Albanian j. agric. sci. 2017;16 (3): 114-120 *Agricultural University of Tirana*

RESEARCH ARTICLE **(Open Access)**

Static analysis of supporting structure – casing of threshing machine in a Polish combine harvester

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Abstract

The article describes static analysis of threshing machine casing, which is at the same time a supporting structure of a combine harvester manufactured by New Holland Bizon Ltd. The model of the examined machine with Z110 thresher rotor was being produced until 2004, and further modifications of the threshing system are being made to date [1]. To conduct the study, the authors used programs: Inventor 2017 to create a digital model, and Nastran In-CAD 2017 to perform simulation calculations. The purpose of this article was to scrutinize and assay the construction of supporting structure with its damages arising from defects of design, as well as to conduct a static analysis of the object of study with the employment of finite elements method (FEM), using CAD/CAE software and ideas for solving constructional problems by reducing stresses, which may cause failures and, in consequence, unplanned downtime during harvesting. Ultimately, static analysis of proposed solutions was conducted.

Keywords: combine harvester, statistic finite elements analysis, internal stresses.

1. Introduction

The first Polish self-propelled combine harvester known as Vistula was manufactured in the 1950s in Fabryka Maszyn Żniwnych in Płock [2]. As far as separation of grains from the chaff is concerded, its mechanism has not changed considerably since then. However, the construction of threshing machine has been modified substantially. One example is that straw walkers have been replaced by rotor shakers. This solution contributed to much quicker conveying of the straw separated from grains outside the combine harvester or to straw chopper, depending on the combine harvester equipment. An additional benefit is more careful threshing of plants remnants, nevertheless, it happens at the expanse of straw quality. Constructions manufactured in the previously mentioned factory had design faults which needed to be studied with the use of contemporary simulation technology. The geopolitical situation of Poland – one of the Eastern Bloc countries until 1989, did not foster innovations.

On the grounds of users' opinion, as well as empirical observation, a failure of the front part of a threshing machine casing was detected. The object of study is the combine harvester New Holland Bizon BS Z110 (Fig. 1).

2. Methodology

The problem is cracking and detaching of a profile which serves both as a tank bearer and clamping of a feeder house. The machine users try to fix the damage with butt and fillet welding, however the failure recurs. Basing on the literature research, the authors did not observe previous ideas or trials of finding reasons of this problem. The element which is being damaged is shown on the (Fig. 2b). After divagations it was decided first to conduct a static analysis of the object of study for its actual working conditions, in order to test what is the dependence between the total mass of the machine with empty and full grain tank and the structural strength. With the aim of making the static analysis in the program Nastran In-CAD, a digital model of the threshing machine casing was

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(Accepted for publication September 20, 2017)
***ISSN**: 2218-2020, © Agricultural University of Tirana*

created in Inventor 2017 (Fig. 2a). The model is an accurate depiction of every dimension on a scale of 1:1. Apart from the static analysis, there is also a necessity to conduct a dynamic analysis examining the influence of vibrations generated by threshing, separating-cleaning, and chopping unit (despite their static balance), which will be the topic of further publications.

Figure 1. Overview picture of the object of study. Source: own elaboration.

The examined part of the construction is presented with explanation on the digital model on (Fig. 2b). Empirical observations showed that the cracking occurs on junction of square profile (1) and rectangular profile (2) with feeder house (3). As to obtain previously mentioned aims, it was necessary to do a series of static calculations and perform a simulation of strength on a digital model of combine harvester supporting structure.

