Lecture Notes in Mechanical Engineering

Eugeniusz Rusiński
Damian Pietrusiak *Editors*

# Proceedings of the 14th International Scientific Conference: Computer Aided Engineering

Springer

# Lecture Notes in Mechanical Engineering

**Lecture Notes in Mechanical Engineering (LNME)** publishes the latest developments in Mechanical Engineering—quickly, informally and with high quality. Original research reported in proceedings and post-proceedings represents the core of LNME. Volumes published in LNME embrace all aspects, subfields and new challenges of mechanical engineering. Topics in the series include:

- Engineering Design
- Machinery and Machine Elements
- Mechanical Structures and Stress Analysis
- Engine Technology
- Aerospace Technology and Astronautics
- Nanotechnology and Microengineering
- Control, Robotics, Mechatronics
- Theoretical and Applied Mechanics
- Dynamical Systems, Control
- Fluid Mechanics
- Engineering Thermodynamics, Heat and Mass Transfer
- Precision Engineering, Instrumentation, Measurement
- Materials Engineering
- Tribology and Surface Technology

To submit a proposal or request further information, please contact the appropriate Springer Editor:

Li Shen at li.shen@springer.com (China)
Dr. Akash Chakraborty at akash.chakraborty@springernature.com (India)
Dr. Leontina Di Cecco at Leontina.dicecco@springer.com (all other Countries)
Please check the Springer Tracts in Mechanical Engineering at http://www.springer.com/series/11693 if you are interested in monographs, textbooks or edited books. To submit a proposal, please contact Leontina.dicecco@springer.com and Li.shen@springer.com.

**Indexed by SCOPUS. The books of the series are submitted for indexing to Web of Science.**

More information about this series at http://www.springer.com/series/11236

Eugeniusz Rusiński · Damian Pietrusiak
Editors

# Proceedings of the 14th International Scientific Conference: Computer Aided Engineering

Springer

*Editors*
Eugeniusz Rusiński
Faculty of Mechanical Engineering
Wrocław University of Science
and Technology
Wrocław, Poland

Damian Pietrusiak
Faculty of Mechanical Engineering
Wrocław University of Science
and Technology
Wrocław, Poland

ISSN 2195-4356 ISSN 2195-4364 (electronic)
Lecture Notes in Mechanical Engineering
ISBN 978-3-030-04974-4 ISBN 978-3-030-04975-1 (eBook)
https://doi.org/10.1007/978-3-030-04975-1

Library of Congress Control Number: 2018967408



This Springer imprint is published by the registered company Springer Nature Switzerland AG
The registered company address is: Gewerbestrasse 11, 6330 Cham, Switzerland

# Contents

# Numerical Analysis and Tests on Selected Dynamic Parameters of Shooting Stand Frame

Paweł Abratowski(✉)

Institute of Aviation, Transport Systems Departament, Al. Krakowska 110/114, 02-256 Warsaw, Poland
pawel.abratowski@ilot.edu.pl

**Abstract.** To build a machine gun on a helicopter board sufficiently rigid frame of shooting stand is required. The construction of the frame ensures changing a shoot direction in the vertical and rotation in the horizontal plane. The frame is hinged and can be pivoted to the inside of a helicopter. The frame design requires dynamic analysis and appropriate laboratory tests. The paper presents calculated results of the systems response to the applied dynamic loads in comparison to the laboratory tests results.

**Keywords:** FEM dynamic analysis · Stand frame tests · Clearance modeling

## 1 Introduction

One of the topics implemented by Institute of Aviation was laboratory tests and numerical analysis of the frame of a shooting stand designed by customer of the institute. Preliminary analysis of a similar new designed column stand for aircraft multi-barrel machine gun was also carried out and published in [2]. Similar shooting stand frames are described also in [1]. The frame, which is the subject of this article, is mounted in emergency exit of Mi-17 helicopter. The frame consist of welded steel pipes, steel sheets and hinges as one part. The hinges allow the frame to be pivoted to inside of the helicopter. On the other side the special pin (on a spring) makes it possible to lock the frame in working position. A rotary base is attached to the frame on which the machine gun is mounted. 7.62 mm multi-barrel machine gun weights 30 kg. The frequency of shots is 70 Hz. The steel 30HGSA is used as the material of the structure. The frame construction with a machine gun is shown on Fig. 1.

Laboratory frame test included:

- Static test
- Dynamic test – damped free vibrations test. The machine gun replaced by mass substitute
- Dynamic test – load of dynamic force. The loads has been realised in 3-seconds cycles. The assumed firing frequency of the machine gun was 70 Hz but there was


E. Rusiński and D. Pietrusiak (Eds.): CAE 2018, LNME, pp. 1–7, 2019.
https://doi.org/10.1007/978-3-030-04975-1_1

applied 35 Hz due to the limitations of the test stand. The structure was loaded calculated force as if the machine gun was mounted, therefore the tests were performed without mass of the gun.

The purpose of the analysis is to examine the system's response to a static and dynamic load and compare it to the test results. Fatigue tests and calculations are not taken into account in this article.

