Katarzyna SZWEDZIAK, Przemysław WINIARSKI
Opole University of Technology, Faculty of Production Engineering and Logistic, Department of Biosystems Engineering
e-mail: k.szwedziak@po.opole.pl

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ANALYSIS OF VIBRATION OF COMBINE HARVESTER STRUCTURE FOR THE ELEMENTS EXCESSIVELY EXPOSED TO STATIC LOAD

Summary

The article presents the static analysis of threshing machine frame and the bracket of rear axle beam of a combine harvester. To conduct the study, two programs were used: Inventor 2017 to create the model and Nastran In-CAD 2017 to perform simulation calculations. The purpose of the article is to present the design of the support frame of a combine harvester along with its defects, and to carry out research using the finite element method by means of CAD/CAE software. A solution to design defects has been proposed, which involves the reduction of stress. In addition, vibration was measured in places that were excessively exposed to static load.
Key words*: vibration value analysis, static analysis, combine harvester*

BADANIA I ANALIZA DRGAŃ KONSTRUKCJI KOMBAJNU ZBOŻOWEGO DLA MIEJSC NADMIERNIE OBCIĄŻONYCH STATYCZNIE

Streszczenie

Artykuł prezentuje analizę statyczną kadłubu młocarni oraz wspornika belki tylnej kombajnu zbożowego. Dla przeprowadzenia badań użyto program Inventor 2017 dla budowy modelu oraz program Nastran in- CAD 2017 dla przeprowadzenia obliczeń symulacyjnych. Celem artykułu jest przedstawienie konstrukcji ramy nośnej kombajnu zbożowego, jej wad oraz przeprowadzenie badań z wykorzystaniem metody elementów skończonych przy użyciu oprogramowania typu CAD/CAE. Zaproponowano rozwiązanie wad konstrukcyjnych zmniejszające naprężenia. Dodatkowo dokonano pomiaru drgań w miejscach nadmiernie obciążonych statycznie.
Słowa kluczowe*: analiza wartości drgań, analiza statyczna, kombajn zbożowy*

1. Introduction

It is well known that the agricultural sector is relatively resistant to mechanization. One example of this is a combine harvester. The first combine harvesters were created in the 1940s in the US, while, as regards the European market, the first German self-propelled combine harvester was constructed in the 1950s in the Federal Republic of Germany [1]. Since then, their principle of operation has not changed significantly in terms of separation grain from chaff. However, the main changes were related to the method of threshing. The replacement of straw walkers with rotary separators is an example of these changes. This solution contributed to much faster feeding of mostly separated straw from grain outside the combine or towards the shredder, depending on whether the combine was equipped with such equipment. As it turned out, the additional advantage was that plant residues could be threshed more precisely, however at the expense of the quality of straw. Another example is the fact that Bizon Z060 was rarely equipped with a screw conveyor to the hopper [2], a construction of which was removed due to weak design and replaced with a paddle conveyor, and now it is still used in high-performance machines of this type, manufactured all over the world [3]. The structures produced by the aforementioned factory were not free from structural defects, which required testing by means of simulation technologies that were modern at the time. The geopolitical situation of the country, which was under the Soviet regime until 1989, did not encourage the improvement of the situation of implementing new technologies. Due to difficulties in accessing computer hardware and specialized software, the process of optimization of combine harvesters' design was very slow or even stopped. Therefore, despite the fact that the structure of Bizon combine harvesters was considered to be very successful, it was not free from defects or shortcomings.

On the example of the last manufactured combine harvesters - Bizon Z110, one can observe the occurrence of problems related to cracking of the fixing system of suspension of the rear axle of the combine harvester, as well as the front part of the threshing machine frame. The problem was so serious that farmers repeatedly performed current repairs to the fixing of the rear axle or modified the original design. It should be mentioned that repairing the damage by means of welding does not bring the desired results. Shortly after the repair, the failure appears again. What is more, the described problems intensify when mounting a straw shredder to the rear part of the combine harvester and a rapeseed attachment to the front of the machine. Therefore, this is the weak design of the harvesters that can be considered as the source of these problems. The conducted literature analysis indicates that the manufacturer has not attempted to solve the problem described. Due to the above, the authors decided to conduct numerical studies for an exemplary Polish combine harvester and determine the real cause of problems with fixing the rear axle suspension. The discussed combine harvester is shown in Fig. 1.