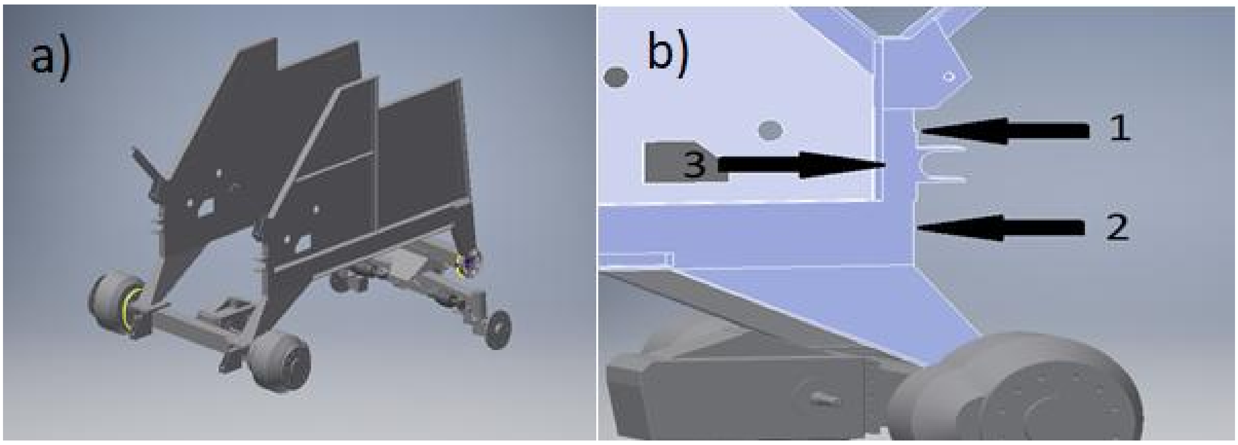

Figure 2. Digital model of supporting structure (a) and the analyzed parts of the combine harvester (b). Source: own elaboration.

Digital simulations of stress distribution were created in Autodesk Nastran In-CAD 2017. This program bases on linear static analysis and enables making calculations with the employment of finite elements method (FEM). Its default model is a linear elastic modulus described by Von Mises equation. In order to implement supports absorbing degrees of freedom (Fig. 4), fixed block joints were put in place of front wheel drive hub (2) and conjunction of rear suspension with the vehicle frame (1), and it set a degree of freedom only in the horizontal direction, which proceeds from possible vertical movements of the machine on fractional distance. This is the result of working of the elements which are propelled to reciprocating motion (e.g. sieve or chaff riddle). Basing on divagations, it was assumed that this solution is not necessary, nevertheless it can influence the outcome of strength simulation. On the grounds of empirical observation, the authors presumed that there is no reason to examine hubs, as no defect of these elements was detected, what is connected with massive cast-iron construction. Moreover, hydraulic cylinders, their clamping, transmissions, and feeder house, which mass and dimensions were used to calculate torque affecting frame profile, were also excluded from the analysis.

According to technical specification of the combine harvester [5], the mass distribution is 70% on the front axle and 30% on the rear axle, considering a variant with a straw chopper and auger header, which is presented on Figure 3. For the purpose of performing calculations, it was determined that the grain in tank is wheat and its average mass is 0.76 kg for 1 dm^3 [3], which gives the mass of 3800 kg for the tank capacity 5000 dm^3. The combine harvester mass with an empty grain tank equals 9940 kg, and the mass of the auger header is 1420 kg. On the basis of the evaluated masses, it was calculated that their total amount is 11360 kg for an empty grain tank and 15160 kg for a full grain tank. Subsequently, the mass loading rear suspension was estimated: 3408 kg for the variant with full tank and 4548 kg for the variant with empty tank; and the load of front suspension: 7952 kg for an empty tank and 10612 kg for a full hopper (Fig. 3). Values of masses and forces are presented on Table 1, and mass distribution is shown on Figure 3. Forces which are employed to the analysis in order to load the digital model were calculated basing on the mass, and their values are 79520 N and 106120 N respectively, with regard to proper force distribution towards mass distribution for the front and rear axle. So as to obtain value of torque affecting the frame front profile, it was estimated that the length of feeder house from the clamping on the supporting frame to the clamping of auger header equals 2540 mm, and the mass of auger header with rapeseed cutter bar is 1890 kg [5]. On the grounds of these physical quantities, the torque of value 48006 Nm was obtained, in the positive clockwise direction, when observing the object from the right side. Forces (4) were placed on the top flat surface of the tank in compliance with their direction (Fig. 4), and the torque (3), generated by feeder house, was applied to the place of movable support which holds the combine harvester feeder house. In accordance with [5], the frame material is high-strength low-alloy steel St35/S235JR, which values are [6]:

