Innovative Solutions for Increasing the Service Life of Cutting Head Components Using Finite Element Method Analysis

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Chipless tree cutting is increasingly becoming more popular because of its low weight and greater throughput due to the small dimensions of the entire cutting head. This paper is focused on the design of the crank plate of the cutting mechanism of a chipless cutting head. It was necessary to calculate the magnitude of the force required to cut the tree in the direction perpendicular to the growth of the tree fibers. In addition, the tree and cutting mechanism parameters were specified. Due to the size of the cutting force, it was possible to further deduce the amount of the discharge force from the linear motor that directly acts on the opening of the crank plate. After finite element method analysis, it was found that the crank plate was undersized and it was necessary to increase its thickness from 15 mm to 30 mm, which eliminated the problem of plastic deformation.

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INTRODUCTION

The primary task of a chipless cutting head is to facilitate difficult motomanual work for workers, typically for either cross-sections or collecting fast-growing woody plants. The efficiency of chipless cutting heads compared to motomanual work is always greater and at the same time the complexity of operation and the number of workers needed is considerably lower (Spinelli *et al.* 2002; Cavalli *et al.* 2014; Chakroun *et al.* 2016). Chipless wood cutting as a forestry process is primarily used in machines intended for the delimbing of trees. Increasingly, however, chipless wood cutting is used in the form of chipless cutting heads in single-operation machines, which perform only wood cutting and are included in machine sets together with processors that can subsequently further process the sawn wood into chips, *e.g.*, for the production of pellets, OSB boards, or material for the pulp industry. In addition, this technology is especially used for bioenergy, as a big emphasis is currently being placed on obtaining energy from renewable sources (Kasmioui and Ceulemans 2012; Hanzelka *et al.* 2015; Nathan et al. 2016; Stochlová et al. 2019).

Theory of Chipless Wood Cutting

The transverse division of wood with a knife is based on the ability of the wood to deform. The wood shows a relatively low resistance to deformation when the wedge is pushed into it. When the cutting tool is pressed into the wood, the cutting edge breaks the bonds between the fibers. In such a process, no waste chips are formed and the cutting joint is equal to zero (Marko and Holík 2000; Kováč *et al.* 2017). The basic scheme of chipless cutting with a single-acting knife is shown in Fig. 1. When cutting with a single-acting flat knife, the cutting edge angle (β) is equal to the cutting angle (δ).

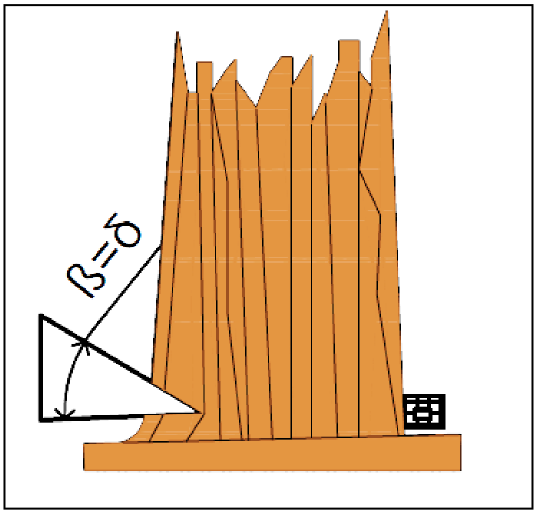

Fig. 1. Schematic of chipless wood cutting (Krilek *et al.* 2018)

For chipless cutting, chipless cutting heads are used, which can cut trunks with a diameter of up to 30 cm. Each cutting head must be comprised of a frame (structure), a hydraulic control for the gripping arms (3), a gripping mechanism (4,6), a cutting knife (5), a rotator (1), and a hydraulic control of the tilt (2), as shown in Fig. 2. The cutting process is ensured thanks to the hydraulic control of the gripping arms, which push the log into the cutting knife of the head (Kováč *et al.* 2013; Jobbágy and Kováč 2014).

