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Design and Finite Element Analysis of Hydrostatic Pressure Testing Machine used for Ductile Iron Pipes

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Abstract

The present work deals with the design, construction and demonstration of a hydraulic pressure testing machine. The machine can be used universally to test a centrifugally cast iron pipe (from 80 to 600 mm diameter) under the applied hydraulic pressure. The developed machine can handle a maximum of 5 MPa pressure for 600 mm diameter pipe. The machine was designed using mild steel flats, plates and bars and was tested under maximum design pressure of 3.5 MPa. Stiffeners were used in proposed design, instead of thick end plates as used by earlier researchers. The modeling of machine was carried out using commercially available SolidWorks 14 and the validation of design was carried out using FE package ABAQUS 6.13. Several trials were carried out on design pressure and precise results of von-Mises stress were obtained well within the specified yield stress limits and significant decrease in total deformation of stiffened end plates. Added stiffeners also resulted in reduced material requirement, weight, higher cost effectiveness and being light in weight helps in easy handling at site.

Keywords: Hydrostatic pressure, finite element analysis, thin walled pipes, hydrostatic leak test, ductile iron pipe testing machine, Abaqus, solidworks, stiffeners

1. Introduction

A usable water supply network is a system of engineered hydrologic and hydraulic components, consisting of regions of various landforms and different environments. It is a means of delivery of water from the source to a point of treatment, water purification plant, transmission network and distribution through piping from storage to consumption areas. The product delivered to the point of consumption is generally fresh water or drinking water. Water is transferred with high pressure, provided in number of ways by pumping or by gravity feed from water source (such as a reservoir or a water tower) at a higher elevation.

Pipelines form a unique mode of transportation (Riyanto & Chuie-Tin Chang, 2009; Riyanto, 2009; Zhang, Feng, & Qian, 2009). They can move large type of certain types of commodities, mainly fluids, over a long distance at relatively lower cost. To ensure safe operation of system, laying and performance of pressure piping must meet the stringent quality standards. Nowadays water mains are made up of variety of materials such as concrete, steel and ductile iron. The operations are environment friendly, dependable and continuous. Pipelines can be laid on variety of terrains without much difficulty (Blizzard, 1994; Subramanya, 1998) Water mains are the important component of any water distribution network and failure of these could pose a lot of problems to users. Beside the loss of treated water and pipe, contamination does occur and this may cause serious health hazard and lead to epidemic (Fawell & Nieuwenhuijsen, 2003). Contaminations sometimes also occur due to the rusting of water supply pipes if they are not of any adequate corrosion resistant material. Corrosion, sometimes known as material wastage, is the root cause of leak in water mains. Crack like defects develop on pipe during the application of internal pressure and sometimes due to manufacturing and installation defects and also sometimes with time, material deterioration occurs as a result of continued operation (Yamini & Lence, 2010). Therefore, the pipes are tested for any leakage by applying adequate pressure before being laid for actual course of operation.

Several big failures of water mains having material as cast iron had been noted in history. There were several failure modes of which most common were blowout holes, circumferential cracking, bell splitting, longitudinal cracking, and bell shearing and spiral cracking. Causes of these failures were corrosion, manufacturing flaws, human error during installation and excessive forces during actual course of operation (Makar, Desnoyers & McDonald, 2001; ASA, 1953; Makar & Rajani, 2000; Makar, 2000).

Most common method used for hydrostatic pressure test of pipes is by using blind flanges. Flanges act as side plates and are used to create and maintain internal pressure. Flanges having same diameter as that of pipe were prepared and then on field test would be conducted. For conducting on field leak proof pressure test on pipes of different diameters, flanges of different diameters would be required. This would create a lot of problems. In order to solve this problem a universal machine would be required which could help in easy leak proof on field testing of pipes.

Earlier a machine had been designed by CSIR-AMPRI, Bhopal (Earlier Design) for hydrostatic pressure testing using ductile iron pipes Figure 1 (Morchhale, Goel & Yagneswaran, 2008). Using this machine, pipes manufactured by Indian Iron and Steel Company, Kolkata were tested for leak proof under the applied hydrostatic pressure. In proposed design the end plates consist of thin base plates with stiffeners built over it which would be much useful for handling the machine with ease. An effort was made to control the stress within the specified yield stress limit and at the same time to optimize the weight of whole machinery. Here, Dassault Systems software Solidworks was used for modeling and finite element analysis software Abaqus (Gardner, Vijayaraghavan & Dornfeld, 2005) was used as a solver of proposed design.

Figure 1. AMPRI designed machine

1.1 Earlier Design

Earlier designed machine consists of flat end plates of 46 mm thickness, twelve tightening studs of 36 mm diameter and a mounting stand consisting of a channel section as shown in Figure 1. End plates had major contribution towards the weight of whole assembly but due to its insufficient thickness, excessive deformation takes place while application of hydrostatic pressure.

