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Investigation of welded protective covers for heat treatment

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ABSTRACT

A welded steel protective cover is investigated, which is used for the heat treatment of steel sheet coils. Protective covers are made of austenitic stainless steel. It consists of three main parts, welded together, which have 1400 mm height each. The plate thicknesses at the lower, middle and upper parts are different. The aim of the investigation was to improve the lifetime, the number of heat cycles of the protective cover, changing the geometry, the thicknesses and the material. We have evaluated the damaged covers, made calculations on stress and deformations and a series of finite element models have been investigated, comparing the behaviour of them. Both horizontal and vertical corrugated covers have been simulated and compared. The original geometry and the horizontal corrugated plate are identical from the stress level point of view. The vertical position of the corrugated plate gave a better result to the horizontal one. The result of the original geometry is close to the measured damage.

1. INTRODUCTION

Description of the process:
In the heat treatment of steel sheet coils there are protective covers employed, which were damaged after several heat cycles.

Description of the heat treatment process:
Heating up to 520-720 C°, keeping time on this temperature between 14-24 hours, cooling with water. The total time cycle of the heat treatment is: 68-70 hours.

Description of the problem:
Due to the heat effects, during the heat treatments, the protective covers destroyed earlier than the assumed lifetime by the owner.

The current layout of the protective covers:
Protective covers are made of austenitic stainless steel No. 1.4571. The bin consists of 3 main parts, which have 1400 mm height each. The plate thickness at the lower part is 6 mm, at the middle part 5 mm, at the upper part 3 mm. At the lower part of the bin there is a skirt and a slightly conical shell covers it (see Figure 1). The protective cover is made of wave like form plate according to Figure 2.

Damage:
Generally the protective covers are damaged at the lower part of the middle section (see Figure 3). At this part the waves are deformed, the material softening occurred and they lost their original form.

The aim:
Investigation of the original corrugated protective cover to find the reason of the damage and the investigation of the new versions to compare the predicted lifetime.