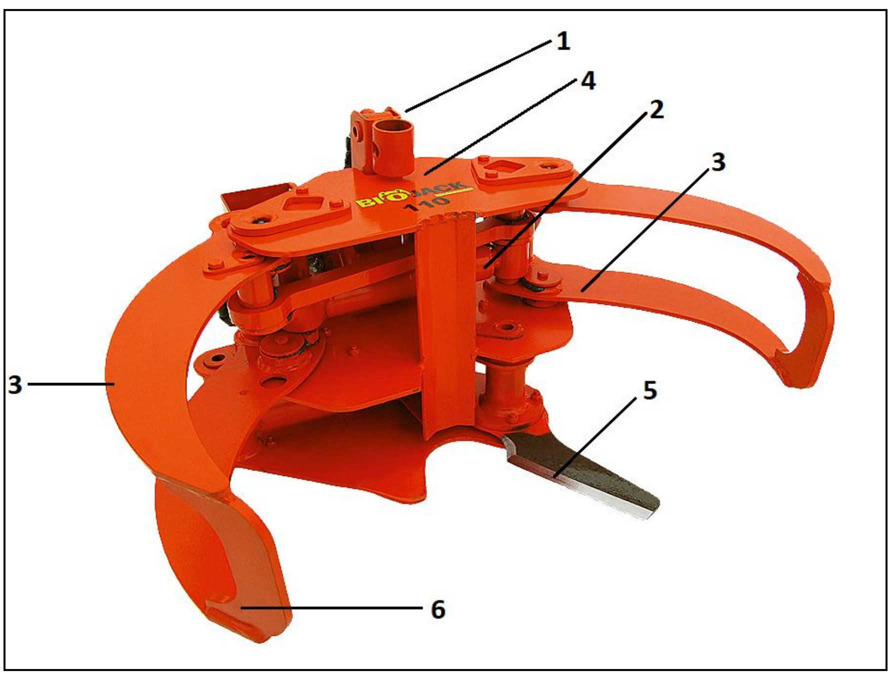

Fig. 2. Schematic of a chipless wood cutting head (Nisula 2022)

One of the most stressed parts of the cutting head is the cutting plate of the cutting arm, which is directly acted upon by the discharge force from the rectilinear hydraulic motor. In order to be able to subject this gear plate to stress analysis, it was necessary to determine the magnitude of the cutting force and the pushing force required from the rectilinear hydraulic motor.

EXPERIMENTAL

Innovative design solutions for cutting heads include new solutions in the field of chipless wood cutting, and an important concern is damage to individual components of the cutting head. Damage to functional components causes the head to malfunction, reducing the efficiency of wood processing. The innovative cutting head considered in this work is designed to ensure frequent use in operation. Compared to the series head, it differs primarily in the shape of the jaw grippers (2) and the method of their attachment through the mounting plates (4), which were tested in the article. The uniqueness of the design is in the change in the shape of the cutting knife (3). The cutting head is constructed of EN 41 1523 (2014) steel, which presents high-quality mechanical properties in terms of the use of the construction of the head, weldability, and machining of the individual components. The tensile strength (R_m) is 510 MPa to 680 MPa, the yield strength (R_e) is 373 MPa to 980 MPa, the carbon content is less than 0.2%, and material density is 7850 kg/m^3.

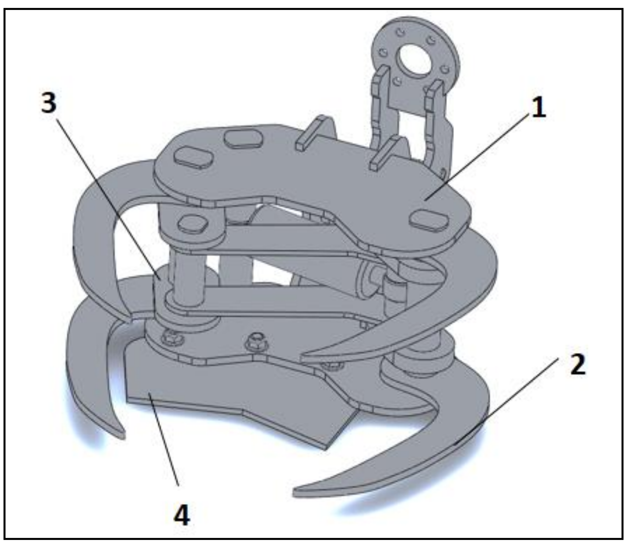

Fig. 3. Inovative design of the cutting head

Calculation of cutting force

The input parameters for calculating the cutting force are as follows: wood - beech; the diameter of the tree at the point of cut - 200 mm; cutting wedge angle - 30°; knife thickness - 8 mm; and used a single-knife cutting mechanism.

