FINITE ELEMENT ANALYSIS OF BALL VALVES

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Abstract. *Finite Element Analysis is accepted in international standards, as an alternative method of designing pressure vessels (including valves). Valves are frequently subject to repeated mechanical and thermal load. When the maximum load varies between yield load and plastic collapse load the valves body can deforms plastically. What happens depends on the load and geometry. Is necessary to integrate FEA into a design cycle as part for safety, quality, reliability and cost effectiveness improvements . This article gives methodology of ball valve design by using CAD design and FEA verification at maximum loading pressure. The main purpose is structural analysis to be carried out of determining stress and strains developed in the valve body and other elements.*

Keywords: *ball valves, Finite Element Analysis*

1. INTRODUCTION

Trunnion ball valves are used especially for large size equipment for the range of pressure and have a more complex construction. In this case the ball is fixed, being guided into the body and the bottom cover through the drive shaft and the sphere support.

The metallic seat has the possibility to slide in the bore processed in the body. A ring of PTFE, Viton or PTFE with hardeners (glass, carbon) is pressed into the seat . Soft sealing material meets the good seal with an easy handling under normal temperatures (<120^{0}C).

For the adequate strength, the design must be based on a calculation method supplemented by an experimental design method or experimental design method, without calculation, if the product of the maximum allowable pressure PS and the volume V is less than 6000 bar x liter or the product PS x NPS is less than 3000 bar [1].

2. DESIGN AND EVALUATION METHODOLOGY

The studied equipment is a closure ball valve NPS 500 mm (10 in), NP 25 (class 150) with movable seats, full bore, split body, double-blocking construction, and discharging inline, supplementary sealing injection, purging and venting, *fire-safe* and antistatic (fig.1).

The design of valves requires, at first, their dimensioning: calculation of wall thickness of envelope under pressure, the dimensions of sealing non-metal / metal and the fire-safe sealing (metal / metal), dimensioning the drive shaft and shutter actuators (lever or gear reducer quarter turn) [3].

Because of the complex distribution of the loads or the complex geometry of the components, calculation methods are not sufficiently precise.

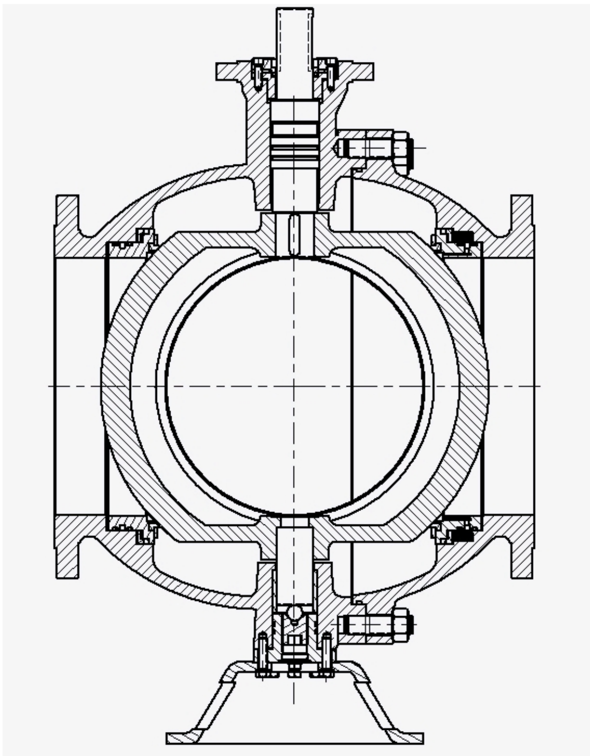


Fig.1.Trunnion ball valve NPS 500, NP 25 (with ball in CLOSED position)

For this reason the design can be used to evaluate experimental methods using strain gauges applied to test prototype [2] or finite element analysis method accepted by the PED rules.

To determine the resistance of the pressure equipment at issue, the calculations must take into account the maximum allowable pressures, the calculation temperatures, all possible combinations of temperature and pressure operating conditions of the equipment reasonably foreseeable, maximum effort and stress concentrations.