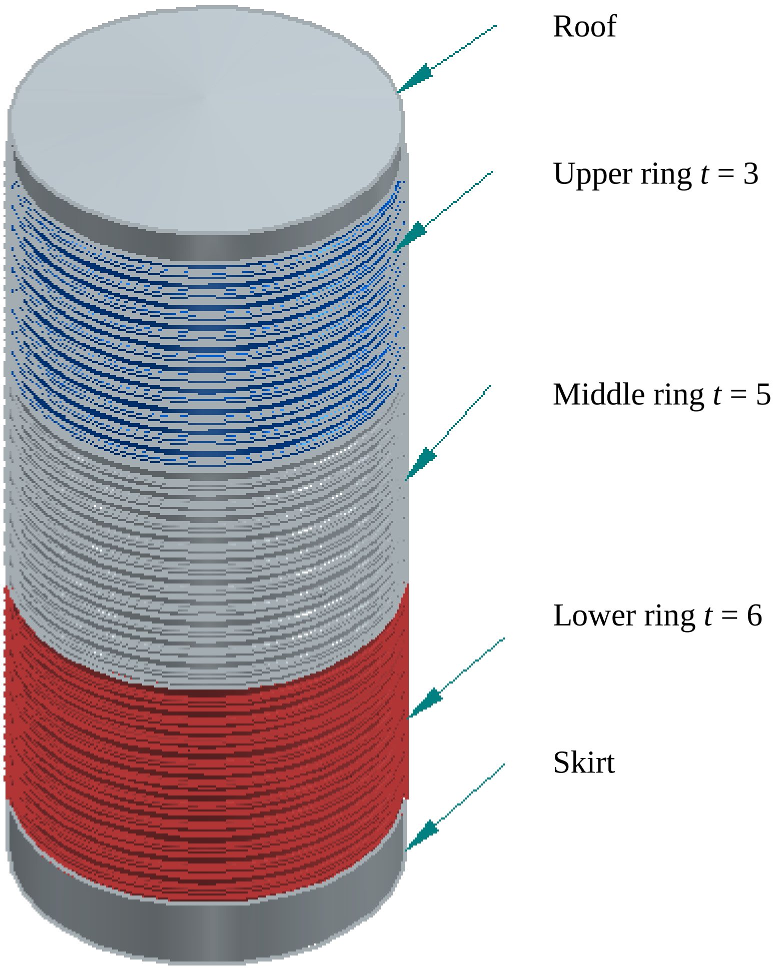

Figure 1. The current geometry of the protective cover

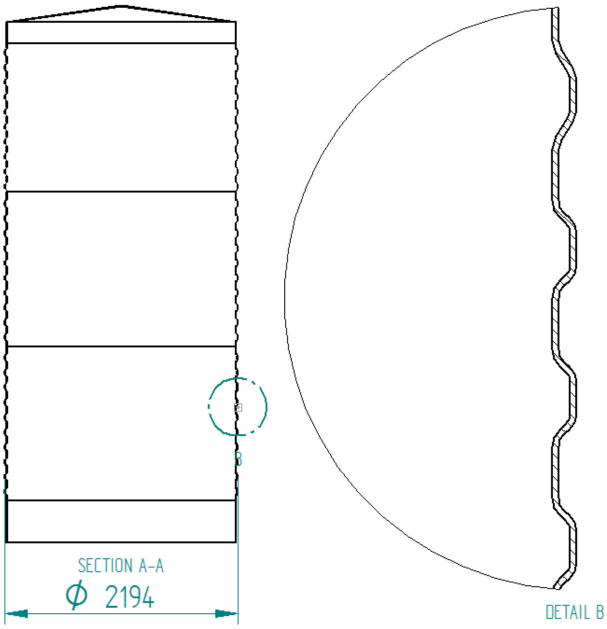

Figure 2. The geometry of the protective cover

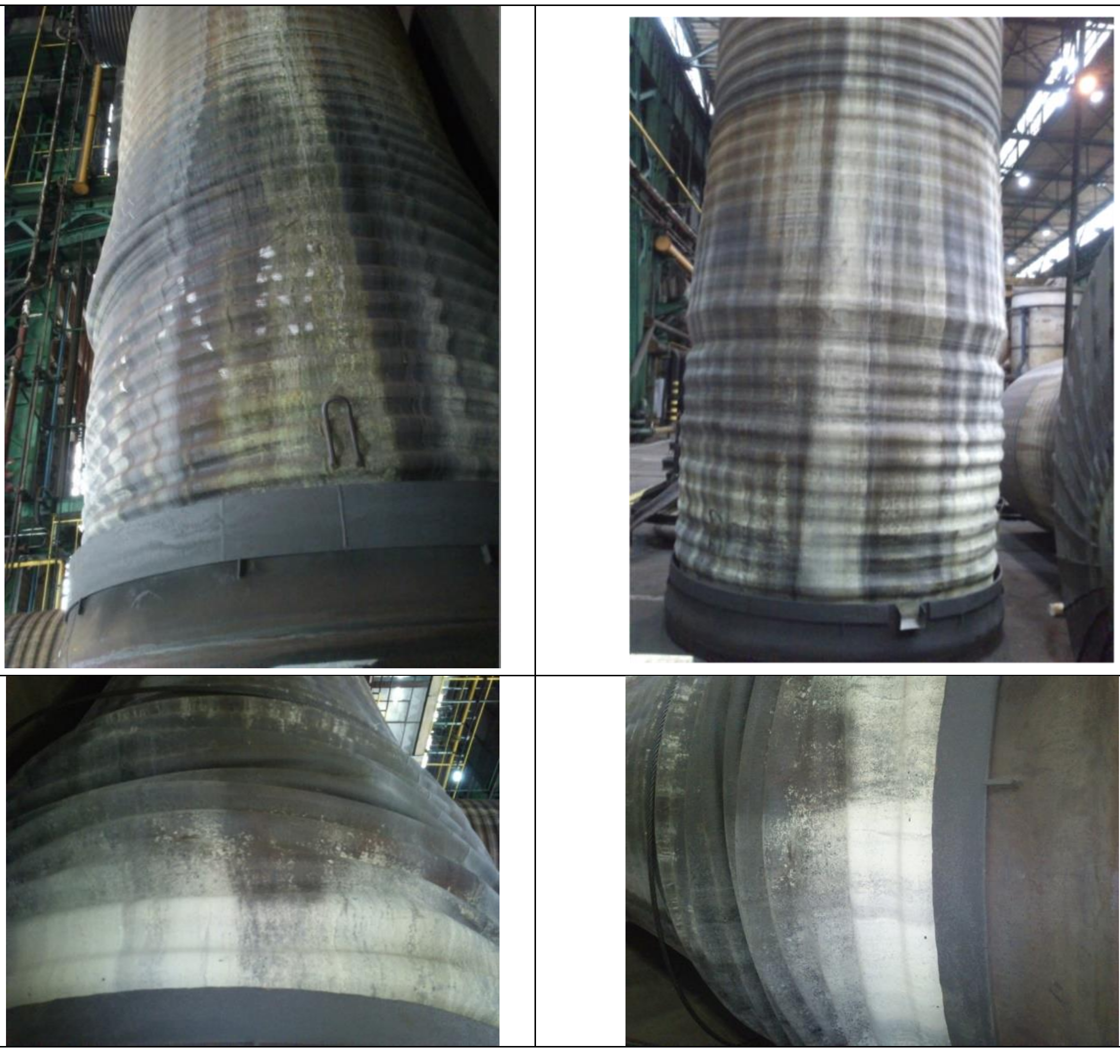

Figure 3. Some damages of the protective cover

2. *STRENGTH CALCULATION OF THE PROTECTIVE COVER*

The protective cover is used for the softening of steel sheet coils in safety gas. The protective cover is heated by gas from outside up to 1100 ^{0}C. The heat convection is made by the protection gas from outside of the wall of the protective cover. They are kept on 520-720^{0}C for 14-24 hours. The time cycle of the heat treatment is 68-70 hours.

The corrugated cylindrical protective cover is made of heat resistant WNr 1.4571 type steel. The three parts are welded together. The diameter of the bin is 2150 mm. The height of each part is 1400 mm. The thickness of the lower part is 6 mm, at the middle part is 5, and the upper part is 3 mm. The inner and outer diameters of the wave are 2150 and 2174 mm. The wave consists of two intermediate sloping parts and a mid-section with 40mm high straight line.

At the top of the protective cover there is a conical roof with 150 mm height and 3 mm thickness. The parameters of the protective cover material are as follows [1]: X6CrNiMoTi 17-12-2 austenitic steel, density 8 kg/m^3, tensile strength 520-670 MPa, 0.2% elongation limit, min $R_{p0.2}$ =220, elongation at break is 40%. Young modulus at room temperature is E = 2.1x10^5 MPa, on 1100 ^{0}C E_1 = 0.0225x210000 = 4725 MPa.

The typical failure of the cover was the deformation of the wave at the bottom of the middle part. Therefore, the purpose of the research is to assess the strength of the waves.

2.1 Loading of the corrugated plate at the lower part of the middle segment

The deadload of the protective cover at the lower part of the middle segment
Volume of the conical roof

$$V = \frac{2\pi t}{\sin\alpha}\int_0^h \frac{x}{\tan\alpha}dx = \frac{2\pi t}{\sin\alpha\tan\alpha}\left|\frac{x^2}{2}\right. = \pi t R\sqrt{R^2+h^2} = 3\pi 1075\sqrt{1075^2+150^2} = 10.9928x10^6 \text{ mm}^3 \qquad (1)$$

the mass $m = \rho V$=8x10.9928 = 88 kg

Volume of the middle and upper segment together

$$V_1 = 2R_{átl}\pi h_1(t_1+t_2) = 2x1081\pi 1400(3+5) = 76.07x10^6 \qquad (2)$$

the mass $m_1 = \rho V_1 = 8x76.07 = 609\,\text{kg}$

The total deadload acting on the lower part of the middle segment
88+609 = 697 kg (3)

The uniformly distributed deadload on the perimeter

$$F = \frac{6970}{2\pi 1075} = 1.032 \text{ N/mm.} \qquad (4)$$

2.2 Stresses in the corrugated plate

The deadload has a compression and a bending effect with 12 mm arm length. The normal stress from compression and bending are as follows:

$$\sigma_n + \sigma_h = \frac{1.032}{5} + \frac{1.032x12}{5^2/6} = 0.2064 + 2.9716 = 3.17856 \text{ MPa} \quad (5)$$

The corrugated bin has bent from the temperature difference between the outer and inner sides.

