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Load Analysis for the Design of Cutting Teeth for Bucket Chain Excavators

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Abstract: *This paper presents an approach to facilitating the design of cutting teeth for bucket chain excavators. The approach combines the theoretical load model and the laboratory determination of the static load capacity of the bucket tooth. Engineering analysis using the FEM was applied in testing the presented approach in the course of development of modular cutting elements for the bucket chain excavator.*

Keywords: *design, cutting teeth, force, load, FEM, bucket chain excavator.*

1. INTRODUCTION

In bucket chain excavators, rock breakage in the course of the excavation process is directly executed by buckets that have cutting teeth along the front edge. The forces generated by the interaction between the bucket teeth and the rock material during penetration and cutting processes are the dominant tooth load values. Overall, load intensity is not constant and is stochastically distributed over time.

Stress and change of stress are the main cause of fatigue, deformation, crack formation and fracture in the cutting tooth. As a rule, stresses are distributed non-uniformly throughout the tooth volume. The degree of non-uniformity is directly associated with the magnitude and distribution of external loads, the form and mode of tooth attachment to the bucket, as well as with the position and orientation of the cutting tooth during the cutting operation. When sizing bucket teeth and selecting the right material to be used for their fabrication, it is necessary to determine the magnitude, characteristics and duration of the load induced by the interaction between the cutting tooth and the rock. However, there are states and processes which have not been fully elaborated in theory or which are too complex to yield satisfactory results in engineering practice [1]. This exactly is the case with the determination of the external load acting on the cutting teeth. The expressions derived until now are based on a large number of interrelated impact factors which have not been fully investigated or defined. Existing theoretical and empirical models can only provide a rough estimate of the load acting on the bucket teeth since the calculated values span a wide range, depending on the model applied and the variables adopted [2].

Given the above considerations, designing cutting teeth for bucket chain excavators involves making decisions based on incomplete data, since assumptions regarding the characteristics of rock materials, the unpredictability of the digging process, the imperfection of the cutting tooth material, etc. are unsuitable in terms of measurement, qualification and documentation. With respect to this uncertainty, the designer must use an idealised computational model with clear boundary conditions describing the process of rock breakage by cutting teeth. The key to success in the light of these three seemingly uncontrolled circumstances is to use scientific uncertainty qualification. This means that the basic settings of digging-related problems and understanding of the necessary assumptions, along with a proper stress analysis of the bucket teeth for different load and boundary conditions, can facilitate making fundamental engineering decisions based on incomplete data.

This paper presents the Evans two-dimensional model for performing bucket tooth load calculations. The Evans model is a simple model of rock breakage i.e. cutting element loading. This model can be used in bucket chain excavators for the simple calculation of external loads acting on the cutting teeth during the digging operation. The main idea is to use this model for performing load calculations for different operating conditions of the excavator in order to obtain starting data to be used in preliminary calculations for cutting tooth design. In addition to these data, the design process would also employ data obtained from the laboratory testing of the maximum static load capacity of existing cutting teeth. Laboratory test data would also be used for the verification of the computational model and implementation of the finite element method. The established model for the maximum loading of existing cutting teeth would be used for the analysis of future similar designs as well as for the optimisation of new cutting tooth designs.

The cross analysis of the data obtained by observing the calculated values of tooth load and maximum load before failure under laboratory conditions can be used in cutting tooth calculations and optimisation with a certain degree of certainty. Accordingly, ensuring a higher degree of certainty would simplify the determination of the cutting tooth load. Moreover, all major impact parameters, such as rock hardness, tooth geometry and tooth movement would be included, and the limit load capacity and tooth behaviour at fracture would be determined.

2. BUCKET TOOTH LOAD

Due to the high complexity of the digging process, and considering the inability to adequately record or measure the spatial and temporal effects of this process, approximation is made by means of appropriate two-dimensional models [3], [4], [5], [6]. The same principle can be use in analysing the total load on the cutting teeth in bucket chain excavators [7], [8].