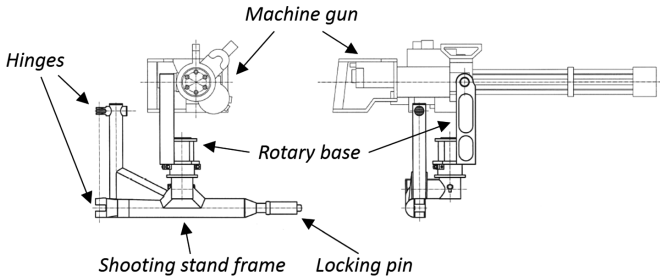


**Fig. 1.** Shooting stand frame with the machine gun

## 2 The Loads and Numerical Model

The dynamic force, shown in Fig. 2, is used directly from the laboratory test stand [3]. The duration of the dynamic force is limited to 1.5 s. Two variants model are made: variant 1 – without clearance, variant 2 – modification of variant 1 with clearance simulation. Geometric model with the method of applying support and load is shown in Fig. 3. The loads variants used for analysis are shown in Table 1. The value of $U_1$ is maximum absolute value obtained in the test. This is initial displacement boundary condition.

**Table 1.** Loads

| Analysis variant | Name | Description | Value |
|---|---|---|---|
| Static analysis, nonlinear | Load 1 | Static force | F1 = −1329 N |
| | Load 2 | Static force | F2 = 818 N |
| Dynamic analysis, free vibrations | Load 3 | Displacement | $U_1$ = −0.253 mm |
| Dynamic analysis, time dependent force load | Load 4 | Course of calculated dynamic force f(t) | Fig. 2 |

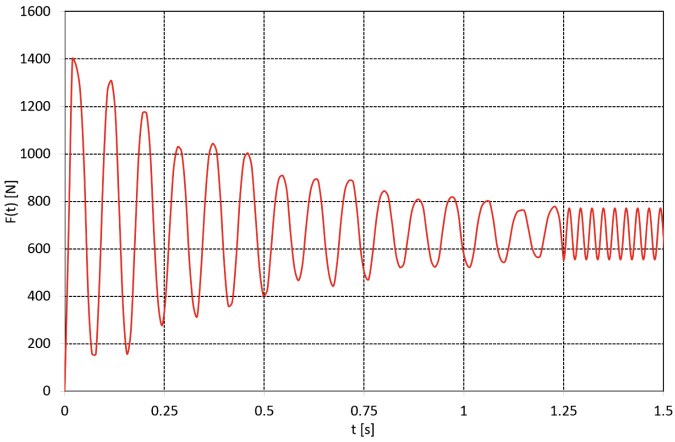


**Fig. 2.** Dynamic force F(t) [N]

The numerical model including the frame together with rotary base is developed using ANSYS program. The mesh model is shown on Fig. 4. The FE models are developed using 4-nodal shell elements (6 degrees of freedom at each node), 8-nodal solid elements (4 degrees of freedom at each node), beam elements (6 degrees of freedom at each nodes), mass element (the machine gun, forced vibrations) and 2-nodal contact element for clearance modeling. Simplified contact model is shown in Fig. 3.