1.2 Proposed Design

The main objective of proposed design as shown in Figure 2 was to control the deformation of end plates and stress within the specified yield stress limit i.e. 250 MPa.

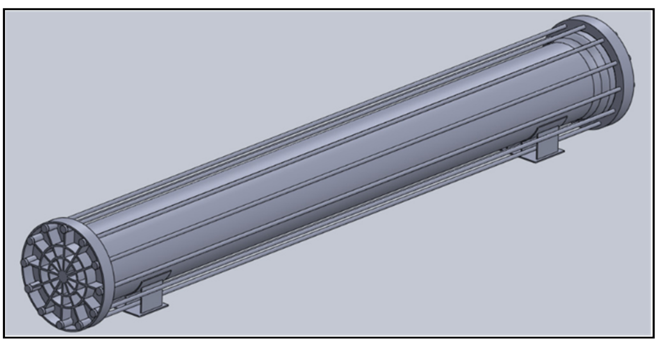

Figure 2. Proposed design of pipe testing m/c

The proposed assembly design consists of 12 tightening studs, 2 stiffened end plates of different dimensions as shown in Table 1, and a mounting stand consisting of a channel section to hold the assembly. Stiffeners were used instead of heavy thick end plates to reduce the weight and to minimize the deformation on application of hydrostatic pressure. Several trials were carried out by varying the thickness of stiffeners built on end side plates.

Table 1. Dimensions used in earlier and proposed design for FE analysis

Parameters	Earlier design	Proposed design			
		Stage 1	Stage 2	Stage 3	Final
Diameter of end plate (mm)	890.6	890.6	890.6	890.6	890.6
Thickness of base plate (mm)	46	10	10	10	10
Height of 15 mm thick stiffeners (mm)	-	50	70	90	110
No. of stiffener rings (mm)	-	-	1	2	2
Central diameter of stiffener ring 1 (mm)	-	-	315	315	315
Central diameter of stiffener ring 2 (mm)	-	-	-	528	528
Length of studs (mm)	6213	6241	6281	6321	6361
Diameter of studs (mm)	36	32	32	32	32

Stiffeners are secondary plates or sections which are attached to beam webs or flanges to stiffen them against out of plane deformations. The stiffeners do not let much stress to develop on base plate and instead distribute it across and along its height (Ismail, Fahmy, Khalifa & Mohamed, 2014; Ahmed & Rameez, 2003). The height of stiffeners varied from 50 mm to 110 mm. In stage 1, 12 straight stiffeners were joined to a cylindrical central surface of 100 mm diameter and 12 hollow cylindrical surfaces to hold the tightening studs. The thickness of base plate and height of stiffeners were taken as 10 mm and 50 mm respectively. In stage 2, another ring of diameter 315 mm was added to the proposed design of stage 1 at central location. The height of stiffeners was also increased to 70 mm. In stage 3, another stiffener ring was added of diameter 528 mm was added at central location and total height of stiffeners was then increased to 90 mm. In stage 4 only the total height of stiffeners was increased to 110 mm as shown in Figure 3.

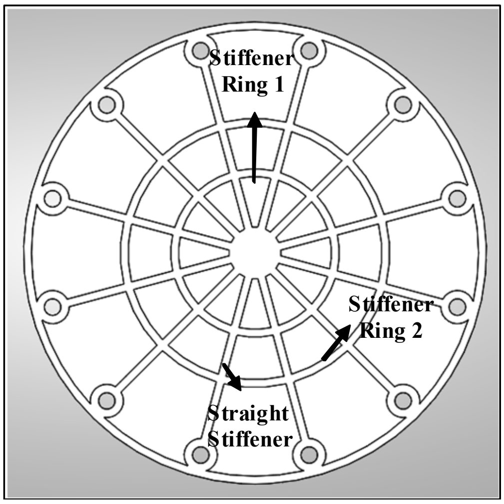

Figure 3. Front view of final stiffened end plate

2. Definition

The ductile iron pipes used in water pipelines are considered as thin walled as shown in Figure 4a, as the ratio of wall thickness to radius was less than 0.1. When the ends of the pipe are closed and pipe is subjected to an internal pressure ' p ', there are various stresses develop in the pipe of which most common are longitudinal stress, hoop or circumferential stress and radial stress. Among these stresses, radial stress is almost considered to be negligible and longitudinal and hoop stresses are considered to be constant across the wall of thin walled pipes (Kumar & Moulik, 2004; Nagesh, 2003; Praneeth & Rao, 2012; Patil, 2013; Ibrahim, Ryu & Saidpour 2015).