The valve body is made of cast steel GP 240GH (EN 10213-2) [4] and the seat and ball are provided to be made of stainless martensitic cast steel GX12Cr12 (EN 10283) [5].

The sphere is designed as a cast part because of its size, reduce material consumptions and therefore a slight rotation of the obturator. The surface flow tube is materialized by rolling a sheet of X6CrNiMoTi17-12-2 (EN 10088-1, 2)[6] to be welded inside the sphere.

Evaluations of each parts behavior and body-seat-sphere assembly in the case of the valve with mobile seats NPS 500, NP 25, were performed by the finite element method with the software CAD-embedded SOLIDWORKS Simulation. To study the behavior of static and mainly, stresses and deformations developed in the sphere, we assembled the three parts and was imposed achieve a sealing contact between the valve seat and spherical surface of the obturator.

3. FINITE ELEMENT ANALYSIS

3.1. Defining constraints

The constraints were applied to the 20 holes surfaces R1 to the flange of the connection *Body – Body 2,* respectively, on the 20 holes R2 in the flange of in touch with the pipeline. Constraints R 3 and R 6 were added,on the guides of sphere coaxial with R4 and R5, those to the body (fig.2) [7].

3.2. Defining contacts

For this assembly the seat and sphere in contact on Sc surface were considered (Fig. 3). In functioning the sealing is done by pressing the upstream seat on the obturator surface.

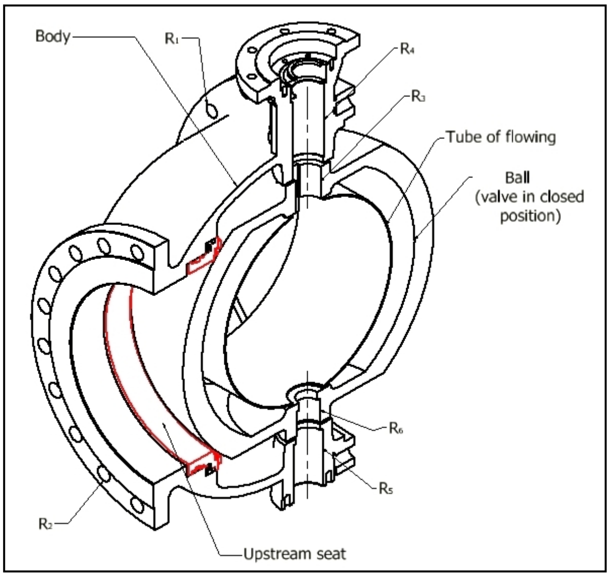


Fig. 2. R1-R6 surfaces which impose constraints

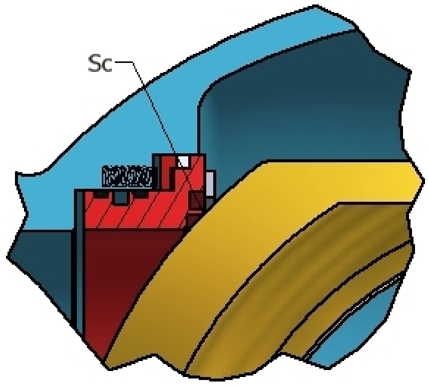


Fig. 3 Sealing detail

3.3 The stages of study

Study 1 was made without considering temperature, an equivalent study with a valve seal test. To verify stress, displacements and deformations equivalent in the presence of mechanical and temperature loads was performed two study: *study 2* using temperature values which were measured during in the fire test an ball valve and *study 3* using minimum temperatures imposed for calorimeter cubes at 15 minutes after initialising the fire (565 ^{o}C), assuming that these cubes calorimeter represent body wall and thus represent its temperature.

3.3.1. Study 1

Application of loads

Loads corresponding to this study are presented in Table 1, and surfaces that apply are those specified in fig. 4.

