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Research Article

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Comparison of DBA vs. DBF by Designing a Storage Tank for Different Head and Control Types

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Abstract The aim of this research work is to compare the design of a water storage tank, using two approaches called design by analysis (DBA) and design by formula (DBF). For this purpose, a water storage tank has been designed for 10 atm. of internal pressure and a temperature of 1200C and capacity of 1500 lt. SA516GR70 was chosen as the material and three different types of head type was chosen as the ellipsoidal, torispherical and hemispherical. Firstly, the required material thicknesses were calculated by the empirical formulas according to COMPRESS programme which is preferred for obtaining quick results in the design of this type of tank. Then the results were examined by the SolidWorks analysis module. COMPRESS programme formulas include high safety factors. Then, the SolidWorks analysis was repeated with the safety factor of 1.5 for the full radiographic control and 2 for the spot radiographic control and the suggested new sheet thickness values for the chosen parts were determined. Finally, the application of spot rt and full rt for the tank made of three different head type was examined in terms of cost analysis. It is shown that the thicknesses suggested by authors are more reliable than those presented numerically. For material thicknesses suggested by authors, weight and the total cost of the storage tank has been reduced by %50.

Keywords Storage tank, Solidworks, NDT, Cost analysis

Introduction

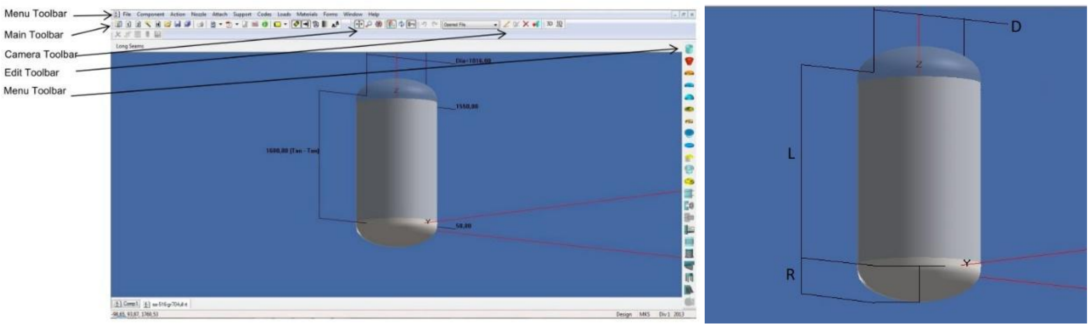
The design of pressure vessels is an important and practical topic which has been studied for decades. The pressure vessels are designed with adequate importance because the breaking of pressure vessels means an explosion that can result in loss of life and property. Pressure vessels are mainly designed to resist high pressures to a certain extent and found wide engineering applications in reactor technology, chemical industry, space and ocean engineering and fluid supply systems in industries. Their mainly task is preservation and transmit of liquids or gases under pressure [1]. As reported by Kumar et al [2] American, Indian, British, Japanese, German and other various standards are available for design of pressure vessels. However, the internationally accepted for design of pressure vessel code is American Society of Mechanical Engineering (ASME). Storage tanks are specific kinds of pressure vessels. Storage tanks are designed for fluid storage for many industrial applications. The design and production of storage tanks are also made taking into account international standards. Storage tanks operate under very little pressure, distinguishing them from pressure vessels. Water storage tanks are used to storage of water for use in many applications such as domestic water, agriculture farming, fire extinguishing applications, both for plants and livestock, chemical manufacturing. Storage tanks are often cylindrical in shape but spheres, cones or ellipsoidal forms can be used also [3]. A common design is a cylinder with end caps called heads. The heads are typically hemispherical, ellipsoidal or torispherical. During the last three decades considerable research effort has been made in the applications of some techniques to analyze pressure vessel and storage tank design problems [4]. The storage tanks can be classified into two main types: Atmospheric storage and Pressured storage. These types of tanks are manufactured that the length of the tank is not greater than six times the diameter. Operation cost and cost effectiveness are the main factors in selecting the type of storage tank [5].

Recently, considerable research effort has been devoted to the analysis, design, and evaluation of the liquid storage tanks by Zingoni [6]. Azzuni and Guzey [7] compared the shell designs for the steel cylindrical liquid storage tanks, based on the three methods as given in API standard 650, for different tank properties: diameter, height and allowable stress. They also developed a stiffness–flexibility method based on thin shell theory that gives the theoretical displacements and stresses at each shell course without any approximation or simplification. Francescato et al. [8] aimed to determine the optimal design characteristic of a type 3 storage tank. According to various damage conditions and different internal pressures. The damage conditions were considered as the first-ply failure (FPF), the progression of damage and the final failure (FF). Minimum tank weight was considered the optimal design parameter. Mandal and Maity [9] performed the nonlinear hydrodynamic finite element analysis of elastic water storage tanks. He used two dimensional eight-node isoperimetric elements for modeling the tank wall. Wang conducted two similar studies. In the first of them the authors Wang et al. [10] experimentally investigated the structural performance of the tank under various loading conditions. The authors also carried out the numerical analysis in order to simulate the tests. In the other one Wang and Xiong [11] it has been performed to simplified methods to reasonably predict the response of water storage tank under blast loading. Firstly, FE model was established and then the Lagrange equation method with combined deflection shape function and varying DIF was presented. Xu *et al* [12] studied on an optimization model using the adaptive genetic algorithm (AGA) in order to minimize the weight of composite hydrogen storage vessel. They concluded that the AGA gives more precision results comparing the simple GA and the Monte Carlo optimization method. To calculate the shell stress accurately and briefly, Chen *et al* [13] proposed a model in such a way that, the first shell regarded as a short cylindrical shell while the others as long cylindrical shells for large tanks. Kumar *et al* [2] designed and analysed a pressure vessel by considering the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1. The stress development analyzed by using ANSYS 15 and an optimized model was modelled to overcome the stresses produced in the vessel. SA516GR60, SA516GR65 and SA516GR70 chosen as design materials and the most suitable material with good design developed in the mentioned study. Gulin *et al* [14] showed the design process of a liquid-storage tank shell according to Eurocode and compared the results obtained using the norms with those from a finite element method (FEM) analysis. The calculations performed for an aboveground vertical steel water-storage tank with a variable thickness wall and stiffening ring on top. The authors presented all the results in tables also in comparable situation. Weight, ease of production and cost analyses of the storage tank design has been performed by Altınbalık and Isencik [15] by using the two different materials and three different head types. In another study Altınbalık and Kantur [16] examined the application of spot rt and full rt for the storage tank made of two different materials in terms of cost analysis. A storage tank can be designed using the rules of design by formula (DBF) and design by analysis (DBA). Diamantoudis and Kermanidis [17] compared the design of pressure vessels of high strength steel P500 with the steel alloy P355 using the rules of DBA and DBF. Murtaza and Hyder [18] aimed to compare the design of the RPV, using two approaches called ‘design by analysis’ (DBA) and ‘design by formula’ (DBF).

On the other hand, non-destructive testing (NDT) are usually used for monitoring and ensuring the integrity of structures especially for tanks used in the oil and gas industry. Inspection of welded structures is essential to ensure that the quality of welds meets the requirements of the design and operation [16]. NDT has two main purposes. One of them are social objective; to save the human and the natural and built environment in case a structure or component fails due to non-detection of a flaw. A failed structure or component can endanger its environment and human life. The commercial duty of NDT is to optimize the productivity of assets, i.e. components or structures of the entire facility being inspected [19]. A variety of NDT are available for identification, and evaluation of defects in welded joints of pipes, being the ultrasound and radiography the most relevant [20].