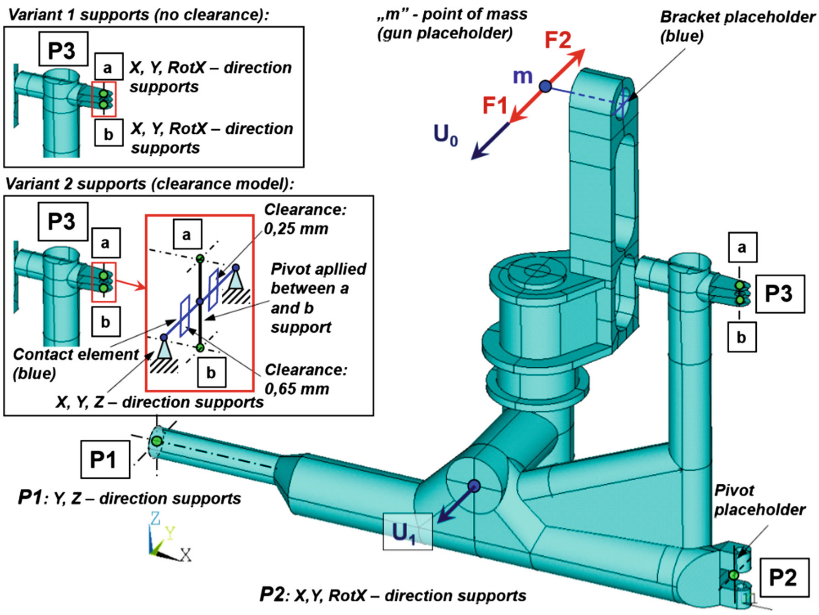


**Fig. 3.** Loads, supports and clearance model

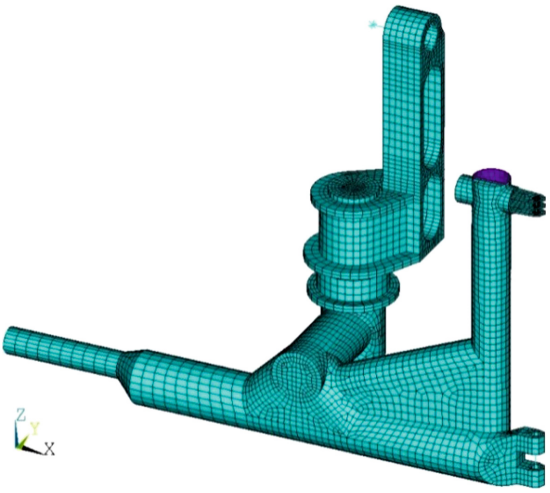


**Fig. 4.** FE discretization

## 3 Static Analysis

Static nonlinear analysis is performed using two models (as mentioned in Chap. 2). First – without clearance in supports and second – the clearance is applied in the P3 support (Fig. 3). The clearance values is determined based on the results of the tests and the model without clearance. The clearance is selected so as obtain approximate $U_0$ displacement to the tested model. The chart shown in Fig. 5 shows displacement $U_0$ (Fig. 3) for the test results [3], model without clearance results and model with applied clearance.

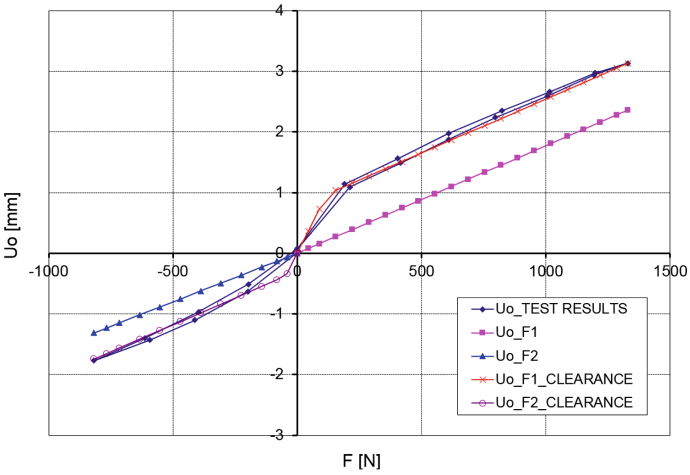


**Fig. 5.** $U_0$ [mm] displacement static results

## 4 Dynamic Analysis

Dynamic transient implicit analysis is performed using Newmark method [4]. Default Newmark parameters in Ansys are applied. In the frame structure the Rayleigh model damping [5] is applied:

$$[\mathrm{C}] = \alpha[\mathrm{M}] + \beta[\mathrm{K}], \tag{1}$$

where, $\alpha = 0$, three coefficients $\beta$ are applied: 0.0007, 0.001 and 0.0015.

Free vibration analysis is performed using model with mass substitute of the gun m = 30 kg (as mentioned in Chap. 1). The initial displacement is applied such that $U_1$ is equal to the maximum amplitude of the test result. The results (shown in Fig. 6) shows $U_1$ time dependent displacements for three $\beta$ damping coefficients in comparison to the test results [3]. The test results are also published in [6].

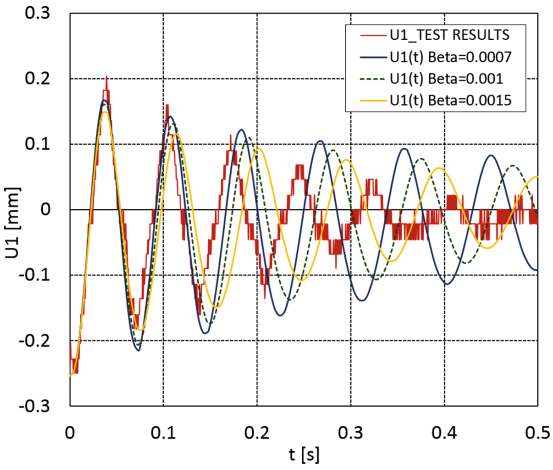


**Fig. 6.** Free vibrations. $U_1$ [mm] displacement results

The chart shows the difference in the course of the damping between the test and calculations results. It can be seen that the result for assumed β = 0.0015 is the nearest to the test result (on the negative side), whereas the vibration period increases at the fastest rate.

The frame analysis with dynamic force load is performed without the mass (as in laboratory tests). During the tests, the load of time dependent force was performed in series. There were performed above 85000 series. Due to the limited volume of this article, there was selected one typical test result for comparison. The test and calculation results for the β = 0.0015 coefficient are shown in Fig. 7.