Table 1

Pressure 1	S1 inner surface of the body , surfaces S1' on seat and surface S1 " on sphere - with pressure **2,5 N / mm ^{2}**	The internal test pressure
Force 1	On the 20th annular surfaces **S2,** in contact with the seat surfaces of nuts, **47 526** N axial force is applied uniformly distributed	The force required
Force 2	**927940 N** force evenly distributed on the surface S3	The force required to ensure sealing of body connection
Force 3	On the eight surfaces S4 was applied axial forces 8455 N	The force given by stuffing box flange fasteners
Force 4	On the 20 annular surfaces, in contact with the nuts, the axial force **74 720 N** is applied uniformly distributed	The force given to the fasteners between the body and the body 2
Force 5	The force of **262 540 N**, evenly distributed on the surface that makes sealing of the body with Body 2	The force calculated,necessary to ensure the seal body - Body 2

Force 6	The force of 308 N, evenly distributed over the 20 seat surfaces of the elastic elements	Sums up the force given by elements elastic preloaded
Force 7	On the seat surface S3' the force from **6160 N** is applied evenly distributed	Sums up the force given by elements elastic preloaded

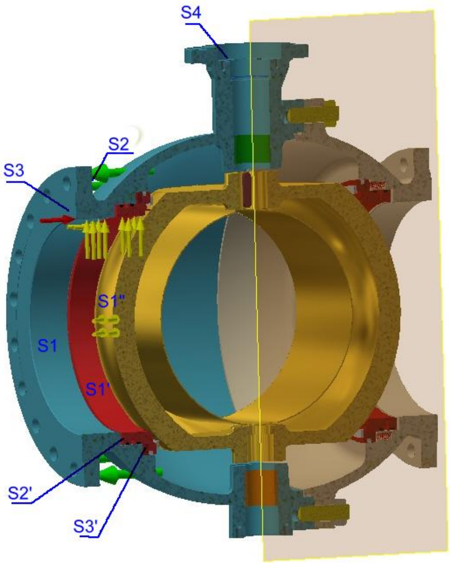


Fig. 4. The surfaces on which the mechanical stress applied

Meshing structure was done with solid type finite element, tetrahedral with curved sides (mesh details are contained in table2).

Table 2

The type of finite element	Tetrahedral finite element
The number of nodes in which to make determinations (Jacobian Check)	4 puncte
The dimensions of elements	25 mm
Tolerance	1.75 mm
The number of elements	87529
Number of nodes	153073

Defining materials

For each piece a material was accordingly assigned and they were defined by the corresponding mechanical characteristics (constant or variable depending on the temperature) according to material standards.

The study results

The results show that maximum demands are on the sphere, in the pivot, which was expected because all loads imposed on the sphere are taken in this area. Total equivalent displacements are maximum in area of flow tube protector of the sphere (Table 3 and icons in fig. 5 and 6).

This study was aimed, in particular, determination of sphere behavior. For this graphically equivalent tensions and displacements total equivalent of nodes located on the surface sphere in contact with seat were established.

Table 3

Name	Type	Max	Location
Static nodal stress	VON: von Mises stress	250.81 (MPa) Node:130914	(-10 mm, 257.636 mm, 35.623 mm)
Static displacement	URES: Resultant displacement	0.00065 m Node: 149608	(245.848 mm, 6.09366 mm, -12.3545 mm)
Equivalent strain	ESTRN: Equivalent strain	0.00109 Node: 130914	(-10 mm, 257.636 mm, 35.623 mm)

Static nodal stress von Mises, in the seal area, have low values (max 56 MPa) relative to the characteristics of the material (for GX12Cr12 material, yield strength is 450 MPa conventional at 20°C).

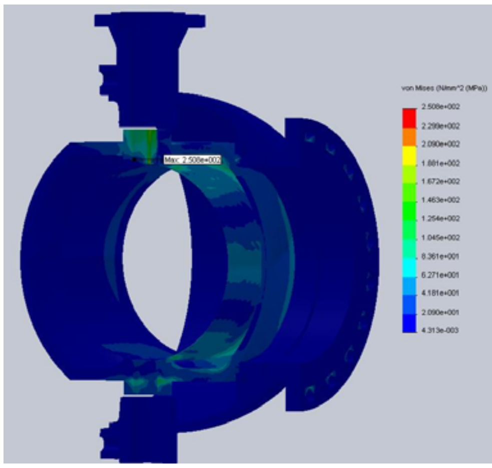


Fig. 5. Equivalent stress (von Mises)

