International Journal of Recent Technology and Engineering (IJRTE)
ISSN: 2277-3878 (Online), Volume-8 Issue-6, March 2020

Experimental and Numerical Determination of Critical Buckling Pressure of thin Cylindrical Shells Subjected To External Pressure

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Abstract**:** ***Thin shell structures have very high load bearing capacity, hence find wide applications in the field of mechanical engineering, structural engineering, sea shore structures, aerospace industries and nuclear engineering structures. The major failure of thin shell structures is buckling. Oil carrying pipelines, hull structures, oil tankers are few examples in which thin cylindrical shell structures fails by buckling under external pressure loading. In order to avoid the buckling failure, prediction of critical buckling pressure is important in thin shell structures under external pressure. But this critical buckling pressure depends on boundary conditions, imperfections, thickness variation of shells etc. To estimate the effects of these parameters on Critical Buckling Pressure (CBP) require a reliable experimental test rig. Hence in our proposed work, efforts are taken to develop a simple cost-effective reliable test rig to determine the effects of these parameter variations on the critical buckling pressure. For developing the test rig two important components to be designed properly namely, external cover cylinder and online pressure measurement system. The external cover cylinder with lid which contains test cylindrical shell inside should be designed in such a way that it should be leak proof and rigid so as to withstand the internal working pressure with negligible deformations. Hence, a ring and stinger stiffened cylindrical shell is taken as external cylindrical shell. The pressure variation in the test rig should be recorded online so as to predict the critical buckling pressure accurately. Hence, PC interfaced microcontroller-based pressure measurement system is developed in our proposed work. The test cylinder considered for this work is made of mild steel of size diameter 456 mm, length 456 mm and thickness 1 mm. The classical (simply supported) boundary conditions are assumed and simulated on both sides of the test cylinders. The experimental critical buckling pressures are compared with the FE results and both the results have good agreement.***

Keywords: ANSYS, Buckling, Critical Buckling Pressure (CBP), FE analysis, thin cylindrical shell,

Manuscript received on March 15, 2020.
Revised Manuscript received on March 24, 2020.
Manuscript published on March 30, 2020.
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I. INTRODUCTION

Circular cylinders under external pressure are usually in the shape of torpedoes, silos, immersed tubes, offshore drilling rigs, food storage cans, tunnels, submarine pressure hulls, missiles, surgical equipment, etc. Such vessels resist both internal or external load, but under uniform external load they can collapse catastrophically which cause failure under internal load. "Buckling phenomenon occurs when most of the strain energy, which is stored as membrane energy, can be converted to the bending energy requiring large deflections" [1]. In long thin cylindrical shells, the buckling resistance can be increased significantly by stiffening thin cylindrical shells with flanges. There are many ways of stiffening thin cylindrical shells. One method is ring stiffening at the outer shell. The overall failure of thin cylindrical shell including the stiffeners is called general instability failure. The failure between the stiffeners is called local instability. The aim of the proposed work is to experimentally predict the CBP of local instability of thin cylindrical shell without stiffener (i.e., bare cylindrical portion in between ring stiffener).

II. LITERATURE REVIEW

The collapse pressure of thin cylindrical shells were mainly influenced by the Young's modulus, D/t ratio (D- diameter, t-thickness), and yield stress of the material in the circumferential direction and wall thickness variations and initial imperfections [2] & [3]. The strength of shell structures under external pressure mainly depends on the nature of imperfections. Geometrical imperfections is the more sensitive parameter which determines the strength of thin shell structures under external pressure. The load carrying capacity is reduced significantly due to the presence of imperfections, hence the classical elastic solutions given by Timoshenko and Gere [4], Dubey [5], Brush and Almroth [6] & Flugge [7] appears to be not adequate. The general instability behavior of ring-stiffened thin cylindrical shells were studied experimentally considering weld induced and inevitable initial imperfections during manufacturing process under external pressure [8]. They also compared experimentally results with theoretical results and found to have good agreement. The CBP of thin cylindrical shells under external loading were studied by Boota et al., [9] in elastic range due to geometrical imperfections caused by manufacturing process and compared numerical and experimental results.