The thermal expansion of the inner wave radius is as follows: $\Delta R_b = \alpha_0 \Delta T_b R_b$, the outer $\Delta R_k = \alpha_0 \Delta T_k R_k$, the difference of the previous two:

$$\Delta R_k - \Delta R_b = \alpha_0(\Delta T_k R_k - \Delta T_b R_b) = 12x10^{-6}(1100x1087 - 720x1075) = 5.06 \text{ mm} \quad (6)$$

The *F* force has this arm length and bending moment and the calculated stress is as follows:

$$\sigma_{h1} = \frac{1.032x5.06}{5^2/6} = 1.2533 \text{ MPa} \quad (7)$$

The value of total stress assuming elastic stress distribution

$$\sigma_n + \sigma_h + \sigma_{h1} = 4.432 \text{ MPa}. \quad (8)$$

2.3 Damage of the corrugated plate

Due to the fact that there is not a given yield stress of the material, we cannot calculate the damage caused by yield point. As an approximation we consider the stresses in the wave as elastic ones on 1100 ^{0}C.The strain of the edge line using E_1 = 4725 MPa value for Young modulus is

$$\varepsilon = \frac{4.432}{4725} = 9.38x10^{-4} \quad (9)$$

We assume that this strain is repeated till damage. We consider the limit strain at tensile strength. The number of thermal cycles is as follows:

$$n = \frac{0.40}{9.38x10^{-4}} = 426\,. \quad (10)$$

It means that the corrugated plate roughly can have 426 thermal cycles. Calculate 20 hours for heating up and keeping on the temperature it means 8500 work hours. Limit strain of the material is 40% at 1100 ^{0}C.

2.4 Investigation of the corrugated bin

A local buckling may occur at the waveless part of the plate. Considering the 1100 ^{0}C temperature, the critical buckling stress is f_y = 0.02x350= 7 MPa, using E_1 = 4725 MPa Young modulus and approximated yield stress. The shell buckling according to Design guide of Det Norske Veritas (DNV) [2] is the following:

$$\sigma_{kr} = \frac{f_y}{\sqrt{1+\lambda^4}}, \lambda = \sqrt{\frac{f_y}{\sigma_E}} \quad (11))$$

$$\sigma_E = (1.5 - 50\beta)\frac{C\pi^2 E_1}{12(1-\nu^2)}\left(\frac{t}{L}\right)^2 \quad (12)$$

$$C = \sqrt{1+(\rho_0\xi)^2}, \rho_0 = 0.5\left(1+\frac{R}{150t}\right)^{-0.5}, \xi = 0.702Z, Z = \frac{L^2}{Rt}\sqrt{1-\nu^2} \quad (13)$$

The shortening of the radius due to the reduction of the circular weld is calculated according to β following the suggestions of [3].

$$\beta = \frac{u_{max}}{4\sqrt{Rt}}, u_{max} = 0.64 A_T \sqrt{\frac{R}{t}}, A_T = 0.844 x 10^{-3} \frac{Q_T}{t} \tag{14}$$

The heat input at the butt weld $Q_T = 60.7 A_W, A_W = 10t$ (15)

u_{max} is the shortening of the radius, A_W is the cross section of the circular butt weld, Q_T is the weld heat input.

Data are as follows: ν=0.3, t = 6 mm, R = 1075 mm, L = 1400 mm

σ_{kr}=3.488 MPa (16)

Stress comes from the deadload

$$\sigma_{n1} = \frac{1.032}{6} = 0.172 \text{ MPa.} \tag{17}$$

It is visible that $\sigma_{n1} << \sigma_{kr}$ so the 6 mm thick wave free shell is protected for local buckling.

2.5 Polygonal cover analysis - plate buckling at high temperature

The question is, how long can be the sheet cover between the corners not to buckle on high temperature?

The classical equation for sheet buckling is as follows:

$$\sigma_{cr} = \frac{k\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 \tag{18}$$

where k is the buckling parameter, the plate is simply supported, the loading is uniform compression: k = 4, ν = 0.3 the Poisson number, b is the width of the plate.

According to Eurocode 3-1-5 [4] no need to calculate effective width, if

$$\bar{\lambda} = \sqrt{\frac{f_y}{\sigma_{cr}}} \le 0.673 \tag{19}$$

From Eq. (19)

$$\sigma_{cr} \ge \frac{f_y}{0.673^2} \tag{20}$$

Combining equations (18) and (20)

$$\frac{b}{t} \le 0.40732 \sqrt{\frac{E}{f_y}} \sqrt{\frac{f_y}{\sigma}} \tag{21}$$

where using Eq. (3) values $\sigma = \frac{6970}{nbt}$ (22)

Due to the fact that in Eq. (22) b is unknown, the determination of b is done by iteration, the approximate value is

$$\sigma \approx \frac{6970}{2R\pi t} \tag{23}$$

If b is known the n-angle cover n value can be calculated from the following equation

$$b = 2R \sin\frac{180^0}{n} \tag{24}$$

In our case from Eq. (23) $\sigma = 0.172$ MPa, from Eq. (21) b = 405 mm, the number of n = 16.6, rounded to n = 18. With this value

b = 373 mm, from Eq. (22), $\sigma = 0.173$ MPa from Eq. (21), b = 403.9 > 373, so there is no buckling using n = 18 polygon cover on 1100^{0}C.

3. FINITE ELEMENT SIMULATION

There are some measurements and calculations made in the field on the effect of thermal treatment to the fatigue behaviour [5]. It can be useful for welded joints. If this thermal effect comes regularly, thermal fatigue occurs [6]. We have made some finite element calculations, to simulate this behaviour.

During the examination the original shape and the new shapes are compared to each other using the same boundary conditions. At the lower part of the protective cover at the sand bed in the bin axis direction the deformation is zero, in the other two directions the deformations are allowed. Loading comes from the deadload. The used temperature of the covers is uniformly 660 °C. During the calculation the stress state and the deformations have been analysed.