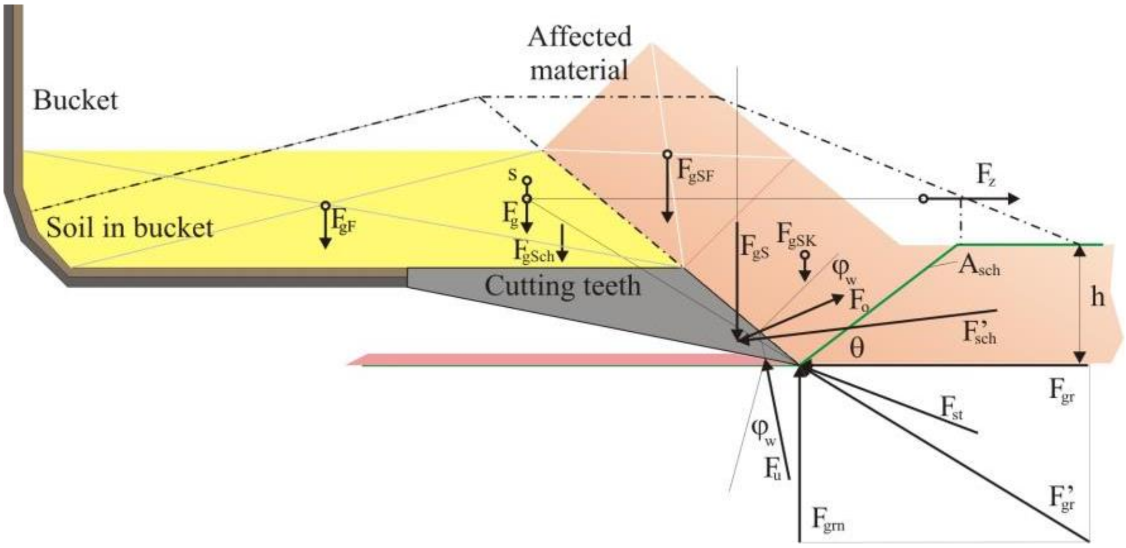


Figure 1. *An in-plane model (two-dimensional) of loads acting on the cutting tooth interacting with the rock material*

Figure 1. shows an in-plane mechanical model providing an analysis of the forces generated during the interaction of the cutting teeth and the bucket with the rock material. The model involves penetration, cutting and scooping processes, where F_{gF}, F_{gSF}, F_{gSK} –weight of the excavated material, s – centre of gravity of the excavated material, F_g – total weight of the excavated material at the centre of gravity, F_{gS} – total weight of the separated material, F'_{sch} – resulting tangential force, A_{sch} – tangential surface, φ_w – angle of external friction, h – depth of the cut, F_{gr} – tangential component of cutting resistance, F_{grn} – normal component of cutting resistance, F'_{gr} – total cutting resistance, F_u – resistance on the back of the tooth, F_o – resistance on the tooth face, F_z –total bucket filling resistance.

The theoretical calculation of digging resistance is very complex, given the dependence of this resistance on a range of interrelated factors, with the interrelation, however, not being clearly defined. The complexity of the processes occurring during the digging operation leads to very intricate and insufficiently clear theoretical explanations. Therefore, an important role in calculating the digging resistance is given to the experimentally derived empirical equations. However, as general laws of simplicity are not applicable in this case, the empirical equations apply only to cases approximately corresponding to the calculated values. The complex correlations among the impact parameters lead to simplification in experimental measurements as well; accordingly, only the most important factors or factors having the greatest influence on the problem analysed are taken into consideration.

The considerations elaborated in [9] have defined the first cutting model used in the description of the process of excavation of coal and similar rock materials. In this research, for the first time in theory, it has been shown that the tensile and compressive strengths of the rock are the dominant characteristics used in calculating the load sustained during rock breakage by a wedge-shaped cutting element. Subsequent experimental studies [10], [11] have confirmed that this theory can also be used for brittle rock materials such as lime and sand. The Evans theoretical model has been further improved and experimentally verified. The improvements of this model have mostly been targeted towards solutions to certain single problems [12], [13], [14], but the basic philosophy and approach have remained unchanged. For its simplicity and the wide range of applications in different rock materials, this model is analysed in more detail below.

3. EMPLOYMENT OF THE EVAN'S LOAD MODEL

The starting point in the Evans plane model is the assumption regarding movement of the ideal symmetrical cutting element and its penetration into the rock material. During the penetration of the cutting element (moving along the path which is normal to the material), the stress generated in the rock material leads to the formation of an initial crack, which further spreads radially starting from the tip of the cutting element to the free surface of the material (arc CD), as presented in figure 2 [15]. The basic Evans model entails the following additional assumptions:

- A force R is acting at the angle δ (external friction angle) relative to the normal of the surface AC of the cutting element (wedge).
- A resultant tensile force of the rock T is acting at the symmetry line of the angle 2θ in the arc CD.
- The depth of penetration of the wedge is very small compared to the cutting depth d.
- The width of the wedge (w) is much greater than the depth (d) ($w>>d$) i.e. the model of the two-dimensional cutting theory can be applied.
- A state of plane stress is applicable.

During its movement, the wedge tends to split the rock, while rotating it about point D. Therefore, it is assumed that the force S acts through point D. Along the fracture line CD, it is assumed that a state of plane stress applies and the equilibrium of forces is considered per unit of width of the wedge.