After reflecting on this matter, it was decided to first perform a static analysis of the object of study for the actual conditions use in order to check what is the relationship between the total vehicle mass with an empty and full grain tank and the strength of the structure. In order to perform static analysis using the Nastran In-CAD program, the numerical model of the combine harvester support frame was created in Inventor 2017 (Fig. 2). The model faithfully represents each dimension in 1:1 scale. In addition to static analysis, it is necessary to measure vibrations in the places of location of work systems that emit vibrations which mainly include: threshing, separating and cleaning, as well as shredding systems - despite their static balancing.

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Fig. 1. New Holland Bizon BS Z110 combine harvester
Rys. 1. Kombajn zbożowy New Holland Bizon BS Z110

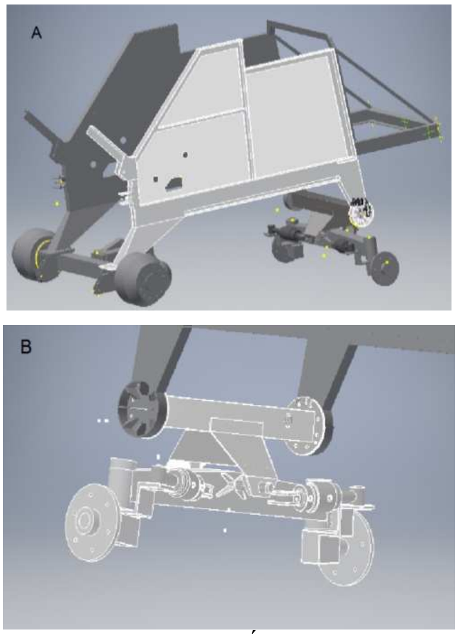

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Fig. 2. Numerical model of the combine harvester frame created by means of Inventor program, A) threshing machine frame, B) suspension of the rear axle
Rys. 2. Model numeryczny ramy kombajnu zbożowego stworzony za pomocą programu Inventor, A) kadłub młocarni, B) zawieszenie tylnego wózka

2. Research methodology

The official Nastran In-CAD 2017 software by Autodesk was used to conduct numerical simulations presenting the stress distribution. This program is based on linear static analysis and performs calculations using the finite element method (FEM), and its default material model is the linear elastic model described by the von Mises equation. In order to create supports reducing the number of degrees of freedom, fixed locking constraints were positioned in the place where hub reduction gears of both front and rear wheels are located. To simplify the analysis, the authors assumed that there is no need to study steering knuckles or hub reduction gears because, as shown by empirical study, no defects were found in this area of the structure. Therefore, the components of hydraulic cylinders, as well as their fixing and the hub reduction gear of the front axle were also excluded from the analysis. Degrees of freedom have been determined only in the horizontal direction, which results from the slight possible movements of the machine in the vertical direction during operation. This is the result of work of the components driven in a reciprocating movement, such as the sieve basket and the grain pan. Based on divagations, it was assumed that such a solution to reduce the number of degrees of freedom is not necessary but it may affect the result of strength simulations. The inclined conveyor whose mass and dimensions were used to calculate the moment acting on the frame profile, was also excluded from the analysis. As regards the rear axle, it was decided to analyze the components of the walking beam and support frame bracket, as shown in Fig. 3b. Fixed brackets were mounted at the place of the steering knuckle and forces of a given value were applied to the place where the threshing machine frame was attached to the rear axis beam [9]. The values of forces are presented in Table 1.

To calculate the value of the moment acting on the front frame profile, it was estimated that the length of the inclined conveyor from the fixing point at the support frame to the coupling of the header is 2540 mm and the mass of the header together with the rapeseed attachment is 1890 kg [4]. Based on these values, a positive moment of 48006 Nm has been calculated. The moment generated by the inclined conveyor was applied at the place of location of the pivot bearing supporting the inclined conveyor of the combine harvester (Fig. 3A).