- Young's modulus = $210 \cdot \cdot 10^3$ MPa,
- Poisson's ratio = 0.3,
- elastic limit (Re) = 235 MPa,
- strength limit (Rm) = 380 MPa,

After input of boundary conditions and other data to the model, a mesh consisting of 85277 elements linked together with 168739 joints was created. The digital model of the threshing machine casing is statically determinate. By virtue of considerable differences in the machine masses, which change along with the grain tank filling ratio, it was decided to conduct two separate analyses for two extreme variants – with full and empty tank. In result, forces acting on construction elements had different values.

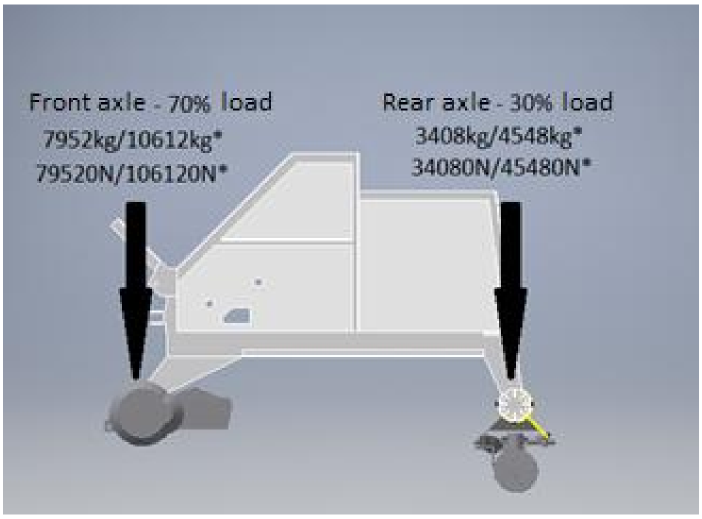

Figure 3. Distribution of the mass and load acting on the front and rear axle, for the variant with empty and full tank (*). Source: own elaboration.

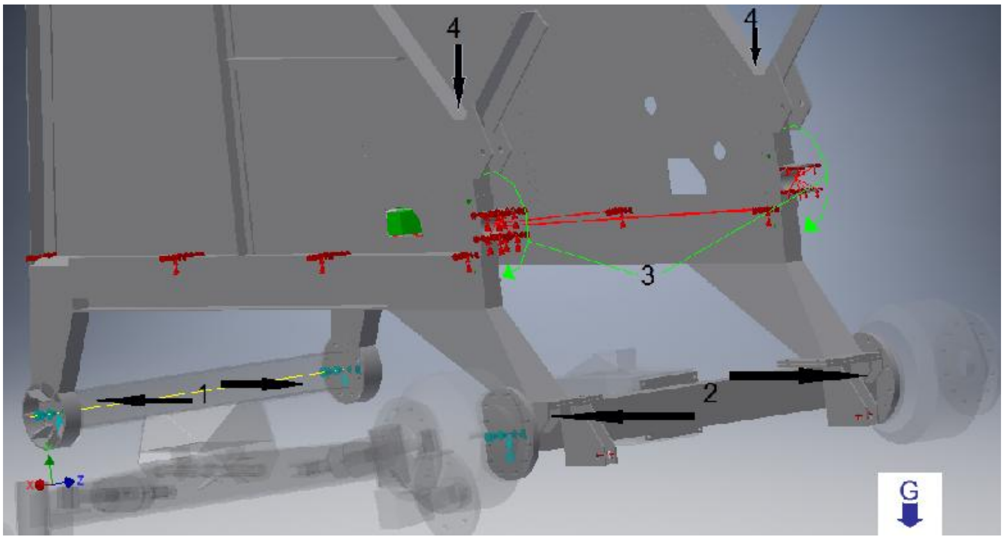

Figure 4. The placement of supports, forces and torques in the digital model. Source: own elaboration.