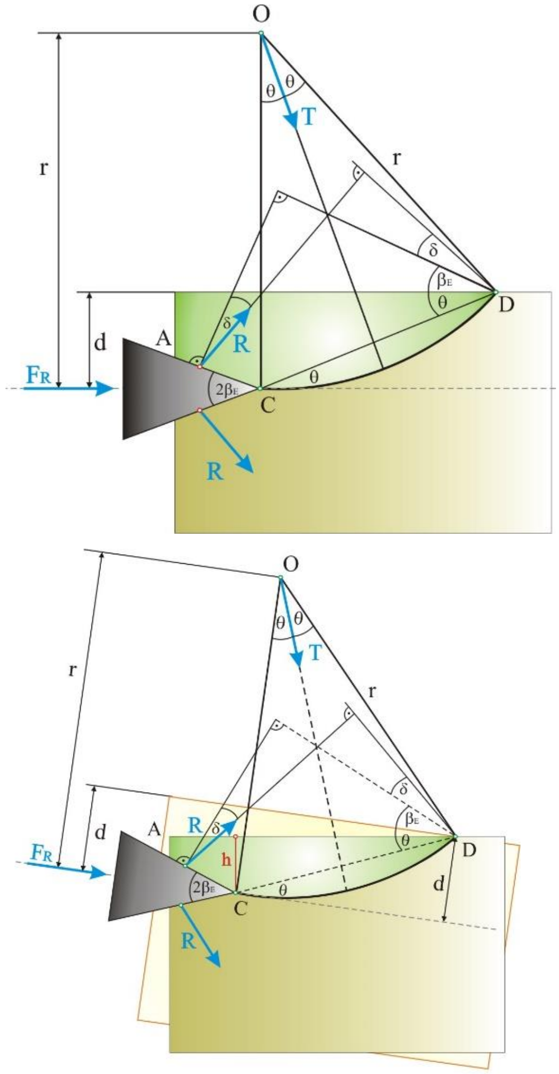
The force due to the tensile strength of the rock (σ_z) is defined by the following expression:

$$T=\sigma_z\cdot r\cdot\int_{-\theta}^{\theta}cos\omega\,\mathrm{d}\omega=2\cdot\sigma_z\cdot r\cdot sin\theta \qquad (1)$$

Where $r\mathrm{d}\omega$ is an element of the arc CD making an angle ω with the symmetry line of the angle. The depth of penetration of the wedge tip into the rock may be neglected in comparison with the depth of the cut *(d)*. This means that the point of application of force R is near point C.

$\sum M_D=0$,

$$R\frac{d}{sin\left(\theta-\beta_E\right)}\cdot cos\left(\theta+\beta_E+\delta\right)=T\cdot r\cdot sin\theta \qquad (2)$$



a)

b)

Figure 2. *a) Tensile breakage theory of Evans, b) Approximation of cutting by a tooth on excavators.*

In accordance with $r\cdot sin\theta=\dfrac{d}{2sin\theta}$, it follows that

$$R=\frac{\sigma_z\cdot d}{2sin\theta\cdot cos\left(\theta+\beta_E+\delta\right)} \qquad (3)$$

The horizontal component of R is $R\cdot sin(\beta_E+\delta)$, and due to the symmetry of the forces acting on the wedge the total cutting force is:

$$F_R=2\cdot R\cdot sin\left(\beta_E+\delta\right)=\sigma_z\cdot d\cdot\frac{sin\left(\beta_E+\delta\right)}{sin\theta\cdot cos\left(\theta+\beta_E+\delta\right)} \qquad (4)$$

The normal force is (for one side of wedge):

$$F_n=R\cdot cos\left(\beta_E+\delta\right)=\sigma_z\cdot d\cdot\frac{cos\left(\beta_E+\delta\right)}{2\cdot sin\theta\cdot cos\left(\theta+\beta_E+\delta\right)} \qquad (5)$$

The angle θ is defined at the minimum value of F_R, and therefore:

$$\frac{\mathrm{d}F_R}{\mathrm{d}\theta}=0,\quad \frac{\sigma_z\cdot d\cdot sin\left(\beta_E+\delta\right)}{sin\theta\cdot cos\left(\theta+\beta_E+\delta\right)\cdot\mathrm{d}\theta}=0 \qquad (6)$$

$$cos\theta \cdot cos\left(\theta+\beta_E+\delta\right)-sin\theta \cdot sin\left(\theta+\beta_E+\delta\right)=0,$$

$$cos\left(2\theta+\beta_E+\delta\right)=0 \qquad (7)$$

Resulting in the angle value: $\theta=\frac{\pi}{4}-\left(\beta_E+\delta\right)$

$$sin\theta \cdot cos\left(\theta+\beta_E+\delta\right)=$$

$$=sin\left(\frac{1}{2}\left[\frac{\pi}{2}-\beta_E-\delta\right]\right)cos\left(\frac{1}{2}\left[\frac{\pi}{2}+\beta_E+\delta\right]\right)=$$

$$=\frac{1}{2}\left[sin\left(\frac{\pi}{2}-sin\left(\beta_E+\delta\right)\right)\right]$$

According to previous, the total cutting force is:

$$F_R=2\cdot\sigma_z\cdot d\cdot\frac{sin\left(\beta_E+\delta\right)}{1-sin\left(\beta_E+\delta\right)} \qquad (8)$$

The normal force for one side is (in this case, total normal force is zero due to the symmetry):

$$F_n=\sigma_z\cdot d\cdot\frac{cos\left(\beta_E+\delta\right)}{1-sin\left(\beta_E+\delta\right)} \qquad (9)$$

For cutting teeth of bucket chain excavators, two assumptions can be achieved:

• the path of the cutting tool has the same direction as the fracture.

