Method of increasing the efficiency of forest machine roll-over protective structure

I G Skobtsov*, V N Shilovskiy

Institute of Forestry, Mining and Construction Sciences, *Petrozavodsk State University, 33 Lenin Street, Petrozavodsk 185910, Russian Federation

*Corresponding email: iskobtsov@mail.ru

Abstract. Current standards in the field of self-propelled machinery for forestry require equipping tractor cabins with roll-over protective structures (ROPS). It is necessary to reduce the risk of operator’s injuries caused by tractor rollovers or falling trees. This article deals with the way of increasing the energy absorption capacity of forest machine cabin roll-over protective structure. The ability to absorb certain amount of potential energy during deformation is one of basic requirements to cabin of a forest machine. The risk of injuries to the operator is reduced by using energy-absorbing cab support in the construction of roll-over protective structure. The protective effect is attained by plastic strain of cab support elements, that provides operator’s protection during an accidental roll-over. The article presents the design of the cab support, describes its functional principle in the event of an emergency.

1. Introduction

Forest machine cabins have technical distinguishing features and differ from agricultural or earth-moving machine cabins due to their specific work conditions. Current standard ISO 8082–1 [1] regulates static laboratory research of roll-over protective structure efficiency according to the safety requirements and test methods for machines and tractors for forestry.

There are three ways to improve tractor cabin protection properties. The first direction is theoretical determination of protective frame deflected mode by using a mathematical model, and few researchers have addressed to this method. For example, Harris J R et al [2, 3], Powers J R et al [4], Etherton J R et al [5] analysed agricultural tractor rollovers using finite element modelling. Dumitrache P [6] investigated the behaviour of earth-moving machine protective structures under load by using the method of parametric modelling. This way of ROPS efficiency evaluation is connected with the solution of an optimization problem to choose substantiated structural parameters (geometrical dimensions, tolerances, defect sizes etc). The statement of an optimal design problem with the catastrophe theory application was presented in [7]. The second way to increase ROPS efficiency is to investigate the availability of new perspective materials to produce protective constructions. And, the last way is connected with the application of additional safety devices. This article contains the design of energy-absorbing cab support to provide additional energy absorption effect in case of rollover. Therefore, the protective effect is attained by plastic strain of cab support elements to ensure the safety of operator during an accidental roll-over.
2. Materials and methods

The evaluation of forest machine safety is an important problem attracting interest of many experts both in Russia and abroad [7–10]. The authors [8] investigated the stress-strain state of wheeled skidder TLK-4-01 ROPS during the lateral loading by using the mathematical model based on the calculation of the variable parameters of elasticity. The design of this ROPS system is presented as an enclosed protective frame 2 (Figure 1). It is located in the transverse plane of symmetry of the cabin 1 and secured by bolts to the tractor frame.

According to [1], the standard values of the lateral force $F$ and the potential energy $U$ absorbed during the lateral loading depend from the mass of the machine $m$. For wheeled forestry machine

$$ F = 60000 \left( \frac{m}{10000} \right)^{1.2} ; U = 12500 \left( \frac{m}{10000} \right)^{1.25} . $$

Machine TLK-4-01 mass is $m = 14500$ kg. Thus, the standard lateral force is $F = 93.71$ kN and the standard potential energy absorbed during the lateral loading is $U = 19.89$ kJ. ROPS construction absorbs about 3 kJ under elastic deformations [8], therefore, the most quantity of the potential energy is absorbed under plastic deformation. Moreover, the energy absorption under plastic deformation of the protective frame (ROPS) is undesirable when the construction under the load reaches the point of deflection-limiting volume (DLV – the protection zone of the operator defined by ISO 3411). The experimental data [8] suggests that, although the comparison between the quantity of this energy and its standard value showed that the size of this criterion meets the standard requirements, in some cases (for example, in the case of presence of crack-like defects in ROPS material) this condition will not be fulfilled.

Consequently, the supplementary energy-absorbing device should be used in the construction to provide operator safety and to increase the ROPS reliability.

The design of energy-absorbing cab support (mount). Energy-absorbing cab support was developed to provide additional energy absorption effect in case of tractor overturn. The protection is attained by plastic strain of cab support elements. The structure of mount (Figure 2) consists of draw die 1, mounting bolt 2, screw nut 3, damping element 4, guide 5 and side panels 6.

Draw die 1 includes (Figure 3): die entrance $I$, operating area $II$, bearing die $III$ and closing die $IV$. Draw die 1 is rigidly fixed at the bottom of the protective frame of the forestry machine cabin in such a way that the axis of its convergent channel aligns with the axis of the mounting holes in the cabin. The mounting bolt 2 is installed in the die channel 1 and fixed in the screw nut 3 by threaded connection. Screw nut 3 and damping element 4 are installed in guide 5. It is rigidly fixed with the machine frame and equipped with side panels 6. Side panels 6 exclude displacement of the construction in a plane perpendicular to the advancing direction of the forestry machine.