The following relation was used to calculate the cutting force,

$$F_C = (1{,}2 * D + 4 * D^2) * (0{,}2 + 100 * t) * \frac{\beta}{30} * a_p * a_T * 10^5 \ [N] \qquad (1)$$

where D is the tree diameter (m), t is the knife thickness (m), β is the cutting wedge angle (°), a_P is the coefficient including the effect of wood composition on the cutting force for beech a_P= 2.4, and a_T is the coefficient including the effect of ambient air temperature (for summer a_T equals 1 and for winter a_T equals 1.2 to 1.3 at -20 to -30 °C (Mikleš 1993,Hatton *et al*.2016). The cutting force in the summer was calculated according to Eq. 2,

$$F_c = (1{,}2 * 0 + 4 * 0.2^2) * (0.2 + 100 * 0.008) * \frac{30}{30} * 2.4 * 1.0 * 10^5 \quad (2)$$

where F_c equals 96000 N or 96 kN. The cutting force in the winter was calculated according to Eq. 3,

$$F_C = (1{,}2 * 0{,}2 + 4 * 0{,}2^2) * (0{,}2 + 100 * 0{,}008) * \frac{30}{30} * 2{,}4 * 1{,}3 * 10^5 \quad (3)$$

where F_c equals 124800 N or 125 kN, which was F_{cmax}.

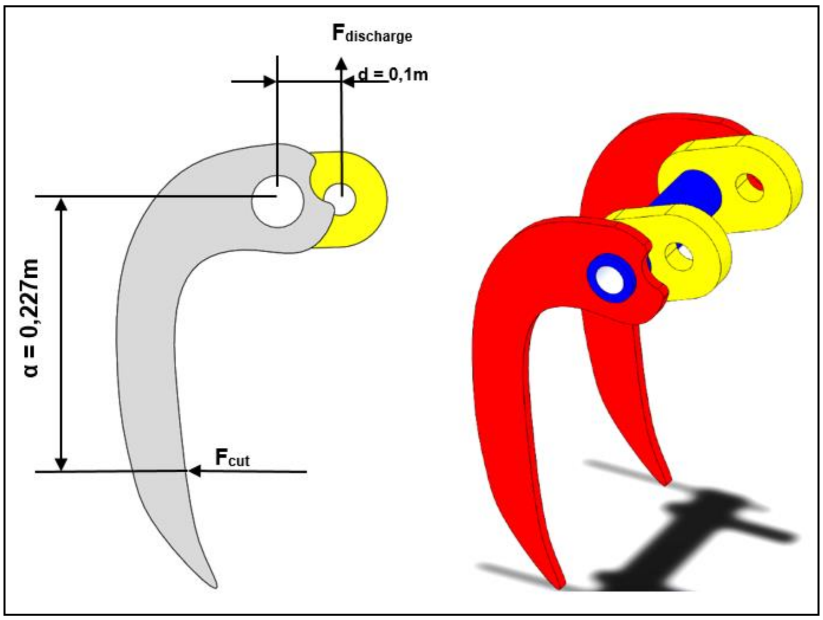

Fig. 4. Force analysis of the cutting mechanism

According to Fig. 3, a moment equation of equilibrium to the point of rotation of the arm was made. According to the moment equation, the magnitude of the discharge force required to design the hydraulic equipment was determined, as shown in Eq. 4 through Eq. 6,

$$\sum M_0 = 0 : F * d - F_c * a = 0 \quad (4)$$

$$F * d = F_c * a = 0 \quad (5)$$

$$F = \frac{F_c * a}{d} = \frac{125000 * 0.227}{0.1} \quad (6)$$

where F equals 284.9 kN.