• the maximum thickness of the chip is almost the cutting depth.

With the aid of figure 2.a, and with two above assumptions:

$h=2\cdot r\cdot sin\theta\cdot sin\left(\theta-\alpha\right)$ and

$$d=2\cdot r\cdot sin^2\theta,\ d=h\cdot\frac{sin\theta}{sin\left(\theta-\alpha\right)} \qquad (10)$$

The total force (per unit of width) will be:

$$F_R=2\cdot\sigma_z\cdot d\cdot\frac{sin\left(\theta+\delta\right)}{1-sin\left(\theta+\delta\right)}=$$

$$=2\cdot\sigma_z\cdot h\cdot\frac{sin\left(\theta+\delta\right)}{1-sin\left(\theta+\delta\right)}\cdot\frac{sin\theta}{sin\left(\theta-\alpha\right)} \qquad (11)$$

The cutting force (horizontal component) is:

$$F_c=F_{RH}=2\cdot\sigma_z\cdot h\cdot\frac{sin\left(\beta_E+\delta\right)}{1-sin\left(\beta_E+\delta\right)}\cdot\frac{sin\theta}{sin\left(\theta-\alpha\right)}\cdot cos\alpha\cdot w \qquad (12)$$

The penetration force (vertical component) is:

$$F_p=F_{RV}=2\cdot\sigma_z\cdot h\cdot\frac{sin\left(\beta_E+\delta\right)}{1-sin\left(\beta_E+\delta\right)}\cdot\frac{sin\theta}{sin\left(\theta-\alpha\right)}\cdot sin\alpha\cdot w \qquad (13)$$

In terms of the type of the material being excavated and the form of its breakage, digging by bucket chain excavators can vary widely between highly ductile failure of soft rock materials such as clay and loam to brittle failure of very brittle materials such as coal and rocks of different composition. If the problem is viewed from the standpoint of rock material strain, the following is observed: in rock materials such as clay and lignite, the cutting edge of the bucket blade serves for cutting purposes in the plastic fracture mechanics; in hard rocks, bucket teeth serve for facilitating the penetration process i.e. for ensuring that normal and tangential stresses are large enough for the initial fracture of the rock material to occur. This very fact justifies the use of the Evans model in designing bucket teeth. When designing bucket teeth, loads potentially leading to their critical state (failure and deformation) during the exploitation process should be taken into consideration. These loads are short-time and mostly generated during the interaction with the rock material of substantial hardness. The Evans model is appropriate for load description and force determination.

Table 1. Physical and mechanical characteristics of the rock material at the Tamnava West Field Open Cast Mine in Serbia

Lithological constituents	Moisture [%]	Bulk density [kN/m^3]	Angle of internal friction ϕ [°]	Cohesion c [kN/m^2]
Quaternary clay	25.0	19.9	25°	16.0
Alluvial sand	22.0	18.1	26°	1.0
Alluvial gravels	30.0	19.0	28°	0.0
Alluvial gravels and sand	29.0	18.6	30°	2.0
Pontian clay	31.0	18.6	25°	20.0
Coal	-	11.5	39°	30.0
Pontian sand	20.0	17.1	28°	4.0
Coaley clay	58.0	16.4	16°	15.0

Table 1. gives an overview of average physical and mechanical characteristics of the rock material at the Tamnava-West Field Open Cast Mine in Serbia. These characteristics apply to average operating conditions of the existing ERS 1000 bucket chain excavator teeth, which are analysed below.

Table 2. Values of parameters for existing excavating conditions used in tooth load calculations by the Evans model

Parameter		Designation	Value
Width of the cutting tooth (new-partially worn)		*w*[mm]	60÷120
Depth of the cut (cutting engagement of the bucket tooth)		*d*[mm]	40÷80
Wedge angle		$\beta=2\beta_E$[°]	22÷30
Uniaxial compressive strength of the rock material		σ_p[N/m^2]	5÷10
Uniaxial tensile strength of the rock material		$\sigma_z= (0.1)\cdot\sigma_c$	0.5÷1
Coefficient of external friction		μ	0.4÷0.5
Angle of external friction		$\delta=$ arctg(μ) [°]	21.8÷26.6°
Calculated value of force	Cutting	F_C[kN]	4.2÷75.0
	Penetration	F_P [kN]	0.8÷20.0
	TOTAL	F_{uk}[kN]	4.2÷77.0

Table 2. presents the values of parameters applicable to current excavating conditions and used in tooth load calculations. The analysis of the results in Table 2. shows a large dispersion of the load values. Due to the complexity of the problem of defining cutting tooth load, the adoption of a large number of starting assumptions and the inability to ensure the exactness of calculations, the obtained values should not be decisive in making engineering decisions. As already highlighted, the maximum calculated load values should be used as a guideline in the design process, as well as in the sizing and shaping of the cutting tooth.