Considering that the pipe ends are closed and the pipe is subjected to an internal pressure, the pipe may fail in longitudinal direction. Elements subjected to this type of failure would be subjected to stress and the direction of this stress is parallel to the longitudinal direction of the pipe as shown in Figure 4b, and is given by

$$\sigma_l = \frac{pd}{4t} \tag{1}$$

The effect of hoop stress as shown in Figure 4c may split the pipe into two halves. The failure of pipe in two halves is in fact possible across any plane, which contains diameter and axis of pipe. Elements resisting this type of failure would be subjected to stress and direction of this stress is along the circumference and is given by

$$\sigma_h = \frac{pd}{2t} \tag{2}$$

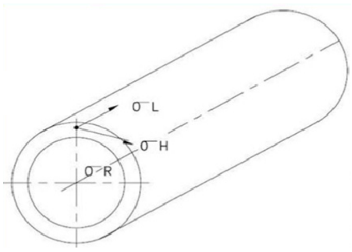

Figure 4a. Thin walled cylinder

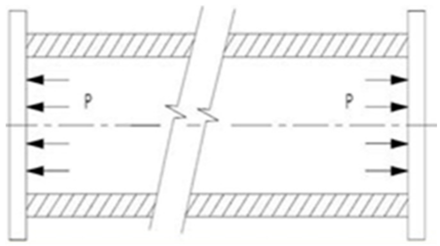

Figure 4b. Longitudinal stress

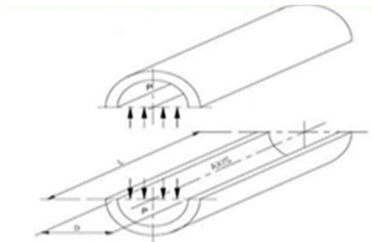

Figure 4c. Hoop stress

Radial stress is in direction coplanar but perpendicular to the symmetry axis. Radial stress is only considerable in thick walled cylinders. The radial stress for thick walled pipe is equal and opposite to the gauge pressure on internal surface, and zero on external surface and is given by

$$\sigma_r = p \tag{3}$$

Where,

Internal diameter of given pipe $= d$

Pressure applied $= p$

Wall thickness of given pipe $= t$

3. Finite Element Analysis of the Machine

The 3D-CAD geometries of all above mentioned designs were constructed and assembled in Solidworks 14 and then imported in FEA solver Abaqus in parasolid file format (*.x_t). The FEA scheme involved was of implicit type as the static analysis of all the proposed designs was done and more importantly the deformation of models does not vary with time. The design was optimized on maximum diameter of pipe i.e. 600 mm under different pressures - 1.5 MPa for water line pressure, 2.5 MPa for quality check on full length of pipe in field, 3.5 MPa for pipe testing at manufacturing work site and finally factor of safety was considered as 2 and machine designed pressure was taken as 5 MPa (IS 1536, 2001; IS 800, 2007).

The following assumptions were made in the stress analysis of machine: homogenous isotropic elasto-plastic material having non-linear behavior and steady static loading condition. These effects were not taken into consideration: external pressure, moments and the roughness of internal surface of pipe.

Centrifugally cast iron pipe of 600 mm diameter was taken for analysis. Cast Iron is considered to be ductile, brittle, damping and more importantly it absorbs vibration and noises better than any other metal. It is slightly destructive upon drilling, produces powder, and does not bend or dent because it is very hard, but it breaks easily unlike steel. It has more compressive strength therefore it can handle high pressure and can be used in pipelines. The machine components were considered of mild steel because of its high tensile strength and machine ability. Mild Steel is the most popular and the cheapest form of steel available. It is common in most of the machine elements because of its ductile properties.

This type of steel contains less than 2 percent of carbon, which makes it magnetize very well. Mild Steel is inexpensive and therefore is used at place where large amount of steel is required. Mild Steel does not have great amount of strength, making it unsuitable to build girders or structural beams. Mild Steel has weak resistance towards rusting and so it must be kept sealed or painted with oil, grease or paint regularly in order to keep it safe from rust. Mild Steel is a soft material and so it is easy to weld as compared to other high carbon steels. Mild Steel is a variant of hard steels, which makes it much less brittle and enhances its flexibility (Seidu, 2014; William, 2007).

The proposed assembly of whole pipe testing machine consist of 12 similar tightening studs of 32 mm diameter, 2 stiffened end plates of different dimensions and mounting stands. Undercut of 2mm was given on internal face of both side plates for proper sitting of pipe collar and tail end. In assembly all parts were made independent of each other, so that different mesh sizes could be assigned to the different instances for more precise results. The coefficient of friction was considered to be 0.3 wherever the frictional contact was established due to the application of hydrostatic pressure on internal surfaces of pipe and side plates. Two major frictional interactions developed were between:

(i) Plates and pipe
(ii) Plates and tightening studs

Boundary conditions were applied on the full geometry of machine. Internal pressure of 1.5 MPa, 2.5 MPa, 3.5 MPa and 5 MPa were applied as uniformly distributed load on inner walls of end plates and pipe. To avoid the change in shape and elongation of pipe and studs, the displacement value maintained in subsequent steps was considered to be zero in all the three directions (x, y and z) at center. To avoid the bending of studs due to the application of pressure, the rotation in studs was also considered to be zero in the 3 principle planes (xy, yz and zx).