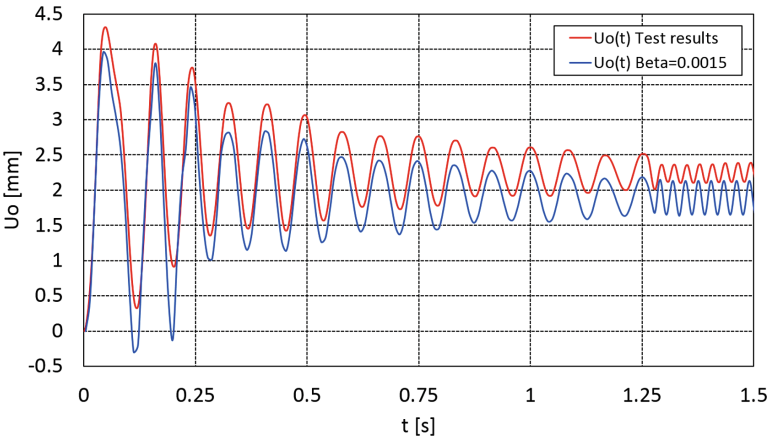


**Fig. 7.** $U_0$ displacement results [mm]

The chart shows the difference in the maximum amplitude values between test and calculations results. The minimum difference for the maximum absolute displacements is 0.241 mm at about t = 0.05 s. At steady state (from about t = 1.25 s) the difference in maximum displacement is 0.227 (4%), whereas the amplitude of the calculated results is approximately twice the tests.

The Fig. 8 shows the frame on the test bench with marked characteristic measuring points.

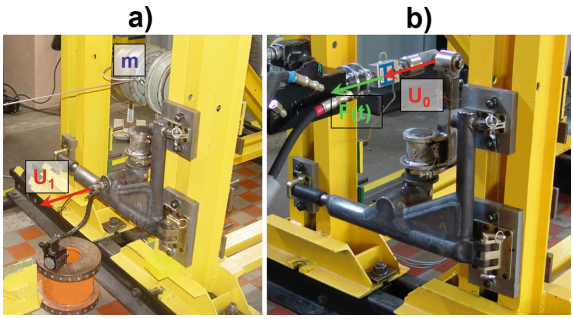


**Fig. 8.** The frame mounted on the test bench. Configuration of the test stand for: (a) – free vibrations test, (b) – tests with dynamic loads

## 5 Summary and Conclusions

1. Static analysis shows high accuracy in comparison to test results, especially model stiffness. There is possible to set the clearance value in the frame at a high level of accuracy basing on comparison of the calculations and the test results.

2. Free vibrations analysis shows the differences. It is possible to set the damping coefficient, whereby with a higher damping factor, the vibration period increases faster. In the next step of the research, a non-linear damping coefficient will be applied.
3. Although the value of the clearance may be set with the high accuracy, the calculated results differ from the test results. The differences between maximum and minimum displacements may be caused the different masses between real and numerical model. The mass and stiffness of the dynamometer (shown in Fig. 8b) are not take into account. It can affect the dynamic calculation results. The influence of mass and stiffness of the dynamometer on dynamic results should be checked.

## References

1. Сушков С, Струцкий ВГ, Данеко АИ, Полковников ВА, Тимашев ИВ (1998) Авиационные артиллерийские установкиl, Москва, Издательство МАИ. ISBN 5-7035-1383-9
2. Abratowski P, Krasoń W, Barnat W, Gnarowski W (2017) Dynamic analysis of column stand for aircraft multi-barrel machine gun with consideration of bearing clearance. In: Lecture notes in mechanical engineering. Springer International Publishing AG
3. Gębski M, Raport z badań próby stanowiska strzeleckiego 7.62 mm WPKM (7.62 Wielolufowy Pokładowy Karabin Maszynowy), raport wewnętrzny Instytutu Lotnictwa nr CBMK/LM1/51073.01-04/1/2010
4. Bushby HR, Staab GH (2008) Structural dynamics: concepts and applications. CRC Press
5. ANSYS 14 (2005) Manual
6. Abratowski P, Krasoń W, Barnat W, Gnarowski W (2017) The overview of construction and selected aspects of testing mounting frames of aircraft machine guns. J Sci Gen 49(1): 183. Tadeusz Kosciuszko Military Academy of Land Forces, Wrocław

# The Numerical-Experimental Studies of Stress Distribution in the Three-Arm Boom of the Hybrid Machine for Demolition Works

Jakub Andruszko(✉) and Damian Derlukiewicz

Faculty of Mechanical Engineering,
Department of Machine Design and Research,
Wroclaw University of Science and Technology, Łukasiewicza 5, 50-370
Wroclaw, Poland
{jakub.andruszko,damian.derlukiewicz}@pwr.edu.pl

**Abstract.** An electric machine for construction works is a multi-functional, remote-controlled demolition robot which is designed to be operated in hard work condition where the human being is not recommended due to high risks resulting, for example high dust content, high temperature or noise. In such machines, their parts wear out very quickly. The article presents the approach to validate the project assumptions using experimental and numerical studies. The experimental studies provided the information about the adverse efforts and frequency states of the machine. The numerical calculation consigns the information about the stress distribution in the whole arm working system.

