thickness of saddle supports etc… There exist few analytical techniques for determination of failure of pressure vessels. This paper aims towards arriving at more exact determination of failure of cylindrical jet dying machine. It includes the effects of strain ratio at the junction of the shell and torispherical head due to varying thickness. It also includes the Finite Element Method for calculating the displacement of various parts of cylindrical jet dying machine and the comparisons of the displacement obtained from Finite Element Method and experimentally measured displacement of jet dying machine in the graphical form.

Key words: Strain ratio, Cylindrical jet dying machine, Finite Element Method (FEM)

1. INTRODUCTION
Pressure vessels are leak proof containers. They may be of any shape ranging from milk bottles, shaving cream cans, automobile tires or gas storage tank. The ever increasing use of vessels for storage, industrial processing and power generation under unusual conditions of pressure, temperature and environment has given special emphasis to analytical and experimental method for determining the operating stresses. For instance, if the stresses or strains in a structure are unduly low, its size becomes larger than necessary and the economic potential of material is not reached. Chemical engineering involves the applications of process industries which are primarily concern with the conversion of one material into another by chemical or other means. These processes require the handling and storing of large quantities of materials in container of varied construction, depending upon the existing state of the material, it's physical and chemical properties and the required operations which are to be performed. For handling such liquids and gases a container or a "vessel" is used. The vessel is the basic part of most types of processing equipments.

Cylindrical jet dying machine is also one type of pressure vessel used in textile industries. Fig 1 shows the detailed drawing of cylindrical jet dying machine. This jet dying machine was failed thrice during the hydraulic testing. The observation of the failed dished end indicated an inward stretching of the plate. This seemed quite surprising if one sees the fact that the vessel was under pressure, on the contrary failure would have been towards outer side in the form of bulging. This paper deals with the detailed analysis of failure of jet dying machine. The design as used here does not mean only the calculation of the detail dimensions of a jet dying machine but rather is an all inclusive term incorporating the reasoning that established the most likely mode of damage or failure. Also the Finite Element Analysis on IDEAS software was carried out to investigate the failure of jet dying machine.

2. LITERATURE
Pressure vessels are the cylindrical or spherical shaped vessels meant to store fluids under pressure. A lot of researches have been made to check the reliability of the pressure vessel because its failure may lead to loss of life as well as money. So utmost care should be taken while designing the pressure vessels. J.H.Gross [1] and Yoshio Ando [2] have done their work on selection of materials used for fabrication of pressure vessels, where heavy section steel pressure vessel for light water reactor and aluminum tank for liquefied natural gas carrier are chosen to review the fabrication of pressure vessels. As the pressure vessel can not be

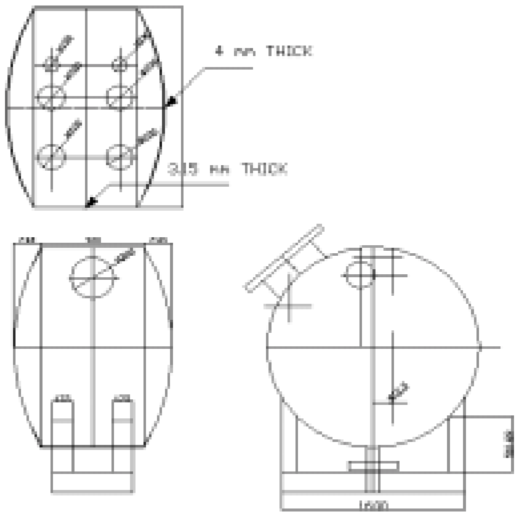

Fig 1 Cylindrical Jet Dying Machine

created in single piece some kind of welding is always necessary to join various parts and that too vary carefully because any errors may lead to leakage of the fluid from the vessel. E.Friedman [3] has developed models for calculating temperatures, stresses and distortions resulting from welding process. The models are implemented in finite element formulations and applied to a longitudinal butt welding. J.Reyenen [4] has described algorithms and their computer implementations to calculate effectively strain energy release rates by means of FEM, making use of sub structuring techniques. A.S.Kobayashi [5] has done his work on inner and outer cracks in internally pressurized cylinders. Stress intensity factor of pressurized surface cracks at the internal surface and unpressurized surface cracks at the external surface of an internally pressurized cylinder are estimated from stress intensity factors of semi elliptical cracks in a finite thickness flat plate. D.Bushnell and G.D.Galletly [6] have done their work on stress and buckling of internally pressurized elastic-plastic torispherical heads. Several aluminum and mild steel torispherical heads were tested by Galletly and by Kirk and Gill [7] and subsequently analyzed by Bushnell with the use of BOSOR 5 computer program. Bushnell [8] and J.S.Porowski [9] have done their work on buckling failure of various shells and weight saving plastic design of vessels.