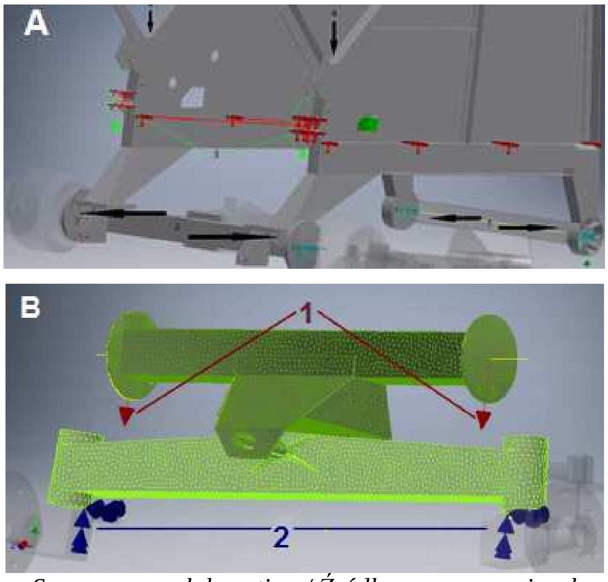

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Fig. 3. A) threshing machine frame – presentation of the application points: supports (1, 2), moments (3), force caused by the mass of grain in the tank (4), B) rear axle - forces (1), supports (2)
Rys. 3. A) kadłub młocarni - ukazanie miejsc przyłożenia: podpór (1, 2), momentów (3), siły powodowanej masą ziarna w zbiorniku (4), B) wózek tylny – siły (1), podpory (2)

On the basis of the technical specification of the combine harvester [4], it was determined that the mass distribution is a ratio of 70% of the mass distribution on the front axle and 30% on the rear axle, which is presented in Fig. 3, including the version with the attached header and straw shredder. Wheat grains were employed for the purpose of calculation as an exemplary type of grain in the tank and it was determined that 1 dm^3 of the quality grain has an average mass of 0.76 kg [5], and when considering the tank capacity of 5000 dm^3, it results in a mass of 3800 kg. The mass of the combine with empty tank is 9940 kg and the mass of the header is 1420 kg [4]. Based on the determined masses, it was calculated that their sum is 15160 kg for the variant with a full tank and 11360 kg for the one with an empty tank. Subsequently, the authors estimated the mass acting on the rear suspension, which is 3408 kg for the variant with an empty tank and 4548 kg for the variant with a full tank, as well as the mass acting on the front suspension, which is 7952 kg for the variant with an empty tank and 10612 kg for the variant with a full tank (Fig. 4).

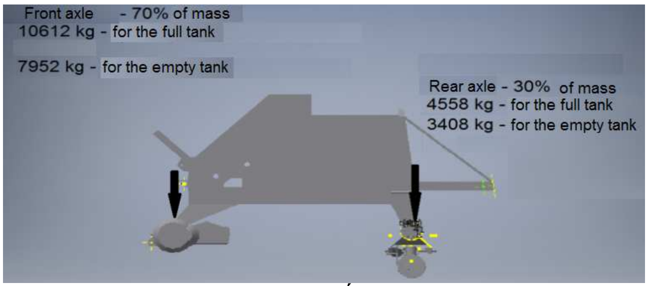

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Fig. 4. The distribution of mass in the combine harvester
Rys. 4. Rozkład mas w kombajnie zbożowym

The values of masses and forces are presented in Table 1. In order to load the numerical model of the frame, the forces used in the analysis, calculated on the basis of the mass and the values obtained, were as follows: 34080N, 45480N 79520 N, and 106120 N, maintaining appropriate force distributions with respect to mass distributions for the front and rear axle [9].

Table 1. Values of masses, forces and moments in the combine harvester
Tabela 1. Wartości mas, sił i momentów w kombajnie zbożowym

Specified masses	Value of loading force (N)	Value of acting moment (Nm)	Value of mass (kg)
Mass of the combine harvester	-	-	9940
Maximum mass of wheat in the grain tank	38000	-	3800
Mass of the header	14200	3606,8	1420
Mass of the combine harvester with the header (empty tank)	113600	-	11360
Mass of the combine harvester with the header (full tank)	151600	-	15160
Mass loading the front axle (empty tank)	79520	-	7952
Mass loading the front axle (full tank)	106120	-	10612
Mass of the header with the rapeseed attachment	18900	48006	1890

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On the basis of [4], the material used for the construction of the frame and, consequently, for the modeled object is the low alloy structural steel St35/S235JR, which is characterized by the following physical properties [6]:
- Young's modulus= $210 \cdot 10^3$ MPa,
- Poisson's ratio = 0,3,
- yield strength (Re) = 235 MPa,
- ultimate strength (Rm) = 380 MPa.