**Keywords:** Demolition machine · Electric machine · Arm working system · Numerical-experimental studies · Construction works · High-speed camera · Testing · Finite element method

## 1 Introduction

As a result of the growing demand for automating the process of demolishing building structures and for the removal of rocks and spoil in the mid-nineteenth century, the development of specialized demolition machines took place. Over time and the need to automate the process of demolition, the strength of human muscles has been changed to light machines and then to heavy machines such as excavators equipped with specialized equipment for demolition. Over the years, entrepreneurs have at their disposal cranes or excavators with a suspended sphere, excavators with specialized arms and a mounted hydraulic hammer. The most crucial moment was the provision of an electric demolition machine, whose dimensions allowed for demolition of elements inside buildings. The use of electric demolition machines allows to optimize the working environment of the machine in terms of operator safety. The use of electric drive allows the machine to move in closed rooms without causing it to become smoky or producing harmful exhaust gases as in the case of machines with the use of an exhaust system [9] (Fig. 1).


E. Rusiński and D. Pietrusiak (Eds.): CAE 2018, LNME, pp. 8–14, 2019.
https://doi.org/10.1007/978-3-030-04975-1_2

**Fig. 1.** Visualization of the electric D-REX demolition machine

## 2 Experimental Studies

Several measurement methods can be used for measurements in the machine's working environment. The most often used is a strain gauges, and to eliminate measurement errors in its application resulting from environmental conditions and for validation of the computational model to determine resonant frequencies a high-speed camera can be used [1, 2]. Comparing the determined resonance frequencies with those calculated in the FEM analysis, we gain assurance of the adopted assumptions and simplifications in the construction of a computational model [7]. Due to the need to eliminate the harmful effects of the environment on the results of measurements, a high-speed camera was used.

The measurements were carried out in the working environment of the machine and the element which has been subjected to the breaking process was a block of reinforced concrete, which was a fragment of the foundation of the building. In the case of such positioning of the machine, the maximum possible extension of the cylinders was chosen so that the position of the arms could generate the greatest possible torque at the rotating element of the working system. This case is the worst possible case of the work system in this environment. Figure 2 shows the machine in the work environment together with the prepared measuring stand and the basic parameters of the measurement.

**Speed of the camera:** 10 000 frames per second

**Density of sampling of displacement signal:** 0.001 m

**Fig. 2.** Work environment of the machine, test stand and parameters of the measurement

The measurement concerned the work of the hammer, which hit the concrete block, destroying the forged material. The obtained graph of displacement in relation to time presents the vertical component of motion, the horizontal one was negligibly small, so it was decided not to plot its dependence and not to apply it in the simulation process (Fig. 3) [3].

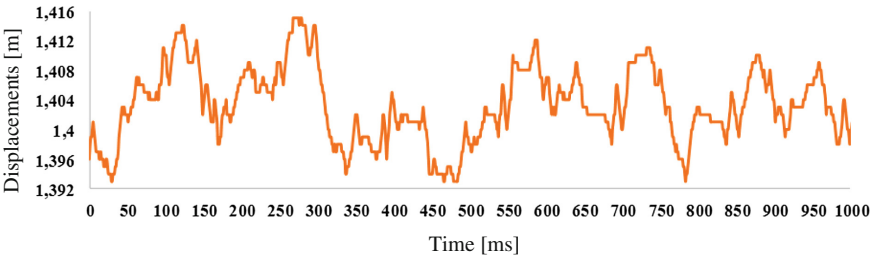


**Fig. 3.** Displacement of the fastening system point during breaking

To obtain the correct results, image scaling was performed by determining the actual distance between two points in the mounting system (distance between pins). Thanks to the TEMA Motion software and the registered image was analyzed. Having the measured displacement of the system and the time of the measurement, the speed of the desired point was determined and then its acceleration. The minimum size of the pixel can be 20 um, and a change in its size can affect the measurement error. In the measuring the natural frequency of the system, acceleration diagrams were determined from time, and then the relationship between acceleration and frequency was plotted thanks to FFT analysis [4] (Fig. 4).

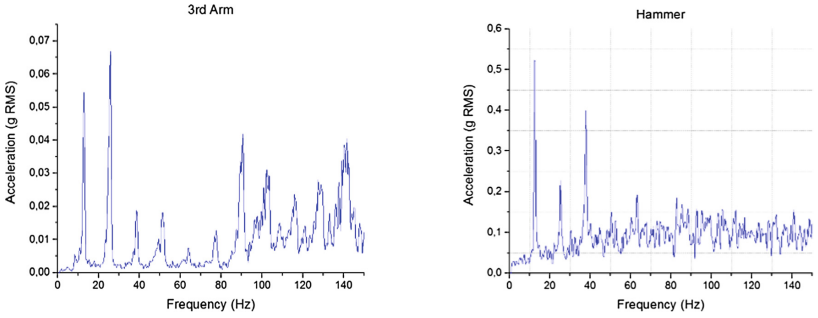


**Fig. 4.** The spectrum of the signal recorded on the last arm and hammer during operation in the range up to 150 Hz

## 3 Computational Model

In order to check the strength of the designed working system, a discrete model was built [6, 8]. Discrete model was built on the basis of shell and beam-rod and rigid elements. The geometrical assembly model is built without pins, actuators and additional connecting elements that is modeled as connections in the process of building a computational model [5]. The discreet model with simplifications used in the form of additional elements and named parts is shown in Fig. 5.