The mesh quality was conclusively determined based on the following factors rate of convergence, solution accuracy and CPU time required to solve the given problem. The greater the convergence better will be the mesh quality and so more accurate results can be achieved easily. Solution accuracy can be achieved on the basis of mesh quality only. Refining meshes at certain areas of geometry where the gradients are high, thus increasing the fidelity of solution in the region. Thus, the mesh quality is dictated by the required accuracy. CPU time is a necessary yet one of the most undesirable factor as time will generally be directly proportional to the number of elements. Tetrahedron elements has high absolute residual value, high pressure drop and so more number of iterations and more runtime are required to get the accurate results. The mesh size of different instances was decided on the basis of size and shape of instance and the type assigned was quadratic tetrahedron elements (C3D10) due to its easy adjustment quality with complex shapes on bends, corners and crevices. Mesh refinement of model was done by varying the number of elements in order to get optimum and more precise results. Local seeds were assigned on stiffeners for increasing mesh quality and reducing distorted elements. Final mesh of proposed design is shown in Figure 5, and size & nos. of elements are shown in Table 2.

The job was then submitted for analysis and results were obtained on different internal pressures and analyzed.

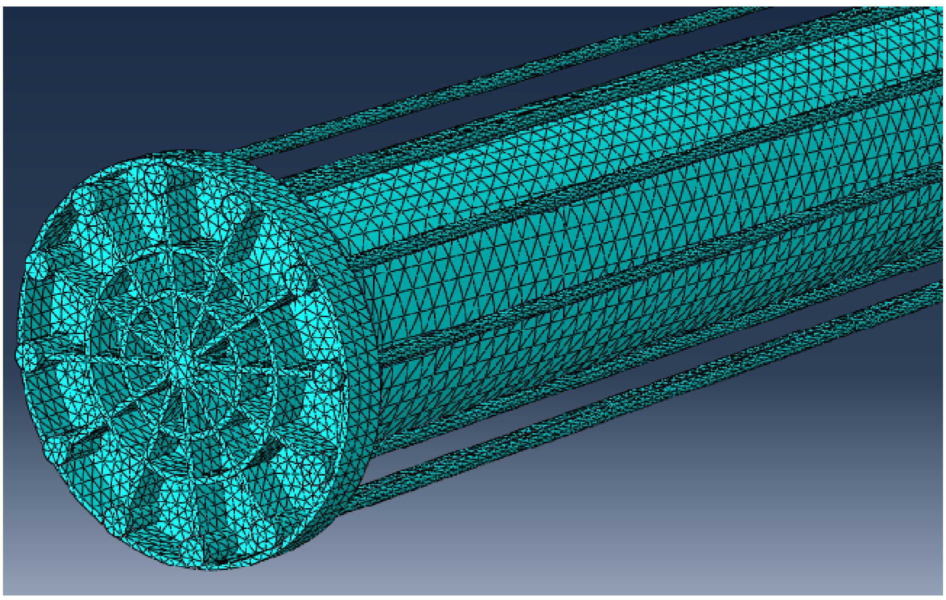

Figure 5. Meshed components of proposed design

Table 2. Size and number of mesh elements assigned to components.

Component	Earlier design		Proposed Design	
	Elements size	Nos. of element	Elements size	Nos. of element
Pipe head end plate	30	20753	30	16503
Pipe tail end plate	30	23193	30	16878
Pipe	50	33160	50	33160
Tightening Bolts	30	5774	30	6107

4. Results and Discussion

The finite element analysis of earlier as well as proposed designs at several stages was done on full scale model and hence the results were obtained. The loading was assumed to be steady within the static structural analysis and two output variables were extracted: (i) the von-Mises stress which takes into consideration of all the 3 components of principal stresses and derives a maximum value upon yield strength formula and (ii) the total deformation when operated under full loading conditions.

All the results demonstrated that the maximum stress, strain and total deformation have an involvement of side flat plates either directly or indirectly irrespective of the design taken into consideration. Elements at the inner wall of machine end plates would experience compressive deformation while elements at the outer wall would experience tensile deformation, causing the change in shape of assigned 3-D tetrahedral mesh elements and hence resulting in maximum stress and strain.

It can be observed from Table 3 that on application of 3.5 MPa internal pressures the von-Mises stress and total deformation in different stages decreased with increase in number and height of stiffeners. Minimum deformation as well as von-Mises stress was observed in final proposed design.