After entering all boundary conditions and other data into the model, a finite element mesh was generated, consisting of 89217 elements connected in 175719 nodes. The numerical model of the support frame of the presented combine harvester is statically determinate. Due to the large differences in machine mass which change depending on the amount of grain in the tank, it was decided to carry out two separate analyzes for two extremely different variants - with an empty and a full tank. This resulted in varied values of forces acting on structural elements used in the static analysis.

Stresses exceeding the limits of strength (Rm) lead to destruction of the component and, consequently, to damage to the structure of the combine harvester and a break in the system's operation.

Fig. 5. Accelerometer with a Vibrotest Shenck 60 reader [7]
Rys. 5. Akcelerometr wraz czytnikiem firmy Vibrotest Shenck 60 [7]

The Vibrotest 60 Schenck device (presented in Fig. 5), equipped with an accelerometer allowing for acceleration testing in 3 axes, was used to perform the vibration measurement. The purpose of the measurement is to show the value of vibration acceleration in places that are excessively exposed to static load. The measurement of vibrations was conducted at a designated point on the supporting frame of the post-harvest residue shredder, which is the closest point to the component that emits vibrations (Fig. 6A), as well as on the square profile of the threshing machine frame which is subject to failure (Fig. 6B). It was decided to measure the vibrations during machine operation at the maximum speed. This is justified by the fact that working parts of the combine harvester work only at full speed during harvesting. Another aspect is the relationship between the value of vibration acceleration and the working speed. During the measurement it was observed that high acceleration values are directly associated with high rpm values. The measurement was conducted in three axes: X, Y, and Z.

Source: own elaboration / Źródło: opracowanie własne

Fig. 6. Places of conducting measurements of mechanical vibrations for the components that are excessively exposed to static load, A) measurement of vibrations caused by a post-harvest residue shredder, B) measurement of vibrations caused by the threshing drum

Rys. 6. Miejsca dokonywanych pomiarów drgań mechanicznych dla elementów konstrukcyjnych nadmiernie obciążonych statycznie, A) pomiar drgań wywoływanych przez rozdrabniacz resztek pożniwnych, B) pomiar drgań wywoływanych przez bęben młócący

3. Research results

Numerical study of stress distribution was conducted for a static problem of the rear axle of the combine harvester. As a result of the calculations, it was possible to determine the value of maximum stresses (von Mises) at the critical point of the analyzed structure. For the node shown in Fig.s 7A and 7B, this value is respectively: 235.5 MPa for the combine harvester with a full grain tank and 160.1 MPa for the machine with an empty grain tank. Accumulation of stresses in the considered place for a fully loaded machine slightly exceeds the value of yield strength of the material which may lead to damage to the node. It is worth noting that the calculated values are the result of a linear static analysis. The authors allow the possibility of occurrence of much higher values of stresses when operating under real harvesting conditions or during vibrations caused by the operation of the combine subassemblies. To achieve structural safety, stress values should be referred to the appropriate safety factors. Based on the accepted standards in the selection of the safety factor [4], and allowing the possibility of measurement and material uncertainty, it is necessary to multiply the results by a factor of 1.5 to 1.7. Such a factor is used in case of the average precision calculations and the ability to determine forces and stresses [4]. For the purpose of the calculations, the factor = 1.7 was adopted. As a result, the calculated stress values increase accordingly to 400.35 MPa for the full grain tank variant and 272.17 MPa for the empty grain tank variant [9].