Theoretical Analysis and COMPRESS

COMPRESS is a user-friendly programme and prevents loss of time. It was developed to be calculating and creating reports for pressure vessels and heat exchangers according to ASME standards. While designing a new storage tank or pressure vessel, COMPRESS selects sizes, calculates the thickness values of the main body and the heads according to ASME sec.VIIIDiv.I. After this stage the results are controlled by the Authorized Inspector (AI) and if there is no problem, accepted. COMPRESS also calculates the MAP (Maximum Allowable Pressure) and MAWP (Maximum Allowable Working Pressure). Maximum allowable pressure (MAP) value is the maximum unit pressure permitted in a given material used in a vessel constructed under ASME Design rules. Maximum allowable working pressure (MAWP) for a vessel is the maximum internal or external pressure permissible at the top of the vessel in its normal operating position at the designated coincident temperature specified for that pressure. The COMPRESS main screen has several components and shown in Figure 1.a.As mentioned above one type of material and three different head types has been chosen for the tank which has 10 atm. internal pressure and 1200 mm. inside diameter and capacity of 3000 lt. Head types has been chosen ellipsoidal, torispherical and hemispherical. Storage tanks are often cylindrical in shape and perpendicular to the ground. Schematic representation of the tank and related dimensions are given in Fig.1.b and Table 1.



a) *b)*

Figure 1: Sample screenshot of COMPRESS

a) Main screen of COMPRESS and components b) Schematic view of the tank and related sizes

Table 1: Geometrical dimensions of the tank for different head types

Dimensions / **Head**	**L_{tank} (mm)**	**D (mm)**	**R (mm)**
Ellipsoidal	2300	1216	308
Torispherical	2300	1216	242,53
Hemispherical	1800	1216	611

Related Equations

In a cylindrical shell the minimum required thickness of shell is given as;

$$t = \frac{PR}{SE - 0.6P}$$

(1) On the other hand, the minimum required thickness at the thinnest point after forming of ellipsoidal, torispherical and hemispherical heads under pressure is calculated by appropriate formulas given in literature. For ellipsoidal heads the thickness is calculated as;

$$t = \frac{PD}{2SE - 0.2P} \tag{2}$$

For hemispherical heads the thickness is calculated as;

$$t = \frac{PR}{2SE - 0.2P} \tag{3}$$

and for torispherical heads the thickness is calculated as;

$$t = \frac{PLM}{2SE - 0.2P} \tag{4}$$

where:

P= Internal design pressure
R= Inside radius of the shell course under consideration
D= Inside diameter of the head
S= Maximum allowable stress value
E= Joint efficiency (equal to 1.00 for full radiography and equal to 0.85 for spot radiography)
L= Inside spherical or crown radius for torispherical heads
M= A factor in the equations for torispherical heads depending on the head proportion L/r

Material Selection and Design

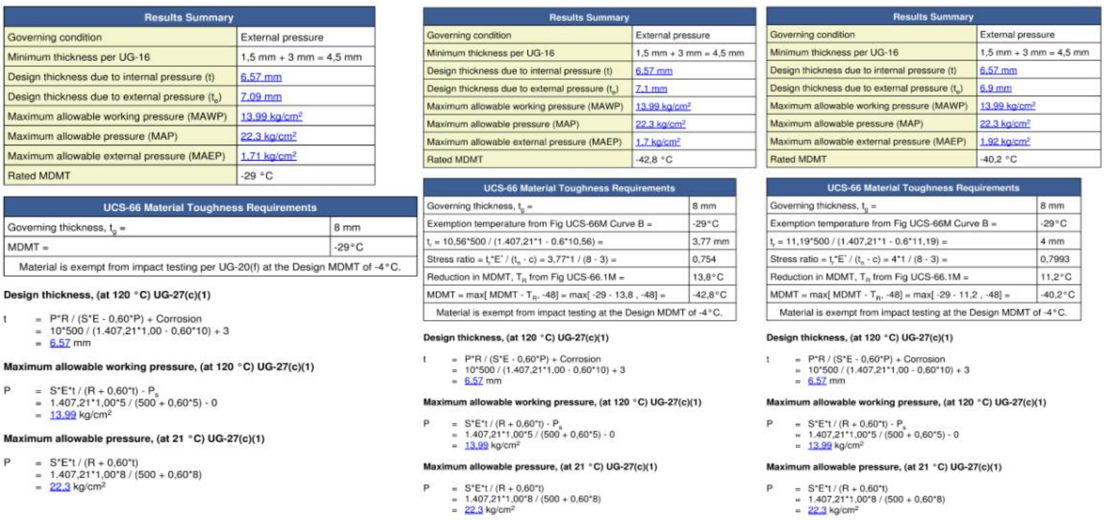
SA-516 GR70 was chosen in order to manufacturing the vessel. SA-516 GR70 is one of the most popular steel grades. It offers greater tensile and yield strength when compared the others. It is primarily intended for use in welded pressure vessels and has excellent notch toughness and is used in both pressure vessels and industrial boilers. After choosing the materials design parameters was determined and these parameters was entered the COMPRESS program screen. Maximum allowable stress values of the chosen materials for 120°C were read from ASME-BPVC 2017 Sec II Part D.

Results and Discussion

Sheet Thicknesses for Full Radiographic Control

In the company in which the presented study performed, all welded joints to be radiographed is examined in accordance with ASME BPVC Sec VIII Div.I. This NDT procedure can be performed by two ways: Full or spot radiography. Full radiography means that every inch of weld length be radiographed. For a big vessel this would mean hundreds of shots and a long process to complete, but the manufacturer could then be assured that there are no flaws. Spot radiography on the other hand would use a particularly critical spots like junctions to get 10% of the length. If there are no flaws found, then pass. If a flaw is found, then do another 10% just to make sure it is a one off. If the vessel is a water tank, spot radiography is generally good enough. Technical details on the radiographic control are given in ref [16].

Body sheet thickness calculation values of tanks to be controlled fully radiographic for three different types of header are shown in Figure 2.a–2.c.The same sheet thickness has been calculated for all three head types as 6.57 mm because body plate thickness calculations are independent of the head type. The program requires the body plate thickness value to be chosen, and this main body value is entered as 8 mm for all head types in accordance with ASME BPVC standards and from experience. The program calculates MAP and MAWP values using this input and presents them to the user.



Results Summary	
Governing condition	External pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	6.57 mm
Design thickness due to external pressure (t_e)	7.09 mm
Maximum allowable working pressure (MAWP)	13.99 kg/cm²
Maximum allowable pressure (MAP)	22.3 kg/cm²
Maximum allowable external pressure (MAEP)	1.71 kg/cm²
Rated MDMT	-29 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	8 mm
MDMT =	-29°C
Material is exempt from impact testing per UG-20(f) at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*500 / (1.407,21*1,00 - 0,60*10) + 3
= 6.57 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*1,00*5 / (500 + 0,60*5) - 0
= 13.99 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*1,00*8 / (500 + 0,60*8)
= 22.3 kg/cm²

a)

Results Summary	
Governing condition	External pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	6.57 mm
Design thickness due to external pressure (t_e)	7.1 mm
Maximum allowable working pressure (MAWP)	13.99 kg/cm²
Maximum allowable pressure (MAP)	22.3 kg/cm²
Maximum allowable external pressure (MAEP)	1.7 kg/cm²
Rated MDMT	-42,8 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	8 mm
Exemption temperature from Fig UCS-66M Curve B =	-29°C
t_r = 10,56*500 / (1.407,21*1 - 0.6*10,56) =	3,77 mm
Stress ratio = t_r*E* / (t_n - c) = 3.77*1 / (8 - 3) =	0,754
Reduction in MDMT, T_R from Fig UCS-66.1M =	13,8°C
MDMT = max[MDMT - T_R, -48] = max[-29 - 13,8 , -48] =	-42,8°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*500 / (1.407,21*1,00 - 0,60*10) + 3
= 6.57 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*1,00*5 / (500 + 0,60*5) - 0
= 13.99 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*1,00*8 / (500 + 0,60*8)
= 22.3 kg/cm²

b)

Results Summary	
Governing condition	External pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	6.57 mm
Design thickness due to external pressure (t_e)	6.9 mm
Maximum allowable working pressure (MAWP)	13.99 kg/cm²
Maximum allowable pressure (MAP)	22.3 kg/cm²
Maximum allowable external pressure (MAEP)	1.92 kg/cm²
Rated MDMT	-40,2 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	8 mm
Exemption temperature from Fig UCS-66M Curve B =	-29°C
t_r = 11,19*500 / (1.407,21*1 - 0.6*11,19) =	4 mm
Stress ratio = t_r*E* / (t_n - c) = 4*1 / (8 - 3) =	0,7993
Reduction in MDMT, T_R from Fig UCS-66.1M =	11,2°C
MDMT = max[MDMT - T_R, -48] = max[-29 - 11,2 , -48] =	-40,2°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*500 / (1.407,21*1,00 - 0,60*10) + 3
= 6.57 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*1,00*5 / (500 + 0,60*5) - 0
= 13.99 kg/cm²