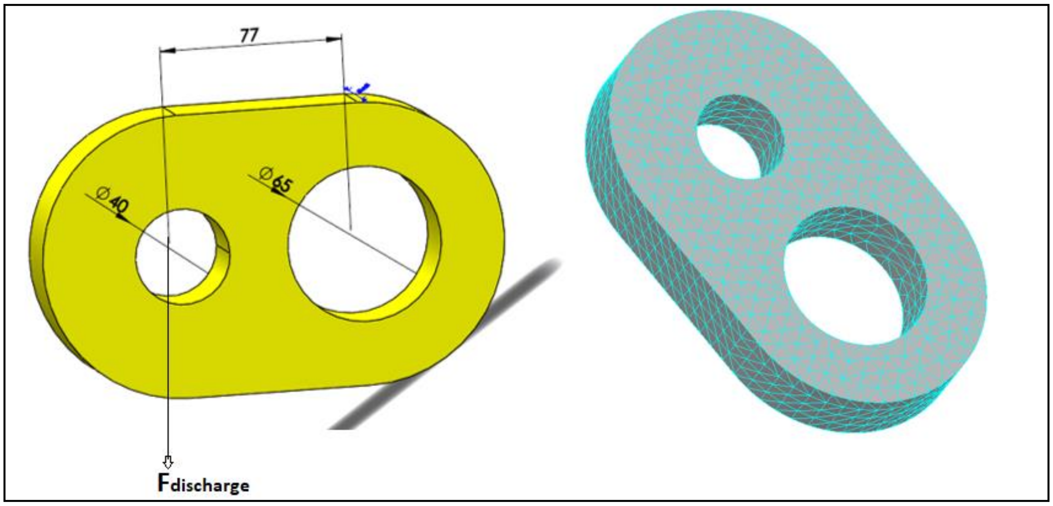

Fig. 5. Load diagram of the gear plate and transfer plate networking

In Fig. 6 it can be seen how the gear plate of the arm is loaded, on which the strength stress analysis was performed. Steel, either according to EN 41 1523 (2014) was used for the design and production of the cutting head. It is an unalloyed structural steel with a strength limit (R_e) of 360 MPa. It is extremely suitable for welding, as the carbon content (C) for a thickness t greater than 100 mm is 0.24. Its usage is possible for bridge structures, welded structures, parts of cars, machines, motorcycles, as well as pressure vessels (U.S. Steel Košice 2022).