Guggenberger [10] in his work studied about the thin cylindrical shell (with dent) under external pressure experimentally and numerically. The experimental results are compared with FE analysis. The plastic tube considered in their study has dimensions of 400 mm (diameter) x 400 mm (length) x 1 mm (thickness). Park and Kyriakides [11] studied about the decrease in the CBP on dented stainless tube both numerically and experimentally. In their work they experimentally studied about three different size tubes of Dc/t ratio 33.6, 24.2, 18.9.

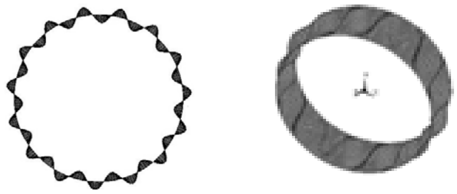
Ross and Sadler [12] carried out experimental work under external hydrostatic pressure on nine thin-walled circular cylindrical shell. Using the results obtained they developed a design chart based on thickness ratio which could be used for designing the various shell instability conditions. Barlag and Rothert [13] developed a nomograph considering the stability criteria for an orthotropic cylindrical shell under external pressure to estimate shell instability and global instability of a cylindrical shell with ring stiffeners. Cosham and Hopkins [14] analyzed the performance of pipe under static and cyclic loading with plain dents, dents on welds and dents containing defects. From experimental and theoretical analysis, they proposed a procedure to consider dent defects on pipe lines in pipeline defect assessment manual. Schneider and Brede [15] numerically studied about the collapse pressure of steel shells under external pressure considering material and geometric nonlinearities and imperfections into account. They proposed the shape of geometrical imperfections which can give collapse pressure nearer to experimentally determined buckling resistances. Instead of eigen affine imperfections they used single longitudinal imperfections pattern mode shape in their analysis. From the numerical results obtained they suggested few modifications in the regulation of Eurocode. Frano and Forasassi,[16] in their work compared the numerical, analytical and experimental works with various imperfections namely manufacturing and weld induced imperfections on buckling of imperfect thin cylindrical shell. It was ultimately concluded that the load bearing capacity is reduced significantly by the presence of imperfections. Grognec et al., [17] compared the results of geometrical imperfections and residual stresses, experimentally and numerically on CBP of shell structures under external pressure loading condition. Rathinam and Prabu [19] numerically studied about the critical pressure variation of simply supported centrally dented structural steel cylindrical shell under lateral pressure due to size variation and dent orientation. The same author in their other work (Rathinam and Prabu [20]) numerically studied about critical pressure variation, shell parameters and material yield strength variation in addition to dent size and orientation parameters. In the works of Rathinam and Prabu ([18], [19], [20]) the buckling behaviour of dented cylindrical shell is considered using elastic and perfect plastic behaviour without strain hardening effect is assumed for analysis which is a valid assumption for material like structural steel and carbon steels. It can be noted from the above literatures that geometrical imperfections effect on critical buckling pressure is dominant and it is necessary to develop a reliable test rig to determine critical buckling pressure accurately. Hence, efforts are taken in our work to develop a test rig to experimentally study the CBP of thin cylindrical shell under external pressure and its suitability of the test rig is verified by conducting experimental work using six mild steel test specimens (cylindrical shells).

III. BUCKLING ANALYSIS

Eigen buckling analysis is used to determine theoretical collapse load of perfect cylindrical shell model under external pressure load. Shell 281 eight noded quadrilateral shell element of ansys was used for modelling as it can able to model the curved surface of cylindrical shell than quadrilateral shell 181 element. This element is able to calculate membrane, bending and certain degree of transverse shear effects. Also, this element can able to handle plastic material modelling and large deformation geometrical non-linearity required for the analysis.

IV. FE MODEL VALIDATION

To determine the load bearing capacity theoretically, FE model is used in the present study and the validation model is taken from Ref. Schneider and Brede [15]. The same boundary conditions mentioned in the Ref. [15] is taken for study. The parameters of the cylindrical shell are Young's modulus = 2.1 x 105 N/mm^2, Radius = 1.9863 m, Height = 1.45 m, Thickness = 4.96 mm, Yield stress = 240 N/mm^2, Poisson's ratio = 0.3. Fig. 1(a) and (b) shows the FE model of Eigen buckling mode shape in front and isometric view. In order to determine, optimum number of elements along circumferential and longitudinal direction, several runs are carried out and it is found that 200 and 23 elements respectively on both circumferential and longitudinal direction gives accurate result compared with that of published literatures as follows. The buckling pressure obtained numerically by Schneider and Brede [15] is 111.2 kN/m^2 and the buckling pressure obtained by FE Eigen is 110.6 kN/m^2. The buckling pressure obtained by present analysis closely matches with the buckling pressure given in Ref [15].