3.1. Analysis of the original protective cover

The original protective cover can be seen in Figures 1 and 4. Using the given load and boundary conditions the results are the followings:

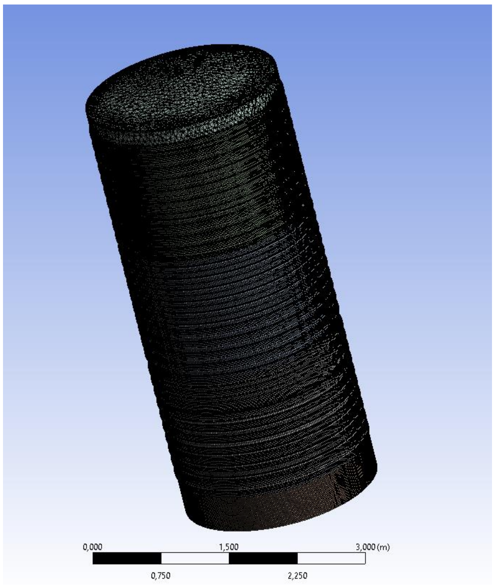

Figure 4. The mess of the original protective cover

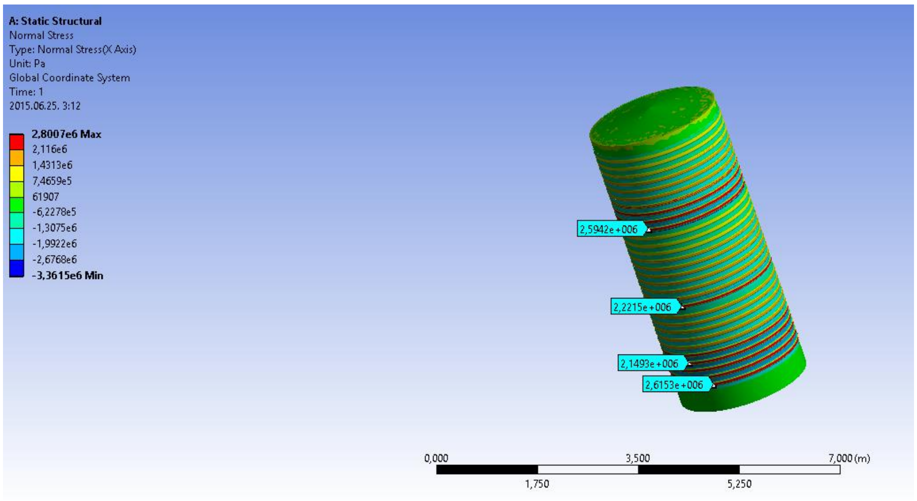

Figure 5. Normal stress

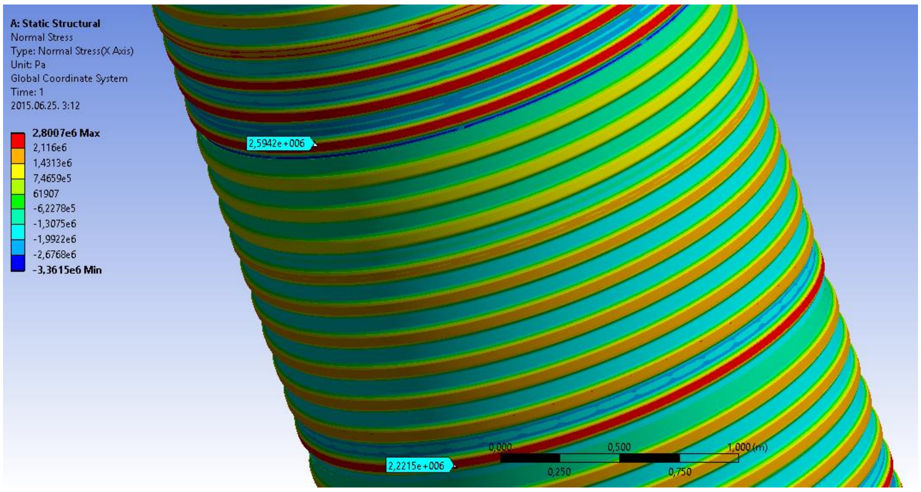

Figure 6. Normal stress details

The results of the FEM calculations (Figs. 5, 6) show good agreement with the hand calculation at the critical point ~2,5 MPa.