Table 1. Evaluation of mass of the combine harvester, its accessories, grain tank content, axles load and forces, and torques used in the static analysis [5]. Source: own elaboration.

Mass type	Mass value (kg)	Loading force value (N)	Torque value (Nm)
Combine harvester mass	9940	-	-
Maximum mass of wheat in grain tank	3800	38000	-
Auger header mass	1420	14200	3606.8
Mass of the combine harvester with auger header (empty tank)	11360	113600	-
Mass of the combine harvester with auger header (full tank)	15160	151600	-
Mass loading the front axle (empty tank)	7952	79520	-
Mass loading the front axle (full tank)	10612	106120	-
Mass of the auger header with rapeseed cutter bar	1890	18900	48006

Passing the elastic limit (Rm) by stresses leads to failure of the element and, in consequence, to damage of the combine harvester construction and unplanned downtime during harvesting. In order to reduce stress, the authors propose several solutions which base on the construction reinforcement by welding profiles to the most troublesome spots. The digital model was supplemented by elements dissipating stress (Fig. 5). Then, a mesh for each variant was modified and further analyses were conducted. Weld modelling was excluded from the study, as its strength was presumed to be equal to the strength of the vernacular material [4]. Supporting elements were made of high-strength low-alloy steel St35/S235JR in every variant, and this material was also set in simulation. The width of each element in section is 10 mm. So as to reinforce the construction, the elements of the following shapes and dependences are employed:

- angle profile of dimensions given on (Fig. 5a). This element is joined with the threshing machine casing with butt weld laid on the profile's perimeter. The modified mesh consists of 85612 elements and 169417 joints.

- pentagonal profile (Fig. 5b), where the conjunction is made with the use of butt weld applied on the perimeter. After changes, the mesh is made of 86521 elements and 171189 joints. However, it should be noted that in this solution a side hole of thresher apperture is marginally covered, which results in restricted access to this element.

- triangular profile, which functions as a support for the feeder house (Fig. 5c). The method of joining it to the construction is fillet welding. This solution in combination with the foregoing construction generated 101528 elements and 152833 joints.

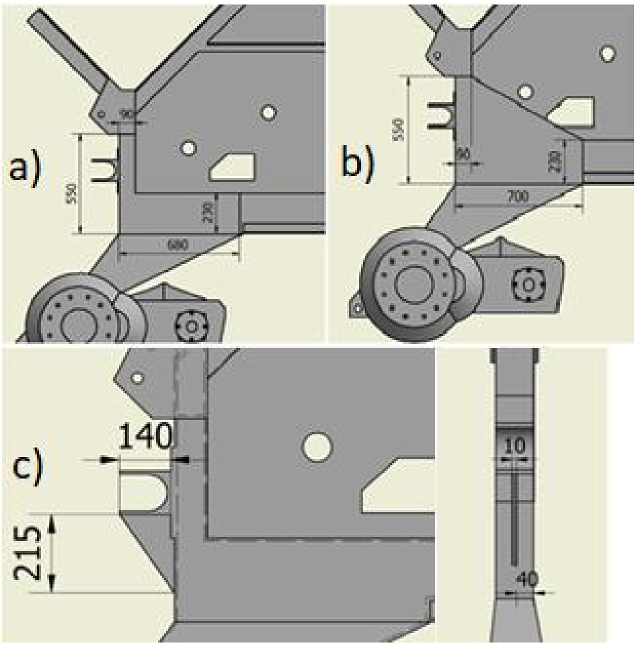

Figure 5. Overview drawing presenting the ideas of modification of the analyzed construction part. Source: own elaboration.