(a) Front view and (b) Isometric view

Fig. 1 FE Eigen buckling mode shape for validation of FE model [15]

(a) Front view and (b) Isometric view

Fig.2 FE first Eigen buckling mode shape of test cylinder

Retrieval Number: F9704038620/2020©BEIESP
DOI: 10.35940/ijrte.F9704.038620
Journal Website: www.ijrte.org

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V. TEST CYLINDRICAL SHELL CONSIDERED FOR PRESENT STUDY

The internal cylinder is made of mild steel, easily rolled and welded. Six test cylinders of mild steel material with dimension 1 mm (thickness) x 456 mm (diameter) x 456 mm (length) is taken for study. The mild steel sheet metal of 1 mm is rolled and welded longitudinally. It is covered on both top and bottom ends by a circular plate and on the entire surface of the test cylinder the pressure is applied.

The CBP obtained from FE Eigen buckling analysis with simply supported boundary condition is 2.21 bars and the mode shape obtained is shown in Fig. 2.

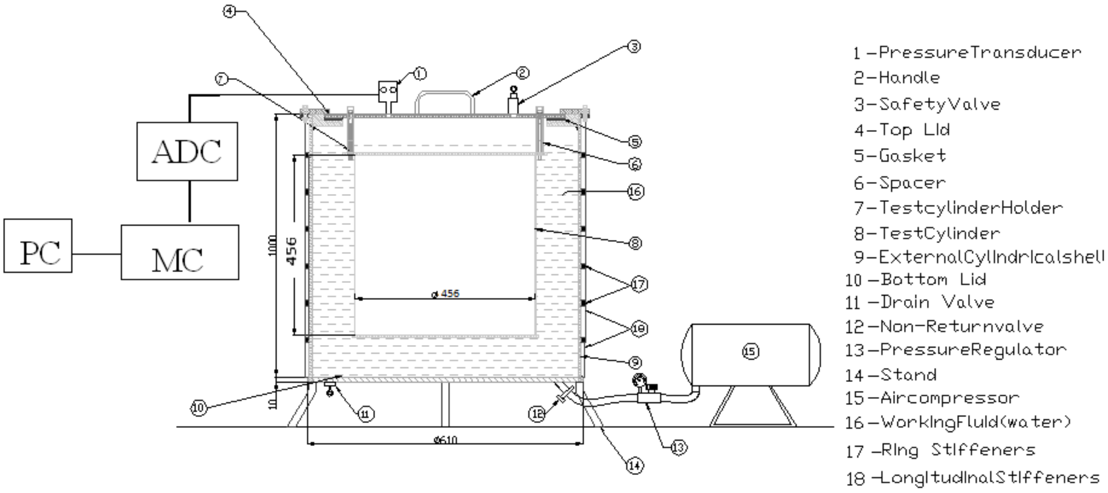

Fig.3 Schematic diagram of test rig

VI. DESIGN OF TEST RIG

Initially for safe design of test cylinder, design pressure is taken as 8 bars. Considering the design pressure, the design parameters required for the test rig is calculated. Fig. 3 shows the 2D view of the test rig. The experimental setup consists of external stiffened cylindrical shell which can resist 8 bar pressure. It has two lids at both ends of the cylindrical shell. A holding arrangement is provided at the top of the test cylinder to hold the test cylinder. A drain valve is located at the bottom of the lid to drain the water. A pressure transducer is used to measure the online pressure. A non-return valve is used to impart external pressure to the test cylinder. A relief valve is provided to release the excess pressure. An air compressor with a pressure regulator is provided to apply pressure to the test cylinder via non-return valve. The external cylindrical shell is stiffened with both longitudinal and ring stiffeners in order to increase its rigidity. The top lid is clamped to the external cylinder by leak proof fastener arrangement. The bottom lid is welded permanently to the bottom of the external cylindrical shell. The top lid has provisions to mount pressure transducer to measure the pressure inside the test rig and a safety valve. It also provided with two handles for convenience of lifting the lid. The bottom lid has a provision for drain valve and also provided with non - return valve to avoid the back flow of water from the test rig. The whole test rig is placed over a stand. The non–return valve is connected with air compressor through flexible hose. Since the diameter of the test cylindrical shell is 456 mm the external cylinder diameter is taken as 600 mm so as to get annulus clearance 72 mm around the test cylindrical shell. The design load for external cylinder is taken as 8 bars. The thickness of the outer cylinder is 1.2 mm and its length is 1000 mm. Top lid has provision to mount safety valve and pressure transducer and also has provision to hold the test cylinder. Bottom lid has provision for drain valve and a non – return valve which in turn connected with air compressor through a pressure regulator via braided hose. The braided hose can withstand 10 bars. Stiffeners are placed orthogonally on top and bottom lids. Test specimen holder consists of bolting screws with nuts, spacers and a mild steel circular plate on which top cover of the test cylinder is welded. The size of the spacer decides submersible height of the test cylinder in water. The pressure regulator is used to regulate the pressure of compressed air from the compressor to the experimental setup. The micro controller-based pressure measurement system is used in this work. Stainless steel sanitary diaphragm type pressure transducer of input 10-30 V DC, Pressure 0-10 bar and output 1-5 V is used to measure the buckling pressure and it is fitted on the top lid of the external cylinder. The 8051 family of microcontroller is used in this work because it has facility to interface the output from the pressure transducer. The pressure transducer is in turn connected to the ADC thereby the pressure readings are converted into digital value and it can be read in computer.