3. OBJECTIVES OF PRESENT INVESTIGATIONS

The purpose of the present study was to calculate the thickness of cylindrical shell and torispherical head at design pressure, to calculate the strain ratio at the junction of shell to head, to measure the displacement at different pressure value and finite element analysis of jet dying machine. The following data of cylindrical jet dying machine were taken to investigate the failure.

Material of construction : SS316L
Maximum tensile strength : 520 N/mm^2
Allowable stress (S) : 130 N/mm^2
Shell inside diameter (D_i) : 1843.7 mm
Shell thickness (t_s) : 3.15 mm
Head inside diameter : 1843.7 mm
Head thickness (t_d) : 4 mm
Inside crown radius (L) : 1843.7 mm
Inside knuckle radius (r) : 210 mm
Design pressure (P) : 0.5 N/mm^2
Modulus of elasticity (E_e) : 2.08x10^5 N/mm^2
Poisson's ratio (μ) : 0.3
Weld joint efficiency (E) : 0.9

In calculations, it was assumed that the shell was of constant thickness and was free from imperfections and residual stress. Fig 2 shows the detail geometry of torispherical head.

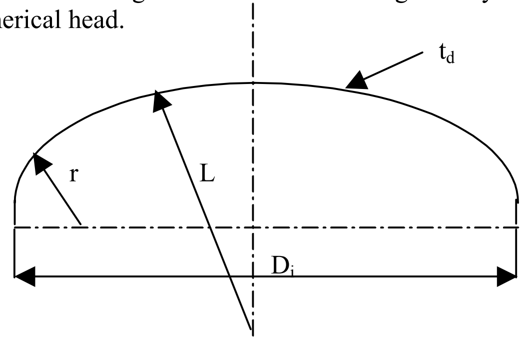

Fig 2 Geometry of torispherical head

4. DESIGN AND ANALYSIS

ASME design equations are used to calculate the thickness of the cylindrical shell and torispherical head of the jet dying machine.

4.1 Design of shell

$$t_s = \frac{PR_i}{SE - 0.6P} \qquad (1)$$

4.2 Hoop strain in shell

$$e_s = \frac{PD_0[2-\mu]}{4t_s E_e} \qquad (2)$$

4.3 Design of torispherical head

$$t_d = \frac{PLM}{2SE - 0.2P}$$

Where

$$M = \frac{1}{4}\left[3 + \sqrt{\frac{L}{r}}\right] \qquad (3)$$

4.4 Hoop strain in head

$$e_d = \frac{PD_0[1-\mu]}{4t_d E_e} \qquad (4)$$

The observation of failed dished end indicated an inward stretching of the plate. This seemed quite surprising if one sees the fact that the vessel was under pressure, on the contrary failure would have been towards outer side in the form of bulging. This surprising fact necessaciates strain calculations.

4.5 Strain ratio of shell to head

$$\frac{e_s}{e_d} = \frac{t_s}{t_d}\left[\frac{1-\mu}{2-\mu}\right] \qquad (5)$$

The strain ratio at the junction of shell and torispherical head was calculated based on original thickness (t_s = 3.15 mm and t_d = 4 mm) of failed jet dying machine which turned out to be 0.323 which being much less than unity, the dished end was stretched inside.

To avoid such kind of failure, the strain ratio at the junction needs to be increased. Accordingly revised thicknesses for shell and dished end thicknesses were selected as 5 mm and 3 mm, respectively. Based on these modified value of thicknesses, strain ratio at the junction improved to 0.6862 i.e. differential strain improved by 72.326%.

The new jet dying machine was fabricated with this revised thicknesses (t_s = 5 mm, t_d = 3 mm, D_o = 1750 mm) and tested at design pressure of 0.5 N/mm^2. The inward stretching of the end plate has not been observed there by confirming the concept of calculating the strain ratio at the junction for the design. This design may be accepted as a compromise between ideal design and practical-cum-economical point without much danger of failure. This offers reasonably lower strain level and differential strain level which will not permit inward stretching of end plate. Ideally, strain level at the junction should be same. This will results in higher thickness levels (t_s = 7.29 mm, t_d = 3 mm) of shell and larger difference in thickness of end plate and shell will create fabrication problem and operation at different hoop stress level.