4. ANALYSIS OF MAXIMUM LOAD

In addition to the computational analysis based on the FEM (finite element method) using the calculated load values, the design process should also involve an analysis using the maximum possible load (behaviour, stress and strain) – the load sustained at tooth failure. Such an analysis is not complicated, and it can be performed by laboratory tests.

The analysis comprises two stages. The first stage takes place as part of laboratory testing, involving loading of the cutting tooth until failure. Both the force being generated and the cutting tooth strain are monitored. Here, the direction and course of action of the external load are designed to be identical to those under actual field conditions, whereas the load intensity is increased until the cutting tooth fails. The second stage involves a computational analysis using maximum force readings before the failure of the physical prototype, taking into consideration all laboratory testing conditions. The comparison of the results from these two stages actually provides verification of the established computational model, while at the same time resulting in the maximum static force that the cutting tooth can withstand before failure. The verified computational model is a virtual prototype [16],[17], which is further used in the analysis incorporating the load calculations given in Table 2.

This approach should be employed at the initial design stage and later on before testing the physical prototype. Virtual prototypes are used at the early stage of development to simulate and check the properties of the cutting tooth before the fabrication of the physical prototype, thus achieving substantial savings or eliminating both the costs and the time needed to fabricate and modify the physical prototype. Unlike physical prototypes, virtual models make use of all advantages of the modern computational technique, with the focus of the engineering process shifted from the physical to the virtual environment. This shift offers a great advantage in solving problems in excavators operating in open-pit mines, given that testing of any physical prototype has significant limitations, and is a difficult costly process.

These considerations are particularly important in view of the tendency towards one-off production in the design of cutting teeth for bucket chain excavators. This entails the optimisation of cutting teeth for every single excavator and operation technology used, and rock material being excavated. In such cases, a flexible approach must be ensured not only in the engineering process but also in the fabrication and testing of prototypes.

The paragraphs below present the laboratory testing of two types of cutting teeth for the ERS 1000 bucket chain excavator [18]: a) the existing one-piece cutting tooth and b) the new modular cutting tooth developed to reduce production costs and improve the cutting performance.

Table 3 Mechanical characteristics of the bucket and cutting tooth material [19]

Characteristic	Designation of the material	Yield strength [N/mm^2]	Tensile strength [N/mm^2]
Existing cutting tooth	42CrMo4	710	1.100
Modular tooth (cutting portion)	42CrMo4	710	1.100
Modular tooth (holder)	CK45	530	650
Bucket	EN S355J2G3	300	580
Lifting eyes on the bucket	EN C35E	170	580
Bolt (M20, 10.9)	34Cr4	900	1.000

In contrast to field testing, laboratory testing is strictly controlled and manageable, and as such directly controllable and comparable with the computational analysis. Importantly, in this way, the verification of the virtual prototype design is made significantly easier. In accordance with laboratory testing, a successful virtual prototype also allows incorporation and modelling of the currently used test equipment, that is, the boundary conditions.

Figure 3. show the basic setup for the laboratory testing. The cutting teeth are mounted on the bucket in exactly the same position as on the excavator during the excavation process. Through the hydraulic system and the hydraulic cylinder, the hydraulic generator automatically exerts force. The position, orientation and point of application of the hydraulic cylinder on the bucket tooth are adjusted to portray the action of the external forces acting on the bucket tooth during the excavating operation.

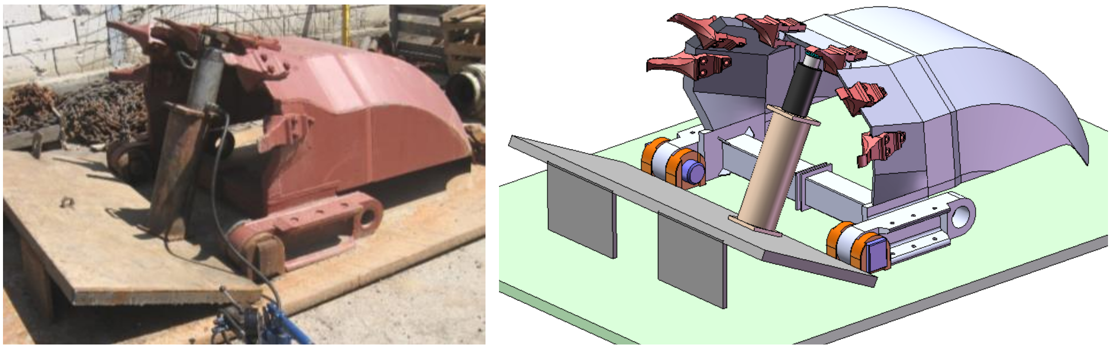
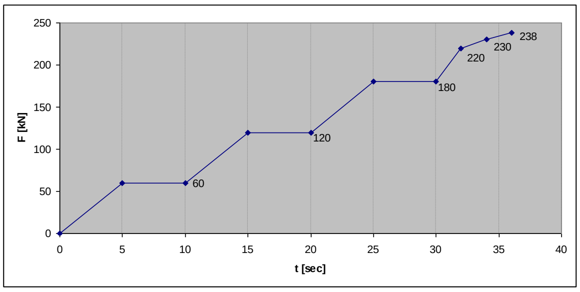