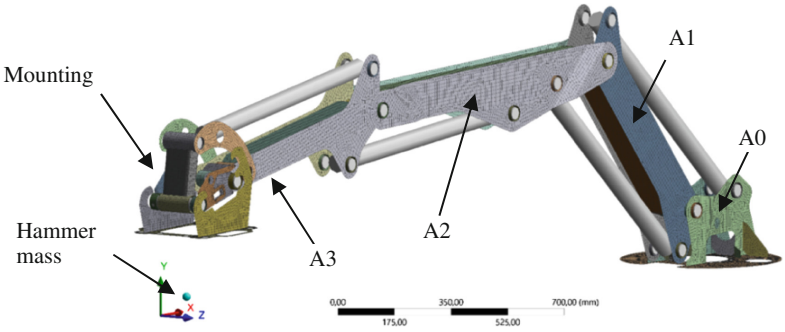


**Fig. 5.** Discrete model of the working system

Numerical analyzes were carried out in two stages: in the first stage, the resonance frequencies of the working system were determined in order to validate the calculation model with the real model, while in the second stage the effort of the working system, with the given displacement, was analyzed.

## 4 Results of the Simulation

To assess the correctness of the preparation of the computational model, the results of measurements using a high-speed camera and modal analysis carried out using computer simulation were used. Spectral analyzes carried out in the working system, regardless of where the measurements were made (hammer, arm tip) showed exactly the same natural frequencies. The graphs for these measurements differ only in the magnitude of accelerations. Table 1 presents the comparison of the natural frequency obtained by experimental and numerical methods together with the error that the calculation model was burdened with. An example of the form of natural frequencies is shown in Fig. 6.

**Table 1.** Comparison of exemplary natural frequencies

| HS Camera (Hz) | FEM (Hz) | Difference (Hz) |
|---|---|---|
| 1,5 | 2,6 | 1,1 |
| 13 | 14,5 | 1,5 |
| 25,25 | 26,8 | 1,55 |
| 38,75 | 44,6 | 5,85 |
| 52 | 56,8 | 4,8 |
| 89,63 | 86,2 | 3,43 |
| 128,75 | 128,1 | 0,65 |

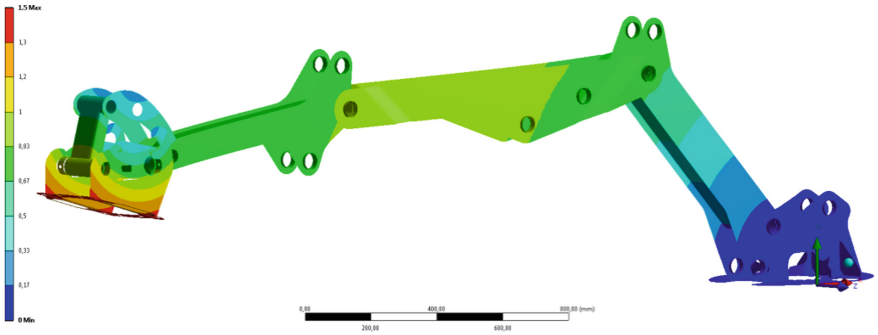

**Fig. 6.** An example of the deformation of the model

After checking the accuracy of the calculation model, the verification of its response to the displacement was carried out. As a result of applying displacement, the dependence of stresses on the time occurring in the structure during the breaking process was obtained, which is shown in Fig. 7.

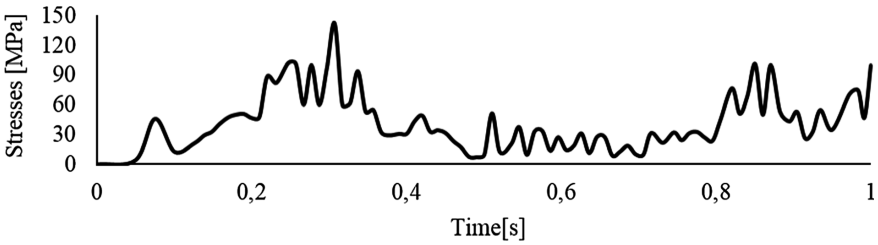


**Fig. 7.** Dependence of stresses from time in the breaking process

With the applied force in the form of displacement and the analysis of the graph in Fig. 7, it can be concluded that the maximum stresses occur in 0.27 s of simulation. Maximum stresses appeared in the places where the A0 was mounted to the robot structure. This place is stiffened in this type of analysis, so after analyzing the whole structure, it was found that the maximum stresses in the structure amount to 122 MPa and also appeared at the junction of two sheets of the A0. Equivalents (using H-M-H theory) stress distribution are shown in Fig. 8, while the distribution of stress in individual components is shown in Fig. 9.