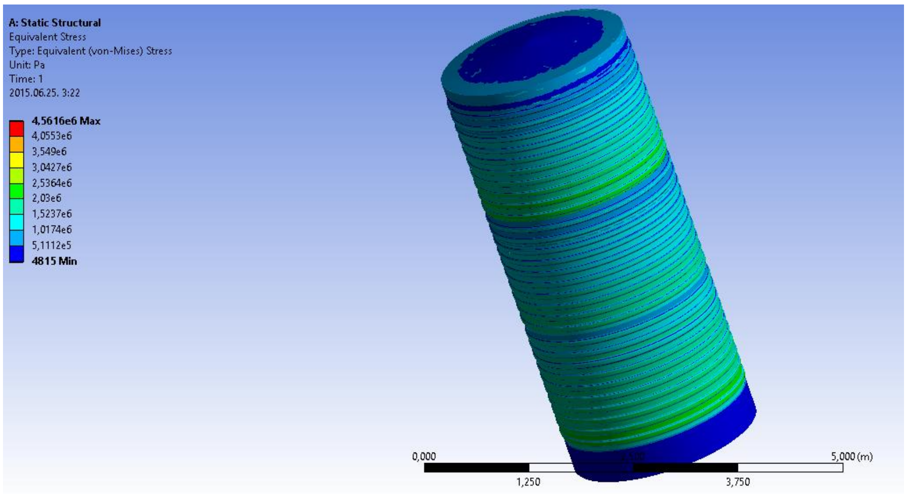

Figure 7. Equivalent stress

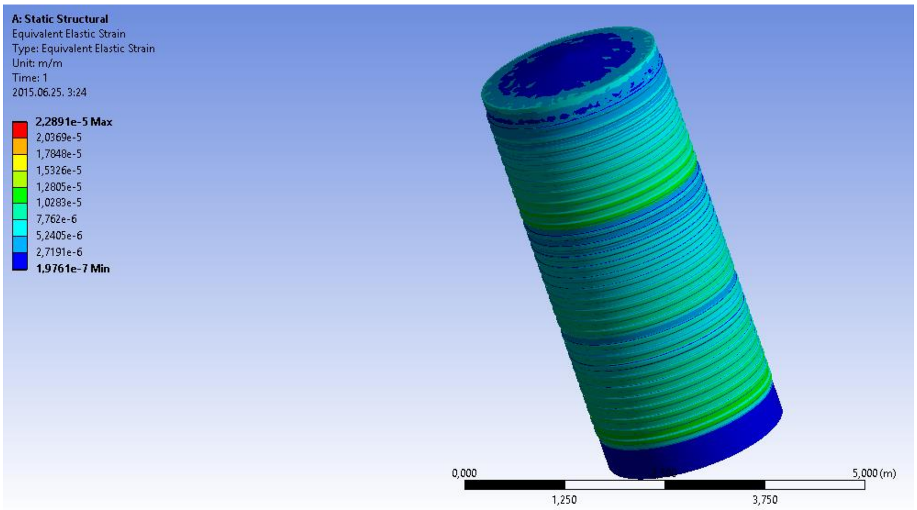

Figure 8. Equivalent elastic strain

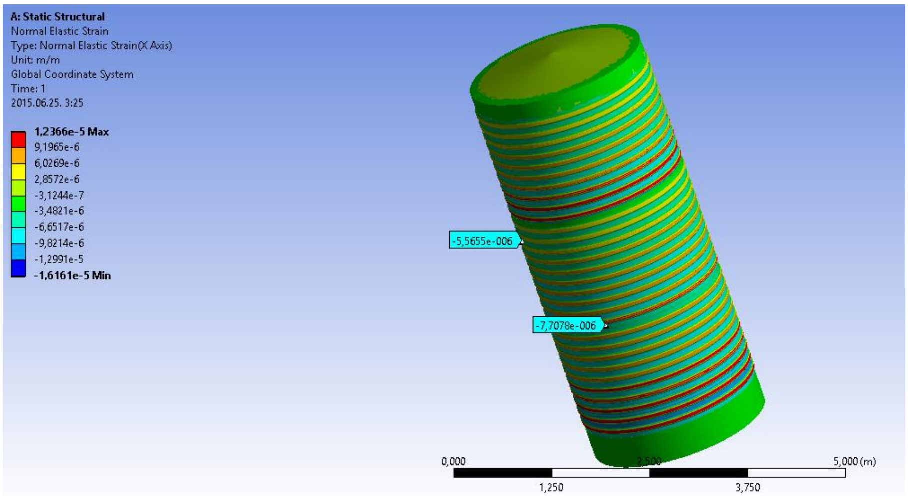

Figure 9. Normal strain

Figs. 7-9 show the equivalent stress, equivalent elastic strain and the normal strain distribution. It is visible, that the lower parts of the segments are in danger.

3.1.1. Remarks

The results of the previous study clearly support the results of manual calculations and are consistent with the actual damage. Based on the analysis, it can be seen that the highest stress and deformation occur in the lower parts of the different segments. The most critical is the middle segment because stress and deformation are larger not only at the peak of the wave, but also in its surroundings, and the values are higher compared to other sections (Figs. 7-9).

3.2. Vertical wave form

In the next structural layout the trapezoidal waveform is used as it described in the previous paragraph, but the position is different, they are vertical and not horizontal (see Figure 10). The thicknesses are: 6, 5, 3 mm, respectively Boundary conditions as described earlier.