Figure 3. *Laboratory testing for maximum bucket tooth load and virtual prototype.*

During the laboratory experiment, loads are exerted through a hydraulic fixture with force gauge showing readings of the instantaneous force value. Loads are gradually applied to avoid dynamic effects that may occur due to a rapid increase in force. During loading, force and strain values were measured for both types of bucket teeth. The force diagram and the average strain diagram for one of the existing bucket teeth tested are presented in figure 4.

a) 

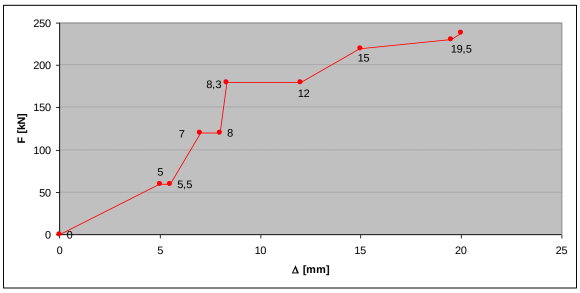
b) 

Figure 4. *a) Load diagram (the force created by the hydraulic cylinder) and b) Value of strain at the tip of the bucket tooth.*

When determining the direction, intensity and location of the point of application of the force acting on the physical prototype, two important facts were considered [20], [21], [22]. The force direction was determined to approximate the most unfavourable rock scooping case in which the excavator bucket is rotated around the external chain wheel to start interacting with the rock material. The force direction in the physical prototype was adjusted by setting the position and orientation of the hydraulic cylinder and its support.

As the experiment focused on bucket tooth fracture, the force applied was gradually increased until failure. The measured value of the force at failure of the physical prototype was taken as the load for the computational analysis using the FEM.

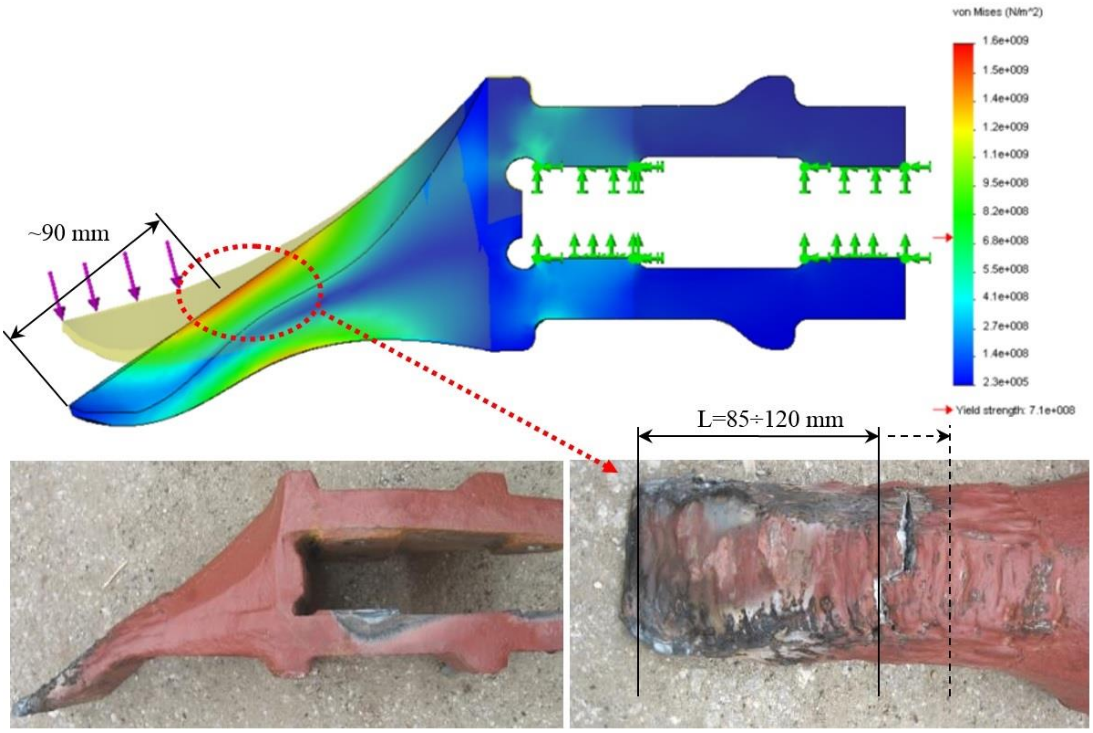


Figure 5. *A stress pattern of the bucket tooth after being subjected to maximum load, and the fractured bucket tooth after being subjected to maximum load under laboratory conditions.*