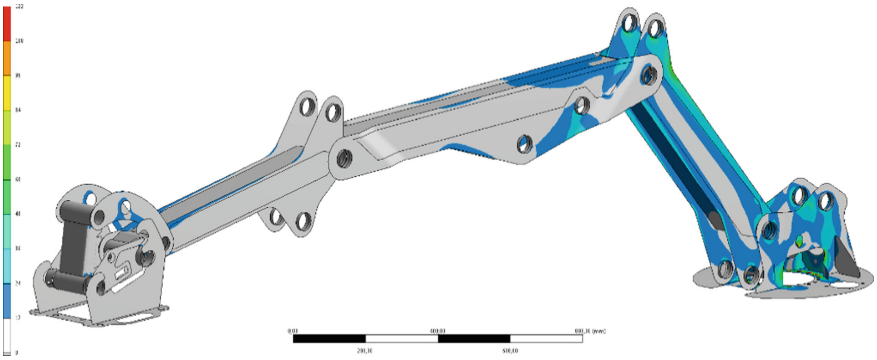

**Fig. 8.** Equivalents (using H-M-H theory) stress distribution along whole model

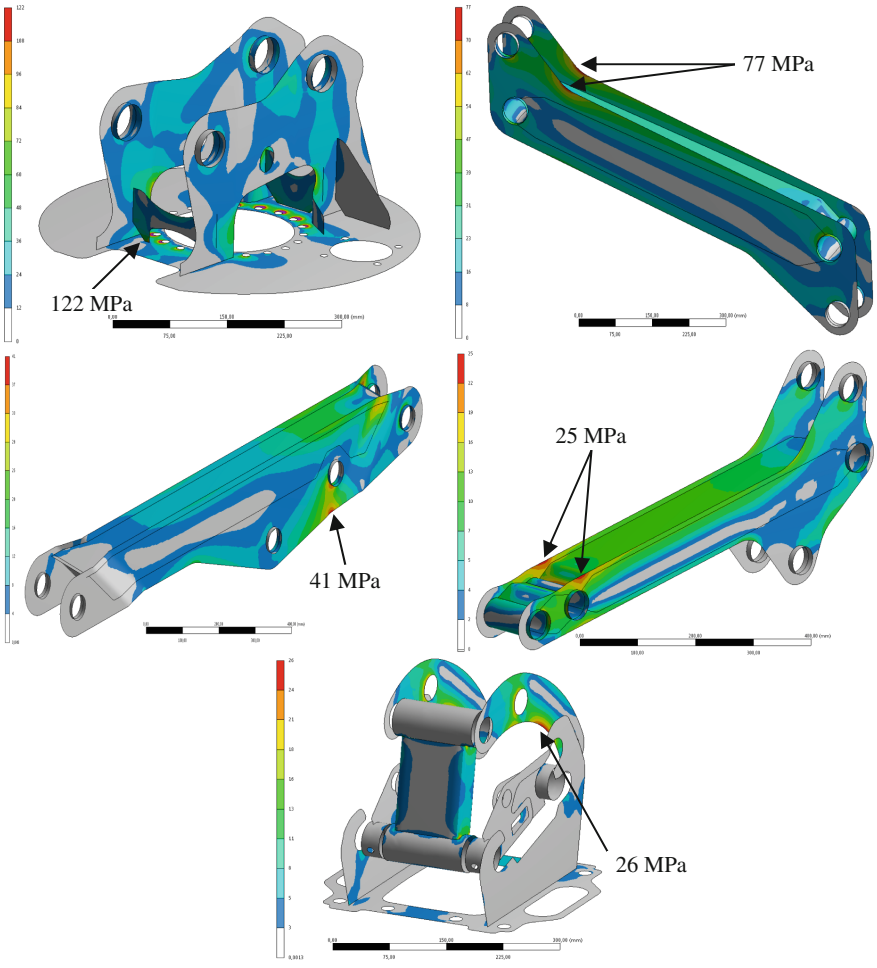


**Fig. 9.** Equivalents (using H-M-H theory) stress distribution in individual components

## 5 Summary and Conclusions

Studies using a fast camera to measure displacements and accelerations, allowed to precisely determine the extortion that worked on the robot's arm during its operation. The comparison of the resonant frequency carried out by high-speed camera and simulation confirmed the correctness of the prepared calculation model. Resonance frequencies obtained during experimental research are similar to the resonance frequencies obtained during modal analysis performed by simulations using the Finite Element Method. The computational model contained simplifications so that deviations between the values derived from the numerical analysis and the experiment could be observed.

Analyzing FEM simulations, it can be concluded that the maximum stresses occur in 0.27 s simulation, which is caused by the increase of displacement in the initial violation of the cohesion of the forged material and local cracks. The maximum stresses in the structure occur in the A0, which was an expected effect, because in this place the largest torque acting on the structure is generated.

It can be stated that the numerical and experimental approach gives sufficient information on the state of effort of demolition machine working systems. The use of a high-speed camera to determine the displacement and at the same time to determine the acceleration of the system in order to validate the calculation model limits the number of measuring instruments used to a minimum.

**Acknowledgments.** The project was carried with the support of the National Centre for Research and Development in Poland under the "Szybka Ścieżka" program no POIR.01.01.01-00-0582/15-00, in cooperation with Advanced Robotic Engineering Ltd. company.