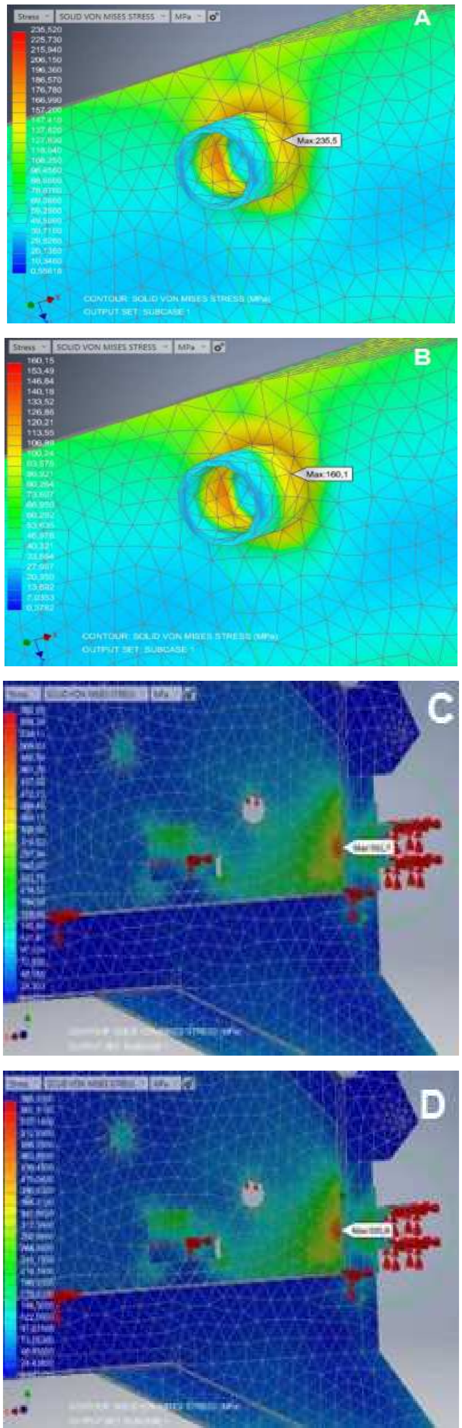

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Fig. 7. A, B) Linear static analysis results for the combine harvester rear axle, full and empty grain tank variant, C, D) results of linear static analysis for a threshing machine frame, a full and empty grain tank variant

Rys. 7. A, B) Wyniki analizy statycznej liniowej dla wózka tylnego kombajnu zbożowego, wariant pełnego i pustego zbiornika zbożowego, C, D) wyniki analizy statycznej liniowej dla kadłuba młocarni kombajnu zbożowego, wariant pełnego i pustego zbiornika zbożowego

The static analysis conducted for the threshing machine frame showed the following values of stresses (Von Mises): 582,7 MPa (Fig. 7C) in the case of a variant with empty grain tank and 585,9 MPa (Fig. 7D) for the full grain tank variant. Considering the fact that in this case the occurrence of calculation or measurement uncertainty is also possible, it is necessary to multiply the obtained results by an appropriate safety factor, based on the adopted standards [8]. For the safety factor of 1.7, the results of the adjusted stress values are as follows: 990.59 MPa for the variant with an empty tank and 906.03 MPa for the variant with a full tank.

On the basis of the conducted study, the values shown in Fig. 8ABC and 9ABC were obtained. On their basis it appears that under real harvesting conditions the straw shredding system generates vibrations at the maximum speed of the shredder shaft (3546.5 rpm which corresponds to 2050 rpm of the internal combustion engine), with a maximum value for the X axis = 62.3 mm/s^2, for the Y axis = 59.8 mm/s^2, and for the Z axis = 63.3 mm/s^2, as shown in Fig. 8ABC. The following results were obtained for the test conducted for the threshing machine frame: the maximum vibration value obtained at the maximum operating speeds is respectively for the X axis = 24.1 mm/s^2, for the Y axis = 18.3 mm/s^2, and for the Z axis = 29.9 mm/s^2. As regards the results, it is necessary to compare them with the applicable standards of allowable level of vibrations emitted by self-propelled machines, acceptable for human health, which will be the subject of subsequent publications.