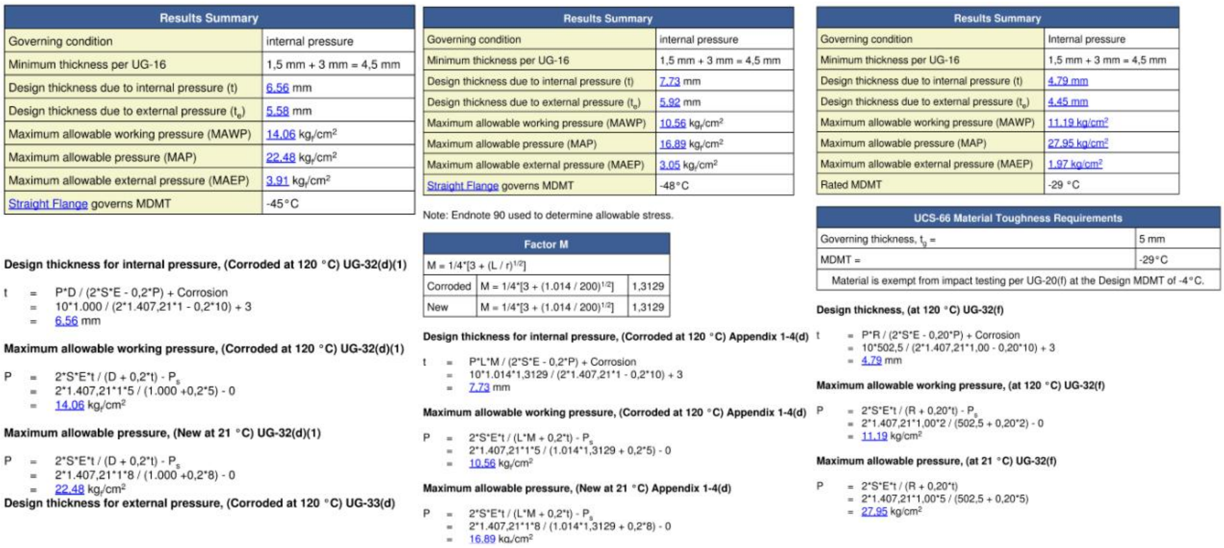
Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*1,00*8 / (500 + 0,60*8)
= 22.3 kg/cm²

c)

Figure 2: Thickness summary screen of main body for full radiography
a) Use of ellipsoidal head b) Use of torispherical head c) Use of hemispherical head

The next step is the calculation of the head sheet thickness, according to the head type. The sheet thicknesses relating to all three head types calculated by the program, and the sheet thicknesses that are entered by the user in the program in accordance with ASME BPVC standards are presented in Figure 3.a–3.c.At this point, it is necessary to evaluate ellipsoidal and torispherical heads in one category and hemispherical head in a different category. While entering these values, the manufacturing characteristics of the heads are also important. Therefore, when the ellipsoidal head is selected, the sheet thickness was calculated as 6.56 mm, but in terms of safety and in the light of experience, it was chosen as 8 mm. If DBF method is to be trusted, the head of the tank will be made of 8 mm sheet metal. The same applies to the use of torispheric heads. COMPRESS program calculated the sheet thickness as 7.73 mm, but the user chose 10 mm sheet thickness. The situation is a little bit different for hemispherical heads. It is convenient enough to choose the standard thickness closest to that calculated by the program. Although it can be selected 5 mm. sheet thickness for the full radiography, depending on the customers' requests of the cooperated company the sheet thickness is selected as 6 mm. Thus, in the case of hemispherical heads, the manufacturing stage can be started with 8 mm of body thickness and 6 mm of head thickness.



Results Summary	
Governing condition	internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	6.56 mm
Design thickness due to external pressure (t_e)	5.58 mm
Maximum allowable working pressure (MAWP)	14.06 kg_f/cm^2
Maximum allowable pressure (MAP)	22.48 kg_f/cm^2
Maximum allowable external pressure (MAEP)	3.91 kg_f/cm^2
Straight Flange governs MDMT	-45°C

Design thickness for internal pressure, (Corroded at 120 °C) UG-32(d)(1)

t = P*D / (2*S*E - 0,2*P) + Corrosion
= 10*1.000 / (2*1.407,21*1 - 0,2*10) + 3
= 6.56 mm

Maximum allowable working pressure, (Corroded at 120 °C) UG-32(d)(1)

P = 2*S*E*t / (D + 0,2*t) - P_s
= 2*1.407,21*1*5 / (1.000 +0,2*5) - 0
= 14.06 kg_f/cm^2

Maximum allowable pressure, (New at 21 °C) UG-32(d)(1)

P = 2*S*E*t / (D + 0,2*t) - P_s
= 2*1.407,21*1*8 / (1.000 +0,2*8) - 0
= 22.48 kg_f/cm^2

Design thickness for external pressure, (Corroded at 120 °C) UG-33(d)

Results Summary	
Governing condition	internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	7.73 mm
Design thickness due to external pressure (t_e)	5.92 mm
Maximum allowable working pressure (MAWP)	10.56 kg_f/cm^2
Maximum allowable pressure (MAP)	16.89 kg_f/cm^2
Maximum allowable external pressure (MAEP)	3.05 kg_f/cm^2
Straight Flange governs MDMT	-48°C

Note: Endnote 90 used to determine allowable stress.

Factor M		
M = 1/4*[3 + (L / r)$^{1/2}$]		
Corroded	M = 1/4*[3 + (1.014 / 200)$^{1/2}$]	1,3129
New	M = 1/4*[3 + (1.014 / 200)$^{1/2}$]	1,3129

Design thickness for internal pressure, (Corroded at 120 °C) Appendix 1-4(d)

t = P*L*M / (2*S*E - 0,2*P) + Corrosion
= 10*1.014*1,3129 / (2*1.407,21*1 - 0,2*10) + 3
= 7.73 mm

Maximum allowable working pressure, (Corroded at 120 °C) Appendix 1-4(d)

P = 2*S*E*t / (L*M + 0,2*t) - P_s
= 2*1.407,21*1*5 / (1.014*1,3129 + 0,2*5) - 0
= 10.56 kg_f/cm^2

Maximum allowable pressure, (New at 21 °C) Appendix 1-4(d)

P = 2*S*E*t / (L*M + 0,2*t) - P_s
= 2*1.407,21*1*8 / (1.014*1,3129 + 0,2*8) - 0
= 16.89 kg_f/cm^2

Results Summary	
Governing condition	Internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	4.79 mm
Design thickness due to external pressure (t_e)	4.45 mm
Maximum allowable working pressure (MAWP)	11.19 kg/cm^2
Maximum allowable pressure (MAP)	27.95 kg/cm^2
Maximum allowable external pressure (MAEP)	1.97 kg/cm^2
Rated MDMT	-29 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	5 mm
MDMT =	-29°C
Material is exempt from impact testing per UG-20(f) at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-32(f)

t = P*R / (2*S*E - 0,20*P) + Corrosion
= 10*502,5 / (2*1.407,21*1,00 - 0,20*10) + 3
= 4.79 mm

Maximum allowable working pressure, (at 120 °C) UG-32(f)

P = 2*S*E*t / (R + 0,20*t) - P_s
= 2*1.407,21*1,00*2 / (502,5 + 0,20*2) - 0
= 11.19 kg/cm^2

Maximum allowable pressure, (at 21 °C) UG-32(f)

P = 2*S*E*t / (R + 0,20*t)
= 2*1.407,21*1,00*5 / (502,5 + 0,20*5)
= 27.95 kg/cm^2

a) b) c)

Figure 3: Thickness summary screen of head for full radiography
a) Use of ellipsoidal head b) Use of torispherical head c) Use of hemispherical head

Sheet Thicknesses for Spot Radiographic Control

In the light of information give above body sheet thickness calculation values of tanks to be controlled fully radiographic for three different types of header are shown in Figure 4.a–4.c. As seen in the figures the sheet thicknesses value for the main body were calculated for all three head types as 8.07 mm because body plate thickness calculations are independent of the head type, as mentioned above. The program requires the body plate thickness value to be chosen, and this main body value is entered by user as 10 mm for all head types in accordance with ASME BPVC standards and from experience.