## References

1. Derlukiewicz D, Ptak M, Wilhelm J, Jakubowski K (2017) The numerical-experimental studies of demolition machine operator work. In: Lecture notes in mechanical engineering. Springer, pp 129–138
2. Derlukiewicz D, Cieślak M (2017) Study of the causes of boom elements cracking of electric demolition machine with use of experimental and numerical methods. In: Lecture notes in mechanical engineering. Springer, pp 109–119
3. Karliński J, Rusiński E, Lewandowski T (2008) New generation automated drilling machine for tunneling and underg round mining work. Autom Constr: 224–231
4. Piszczek K, Walczak J (1982) Drgania w budowie maszyn, wyd. 3. Warszawa
5. Rusiński E (2002) Design principles supporting structures of motor vehicles, in Polish. Wrocław
6. Rusiński E, Czmochowski J, Smolnicki T (2000) Zaawansowana metoda elementów skończonych w konstrukcjach nośnych. Wrocław
7. Rusiński E, Czmochowski J, Pietrusiak D (2012) Problems of steel construction modal models identifcation. Maint Reliab: 54–61
8. Zienkiewicz O, Taylor R (2006) The finite element method for solid and structural mechanics, 6th edn., 3rd repr. edn. Amsterdam
9. Global Construction Robots Market 2016–2020 with Brokk (2016) Fujita & ULC Robotics Dominating. https://www.businesswire.com/news/home/20160524005815/en/Global-Construction-Robots-Market-2016-2020-Brokk-Fujita

# Analysis of the Causes of Fatigue Cracks in the Carrying Structure of the Bucket Wheel in the SchRs4600 Excavator Using Experimental-Numerical Techniques

Jakub Andruszko, Przemysław Moczko, Damian Pietrusiak(✉), Grzegorz Przybyłek, and Eugeniusz Rusiński

Faculty of Mechanical Engineering, Department of Machine Design and Research, Wroclaw University of Science and Technology, Lukasiewicza 7/9, 50-371 Wrocław, Poland
damian.pietrusiak@pwr.edu.pl

**Abstract.** The article presents design faults related to the carrying structure of the bucket wheel of the SchRs4600 excavator working in the brown coal mine. The authors analyzed cases of damages of this type of structure occurred in the past in such machines. In order to determine the damage of the carrying structure of the bucket-wheel excavator, non-destructive examinations were carried out using visual and magnetic-particle methods. The real loads occurring during operation of the machine in the case of fatigue were also determined. These measurements were used to verify the numerical model. A strength analysis was carried out using the Finite Element Method. The cause of fatigue cracks was determined by measurements and numerical calculations.

**Keywords:** Mining · Excavator · Bucket-wheel · Cracks · Non-destructive testing · Strain gauges · Stress analysis · Finite element method

## 1 Introduction

One of the basic methods of exploiting mineral raw materials such as brown coal is the opencast method, and its systems have been described in detail in [1]. In order to conduct brown coal mining, specialized machines operating in appropriate systems are used [1].

One of these machines used in open cast mining are bucket-wheel excavators (Fig. 1) and they are a part of a group of machines called basic mining machines. These machines are working in a continuous manner and equipped with a mining head with a number of elements, which are e.g. buckets with teeth. These machines are an integral part of the basic technological system of the open cast mine, where they constitute its first and one of the most important links. The construction of various types of bucket-wheel excavators is practically the same and is based on very similar or even identical functional systems. The construction of various types of bucket-wheel excavators is practically the same and is based on very similar or even identical functional systems. Small differences are described in [2]. One of the basic functional system of a bucket-


E. Rusiński and D. Pietrusiak (Eds.): CAE 2018, LNME, pp. 15–28, 2019.
https://doi.org/10.1007/978-3-030-04975-1_3

wheel excavator is a mining system consisting of a carrying structure, which is a bucket wheel and a mining boom.

**Fig. 1.** SchRs4600 bucket-wheel excavator

Currently, in the design of carrying structures of basic mining machines, in this case bucket-wheel excavators, the available standards are used. They have been developed based on many years of experience of the manufacturers of these machines. These standards were created at the beginning of the 20th century, and some machines working in brown coal opencast mines were designed according to older standards, which were less precise. The standards have some differences depending on which parts of the world they are used in. Differences between the assumptions occurring in individual standards concerning the design of basic mining machinery are presented in paper [3], where the authors presented differences in terms of static, dynamic and fatigue loads.

Due to the age of these machines, and consequently their large repair history, which is not always correctly carried out, the size of these machines, the complexity of the technological process and the existence of high variable loads that cannot be clearly predicted, these machines are exposed to occurrence of various types of failures [4]. One of the first and at the same time one of the most key elements that is in contact with the material is the bucket wheel of the machine. It is exposed to extreme dynamic loads resulting both from the mining technique, the properties of the material being mined and the hardly-abrasive or non-abrasive materials contained therein. Due to the high variability of loads, sometimes this part of the structure is exposed to various types of failures.

Research articles that have started to appear recently are the result of these failures, and the authors present their individual approaches to determining their causes. One of such failures of the mining system, which was the drive shaft of the bucket wheel in the SRs 2000.32/5.0 + VR92 bucket-wheel excavator, was described in [5], where the