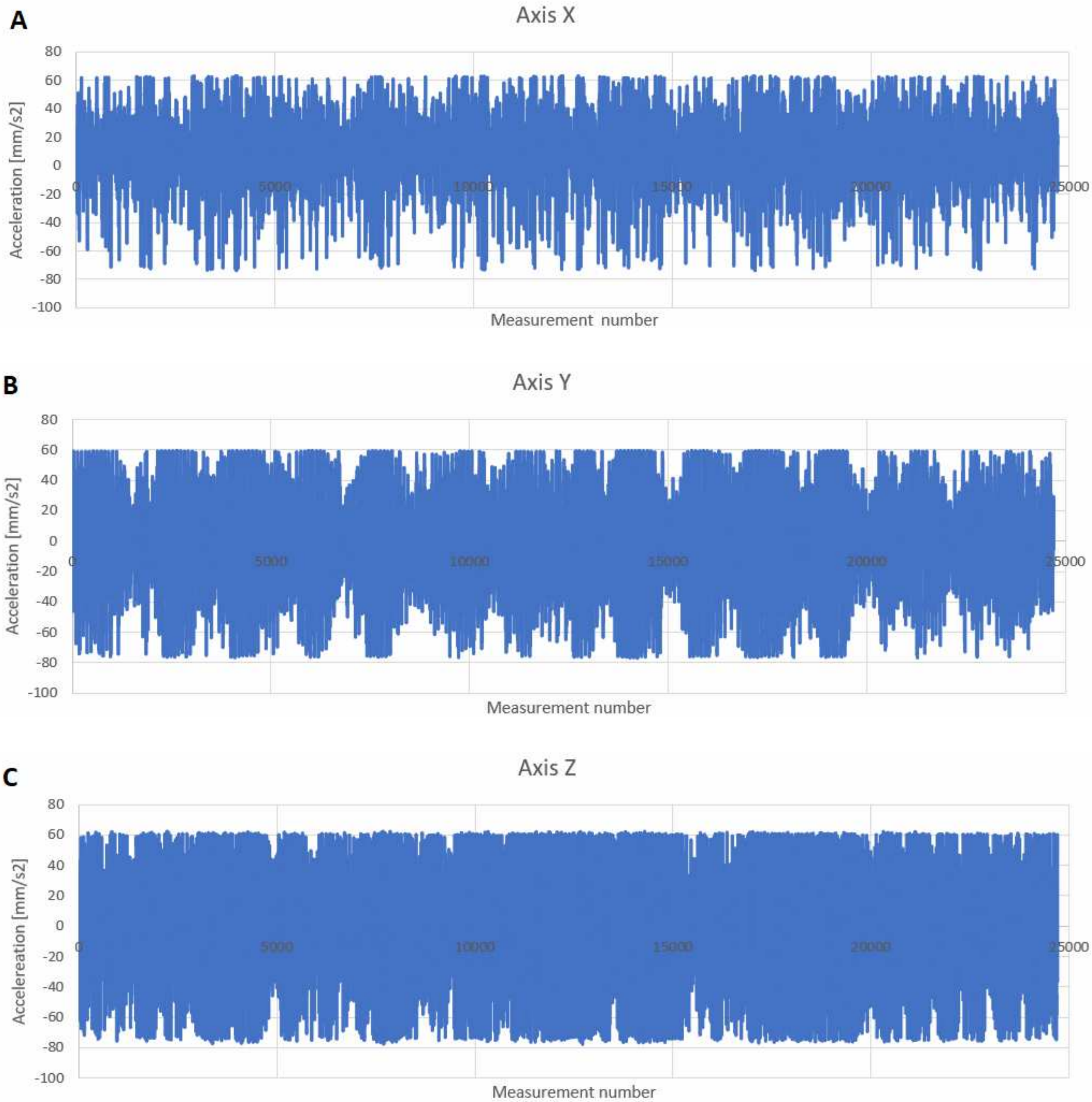

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Fig. 8. A) graph of vibration acceleration values of the straw shredder for the X axis, B) graph of the vibration acceleration values of the straw shredder for the Y axis, C) graph of the vibration acceleration values of the straw shredder for the Z axis
Rys. 8. A) wykres wartości przyśpieszenia drgań rozdrabniacza słomy dla osi X, B) wykres wartości przyśpieszenia drgań rozdrabniacza słomy dla osi Y, C) wykres wartości przyśpieszenia drgań rozdrabniacza słomy dla osi Z