Table 3. FEA results of end plate optimization

Parameters	Proposed design at 3.5 MPa pressure			
	Stage 1	Stage 2	Stage 3	Final
von-Mises stress (MPa)	1144	620	328	240
Total deformation (mm)	9.8	5.3	3.7	3.2

After finalizing the proposed design, further FE Analysis of earlier design as well as of Proposed Design was performed at 1.5 MPa, 2.5 MPa, 3.5 MPa and 5 MPa and results were reported in Table 4.

Table 4. FEA results

Particulars	Pressure (MPa)	Earlier design	Proposed design
von-Mises stress (MPa)	1.5	129.6	103.4
	2.5	215.1	172.1
	3.5	300.8	240.8
	5.0	428.9	343.7
Hoop stress (MPa)	1.5	129.7	106.7
	2.5	215.2	175.9
	3.5	300.8	245.8
	5.0	429.0	350.5
Longitudinal stress (MPa)	1.5	85.1	98.5
	2.5	161.9	162
	3.5	229.8	226.1
	5.0	347.0	321.1
Displacement (mm)	1.5	2.03	1.38
	2.5	3.35	2.31
	3.5	4.67	3.23
	5.0	6.64	4.61

The von-Mises stress is the maximum stress above which the material starts yielding and as a result deformation occurs. It can be observed from contour plot in Figure 6a which is a FE Analysis of earlier design, which Von Mises

Stress is concentrated at centre of end plates which could lead to high deformation resulting in cracks and failure of hydrostatic pressure test machinery. In order to avoid such failures a design of machine was proposed which a base plate with stiffeners had built over it constituting a full end plate.

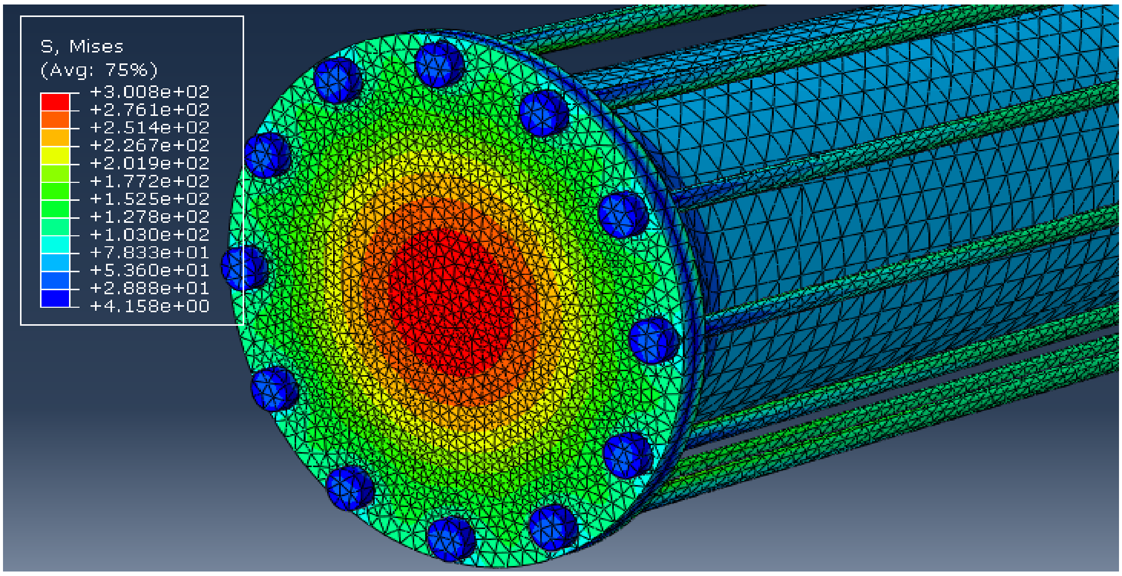

Figure 6a. von-Mises stress at 3.5 MPa pressure in earlier design

The stress was uniformly distributed along and across the height of stiffeners and no permanent deformation was observed and the stress was within the elastic limits as shown in contour plot in Figure 6b. The graph shown here in Figure 6c also displays the difference between the stress in earlier design and in final design proposed. It can be seen that the stresses developed on the machinery till 3.5 MPa pressure were within the specified yield stress limits of 250 MPa, but in case of 5 MPa applied pressure at very few locations, stress was beyond the yield limit which can be ignored considering the overall shape and size of base plate.