5. THE FINITE ELEMENT MODEL

The software which was used for the modeling and analysis is IDEAS master series 8. Finite Element Analysis (FEA) is a process which can predict deflection and stress on the structure. It can also be used to calculate heat transfer, fluid flow or electrical field etc… Finite Element Modeling (FEM) divides the structure into a grid of “elements” which form a model of the real structure. Each of the elements are in a simple shape (such as square or triangle) for which the finite element program has information to write the governing equations in the form of stiffness matrix. The unknown for each element are the displacements at the “node” points, which are the points at which the elements are connected. The finite element program will assemble these stiffness matrices for these simple elements together to form the global stiffness matrix for the entire model. This stiffness matrix is solved for the unknown displacements, given the known forces and boundary conditions. From the displacements at the nodes, the stresses in each element can then be calculated. For detailed structural analysis, cylindrical jet dying machine was modeled and divided into finite elements.

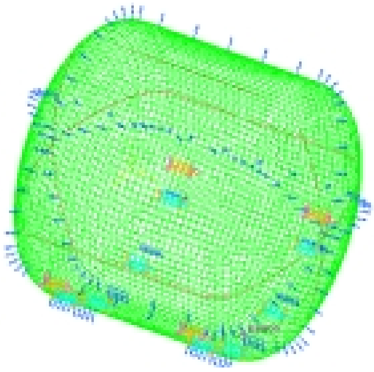

Fig 3 Finite element model of jet dying machine

For the structural analysis, plane stress, four nodded quadrilateral elements (two dimensional, degree of freedom = 8) were considered. In this analysis restraining boundary conditions were imposed on the saddle. A typical finite element model of jet dying machine with all boundary conditions and internal pressure loading is shown in fig 3

6. RESULTS AND DISCUSSION

For detailed analysis of failure of cylindrical jet dying machine, Finite Element Analysis was carried out on IDEAS-8 software. Fig 4 shows the deformed shape of the jet dying machine analyzed at the actual failure pressure of the jet dying machine.

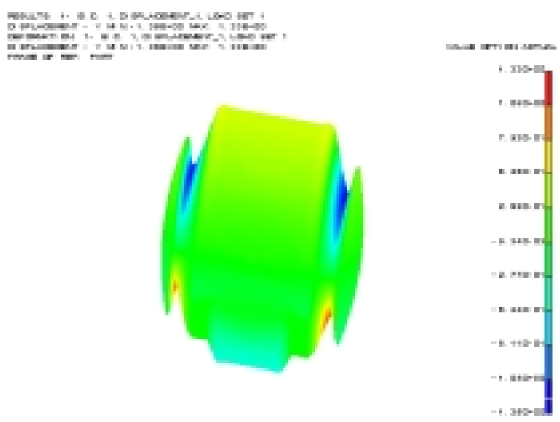

Fig 4 Displacement of jet dying machine

During the finite element analysis, it was observed that the cylindrical shell and the torispherical head of the jet dying machine have been stretched outside while the junction of the shell and the head has been stretched inside. That could be possible only just because of the low strain ratio at the junction.

Fig 5 and fig 6 show the actual comparison of the experimentally measured displacement and the displacement obtained from finite element analysis in the graphical form for different ranges of pressures. For experimental measurement of the displacement, three dial gages were fixed on the jet dying machine at different places. From these graphs one can say that as the pressure increases, the head stretches outward while the junction comes inward

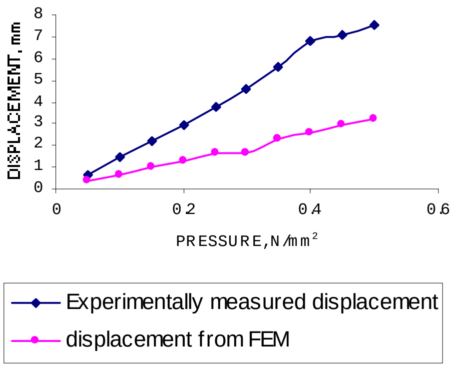

Fig 5 Displacement vs Pressure curve
(Experimentally measured displacement at the torispherical head)