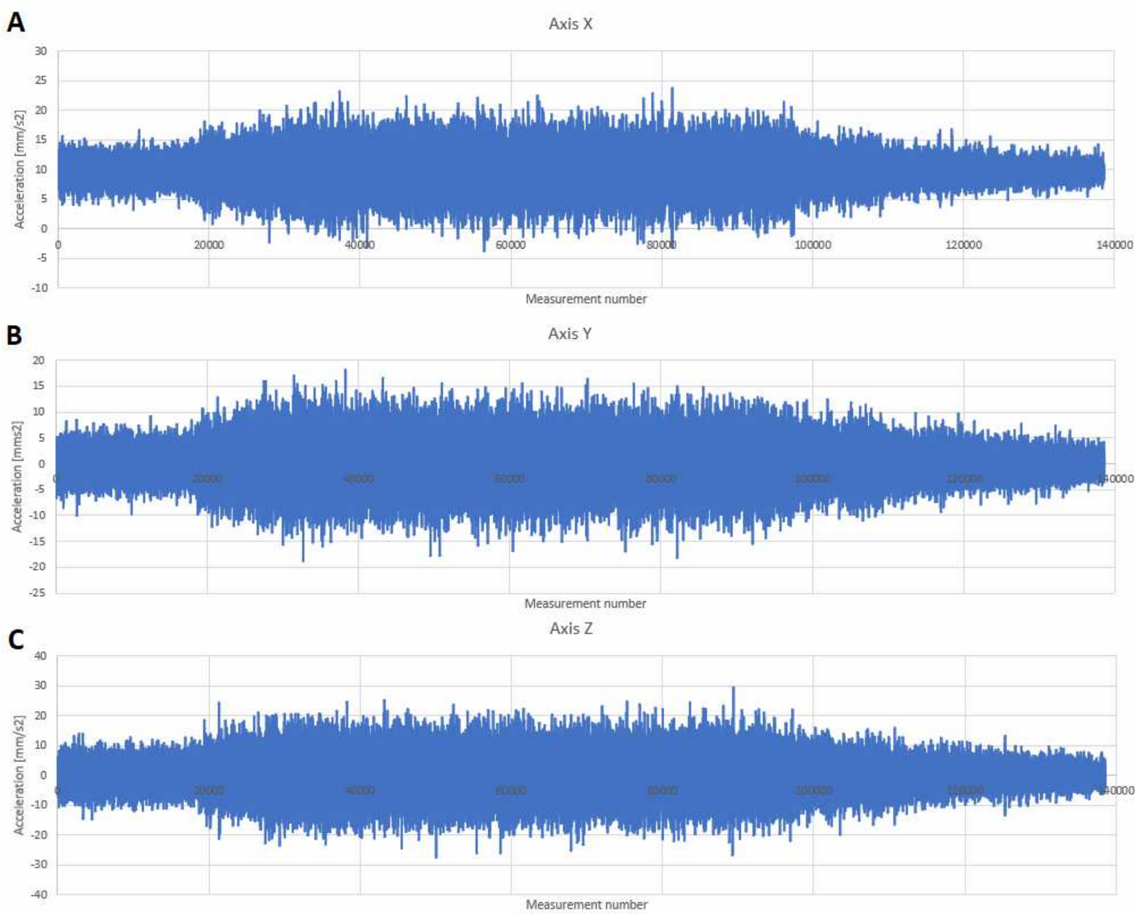

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Fig. 9. A) graph of vibration acceleration values of the threshing machine frame for the X axis, B) graph of vibration acceleration values of the threshing machine frame for the Y axis, graph of vibration acceleration values of the threshing machine frame for the Z axis

Rys. 9. A) wykres wartości przyśpieszenia drgań kadłuba młocarni dla osi X, B) wykres wartości przyśpieszenia drgań kadłuba młocarni dla osi Y, wykres wartości przyśpieszenia drgań kadłuba młocarni dla osi Z

4. Conclusions

The conducted study resulted in the following conclusions:

- The static analysis made it possible to estimate the stress value at the problematic point of the structure of the rear axle of the combine harvester frame, amounting respectively to 235.5 MPa for a loaded combine harvester and 160.1 MPa for the variant without load. In case of the first variant, the value exceeds the value of yield strength of the construction material.

- The static analysis made it possible to estimate the stress value at the problematic point of the threshing machine frame structure, amounting respectively to 585.9 MPa in case of loaded combine harvester and 582.7 MPa for the variant without load. In both cases the values significantly exceed the yield strength value of the indicated material.

- During the operation of the combine under real harvesting conditions, an increase in the acceleration value in the mm/s2 unit was observed when increasing the operating speed of the machine, which can be observed in Fig. 9ABC in the measurement range from 20,000 to 100,000 measurements.

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