Results Summary	
Governing condition	Internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	8.07 mm
Maximum allowable working pressure (MAWP)	13.79 kg/cm²
Maximum allowable pressure (MAP)	19.74 kg/cm²
Rated MDMT	-36,15 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	10 mm
Exemption temperature from Fig UCS-66M Curve B =	-27,75°C
t_r = 13,79*603 / (1.407,21*0,85 - 0.6*13,79) =	7 mm
Stress ratio = t_r*E* / (t_n - c) = 7*0,85 / (10 - 3) =	0,85
Reduction in MDMT, T_R from Fig UCS-66.1M =	8,4°C
MDMT = max[MDMT - T_R, -48] = max[-27,75 - 8,4 , -48] =	-36,15°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*603 / (1.407,21*0,85 - 0,60*10) + 3
= 8.07 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*0,85*7 / (603 + 0,60*7) - 0
= 13.79 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*0,85*10 / (600 + 0,60*10)
= 19.74 kg/cm²

% Extreme fiber elongation - UCS-79(d)

EFE = (50*t / R_f)*(1 - R_f / R_o)
= (50*10 / 605)*(1 - 605 / infinity)
= 0,8264%

The extreme fiber elongation does not exceed 5%.

a)

Results Summary	
Governing condition	Internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	8.07 mm
Maximum allowable working pressure (MAWP)	13.79 kg/cm²
Maximum allowable pressure (MAP)	19.74 kg/cm²
Rated MDMT	-43,55 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	10 mm
Exemption temperature from Fig UCS-66M Curve B =	-27,75°C
t_r = 11,67*603 / (1.407,21*0,85 - 0.6*11,67) =	5,92 mm
Stress ratio = t_r*E* / (t_n - c) = 5,92*0,85 / (10 - 3) =	0,7187
Reduction in MDMT, T_R from Fig UCS-66.1M =	15,8°C
MDMT = max[MDMT - T_R, -48] = max[-27,75 - 15,8 , -48] =	-43,55°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*603 / (1.407,21*0,85 - 0,60*10) + 3
= 8.07 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*0,85*7 / (603 + 0,60*7) - 0
= 13.79 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*0,85*10 / (600 + 0,60*10)
= 19.74 kg/cm²

% Extreme fiber elongation - UCS-79(d)

EFE = (50*t / R_f)*(1 - R_f / R_o)
= (50*10 / 605)*(1 - 605 / infinity)
= 0,8264%

The extreme fiber elongation does not exceed 5%.

b)

Results Summary	
Governing condition	Internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	8.07 mm
Maximum allowable working pressure (MAWP)	13.79 kg/cm²
Maximum allowable pressure (MAP)	19.74 kg/cm²
Rated MDMT	-42,95 °C

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	10 mm
Exemption temperature from Fig UCS-66M Curve B =	-27,75°C
t_r = 11,83*603 / (1.407,21*0,85 - 0.6*11,83) =	6 mm
Stress ratio = t_r*E* / (t_n - c) = 6*0,85 / (10 - 3) =	0,7286
Reduction in MDMT, T_R from Fig UCS-66.1M =	15,2°C
MDMT = max[MDMT - T_R, -48] = max[-27,75 - 15,2 , -48] =	-42,95°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-27(c)(1)

t = P*R / (S*E - 0,60*P) + Corrosion
= 10*603 / (1.407,21*0,85 - 0,60*10) + 3
= 8.07 mm

Maximum allowable working pressure, (at 120 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t) - P_s
= 1.407,21*0,85*7 / (603 + 0,60*7) - 0
= 13.79 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-27(c)(1)

P = S*E*t / (R + 0,60*t)
= 1.407,21*0,85*10 / (600 + 0,60*10)
= 19.74 kg/cm²

% Extreme fiber elongation - UCS-79(d)

EFE = (50*t / R_f)*(1 - R_f / R_o)
= (50*10 / 605)*(1 - 605 / infinity)
= 0,8264%

The extreme fiber elongation does not exceed 5%.

c)

Figure 4: Thickness summary screen of main body for spot radiography
a) Use of ellipsoidal head b) Use of torispherical head c) Use of hemispherical head

The last step is the calculation of the head sheet thickness, according to the head type. At this stage the sheet thicknesses relating to all three head types also calculated by the program, and the sheet thicknesses that are entered by the user in the program in accordance with ASME BPVC standards are presented in Figure 5.a–5.c.As seen in Fig. 5.a. the sheet thickness was calculated as 8.01 mm for ellipsoidal head usage but in terms of safety and in the light of experience, it was chosen as 10 mm. The same applies to the use of torispheric heads. COMPRESS program calculated the sheet thickness as 10.71 mm, but the user chosen 12 mm sheet thickness. In the case of hemispherical heads, the programme calculated 5.54 mm sheet thickness but 6mm was chosen by the user.

Results Summary	
Governing condition	internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	8.01 mm
Maximum allowable working pressure (MAWP)	13.96 kg_f/cm²
Maximum allowable pressure (MAP)	19.9 kg_f/cm²
Straight Flange governs MDMT	-46,75°C

Factor K		
K = (1/6)*[2 + (D / (2*h))²]		
Corroded	K = (1/6)*[2 + (1.206 / (2*303))²]	0,9934
New	K = (1/6)*[2 + (1.200 / (2*300))²]	1

Design thickness for internal pressure, (Corroded at 120 °C) Appendix 1-4(c)

t = P*D*K / (2*S*E - 0,2*P) + Corrosion
= 10*1.206*0,993416 / (2*1.407,21*0,85 - 0,2*10) + 3
= 8.01 mm

Maximum allowable working pressure, (Corroded at 120 °C) Appendix 1-4(c)

P = 2*S*E*t / (K*D + 0,2*t) - P_s
= 2*1.407,21*0,85*7 / (0,993416*1.206 +0,2*7) - 0
= 13.96 kg_f/cm²

Maximum allowable pressure, (New at 21 °C) Appendix 1-4(c)

P = 2*S*E*t / (K*D + 0,2*t) - P_s
= 2*1.407,21*0,85*10 / (1*1.200 +0,2*10) - 0
= 19.9 kg_f/cm²

% Extreme fiber elongation - UCS-79(d)

EFE = (75*t / R_f)*(1 - R_f / R_o)
= (75*12 / 210)*(1 - 210 / infinity)
= 4,2857%

a)

Results Summary	
Governing condition	internal pressure
Minimum thickness per UG-16	1,5 mm + 3 mm = 4,5 mm
Design thickness due to internal pressure (t)	10.71 mm
Maximum allowable working pressure (MAWP)	11.67 kg_f/cm²
Maximum allowable pressure (MAP)	15.51 kg_f/cm²
Straight Flange governs MDMT	-48°C

Note: Endnote 90 used to determine allowable stress.

Factor M		
M = 1/4*[3 + (L / r)^1/2]		
Corroded	M = 1/4*[3 + (1.203 / 123)^1/2]	1,5318
New	M = 1/4*[3 + (1.200 / 120)^1/2]	1,5406

Design thickness for internal pressure, (Corroded at 120 °C) Appendix 1-4(d)

t = P*L*M / (2*S*E - 0,2*P) + Corrosion
= 10*1.203*1,5318 / (2*1.407,21*0,85 - 0,2*10) + 3
= 10.71 mm

Maximum allowable working pressure, (Corroded at 120 °C) Appendix 1-4(d)

P = 2*S*E*t / (L*M + 0,2*t) - P_s
= 2*1.407,21*0,85*9 / (1.203*1,5318 + 0,2*9) - 0
= 11.67 kg_f/cm²

Maximum allowable pressure, (New at 21 °C) Appendix 1-4(d)

P = 2*S*E*t / (L*M + 0,2*t) - P_s
= 2*1.407,21*0,85*12 / (1.200*1,5406 + 0,2*12) - 0
= 15.51 kg_f/cm²

% Extreme fiber elongation - UCS-79(d)

EFE = (75*t / R_f)*(1 - R_f / R_o)
= (75*14 / 127)*(1 - 127 / infinity)
= 8,2677%

b)

UCS-66 Material Toughness Requirements	
Governing thickness, t_g =	6 mm
Exemption temperature from Fig UCS-66M Curve B =	-29°C
t_r = 11,83*606 / (2*1.407,21*0,85 - 0.2*11,83) =	3 mm
Stress ratio = t_r*E* / (t_n - c) = 3*0,85 / (6 - 3) =	0,85
Reduction in MDMT, T_R from Fig UCS-66.1M =	8,4°C
MDMT = max[MDMT - T_R, -48] = max[-29 - 8,4 , -48] =	-37,4°C
Material is exempt from impact testing at the Design MDMT of -4°C.	