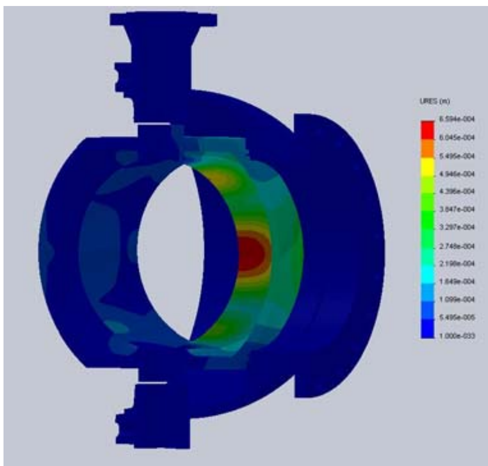
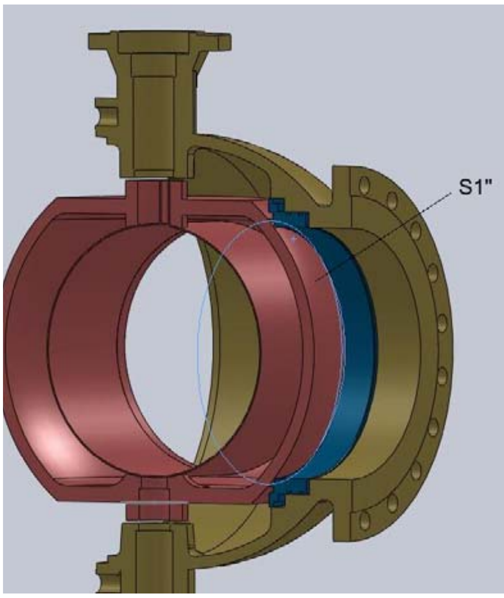
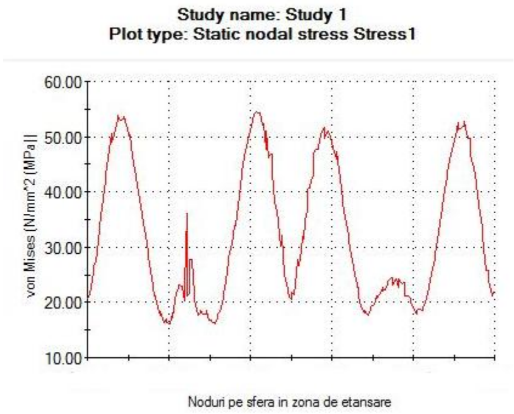


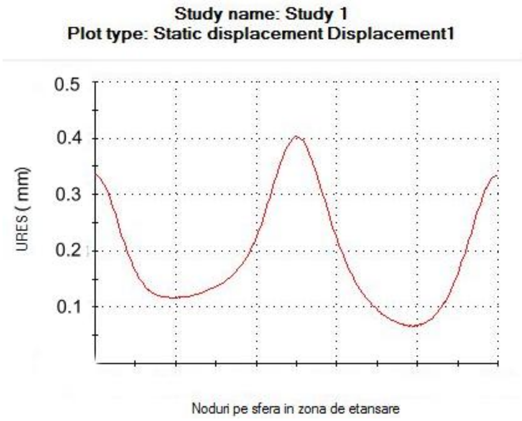
Fig. 6. Total equivalent displacements



a)



b)



c)

Fig. 7. The graphs nodal stress equivalent (von Mises)(b) and total displacements (c) in selected nodes on sealing ring S_1" (a)

3.3.2. Study 2 (the effect of pressure and temperature)

If study 1 was made without taking into account the temperatures in study 2 outside mechanical loads were applied average temperatures measured during the test fire a ball valve subject to a number tests and in study 3, the outer surface have imposed minimum temperature of calorimetry cubes at 15 minutes from init fire, namely the 565 ^{o}C .