In event of an emergency the cabin of the forest machine will be subjected to lateral load impact $F$ (Figure 4). This load will initiate reaction force in the mounts: one of them will work in tension (reaction force $R_1$); the other will work in compression (reaction force $R_2$, Figure 4). The protective structure with draw die 1, mounting bolt 2 and screw nut 3 will begin to apply compressive action to the damping element 4 due to force $R_2$ impact. The damping element 4 will deform until the protective structure with the draw die 1 touches the guide 5. Due to reaction force $R_1$ impact, the protective structure will start stretching the bolt 2 and trying to tear off the left mount. In this case the draw die 1 will start plastic deforming – stretching the bolt 2, moving in the axial direction and creating safety effect of an energy absorption. Therefore, in event of an accidental rollover, lateral load impact $F$ will be directed to the deflection of the bolt 2 and overcoming friction in the die channel 1 and, thereby, the main quantity of strain energy will be absorbed due to bolt 2 plastic deformation by drawing. If the value of lateral force is $F = 93.7$ kN (according to ISO 8082), the value of mount reaction in this case $R_1 = R_2 = 200$ kN (Figure 3) and the value of required plastic deformation $\Delta = (19.89 – 3) / 200 = 0.085$ mm.
Figure 1. The design of forest machine roll-over protective structure

Figure 2. The design of energy-absorbing cab support

2.1. Calculations
Drawing per pass is defined by the relation of cross-sectional areas $S_0 / S_1$ before and after drawing (with constant volume of metal). Obviously, drawing and cold hardening are equally distributed over the thickness of deformation zone. It allows using the method of simultaneous solution of equilibrium and plasticity equations to determine the pressure of metal and the drawing force.

Mounting bolt deforming element with thickness $dx$ is marked by two cross sections in the deformation zone (Figure 3). Element edge normal stresses $\sigma_x$ and $(\sigma_x + d\sigma_x)$ are considered to be uniformly distributed over the element edge area. Apart from these stresses, the element is subjected to pressure $p$ and friction $\tau$ at the point of contact with the draw die. We shall accept that $\tau = \text{const}$.

Condition of equilibrium is the sum of force projections on an axis $x$ (Figure 3).
\[
\pi \cdot (r - \tan \alpha \cdot dx)^2 \cdot (\sigma_x + d\sigma_x) - \pi r^2 \cdot \sigma_x - 2\pi \cdot (r - 0.5 \cdot \tan \alpha \cdot dx) dx \cdot \tau - 2\pi \cdot (r - 0.5 \cdot \tan \alpha \cdot dx) dx \cdot p \cdot \tan \alpha = 0. 
\]  
(2)

Figure 3. Scheme of the drawing process

Figure 4. Scheme of the lateral loading during the accidental rollover \((a = 1800 \text{ mm}; \ d = 810 \text{ mm})\)

After similarity transformations and removing infinitely small higher orders, the equilibrium condition

\[
r \left( \frac{d\sigma_x}{dx} \right) - 2\tan \alpha \cdot (\sigma_x + p) - 2\tau = 0,
\]  
(3)

where, \(r = r_0 - x \cdot \tan \alpha\); \(\alpha\) – operating area angle.
Approximate plasticity equation:

\[ \sigma_1 - \sigma_3 = \sigma_s, \]  

(4)

where, \( \sigma_1, \sigma_3 \) – principal stresses; \( \sigma_s \) – strength of material.

Assuming that axes \( x \) and \( y \) align with the direction of principal stresses \( \sigma_1 \) and \( \sigma_3 \) and value of angle \( \alpha \) is rather small, we can write

\[ \sigma_x + p = \sigma_s, \]

(5)

Simultaneous solution of (3) and (4) gives

\[ \sigma_x = \sigma_0 + \left( \sigma_s + \frac{\tau}{\tan \alpha} \right) \cdot \ln \left( \frac{r_0}{r} \right) \]

(6)

then the relationship between the drawing force and the diameter of mounting bolt deforming element

\[ F(d) = S_1 \cdot \left[ \sigma_0 + \left( \sigma_s + \frac{\tau}{\tan \alpha} \right) \cdot \ln \left( \frac{r_0}{r} \right) \right] = \frac{\pi \cdot d_1^2}{4} \cdot \left[ \sigma_0 + \sigma_s \cdot \left( 1 + \frac{f}{\tan \alpha} \right) \cdot \ln \left( \frac{d_0}{d_1} \right) \right], \]

(7)

where, \( f \) – friction coefficient.

2.2. Experimental part

The aims of the test were confirmation and correction of estimated values of mounting bolt diameter. These values are guide ones and cannot be considered as final because the actual mechanical characteristics (ultimate stress limit, yield strength, friction factor etc.) are random and can differ from the abovementioned.