Design thickness, (at 120 °C) UG-32(e)

t = P*R / (2*S*E - 0,20*P) + Corrosion
= 10*606 / (2*1.407,21*0,85 - 0,20*10) + 3
= 5.54 mm

Maximum allowable working pressure, (at 120 °C) UG-32(e)

P = 2*S*E*t / (R + 0,20*t) - P_s
= 2*1.407,21*0,85*3 / (606 + 0,20*3) - 0
= 11.83 kg/cm²

Maximum allowable pressure, (at 21 °C) UG-32(e)

P = 2*S*E*t / (R + 0,20*t)
= 2*1.407,21*0,85*6 / (603 + 0,20*6)
= 23.76 kg/cm²

% Extreme fiber elongation - UCS-79(d)

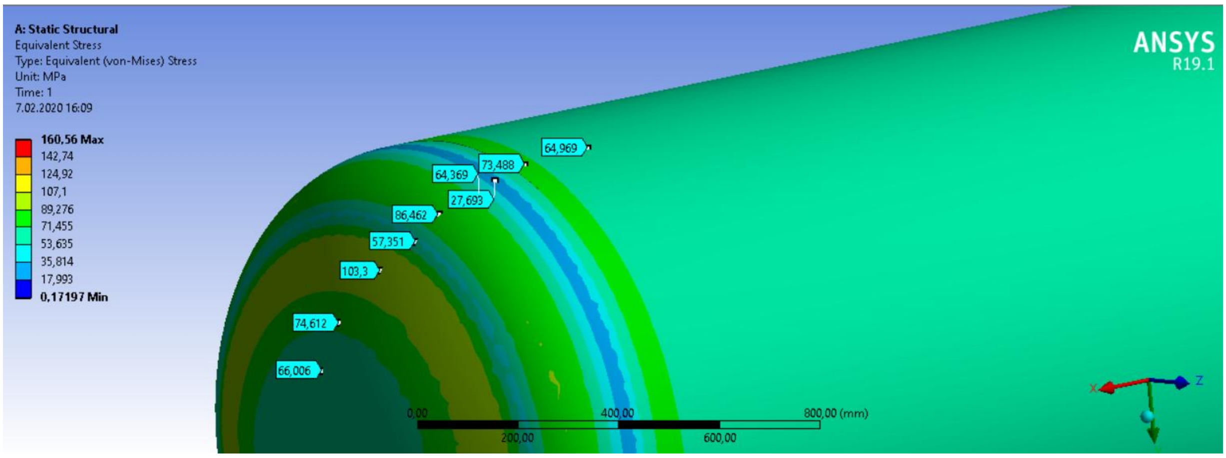
EFE = (75*t / R_f)*(1 - R_f / R_o)
= (75*6 / 606)*(1 - 606 / infinity)
= 0,7426%

c)

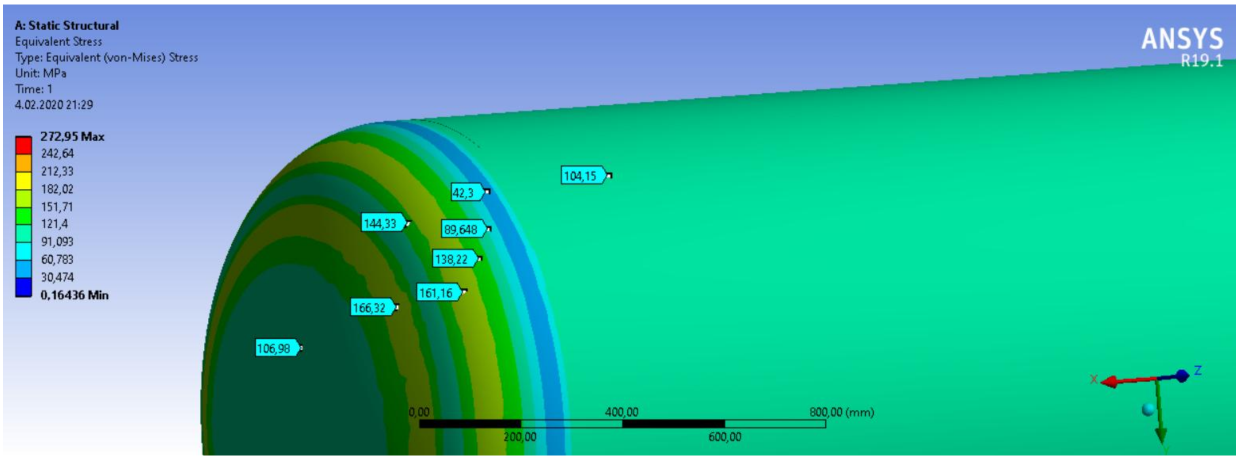
Figure 5: Thickness summary screen of head for spot radiography
a) Use of ellipsoidal head b) Use of torispherical head c) Use of hemispherical head

Comparison of the Results by SolidWorks

Compress programme and the empirical equations have been making calculations in accordance with the large factors of safety. From this point of view thicknesses of components which have been found by means of calculations shall operate and work safely under the current internal stress conditions. However, yet, it is advised and recommended that empirical calculations are to be checked and verified by using SolidWorks.



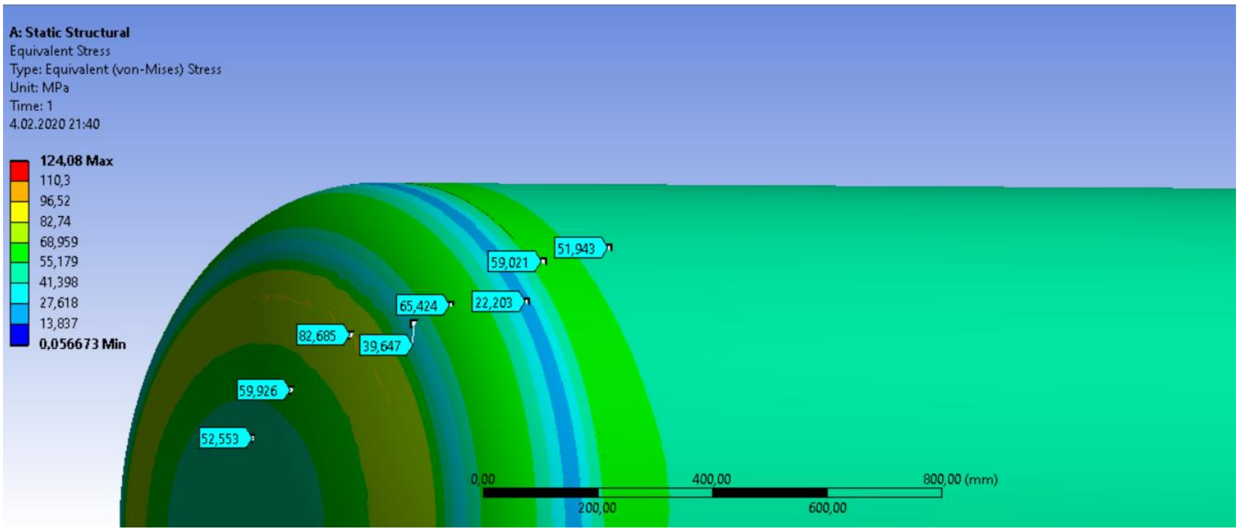
a)



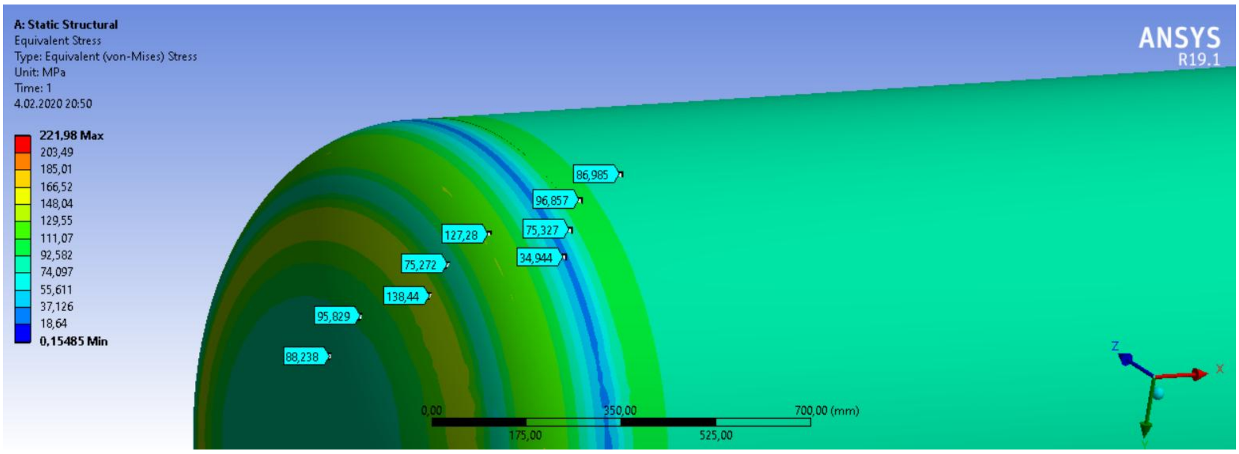
b)

Figure 6: Stress analysis results of main body and ellipsoidal head for full RT
a) s=8 mm (calculated) b) s=5 mm (suggested)