VII. EXPERIMENTAL PROCEDURE ADOPTED

Initially the test cylinder is mounted with top lid using four studs which are welded at the bottom surface of the top lid. For conducting the experiment, the working fluid is taken as water because it is an incompressible fluid, easily and cheaply available. Water is poured up to 1/3rd height inside the annulus region between the test cylinder and external cylindrical shell.

It is noted that the test cylinder is completely immersed inside the test rig so that pressure is applied equally in all directions of the test specimen. After filing the water, the top lid with the test cylinder is clamped in its position by eight bolted joints and gasket. Uniform clamping pressure is applied on all bolts to arrest the leakage. The air compressor is switched on closing the outlet of the air compressor storage drum. The pressure is allowed to build up to 7 bars in the storage drum of the compressor. Then the air compressor is switched off. Gradually the outlet of the air compressor is opened and the pressure is released to the non-return valve. Hence the pressure is raised inside the test rig and the test cylinder experiences hydrostatic pressure on all directions.

The pressure inside the test rig is measured by the pressure transducer attached to the test rig. On reaching the limit pressure condition there is a sudden drop in the pressure with a metal knock sound. The pressure drop is noticed in the online pressure measurement. The sudden drop in pressure is noted as the CBP of the cylindrical shell.

VIII. RESULTS AND DISCUSSIONS

To check the ultimate strength of the test rig developed, the CBP of mild steel thin cylindrical shells are determined using the test rig and compared with finite element results and the same is shown in table 1. From the table it is noted that FE results closely matches with experimental results. The deviations between the FE analysis and experimental results may be due to geometrical and material imperfections present in the test cylindrical shell. The failure mode shape obtained by FE analysis are shown in Fig. 2 consists of six circumferential lobes and one longitudinal lobe. Similar failure pattern of six circumferential lobes and one longitudinal lobe is obtained from experiments also and it is shown in Fig. 4 & 5.

Fig.4 Test cylinder 1

Fig.5 Test cylinder 4

Table- I: Experimental and FE Eigen buckling pressure values

Test Cylinder	FE Eigen buckling pressure value in bar	Experimental buckling pressure value in bar	Percentage error
1	2.21	2.09	5.42
2	2.21	2.09	5.42
3	2.21	2.11	4.52
4	2.21	2.11	4.52
5	2.21	2.12	4.07
6	2.21	2.08	5.88

IX CONCLUSIONS

The suitability of developed test rig to determine the CBP of the cylindrical shell is validated. Further from the experiments conducted it is found that CBP obtained reasonably matches with the FE Eigen buckling pressure. The maximum deviation between the FE analysis and experimental results is 5.88. The deviation may be due to material and geometrical imperfections present in the test cylindrical shells taken for study.

ACKNOWLEDGMENT

This work is carried out by the financial support of UGC -Minor Research Project. F.No. MRP-6752/(UGC/SERO)

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DOI: 10.35940/ijrte.F9704.038620
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