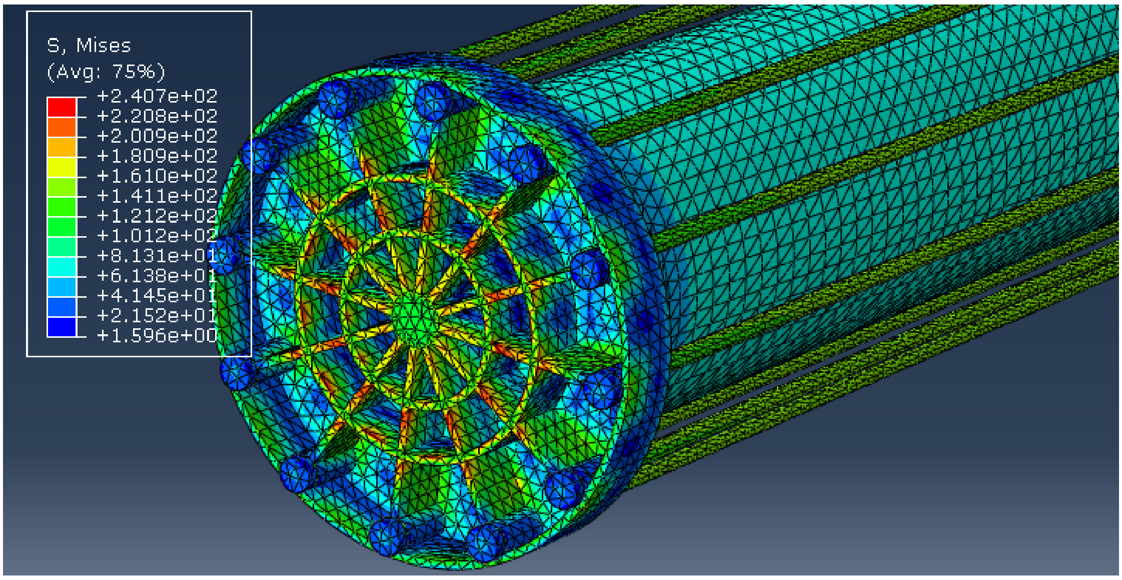

Figure 6b. von-Mises stress at 3.5 MPa pressure in proposed design

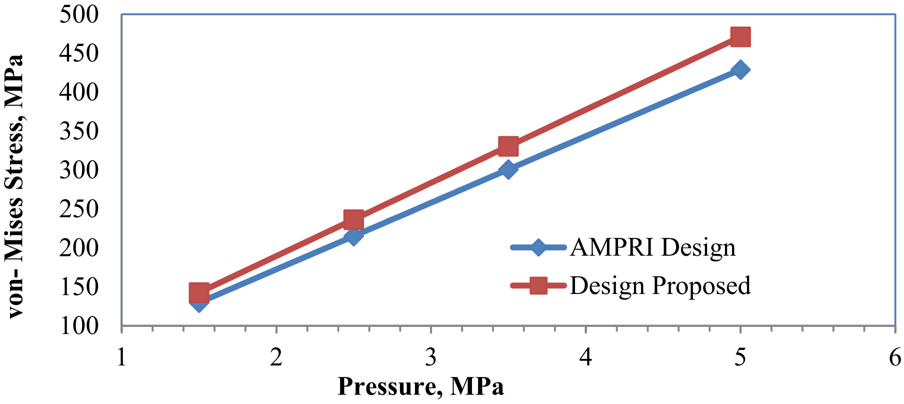

Figure 6c. von-Mises stress vs. pressure applied

Maximum hoop stress and longitudinal stress varying with increasing internalpressure can be observed from graph in Figure 7 and Figure 8 respectively. Hoop stress acts along the circumference of cylindrical structures. Longitudinal stress acts along the longitudinal axis of cylindrical structures. Of all the mentioned designs, minimum hoop stress and longitudinal stress were observed in final design proposed having end plate thickness of 120 mm.