The sheet thickness values selected by the user after the COMPRESS calculation for this type of head are as follows: Both the main body and the head region are 8 mm in thickness. SolidWorks simulation of the main body and ellipsoidal head with a 8 mm thickness, as calculated by the COMPRESS, based on its geometrical dimensions and internal pressure is shown seen in Figure 6.a. Referring to the color code of overall part it is observed that the highest equivalent stress value is approximately 65 MPa. When the stress values at the selected points are examined, it is obvious that the stresses occurring both in the main body and in the head part are at very low levels compared to the flow stress of the material. Besides, there is no need to consider the maximum value of the color scale because there is no red region in the color code on the part. Considering that flow stress of the material is 260 MPa, it can be said that the determined main body and head thicknesses has a safety factor of 4. As it is seen in the figure, the stress in the yellow color of the head part is 103 MPa and even at this design value, there is approximately 2.5 times safety compared to the yield stress of the material. The safety factor for the storage tank which will be controlled fully radiographically was selected as 1.5 and SolidWorks analyzes were performed to determine the new sheet thickness value according to DBA. For the case where the VonMises equivalent stress is 166 MPa, the head thickness was determined as 5 mm and the main body thickness was chosen as 5 mm in order to avoid incompatibility. Thus, the main body is designed as 2.5 times safe and the head is 1.5 times safe. The SolidWorks analysis of the main body and ellipsoidal head with a 5 mm sheet thickness is seen in Figure 6.b. As seen in the figure the highest Von-Misses equivalent stress is about 104 MPa at the main body and 166 MPa at the head region of the tank according to color scale and the tank has a safety factor of approximately 1.5 as determined. In materials with normal flexing properties such as steel, alloy steel, aluminium and copper, the factor of safety can be selected between 1.2-2.0 in case of static loading. Ayvaz et al. [21] examined the safety of hydrogen tanks at different temperatures for two different materials and two different internal pressure values. In this study, the factor of safety for the design temperatures of 100-150 ºC was between 1.5-2.



a)

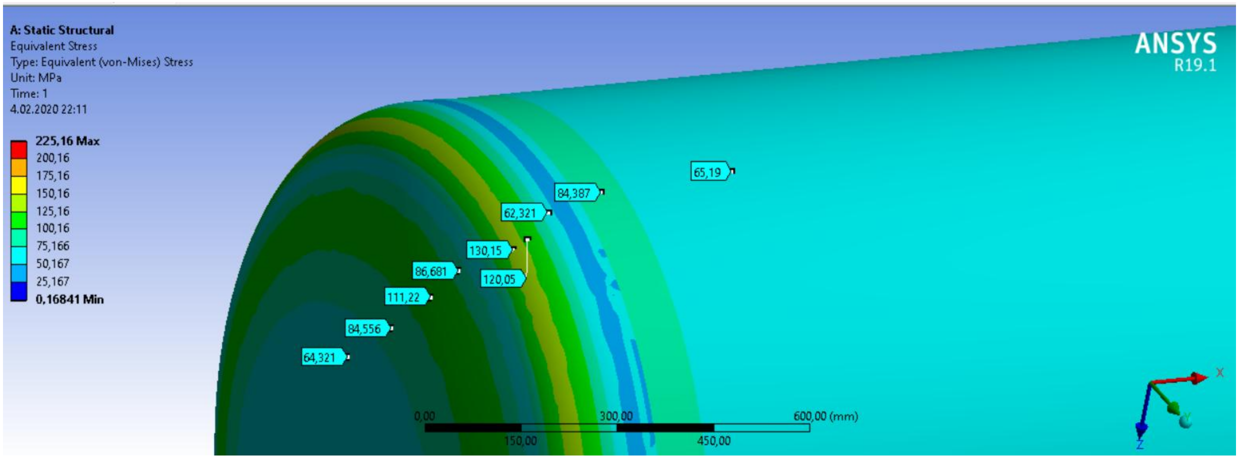


b)

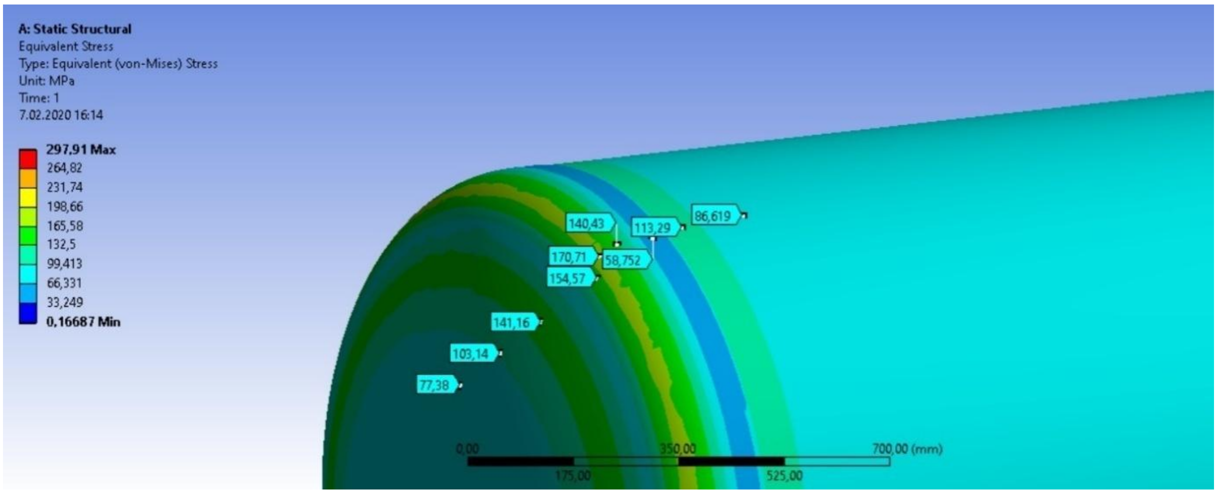
Figure 7: Stress analysis results of main body and ellipsoidal head for spot RT
a) s=10 mm (calculated) b) s=6 mm (suggested)

If spot radiographic control will be applied to the tank, it is first entered to the COMPRESS program that a control will be made in this way. Thus, the joint efficiency increases and the program calculates thicker sheet thicknesses. Results of analysis which have been done according to the thickness of 10 mm calculated by COMPRESS for the main body and 10 mm calculated for the ellipsoidal head by COMPRESS are shown in the Figure 7.a. According to Fig. 7.a. the existing inner pressure creates an equivalent stress of approximately 55 MPa on the main body. The flow stress of the material is 260 MPa as mentioned before. Thus, the cylinder is 4.7 times safely. On the other hand the ellipsoidal head thickness is 4 times safely also when the maximum equivalent stress value on the color scale is taken into consideration.

In the next step, the sheet thickness was gradually reduced from 10 mm. until the Von-Mises equivalent stress caused by the internal pressure reached half of the yield stress of the material in the head region. Then, the SolidWorks analysis of the main body and ellipsoidal head with a 6 mm sheet thickness is seen in Figure 7.b. As seen in the figure the main Von-Misses equivalent stress is about 90 MPa at both the main body and the head region of the tank according to color scale and the number labels. Maximum stress value is 138 MPa and this means the tank has a safety factor of approximately 2 as desired.



a)

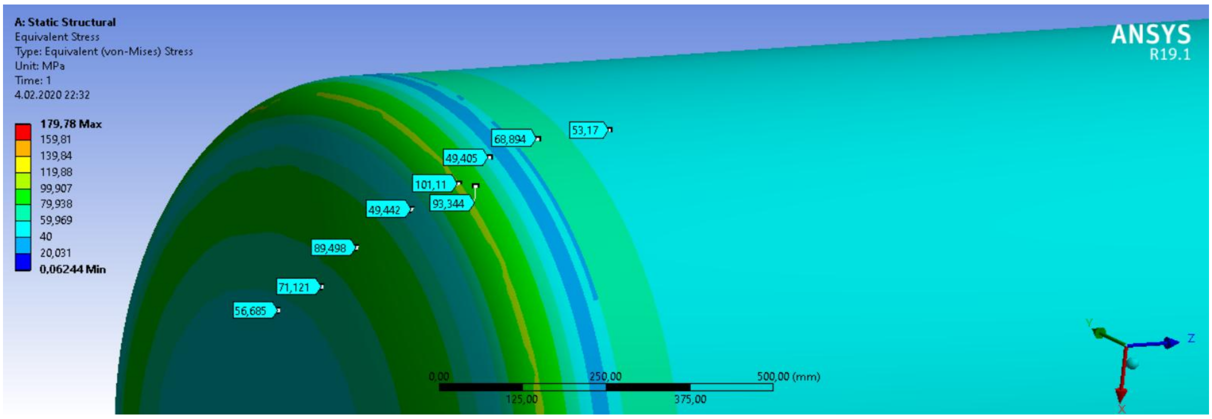


b)