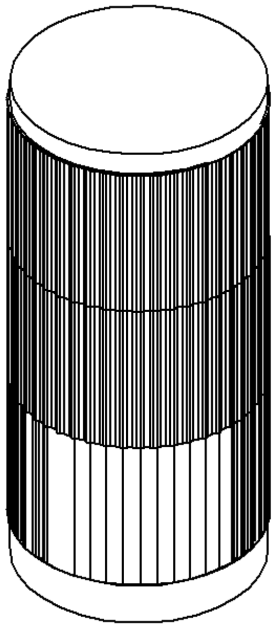

Figure 10. Vertical wave form

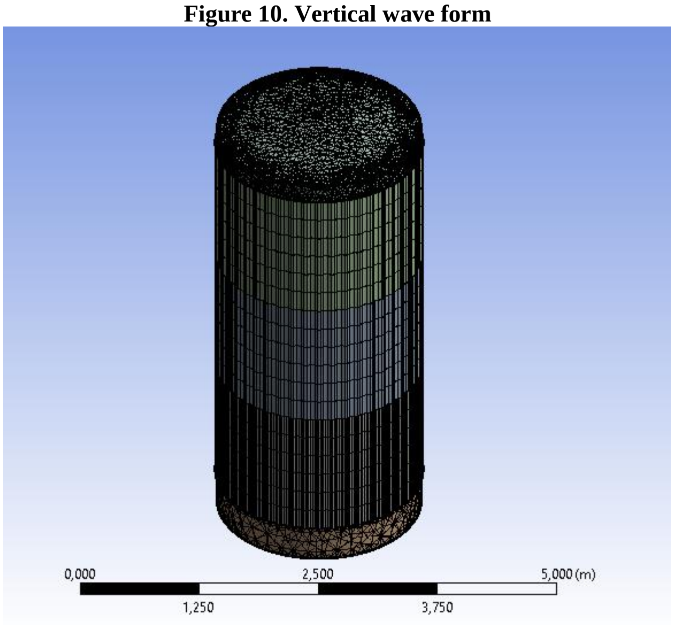

Figure 11. Mess of the cover

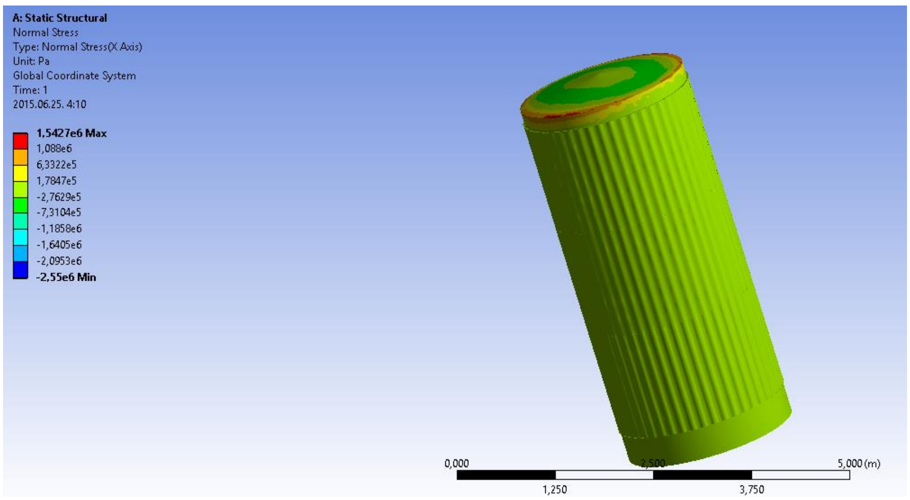

Figure 12. Normal stress

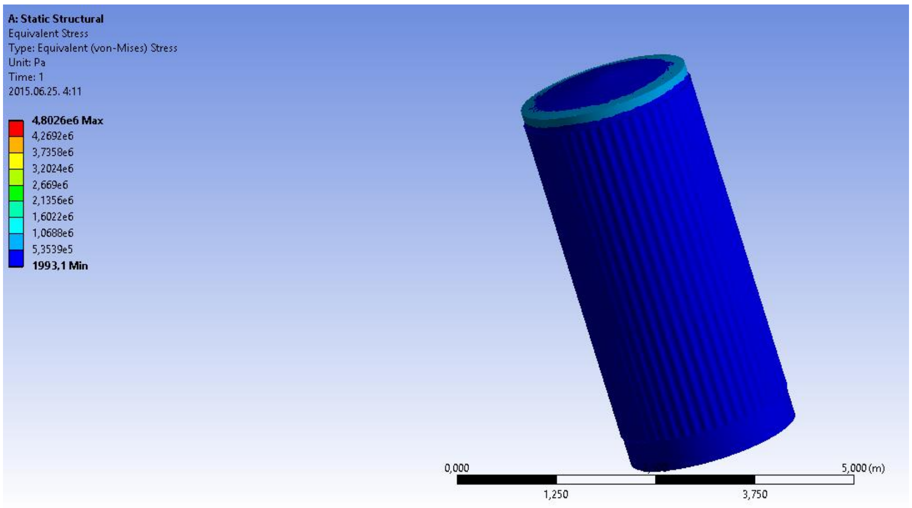

Figure 13. Equivalent stress

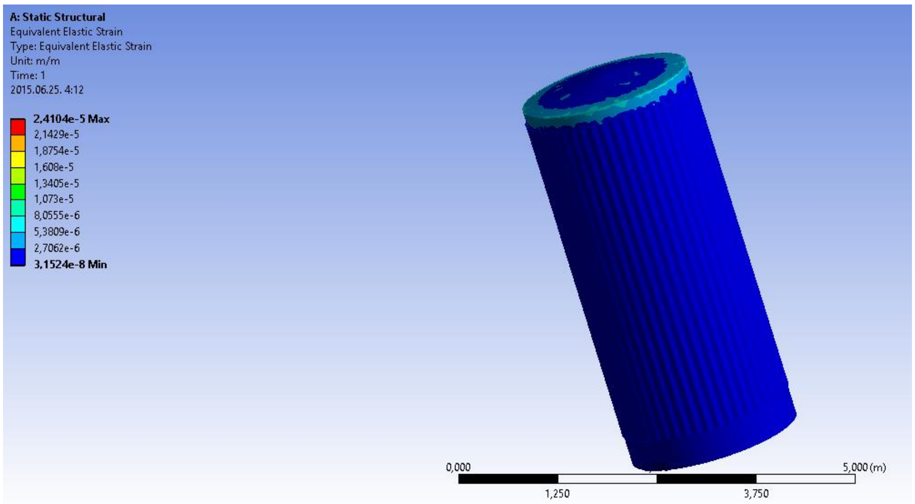

Figure 14. Equivalent deformations

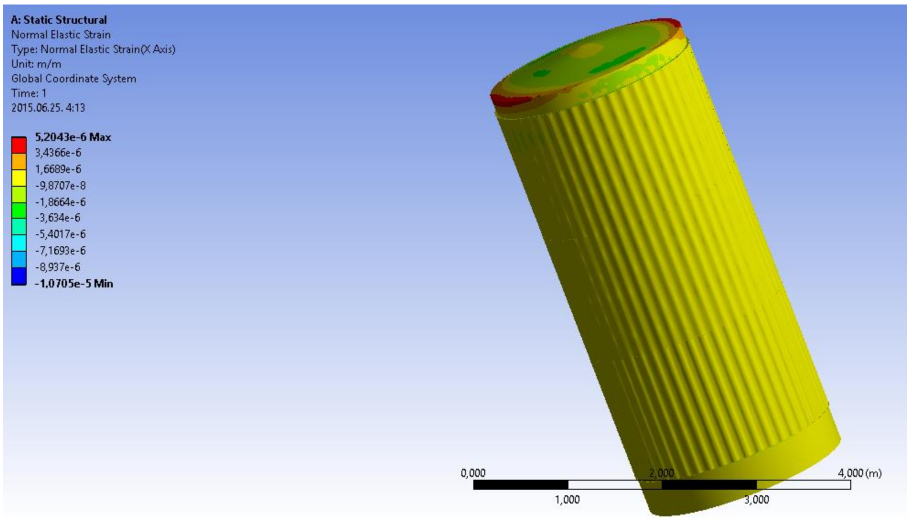

Figure 15. Normal strain

3.2.1. Remarks

The finite element calculations show, that the vertical wave is useful and the strength of the protective cover is larger. Unfortunately, there is a large drawback: joining the different segments with different thicknesses is difficult (Figs. 11-15).

3.3. Bent plate

The next versions are bent plate covers. The following 4 different bent plates are employed in Figure 16. The difference between them is the number of the bents on a given diameter. This way we have chosen the followings:

- Type 17 (17 bents)
- Type 23
- Type 34
- Type 67.

We deal with the first 3 types, because for manufacturing considerations the last type (67) is not optimal. First the 3 types are compared to each other. The plate thicknesses are identical: 6, 5, 3 mm at the lower, middle and upper segments. The end conditions are according to the previous ones (Fig. 16).