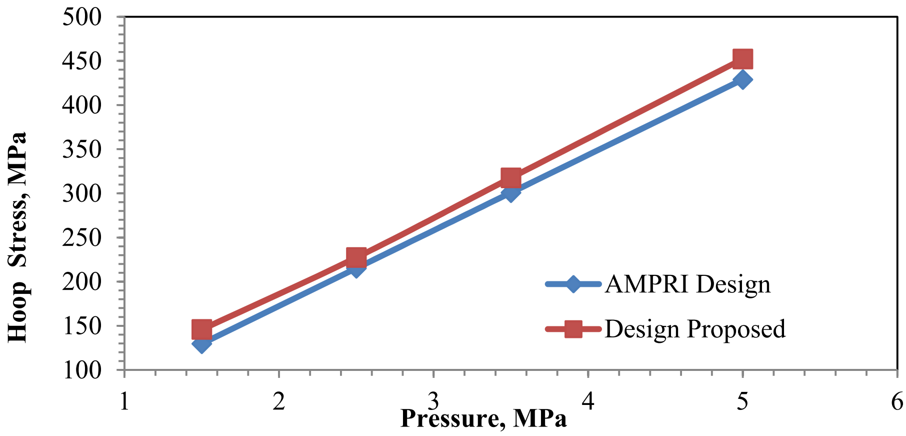

Figure 7. Hoop stress vs. pressure applied

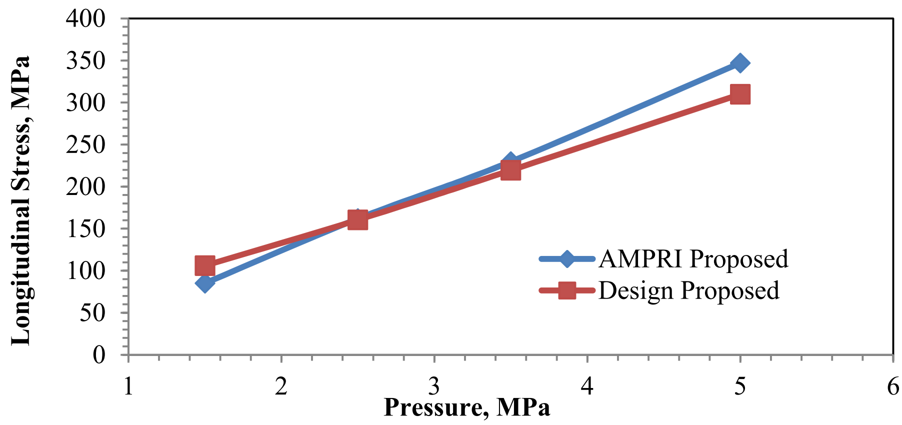

Figure 8. Longitudinal stress vs. pressure applied

It was observed in earlier design in Figure 9a that at 3.5 MPa of applied internal pressure the Total Deformation was concentrated at center of end plates which could be more harmful for long term use of machineries and could lead to permanent deformation in the form of cracks and crevices. This permanent deformation can be defined as change in shape of solid body without fracture under the action of sustainable force, there may be small changes in density of crystals due to plastic deformation.