3. Results and Discussion

The static analysis provided masses values (Von Mises): 582.7 MPa (Fig. 6a) for the variant with empty grain tank and 585.9 MPa (Fig. 6b) for the variant with full grain tank. However, the authors provide for the possibility of occurring of an uncertainty of measurements and calculations, and due to this fact the obtained results should be multiplied by an appropriate safety factor, which is based on accepted standards [4]. In case of standard accuracy of calculations determining force, torque, and stress, a safety factor's value vary between 1.5 ÷ 1.7. Presuming safety factor 1.5 for this study, the new stress values are 874.05 MPa for an empty tank and 878.85 MPa for a full tank. The results of the static analysis are presented on Table 2.

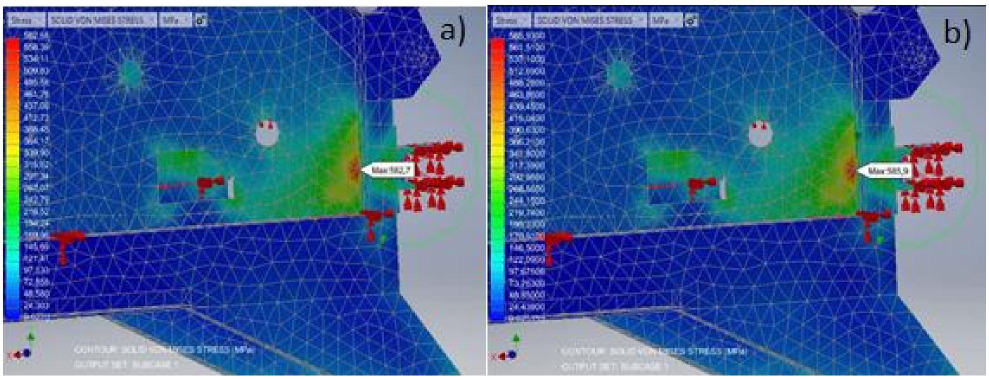

Fig.6. Simulation results for the variant with empty tank (a) and full tank (b). Source: own elaboration.

Table 2. Stress values (Von Mises) obtained in static analysis. Source: own elaboration.

Variant	Stress obtained in static analysis (MPa)	Stress values with regard to the safety factor 1.5 (MPa)
Empty tank	582.7	874.05
Full tank	585.9	878.85

Linear static analysis was conducted again for the three variants of construction reinforcement, and the results were multiplied by the previously determined safety factor. As an outcome, Von Mises stresses has been obtained, which values are:

- angle profile (Fig. 7a) - 210 MPa calculated in the static analysis, and 315 MPa considering the safety factor,

- pentagonal profile (Fig. 7b) – 77.1 MPa and 115.65 MPa respectively,

- triangular profile (Fig. 7c), which functions as the feeder house support - 149.5 MPa and 224.25 MPa respectively.

When scrutinizing the results, it can be noticed that all of them keep within the elastic limit (Re = 235 MPa) and strength limit (Rm = 380 MPa). However, after taking the safety factor into consideration, it is visible that only the variants with pentagonal profile (Fig. 7b) and triangular profile (Fig. 7c) do not pass the elastic limit. The outcomes of the static analysis are presented on Figure 7, and values of the results are described on Table 3.