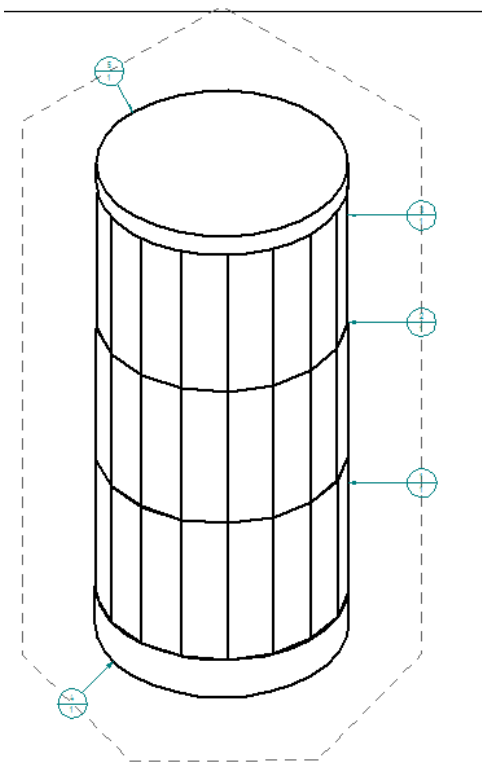

Figure 16. The bent plate geometry

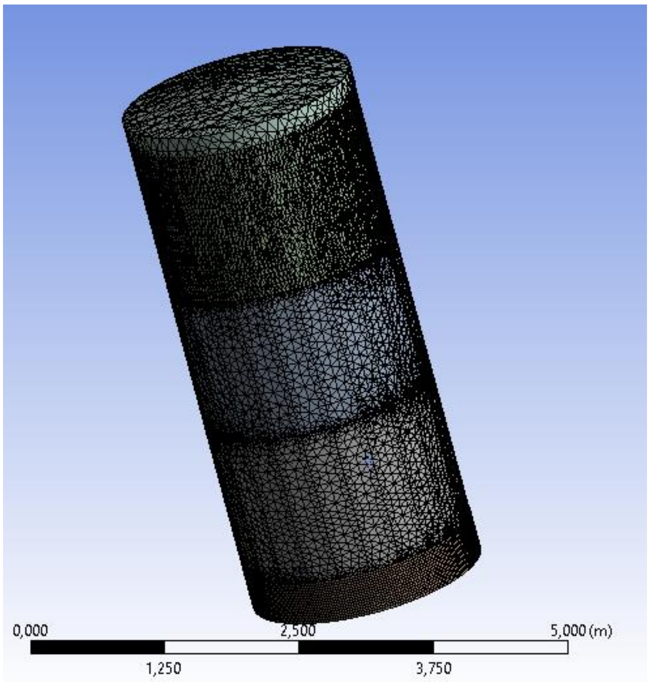

Figure 17. Mess of the cover

The finite element calculation is made for type 17, 23, 34. The equivalent stress, the equivalent elastic deformations are shown for type 17 (Figs. 18, 19), for type 23 (Figs. 20, 21), for type 34 (Figs. 22, 23). In all cases the most critical is the lower part of the different segment, especially the lowest one.