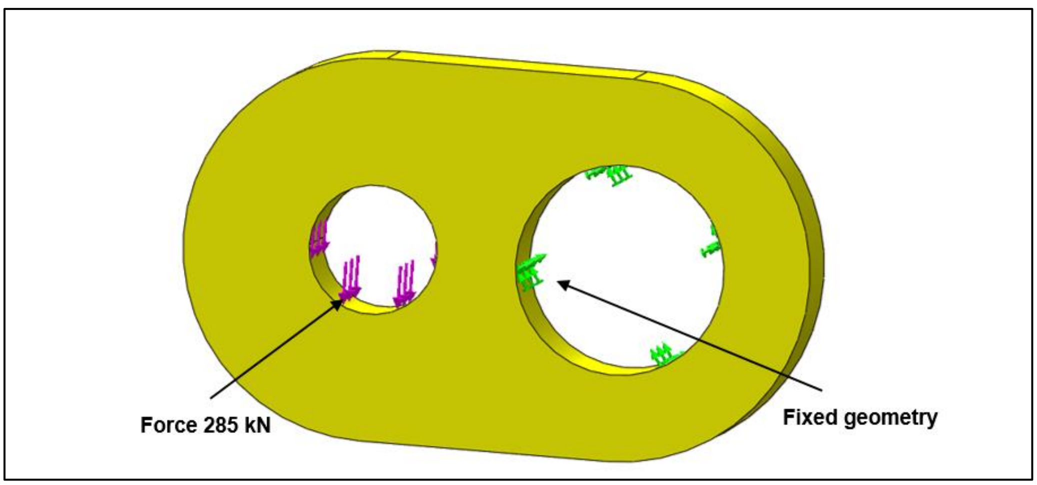

Fig. 6. Loading and fixed of the analyzed component of the cutting head

According to the design in Fig. 3 and mathematical calculations of the cutting force, values were applied to determine the sufficient load and fixation of the examined component. The force that is transmitted at the point of cutting to the examined component represents a value of 285 kN. Fixed holes therefore represent the greatest risk of damage to the cutting head.

Table 1. Chemical Composition of the Steel

EN	Chemical Composition (%)								
	C	Si	Mn	Cr	Mo	Ni	N	S	P
S355J0	0.24	0.60	1.70	-	-	-	0.014	0.040	0.60

EN 41 1523 (2014) is a low carbon, high tensile strength structural steel, which can be readily welded to other weldable steel. With its low carbon equivalent, it possesses good cold-forming properties. The plate was produced *via* a fully killed steel process and supplied under normalized or controlled rolling conditions.

Table 2. Mechanical Properties of Steel EN 41 1523 (2014)

Tensile Strength (MPa)	510 to 680
Yield Strength (MPa)	min. 360
Tensibility, Elongation to Break (%)	22

RESULTS AND DISCUSSION

To check the stressed part of the cutting head, it was necessary to perform a strength analysis of the transmission of the arm, which is directly affected by the force from the cutting rectilinear hydraulic motor. The analysis was performed in Solidworks professional 2019 (Profesinal 2019, Dassault Systèmes, Vélizy-Villacoublay, France) simulations using the finite element method (FEM). This method was also used by the authors in their previous contributions (Mikleš 2012; Hiesl *et al*. 2015; Hatton *et al.* 2017; Mederski *et al.* 2016, 2018; Melicherčík *et al.* 2020). The larger diameter hole is firmly welded to the entire arm, and therefore a wedge-type bond, and thus a stationary bearing, was used. A force of 285 kN acts directly on the smaller diameter hole, but since two plates are used, the resulting force will be only 142.5 kN. However, since the force acts on only half of the hole, it was necessary to divide the hole into two parts. The material was used in the same way as on the whole head - steel class EN 41 1523 (2014) with a yield strength (R_e) that equals a minimum of 360 MPa, with a thickness of 15 mm.

After analysis, it was found that the area of deformation exceeded the limit of elasticity and the part reaches the area of plastic deformation, as can be seen from Figs. 7 and 8. The individual parameters set during the stress analysis were set according to the technical parameters of the material, as well as the Young's module and the post number. The basic parameters were used in the same way as in the simulations published by other authors (Mederski *et al.* 2018; Melicherčík *et al.* 2020).

The stress analysis was determined according to the calculated values and the parameters of the FEM analysis, thus achieving the desired results. The area of greatest damage is indicated in Fig. 6 (as shown in red) as the maximum value, which was 446.6 MPa. This value exceeds the values of the material used and the thickness of the material component by 15 mm.

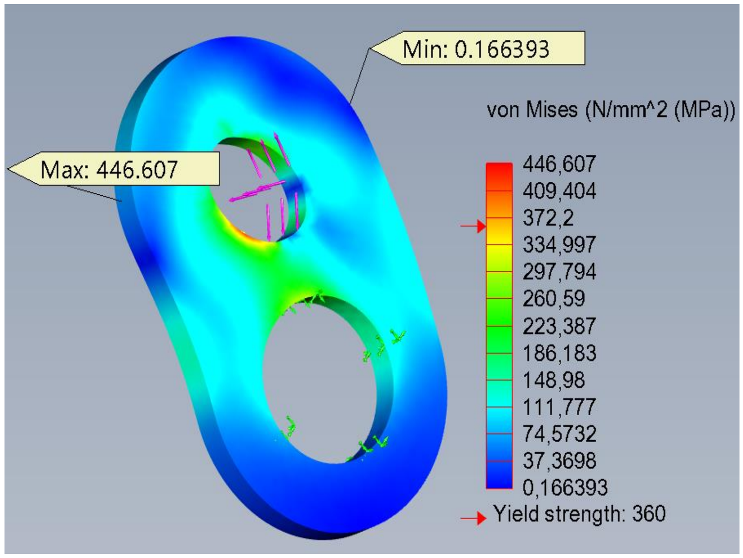

Fig. 7. The stress strain analysis of the arm trasmission (15 mm)