The experimental test included the investigation of mounting bolts with the following diameter dimensions:

\[ d_{01} = 30.8 \text{ mm}, \quad d_{02} = 31.1 \text{ mm}, \quad d_{03} = 31.4 \text{ mm}, \quad d_{04} = 32 \text{ mm}; \]

thread length \( L = 30 \text{ mm}. \)

Alloy tool steel 5XHM was used to manufacture the experimental draw die. Heat treatment schedule:

\- Oil hardening at \( t = 870^\circ\text{C}; \)
\- High-temperature tempering at \( t = 450^\circ\text{C}. \)

Hardness of the material amounted to 50 HRC after the heat treatment.

Draw die dimensions: operating area angle \( \alpha = 14^\circ \), operating area length – 13 mm; bearing die diameter \( d = 30 \text{ mm}, \) bearing die length – 3 mm; die entrance and closing die lengths were 3 mm and 2 mm respectively, die entrance and closing die angles were \( 45^\circ\).

Stress machine (press) IP – 1000-type (limit load – 1000 kN) was used for experimental testing. The experimental values of minimum drawing force for initial mounting bolt plastic strain are:

\[ F_1 = 100 \text{ kN} \quad \text{(corresponds to } d_{01}); \quad F_2 = 113 \text{ kN} \quad \text{(corresponds to } d_{02}); \]
\[ F_3 = 135 \text{ kN} \quad \text{(corresponds to } d_{03}); \quad F_4 = 175 \text{ kN} \quad \text{(corresponds to } d_{04}). \]

3. Results and discussion

Steady (continuous) drawing is possible in condition (at the exit of draw die)

\[ \sigma_1 \leq \frac{\sigma_s}{K} \]

(8)

where, \( K \) – factor of safety.

Factor of safety \( K \) is equal to 1.4 – 2.0 in practice. The value of \( \sigma_s \) is used with regard to actual cold hardening. The definition of mounting bolt diameter was carried out by solving the equilibrium equations. Whereby, the size of mounting bolt diameter comes to \( d_0 = 32.88 \text{ mm} \) (with mount load up
to $F = 200$ kN, draw die diameter $d_1 = 30$ mm, friction coefficient $f = 0.3$, strength of material $\sigma_s = 700$ MPa and operating area angle $\alpha = 14^\circ$.

The combination of experimental and theoretical (plotted by the equation (7)) $F(d)$ relationship graphs are presented in Figure 5.

According to theoretical and experimental data, we propose the design of the deforming element with following original sizes (before drawing): original mounting bolt diameter $d_z = 32.4$ mm; original deforming area length $L_z = 93$ mm. The length of the deforming element after drawing can be calculated on the assumption of constant volume condition and amounts to 108.5 mm.

![Figure 5](image)

**Figure 5.** Experimental and theoretical values of drawing force.

4. Conclusion
The article presents the design of a cab support of forest machine to provide additional energy absorption effect in case of rollover. The protective effect is attained by plastic strain of a cab support element (mounting bolt drawing) to improve ROPS safety abilities and increase job safety. Theoretical research of drawing process was carried out; original sizes of the mounting bolt were calculated. The results of experimental research allowed specifying the properties of mount deforming elements.

Acknowledgments
This work was supported by the Strategic Development Program of Petrozavodsk State University (2017-2022).

References
[1] International Organization for Standardization ISO 8082-1:2009 Self-propelled machinery for forestry – Laboratory tests and performance requirements for roll-over protective structures – Part 1: General machines, IDT
[2] Harris J R, Mucino V H and Etherton J R 2000 J. Agric. Saf. Health 6 215–225
[3] Harris J R, Ronaghi M and Snyder K A 1998 Anal. Solutions 2 24-25
[4] Powers J R, Harris J R, Etherton J R, Ronaghi M, Snyder K A, Lutz T J and Newbraugh B H 2001 Injury Prevention 7 (I) 54–58
[5] Etherton J R, Cutlip R G, Hams J R, Ronaghi M, Means K H and Gillkpie A 2002 J. Agric. Safe. Health 8(1) 119-126
[6] Dumitrache P 2014 About the parametric modeling of the protective structure included in a tractor cabin *The Annals of “Dunarea De Jos” University of Galati, Fascicle XIV Mechanical Engineering* **1** 21-26

[7] Pitukhin A V and Skobtsov I G 2015 *Applied Mechanics and Materials* **709** 530-533

[8] Pitukhin A V, Skobtsov I G, Shilovskiy V N and Dobrynina O L 2016 *Applied Mechanics and Materials* **93** 439-443

[9] Myers M L 2004 *Compactor Overturns and Rollover Protective Structures* (Silver Spring MD: The Center to Protect Worker’s Rights) p 16

[10] Myers M L 2002 *J. Agric. Saf. Health* **8(2)** 185–198