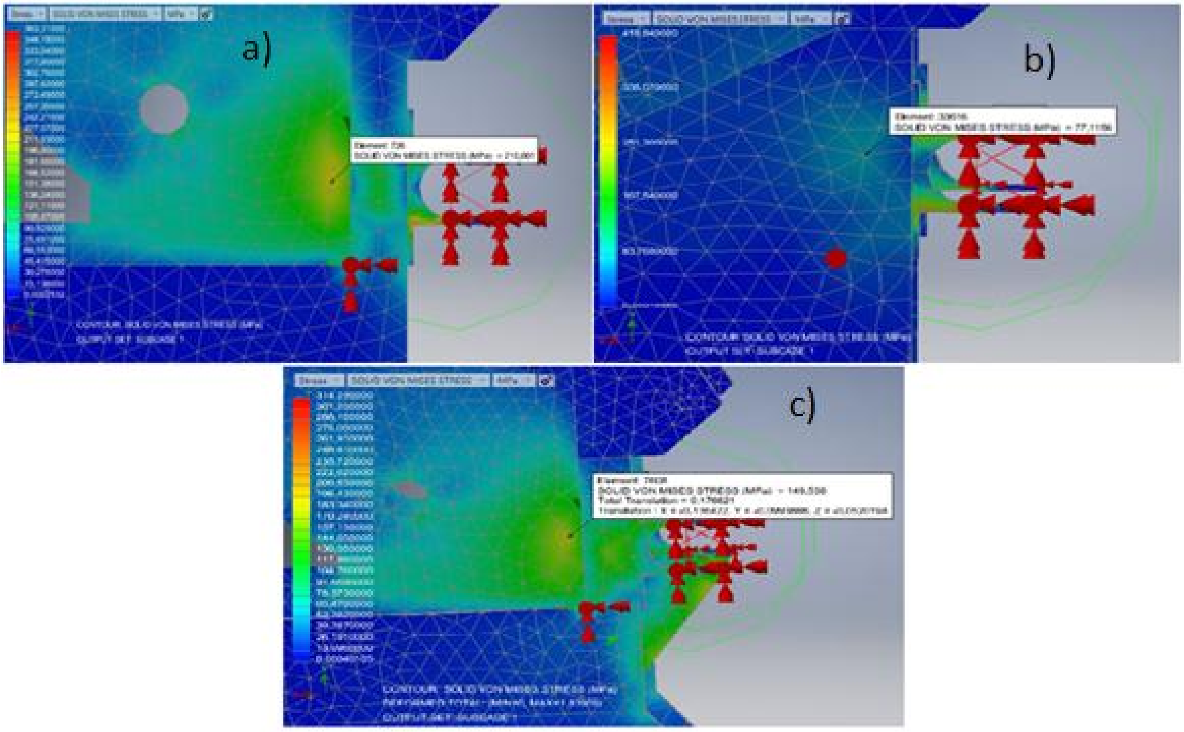

Figure 7. Overview graphic showing the results of static analysis conducted in the program Nastran In-CAD, for: a) angle profile, b) pentagonal profile, and c) triangular profile, which functions as a feeder house support. Source: own elaboration.

Table 3. Stress values (Von Mises) obtained in static analysis of the modified elements. Source: own elaboration.

	Mass value (MPa)	Stress reduction value (%)
Stress value after modification of construction		
angle profile	210	64
pentagonal profile	77.1	86.8
triangular profile	149.5	74.3
Stress value with regard to the safety factor 1.5		
angle profile	315	64
pentagonal profile	115.65	86.8
triangular profile	224.25	74.3

4. Conclusions

On the grounds of conducted analyses and examinations, it can be concluded that:

•The static analysis demonstrated that the construction's stresses are too high (Von Mises): 582.7 MPa for the variant with empty grain tank and 585.9 MPa for the variant with full grain tank. These values pass elastic and strength limits considerably. Considering the safety factor 1.5, the new stress values are 874.05 MPa for an empty tank and 878.85 MPa for a full tank. The results are presented on Table 2.

• Stress reduction percentage value is 64% for the variant with angle profile, 86.8% for the variant with pentagonal profile, and 74.3% for the variant with triangular profile – Table 3.

•Stress values after modifications equal 210 MPa for the first alternative; 77.1 MPa for the second variant, and 149.5 MPa for the third one, and with regard to the safety factor: 315 MPa, 115.65 MPa i 224.25 MPa respectively, which is presented on Table 3.

•The ideas of modifications proposed by the authors improve the strength of construction significantly. After analyses including modifications and the safety factor, stresses for the variants with pentagonal and triangular profile were reduced to the level below the elastic limit. Considering the alternative with angle profile, the stress value passes the elastic limit.

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