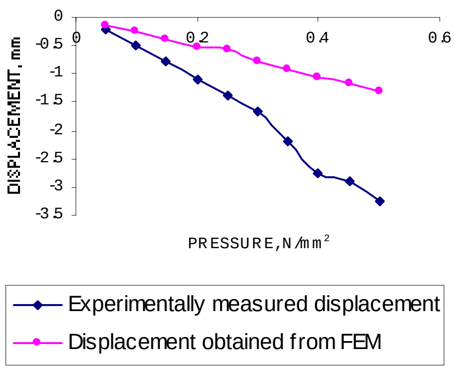

Fig 6 Displacement vs Pressure curve
(Experimentally measured displacement at the junction of the shell and torispherical head)

Fig 7 shows the results of displacement of the torispherical head obtained from finite element analysis. Fig 8 and fig 9 show the results of displacement of shell and the junction of the jet dying machine. From these figures it was observed that the shell and the torispherical head of jet dying machine stretched outward while the junction of the shell and the head stretched inwards.

7. CONCLUSION

The most possible cause of failure of jet dying machine may be due to very low strain ratio at the junction of shell and dished end which may be responsible for the inward stretching of the end plate. The failure of the pressure vessel was practically analyzed for 3.15 mm shell thickness and 4 mm dished end thickness at 0.45 N/mm^2 working pressure using finite element method which confirmed the failure due to low magnitude of strain ratio at the junction.

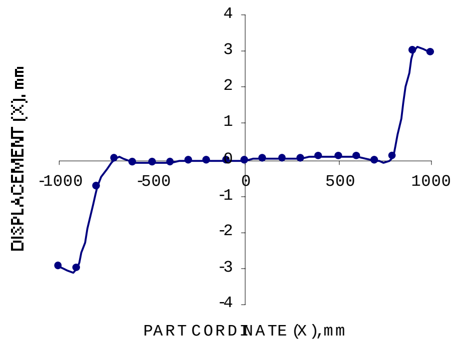

Fig 7 Displacement (x) vs Distance from origin
(results of FEM)

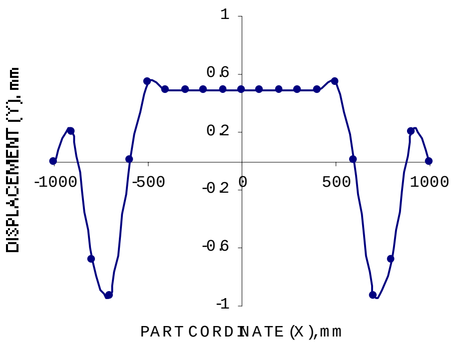

Fig 8 Displacement (y) vs Distance from origin
(Results of FEM)

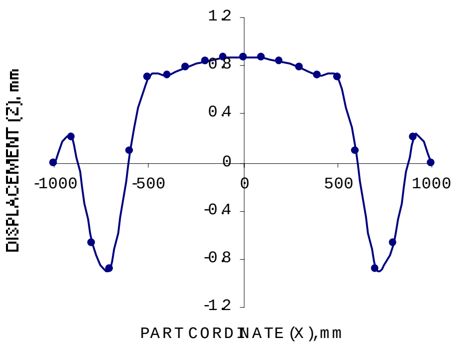

Fig 9 Displacement (z) vs Distance from origin
(Results of FEM)

Based on this realization, the new jet dying machine was fabricated according to the proposed design with strain ratio of about 0.69, shell thickness 5mm and dished end thickness 3mm and tested up to the pressure of 0.5 N/mm^2 design pressure. The absence of inward stretching in this new design conforms that the concept of strain ratio is an acceptable parameter for economical and fail-proof design of cylindrical jet dying machine. It is also believed that this concept may be extended to any category of pressure vessels where two different thicknesses of shell and dished ends are to be employed for obtaining desired level of economy associated with safety

8. REFERENCES

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9. NOMENCLATURE

Symbol	Meaning	Unit
D_i	Inner diameter of the shell	(mm)
D_o	Outer diameter of the shell	(mm)
L	Inside crown radius	(mm)
r	Inside knuckle radius	(mm)
t_s	Shell thickness	(mm)
t_d	Head thickness	(mm)
P	Internal pressure	(MPa)
S	Allowable stress factor	(MPa)
Ee	Modulus of elasticity	(MPa)
μ	Poisson's ratio	
e_s	Hoop strain in shell	
e_d	Hoop strain in head	
M	Ratio of maximum knuckle to crown stress	
E	Weld joint efficiency	