Fire test is performed on a valve mounted in the test installation, with the obturator in the closed position, with horizontally stem and flow bore. The installation is filled with water and pressurized (to a value determined by applicable standard). The flames of the burners (760^{o}C ÷ 980 ^{o}C) is directed on the test valve during a period of 30 min, simulating in this way a possible fire in a plant. Is determined internal and external losses in this period and after extinguishing the fire.

For studies which take into account the pressure and temperature effects, *pressure 1* (din table 1) has a value of 1.875 MPa corresponding to the required standard fire test (75% the maximum allowable working pressure of the valve [1]).

Additional loading of arson and surfaces that apply are given in Table 4 and fig. 8.

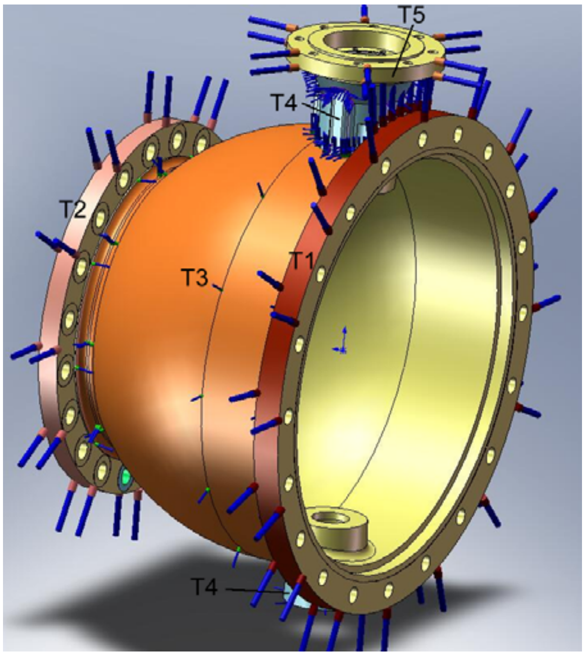


Fig. 8. Distribution of temperatures on the outside surface of the valve body

Table 4.

Additional loading of arson	The surface for applying and the temperature value (fig. 8)
Temperature – 1	The surface T1, with temperature 471 ^{0}C
Temperature – 2	The surface T2, with temperature 408 ^{0}C
Temperature – 3	The surface T3, with temperature 437 ^{0}C
Temperature – 4	The surface T4, with temperature 464 ^{0}C
Temperature – 5	The surface T4, with temperature 419 ^{0}C

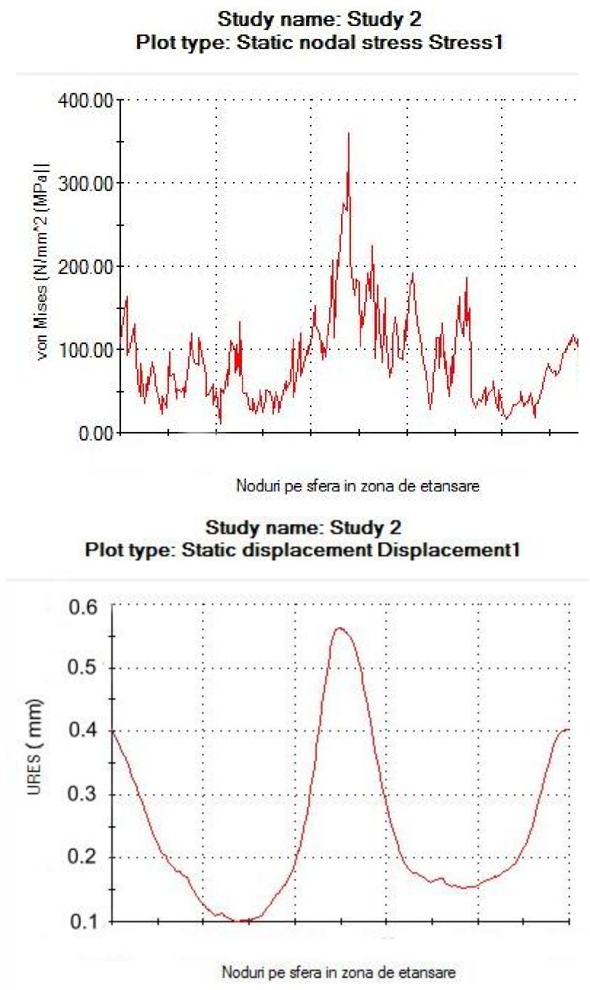


Fig. 9 The graphs nodal stress equivalent (von Mises) and total displacements in selected nodes on sealing ring