3.3.1. Type 17

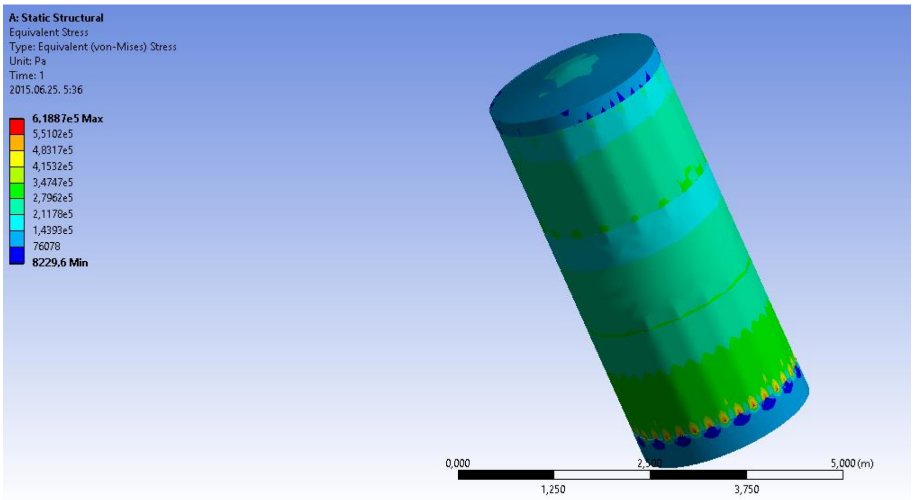

Figure 18. Equivalent stress

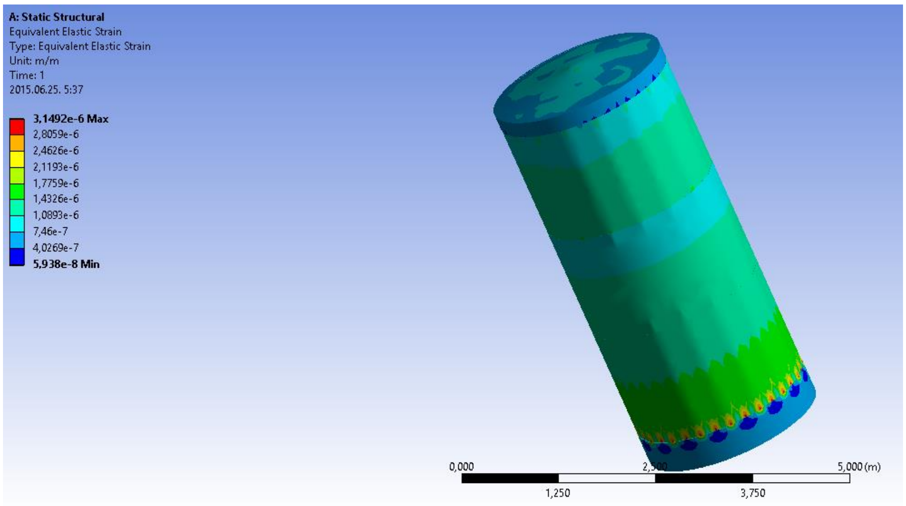

Figure 19. Equivalent elastic deformations

3.3.2. Type 23

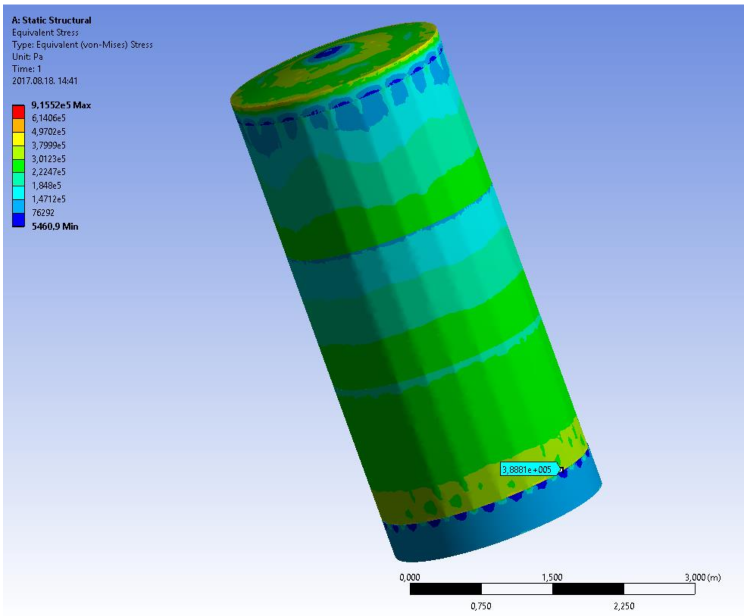

Figure 20. Equivalent stress

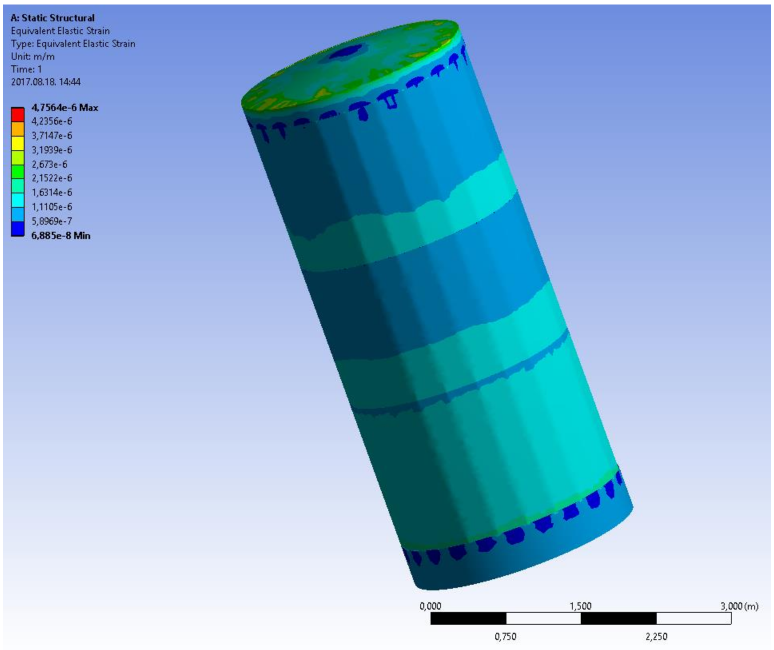

Figure 21. Equivalent deformations

3.3.3. Type 34

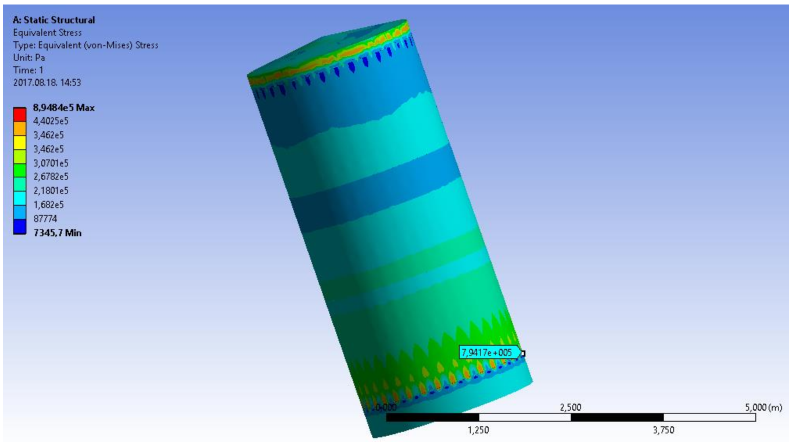

Figure 22. Equivalent stress

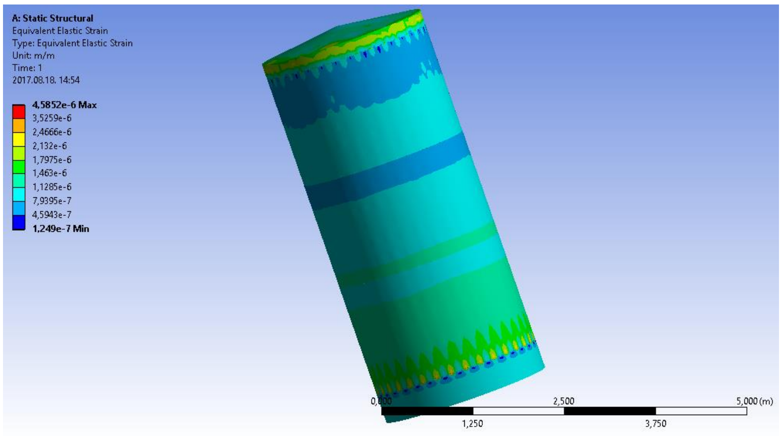

Figure 23. Equivalent elastic deformations

3.3.4. Remarks

Comparing the FEM results of the three versions and also taking into consideration manufacturing aspects, Type 23 is the best solution.

4. CONCLUSIONS

A welded steel protective cover is investigated, which is used for the heat treatment of steel sheet coils. Protective covers are made of austenitic stainless steel. The aim of the investigation was to improve the lifetime, the number of heat cycles of the protective cover changing the geometry, thicknesses and material. We have evaluated the damaged covers, made calculations on stress and deformations and a series of finite element models are investigated, comparing the behaviour of them. According to Chapter 3, the best results have been got are visible and for comparison, we can say, that the smallest stress belongs to type 23 bin protective cover. The original geometry and the horizontal corrugated plate are identical from the stress level point of view. The vertical position of the corrugated plate gave a better result to the horizontal one. The result of the original geometry is close to the measured damage.

Further increase of lifetime belongs to the application of better steel grade. This effect can be validated by experiments, using thermal resistant steel instead of the used stainless steel.

Whatever kind of steel grade is used, it is important to validate the effect of geometrical or material changes by experiments.

Acknowledgement

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