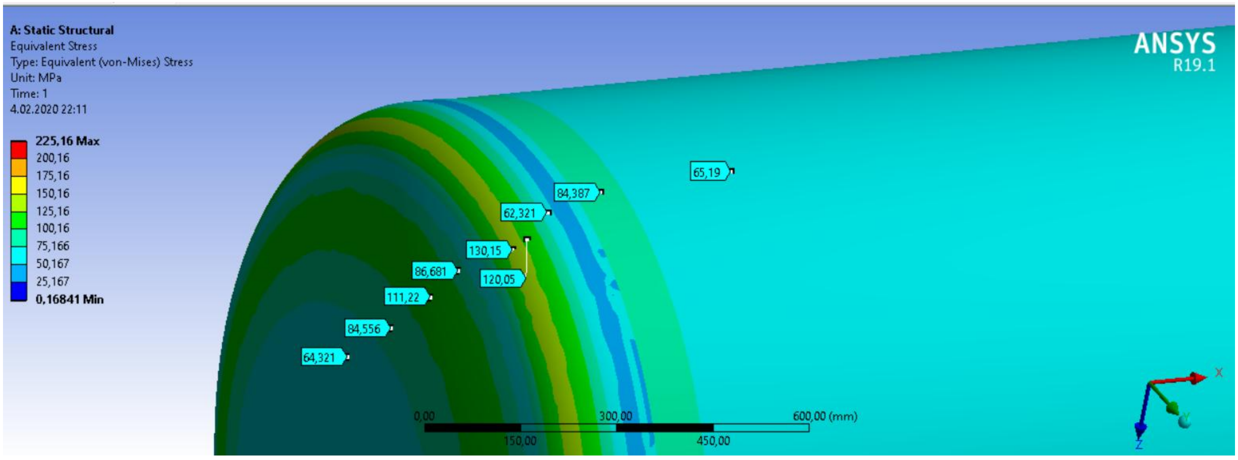
Figure 8:Stress analysis results of main body and torispheric head for full RT

a) s=body: 8mm-head: 10mm (calculated) b) s=body: 6mm-head: 10mm (suggested)

Similar calculations and steps were made for the torispheric headed storage tank and the stress values for the sheet thicknesses determined for the full radiographic control are presented in Figure 8.a and Figure 8.b. The COMPRESS program was calculated a sheet thickness of 8 mm for the main body and 10 mm for the head. Figure 8.a. shows the stress analysis according to the sheet thicknesses determined by the compress program for 10 atmospheres of internal pressure. When the colour scale and number labels are examined it can be said that the determined main body and head thicknesses has a safety factor of 4 and 2.5, respectively. When the tank head is designed to be 1.5 times safe according to the stress analysis, the main body thickness can be 6 mm and the head thickness can be 8 mm. As seen in the figure 8.b. the highest Von-Mises equivalent stress is about 113 MPa at the main body and 170 MPa at the head region of the tank. This means that the main body is designed as 2.3 times safe and the head is 1.5 times safe.



a)

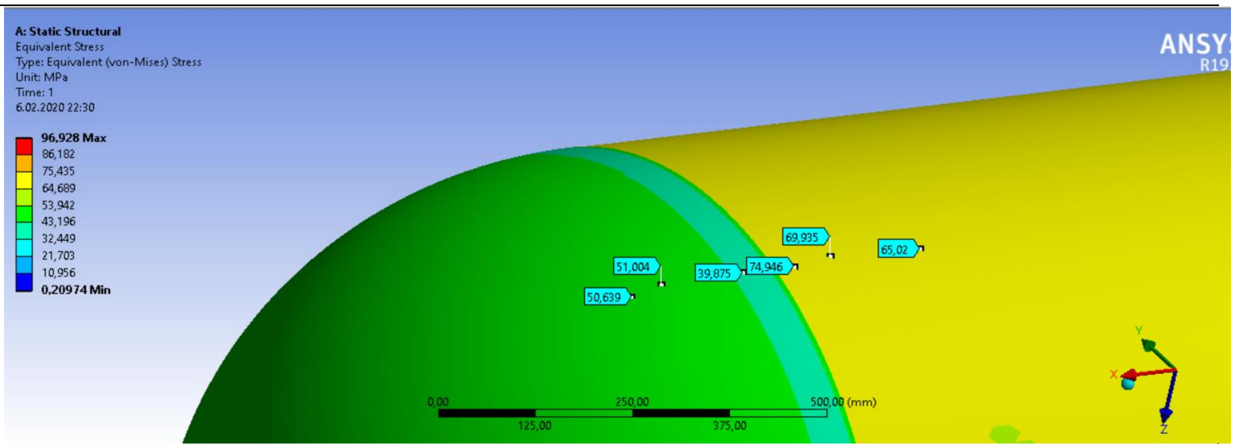


b)

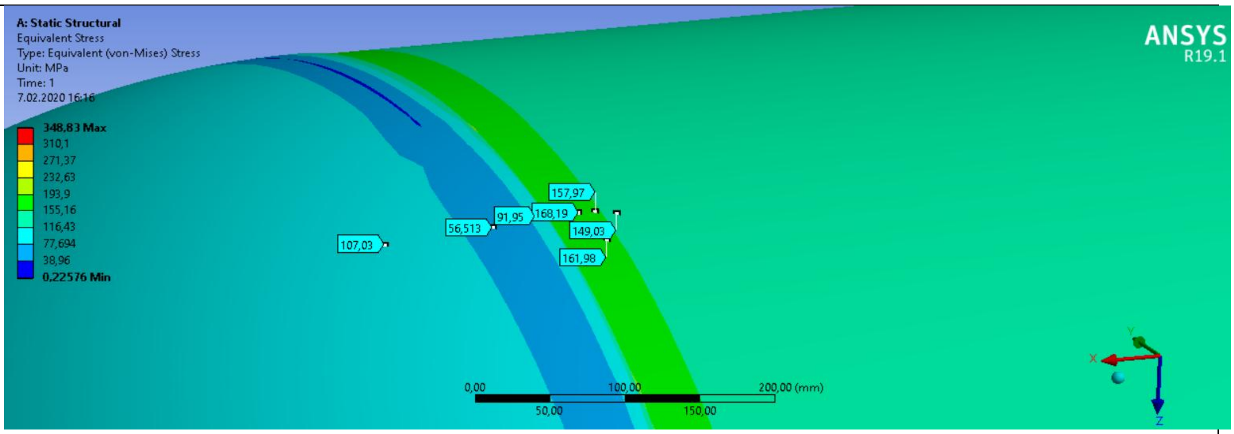
Figure 9:Stress analysis results of main body and torispheric head for spot RT

a) s=body: 10 mm-head: 12 mm (calculated) b) s=body: 8 mm-head: 10 mm (suggested)

When the spot radiographic control is considered for the torispheric headed tank, the COMPRESS programme was calculated a sheet thickness of 10 mm for the main body and 12 mm for the head. So, the existing inner pressure creates an equivalent stress of approximately 53 MPa on the main body and 89 MPa on the head region as seen in Figure 9.a. Thus, the cylinder of the tank is 5 times and the head region is 3 times safely. Therefore, it is possible to reduce the thickness of main body to 8mm and the head to 10 mm. In Figure 9.b., the SolidWorks analysis of the tank with a 8 mm main body thickness and 10 mm head thickness, is shown. According to analysis this thickness values are least 2 times safely at the head region because maximum equivalent stress is 130 MPa while flow stress of the material is 260 MPa.



a)