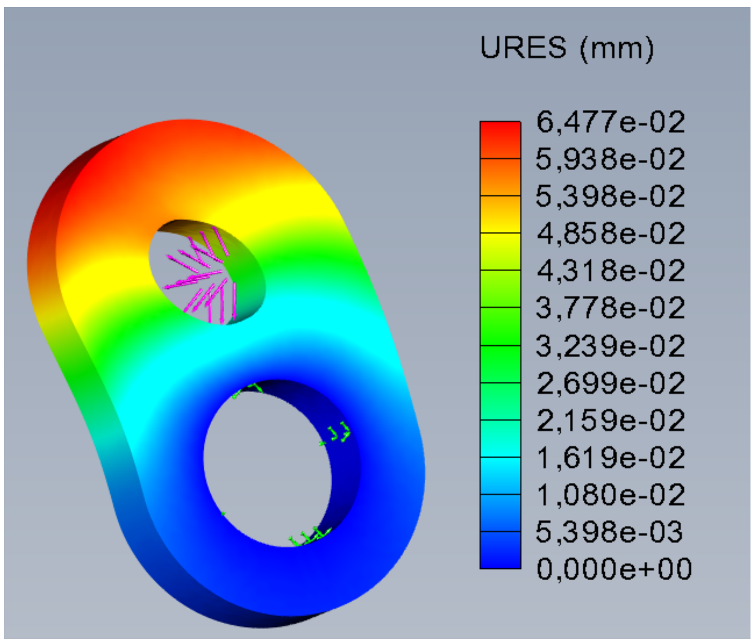

Fig. 8. Stress statics displacement before dimensioning the material to 15 mm

Plastic deformation of the component confirmed excessive damage at the place of functional use, which causes equipment failures during wood processing and limited operation (Fig. 9). With a material thickness of 15 mm, the component was formed at critical points marked in the analysis in red for bending up to 6.477 mm. As it is inadmissible for any part of the assembly to work in the area of plastic (and thus permanent deformation), it was uncompromisingly necessary to oversize the part and perform the analysis again. The material remained unchanged due to good weldability, but its thickness was changed from the original 15 mm to 30 mm. The result of the deformation is shown in Fig. 10. The maximum amount of deformation was 224.8 MPa, which is less than the yield strength (R_e), *i.e.*, 360 MPa, and the safety coefficient remained at the value of k = 1.7, which is the minimum value for steel.