An increase of static nodal stress to a maximum of 360.1 MPa, but with a mean value low (99,71 MPa) is noticed. Equivalent displacements have higher values, reaching up to 0.56 mm.

3.3.3. Study 3 (effect of pressure and temperatures of uncontrolled fires)

Study 3 approximate conditions for fire longer, with possible serious consequences, that temperature imposed on external surfaces of the body is 565^{o}C. It can be seen that static stress and displacement nodal are not changed significantly from the previous study and this may be explained by the large size of envelope and high temperatures do not significantly affect the sealing area.

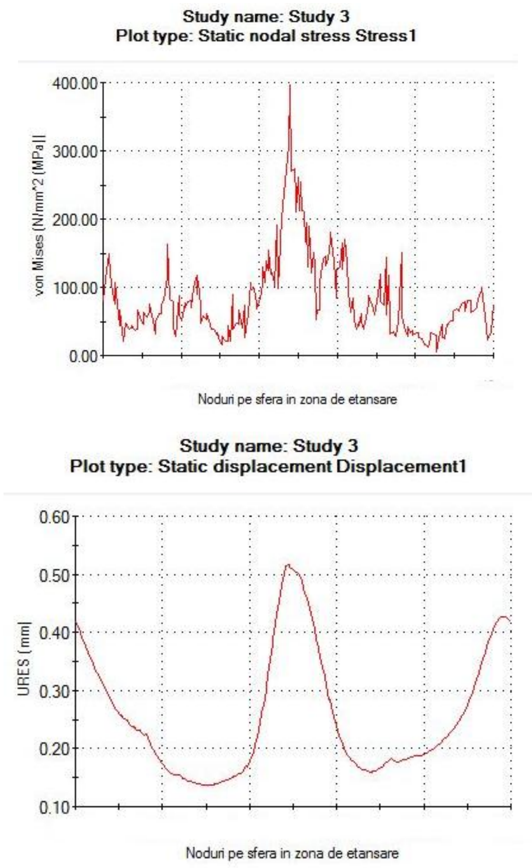


Fig. 10. The graphs nodal stress equivalent (von Mises) and total displacements in selected nodes on sealing ring

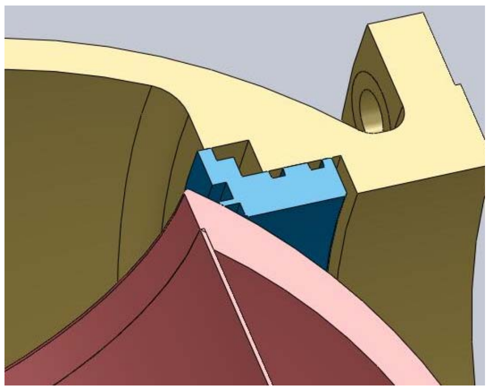


Fig.11. Area of maximum displacement

4. CONCLUSION

The checks which were made on trunnion ball valve with mobile seats NPS 500 (mm), NP 25 (bar) can be seen keeping tensions within acceptable values and equivalent strains for equipment loading concomitant mechanical and thermal stresses, confirming in this way, the design of the product, before its approval by effective test.

Extreme values of static nodal stress (von Mises) obtained in the points in the near to the generatrix plan through hole of the sphere, in the studies 2 and 3 may be subject to a new inspection by decreasing the size of the mesh element or using finite elements by higher order.

The static displacements values on the sealing area is compared with the displacements of nodes selected by the metal sealing ring with the exception of nodes that are in the vicinity of the plane of the generatrix of the passage opening, wherein the wall thickness of the sphere is smallest. (fig. 11).

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