Figure 5 presents the results of failure of the one-piece bucket tooth during the laboratory testing. The figure also shows the stress distribution resulting from the computational analysis. The value of force at failure during the laboratory testing ranged from 210÷250 kN. This range was used when exerting force during the virtual experiment, with the average of 240 kN (for 10 tested cutting teeth) adopted. Computer-assisted analysis (the virtual experiment) was performed in the same manner as during the laboratory investigation. The stress distribution (figure 5) at a given load of 240kN indicates that the stress value in the critical area (red zone) exceeds the tensile strength value of the tooth material. The marked area indicates the location of a possible fracture. The measured values of maximum strain at the tip of the cutting tooth ranged from 15÷20 mm, whereas the computed value was approximately 16 mm.

During the testing of the physical prototype of the modular cutting tooth, the value of force at fracture was 180÷210 kN. Based on the experimental testing of a substantial number of modular bucket teeth, the average force before fracture was 190 kN. The exertion of this load during the computational analysis of the modular cutting tooth resulted in the stress pattern presented in figure 6.

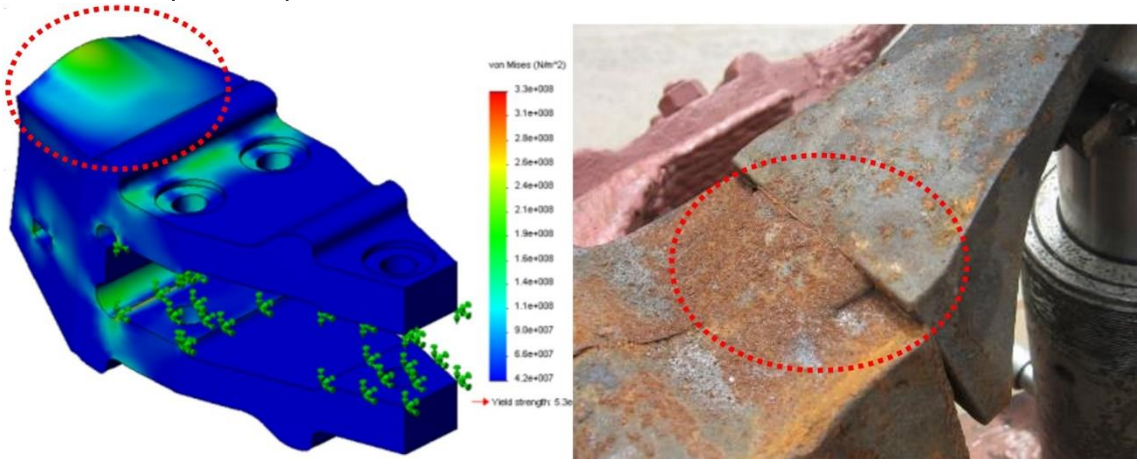


Figure 6. *Stress distribution in the modular tooth holder, and the fractured physical prototype.*

The figure provides an example of the most common type of fracture of the bucket tooth holder during the laboratory experiment. This fracture is due to overstrain in the tooth holder, causing the cutting portion of the tooth to drop out. A very high degree of compliance was found between the location and form of fracture of the tested bucket teeth, and the stress pattern and stress levels in the computational model.

5. CONCLUSION

Chapter 3. provides bucket tooth load calculations for the existing excavating conditions in the Tamnava-West Field Open Cast Mine in Serbia. Loads were calculated for the existing one-piece cutting tooth using the Evans two-dimensional model. In Chapter 4, the maximum static load that the existing one-piece tooth and the newly developed modular cutting tooth can endure were calculated. Models used in the finite element analysis of the maximum load capacity were established and verified.

During the first step of optimisation, the newly developed modular cutting tooth retained the geometrical characteristics and mode of attachment of the existing one-piece tooth. Therefore, the load value calculated in Chapter 3. is also applicable to the analysis of the modular cutting tooth. Part of the results of the finite element analysis of the modular cutting tooth are presented in figure 7. The aim of the analysis was to observe the influence of the assumed direction and distribution of external loads (cutting force FC and penetration force FP) on stresses inside the cutting tooth. When designing a cutting tooth, it is important to identify the most unfavourable stress pattern for the same intensity of penetration and cutting forces.