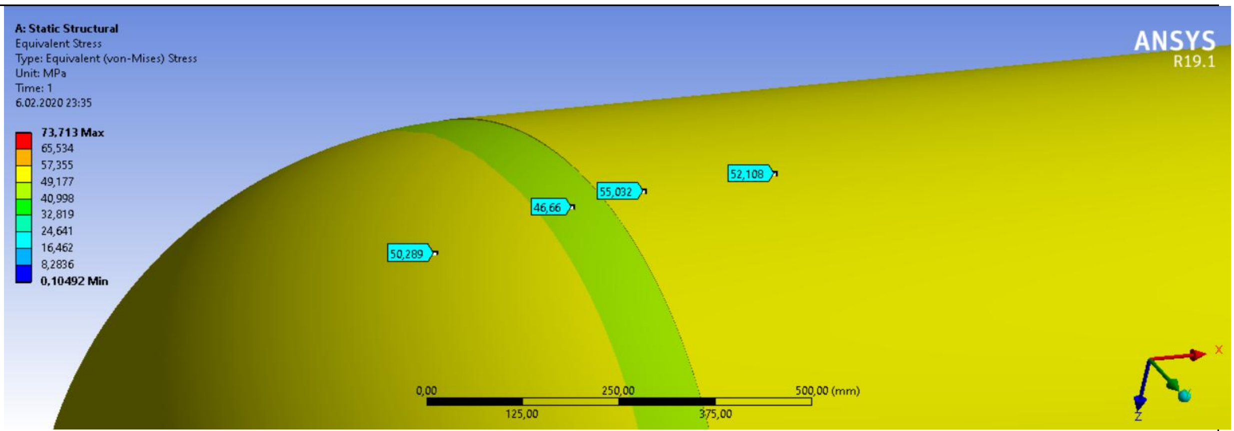


b)

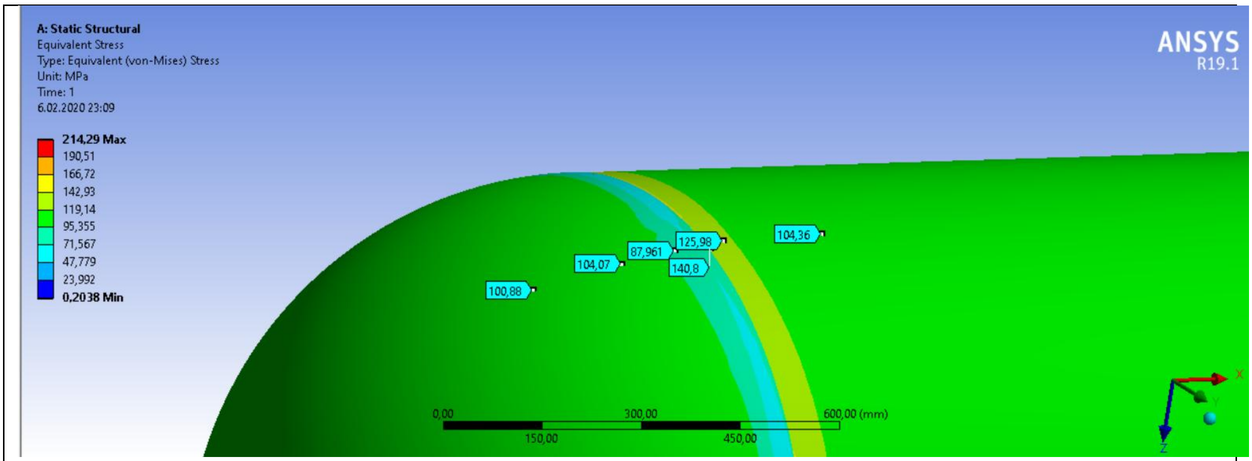
Figure 10:Stress analysis results of main body and hemispherical head for full RT
a) s=body: 8 mm-head: 6 mm (calculated) b) s=body: 4 mm-head: 3 mm (suggested)

It is necessary to evaluate ellipsoidal and torispherical heads in one category and hemispherical head in a different category. For ellipsoidal and torispherical heads, body and head sheet thicknesses are preferred to be as close as possible to each other. When entering these values, the manufacturing characteristics of the heads are also important. As known a hemispherical head has approximately half the thickness compared the others with the same pressure value. For this reason, while the head thicknesses are very close to each other in ellipsoidal and torispheric heads, the situation is slightly different in hemispherical heads. Under these conditions, for the hemispherical headed storage tank and the stress values for the sheet thicknesses determined for the full radiographic control are presented in Figure 10.a and Figure 10.b. The COMPRESS program was calculated a sheet thickness of 8 mm for the main body and 6 mm for the head. When the colour scale and number labels are examined it can be said that the determined main body and head thicknesses has a safety factor of 3.5 and 5.2, respectively. When the tank head is designed to be 1.5 times safe according to the stress analysis, the main body thickness can be 4 mm and the head thickness can be 3 mm. Although these sheet thickness values seem quite low, they are actually safe. While the safety factor for the body is 1.5, this value increases to 2.4 for the head region. As seen in the figure 10.b. the highest Von-Mises equivalent stress is about 168 MPa at the main body and 107 MPa at the head region of the tank.

The last tank type in the presented study is to use hemispherical head and spot rt control. For this case, the sheet thickness values calculated by the COMPRESS program are 10 mm for the body and 6 mm for the head region.



a)



b)

Figure 11:Stress analysis results of main body and hemispherical head for spot RT

a) s=body: 10 mm-head: 6 mm (calculated) b) s=body: 5 mm-head: 3 mm (suggested)

As shown in Figure 11a, the entire tank is subjected to an equivalent stress of approximately 50 MPa and as this means 5 times safe. Thus, it is clear that the thickness of the sheet can be reduced. Sheet thickness can be reduced until the safety factor is 2. Figure 11 b shows the Von-Mises equivalent stress values when the body thickness is 5 mm and the head thickness is 3 mm. As can be seen from the figure, the body of the tank is designed with 2 times safety. The safety coefficient in the header area is 2.5. Although it is possible to reduce the head thickness to less than 3mm, such an option has not been considered due to commercial concerns.

Weight and Cost Analysis

The main structural parts of a water storage tank are the body and the head, and these parts have a comparatively high safety factor, as can be seen from the calculated thickness values according to the COMPRESS programme. It is also shown in the diagrams above that the thicknesses of these parts can be determined with the safety factor of 1.5 for the full radiographic control condition and 2 for the spot radiographic condition with SolidWorks analysis. Weight values and prices are given in the following Table 2 according to results by the DBF and the DBA.

Table 2: Comparison of the weight and total cost values

Part	DBF Weight (kg)	DBA Weight (kg)	Differ. (kg)	Unit Price ($/kg)	Differ. ($)	Differ. (%)
Elliptic Head and Full RT	739	461	278	1	278	37.7
Elliptic Head and Spot RT	926	553	373	1	373	40
Torisph. Head and Full RT	760	589	171	1	171	22.5
Torisph. Head and Spot RT	953	760	193	1	193	20
Spherical Head and Full RT	638	317	321	1	321	50
Spherical Head and Spot RT	744	370	374	1	374	50

Weight values of sheet metal thicknesses calculated according to COMPRESS (DBF) and suggested by authors in order to DBA are given in Table 2. It is clearly seen that there are considerable advantages in design of a storage tank by using DBA. Based on calculations total weight of the tank varies 638 kg to 953 kg according to head and control method. However, considering the suggestion of authors, the tank weight decreases at least 171 kg to 374 kg. Therefore, it is apparent that the sheet metal thicknesses selected according to stresses provide an advantage in terms of weight. On the other hand, material costs of the tank chamber will be reduced also in order to low material usage. Due to the geometry of the torispheric head, the lowest weight gain both in weight and in proportion was obtained in this head type. The highest weight gain was obtained in the hemispherical head tank as 374 kg. Depending on the capacity of the company where the work is carried out, 10 units of elliptical or hemispherical headed tanks can be produced per month. If spot rt is chosen, approximately **3.75 tons** of material will be saved per month. Thus, the material gain is **45 tons/per year** and total cost is reduced as **45,000$/per year**. As a result, it is clearly observed that considering DBA offers great advantages over DBF in terms of lightness and production cost. If the number of tank production teams is increased, the number of tanks produced per month will increase accordingly.

Conclusion

Numerous factors play a role in the production of a tank, aside from the design criteria that are present in the literature. Customer expectations, information obtained by experience and company production routines are some of these factors. Therefore, the aim of this study is to increase people's options, instead of presenting accurate information on a specific design. Design of a water storage tank determination of main body and head region thicknesses of material by means of DBF and DBA have been realized and advantages of DBA for the production of these types of tanks have been well presented in this study. As known, the material thickness values obtained as the result of calculations made according to COMPRESS programme have got rather high safety factors. As it was also seen at the results of SoildWork sanalysis, these coefficient values range from 2.5 to 5.2. Moreover, a safety factor is included in empirical calculations. Thus, it will be determined for these sections in accordance with the agreement and understanding to be reached between manufacturer and customer, for example a safety coefficient between 1.5 and 2 times will also pull down costs to lower level significantly. The thicknesses suggested by authors are more reliable than those presented numerically in the study. The use of DBA instead of DBF decreases the total cost by 50% also.

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