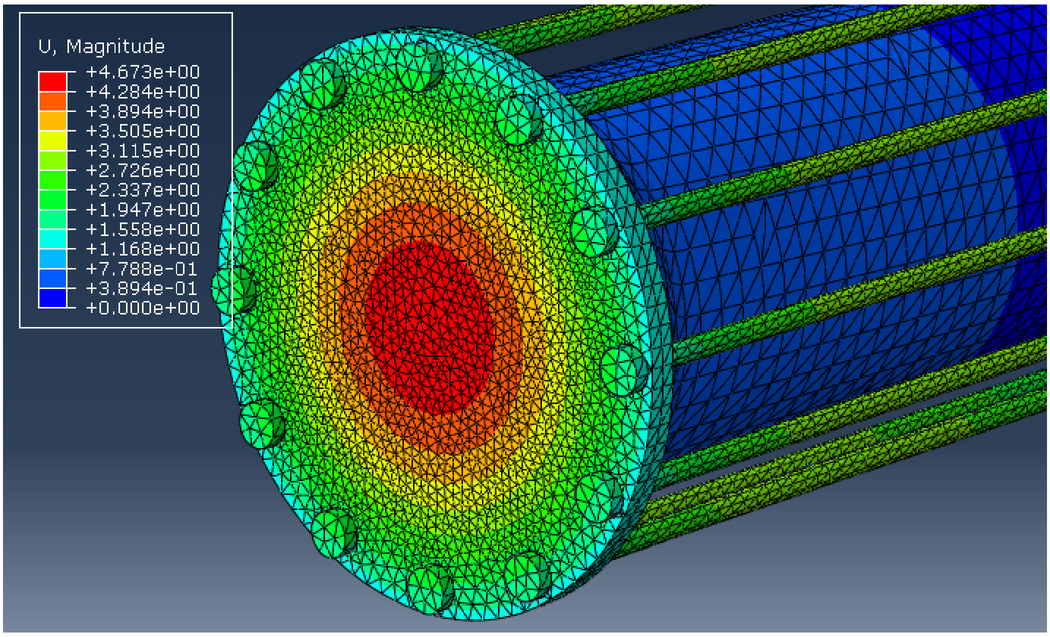

Figure 9a. Total deformation at 3.5 MPa pressure in earlier design

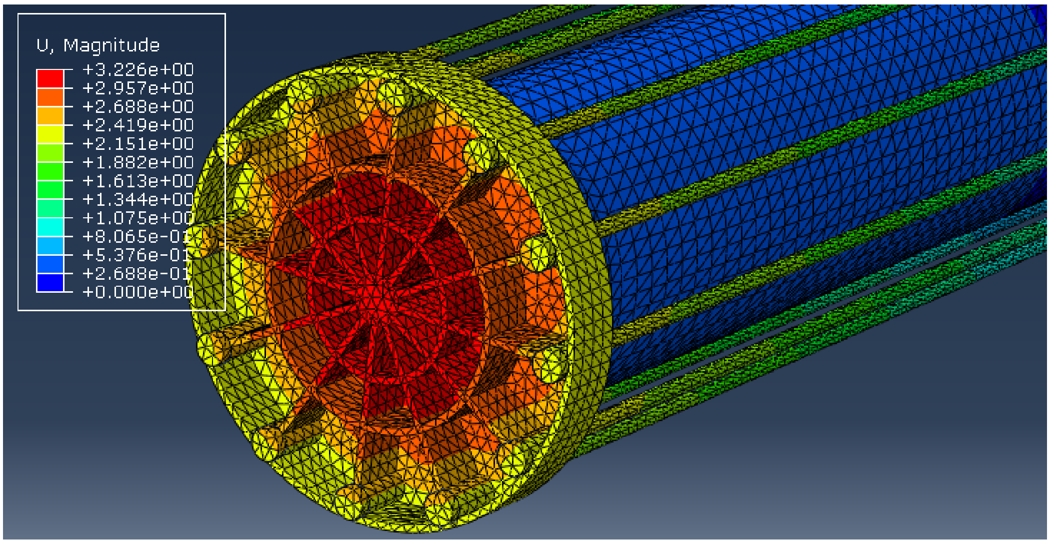

Figure 9b. Total deformation at 3.5 MPa pressure in proposed design

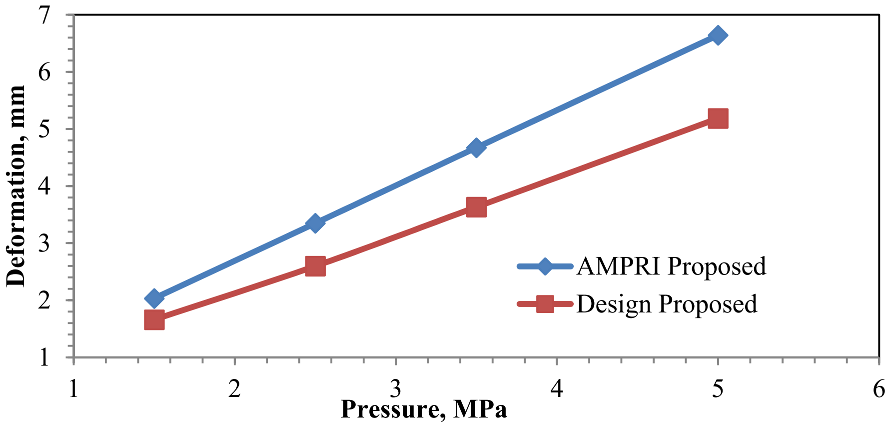

Figure 9c. Total deformation vs. pressure applied

Figure 9b shows the deformation of final end plates at 3.5 MPa of applied internal pressure in proposed design. Less deformation of stiffened end plates was observed as compared to the earlier design. Figure 9c shows the graph of total deformation of assembly with applied internal pressure.

4.1 Weight Optimization

Other priority during the whole research work was weight optimization of machine for easy installation. Several efforts were made at different stages to control the weight as much as possible in order to handle the machine easily for in field operations. Total reduction of 14.92% in weight was observed of final proposed design as compared to earlier design.

Weight reduction process was carried out at several stages by increasing the height and number of stiffeners on end plates. During the whole research work, first priority was to bring the stress within the suggested yield stress limit and then to control the deformation at applied design pressure. The corresponding weight of components in final design was compared to that of earlier design in Figure 10.

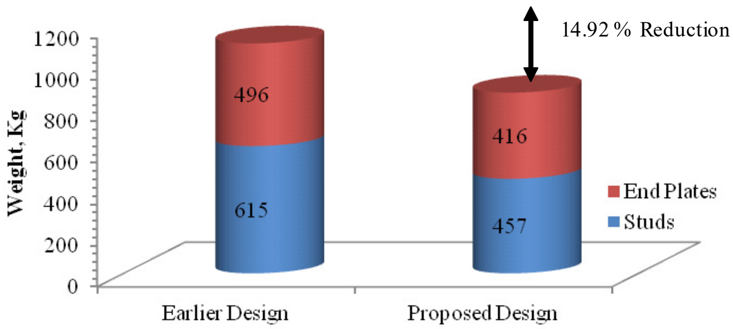

Figure 10. Weight of machine components

5. Conclusion

It can be concluded from results that wherever and whenever weight optimization and reduction in deformation of components on application of pressure or force is required, the best way is to introduce stiffeners of appropriate height and at appropriate place instead of thick flat plates. Stiffeners were welded to thin flat end plates in order to control its out of plane deformation. It is more important to position the stiffeners at right place rather than just increase its number or the dimensions. Added stiffeners reduced the deformation of end plates, material requirement, weight and increased cost effectiveness, age and usefulness of machine at site. More importantly, stiffeners always have larger impact in stress control than the flat plates.

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