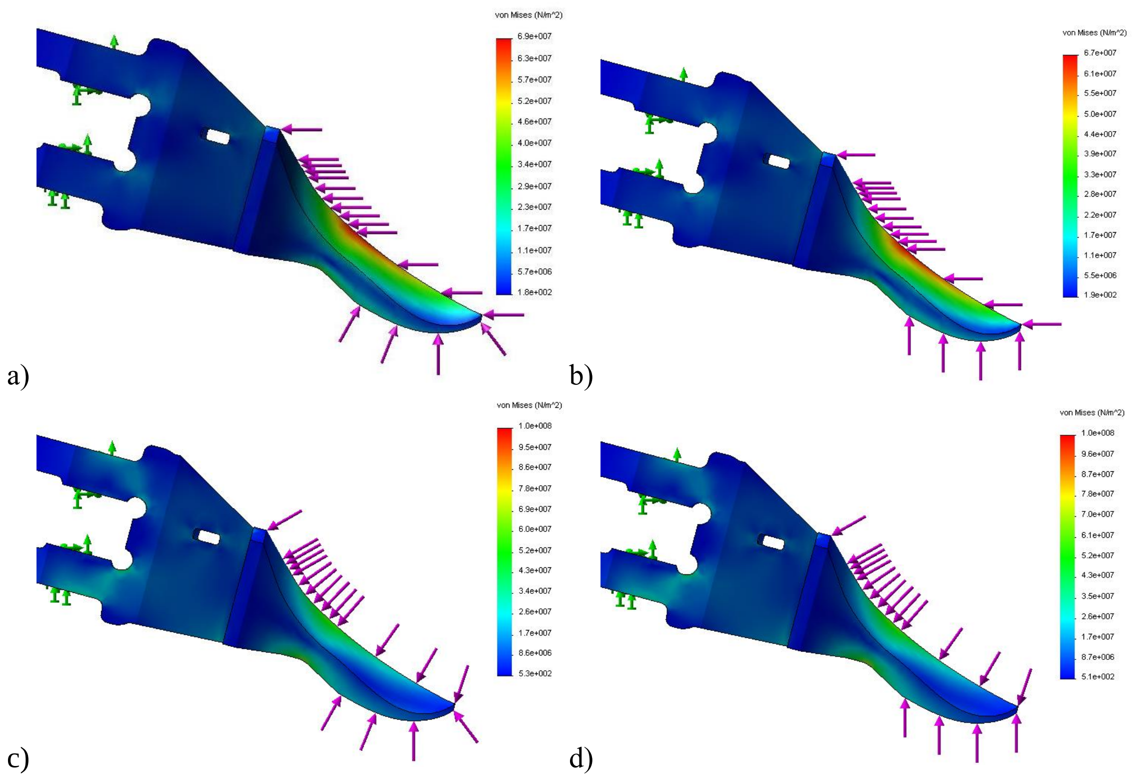


Figure 7. *Stress distribution in the modular cutting tooth for the same support conditions and the same intensity of forces, but for different assumed directions of cutting and penetration forces.*

Maximum values of the calculated cutting resistance FC=75kN and penetration resistance FP=20kN were used in the analysis. The material for the modular cutting tooth was characterised in accordance with Table 3.

The entire analysis (model parameters, attachment to the bucket, number of finite elements, etc.) was based on the verified FEM model for the analysis of maximum loads

Four different cases were discussed, each featuring forces of steady intensity, an identical mode of support for securing the tooth to the bucket, and variations in the direction of cutting and penetration forces. Figures 7. a, b, c and d present the following 4 cases, respectively:

a) The direction of the force FR coincides with the direction of motion; The force FP acts in a direction normal to each point on the back,

b) The direction of the force FR coincides with the direction of motion; The force FP acts in a vertical direction,

c) The force FR acts in a direction normal to each point on the face; The force FP acts in a direction normal to each point on the back,

d) The force FR acts in a direction normal to each point on the face; The force FP acts in a vertical direction.

The results indicate that the maximum stress acting on the cutting tooth varied from 70÷100 N/mm2, for the same load intensity and the same adopted tooth material, but for different load directions. This shows that stress distribution is influenced not only by load intensity but also by the mode of load determination in the finite element analysis. From the point of view of certainty, a less favourable case should be adopted in the load capacity analysis; accordingly, the force acting on the tooth face should be modelled as in the case of c) and d). The assessment of only these two cases suggests that variations in the direction of the force FP have no substantial effect on total stress.

The advantage of the approach presented in this paper is that it reduces the time required to develop and test the cutting teeth of bucket chain excavators. This is of particular importance given the need for uniform optimised cutting tooth designs. Virtual prototyping allows the fabrication of a physical prototype at the very end of the process i.e. after the correction of most of the errors and after optimisation of the cutting tooth design.

The simplified mechanical model provides a rough approximation to the excavating process, while substantially departing from the inclusion of all impact factors associated with the total load value. In particular, load values obtained by direct measurement [23] (e.g. using strain gauges) on the cutting tooth during the excavation operation contribute to the design of more appropriate, more accurate simulation models. However, in general, there is no comprehensive solution to the problem due to a range of non-measurable effects (such as determination of the exact load direction, number of surfaces on cutting teeth across which loads are transferred, characteristics of the scooped rock material, dynamic effects etc.). The design of cutting teeth for bucket chain excavators is mostly based on empirical knowledge. Therefore, the authors believe that, towards a systemic approach to cutting teeth engineering, the approach presented in this paper is the optimal solution among all possible solutions entailing either simplification of the load defining process or inclusion of as many parameters affecting bucket tooth load as possible.

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