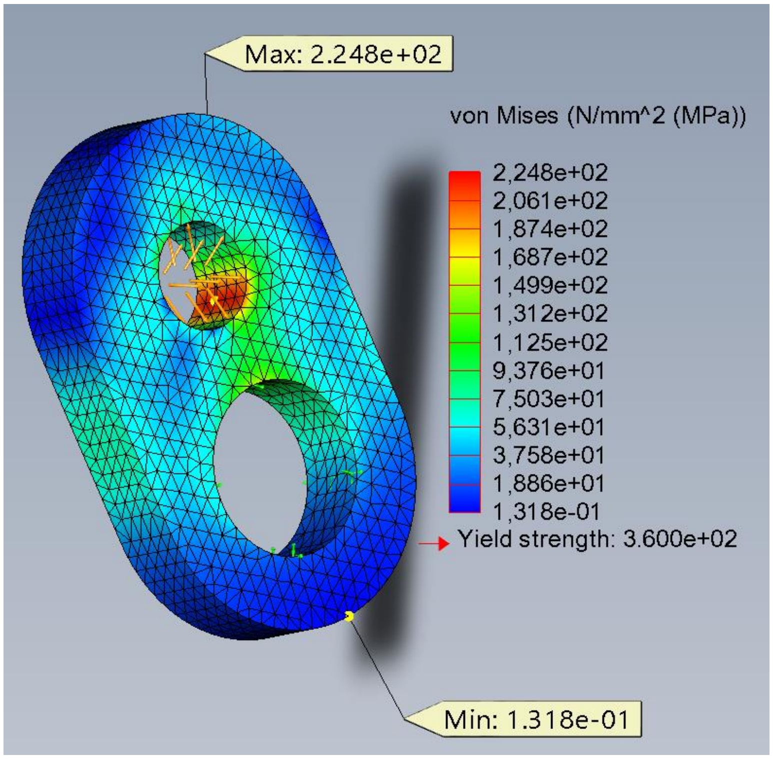

Fig. 9. The stress–strain analysis of arm the transmission

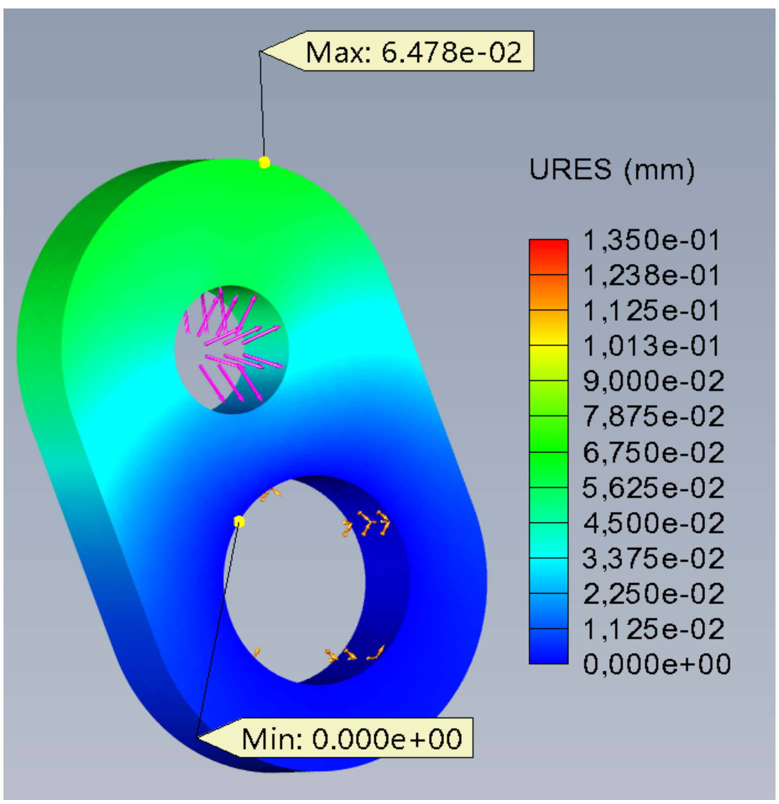

Fig. 10. The stress statics displacement after dimensioning the material to 30 mm

CONCLUSIONS

1. The article describes the theoretical analysis of the forces acting on the cutting mechanism of a chipless cutting head, which reduce the service life of components required for the function of the device.
2. According to mathematical and physical calculations, as well as the use of empirical relations, the cutting force was determined for the “summer” and “winter” seasons, due to different input parameters.
3. The outputs from the stress and strain analysis created *via* FEM confirmed the appropriate material design for the production of a new component with a longer service life. The defined input variables were based on older studies and shielding head parameters.
4. Stress analysis of EN 41 1523 (2014) steel with a material thickness of 15 mm showed a maximum stress of 446 MPa and a deformation of 6.477 mm, which meant excessive damage to the designed component in terms of its functionality.
5. The proposed solution to change the type of material to EN 41 1523 (2014) and the thickness of the material to 30 mm, as confirmed *via* FEM analysis, resulted in the shielding component being eliminated. The deformation Fig. 9 was 1.35 mm, which can be considered negligible.

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