INVESTIGATION OF CONTACT STRESSES IN ENGAGEMENT ELEMENTS OF SCREW CONVEYOR SAFETY CLUTCH

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Abstract. In the article there is presented the design of a safety clutch for a screw conveyor with time-separated modes of slipping and axial displacement of the screw for automatic restoration of the operating condition in the event of overload**.** A calculation has been made using contact stresses in the engagement elements of the safety device, the level of which is the determining indicator that affects the wear rate of the contact surfaces. Using the SolidWorks Premium software diagrams of the distribution of contact stresses during operation of the safety clutch were constructed, by the results of which it was established that the analytical dependencies, presented in the article, can be used in the engineering design of various standard sizes of this protective mechanism.

Keywords: screw conveyor, screw (auger), clutch, groove, semi-clutch, contact stresses.

Introduction

Technological processes for the production of agricultural products involve a large number of labour-intensive loading, unloading and transport operations. In this case, they most often carry out reloading and transportation of grain crops, bulk and lump materials, granulated mineral fertilizers, etc.

Screw conveyors in agro-industrial production are widely used during the transportation of agricultural materials [1-3]. However, when transporting such bulk materials, jamming of the screw working body may occur due to the presence of a gap between the rotation surface of the screw and the inner surface of the conveyor casing [4]. To restore the operability of the conveyor, it is necessary to move the jammed edge of the screw away from contact with the material in the axial direction, and subsequently, after removing the overload, the drive elements must ensure the restoration of the original position of the working element for further transportation of the material to the unloading zone [5; 6].

Many different types of safety mechanisms used for mechanical technological machines, such as screw conveyors, ensure more or less the requirements, placed on them regarding the functioning process when critical loads occur on the conveyor screws [7-9]. However, the traditional safety ball and cam clutches, when activated, cause significant shock dynamic loads, and this subsequently causes destruction of both the couplings and the conveyor drives [10-12]. Friction safety clutches have low operating accuracy and do not allow reliable protection of the working parts and machine drives [13-15].

Therefore, among the current tasks of agricultural production it is important to develop new designs of ball safety clutches with closed circular profiles of the engagement elements to ensure reliable protection of the working bodies and conveyor drives, with insignificant shock loads during the operation of the safety mechanisms.

The purpose of the study is to improve the operating efficiency of a screw conveyor by developing and justifying the optimal parameters of a safety clutch.

Materials and methods

To eliminate jamming of the working body of a screw conveyor when transmitting torque, it is proposed to use a safety clutch with time-separated modes of slipping and axial displacement of the screw to restore the operating condition of the conveyor. Its design diagram is presented in Fig. 1. When transmitting torque, the cylindrical pins 4, which are spherical on the working side and located in the through-passing axial holes in the antifriction bushings 5 of the driven semi-clutch 3, are engaged with the holes 7 of the driving semi-clutch 1, which ensures rotation of the safety clutch and the screw body. Along the diameter of the pins and holes, on both sides of the holes, there are inclined working and reverse grooves made on the end surface of the driving semi-clutch, and the angle of inclination of the working groove is significantly less than the angle of inclination of the reverse groove. When overload occurs, the driven semi-clutch 3 stops; at the same time the driving semi-clutch 1, continues to rotate, which causes the cylindrical pins 4 to come out of the engagement with holes 7, and, at the same time, the cylindrical pins 4 move along the circular working groove 11. As a result of rotation of the driving semi-clutch 1, the cylindrical pins 4 move to their previous position, and the original state is restored. In this case, the cylindrical pins 4 move along the circular reverse groove 12.

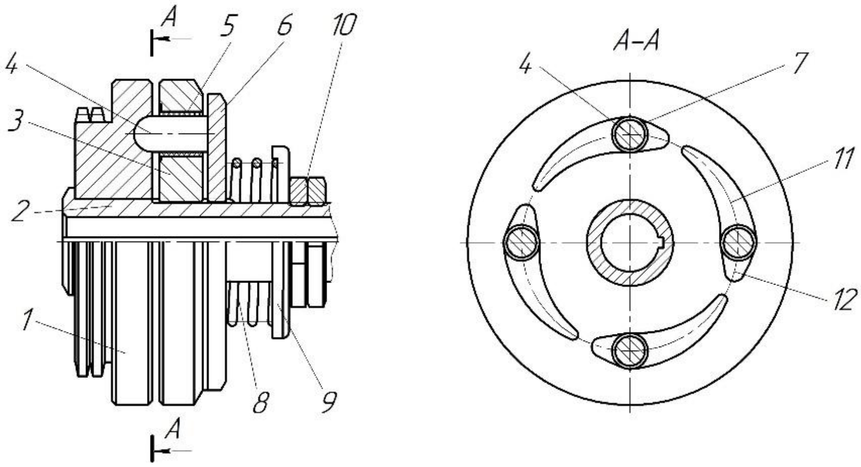

Fig. 1. **Design diagram of the safety clutch**:1 – driving semi-clutch; 2 – hub; 3 – driven semi-clutch; 4 – cylindrical pins;5 – antifriction bushings; 6 – pressure disk; 7 – holes; 8 – spring; 9 – washer; 10 – nut; 11 – circular working grooves; 12 – circular reverse grooves

For the analysis of the change in the value of the contact stresses in the elements of the engagement of the safety clutch from its rotation of the semi-clutch a theoretical calculation was made. On the basis of Hertz's contact problem, a relationship between the magnitude of the contact stresses and the design parameters of the working surfaces of the screw conveyor clutch was established. Determination of the magnitude of the contact stresses depending on the rotation of the semi-clutches of this safety mechanism was carried out at two stages of its operation: when the cylindrical rounded pins have engaged with the holes of the driving semi-clutches, and when they move along the circular grooves.

In a general case, when two bodies interact, the contact plane has the form of an ellipse with semi-axes:

$$a=\alpha\sqrt[3]{\frac{3F(1\text{-}\mu^2)}{E\left(\frac{1}{\rho_1}+\frac{1}{\rho_1'}+\frac{1}{\rho_2}+\frac{1}{\rho_2'}\right)}}\ ;\ b=\beta\sqrt[3]{\frac{3F(1\text{-}\mu^2)}{E\left(\frac{1}{\rho_1}+\frac{1}{\rho_1'}+\frac{1}{\rho_2}+\frac{1}{\rho_2'}\right)}}, \tag{1}$$

where F – contact force of the engagement elements relative to the magnitude of the torque;
μ – Poisson's ratio;
E – elasticity modulus;
α and β – coefficients;
ρ_1 and ρ'_1 – radii of curvature of the first body (cylindrical pin); and are the radii of curvature of the second body (hole or groove).

Radii of curvature are considered to be positive if their centres are inside the body.The value of the coefficients α and β are determined as functions of the auxiliary angle ψ, which is determined according to the formula:

$$\cos\psi=\frac{\pm\sqrt{\left(\frac{1}{\rho_1}-\frac{1}{\rho_1'}\right)^2+\left(\frac{1}{\rho_2}-\frac{1}{\rho_2'}\right)^2+2\left(\frac{1}{\rho_1}-\frac{1}{\rho_1'}\right)\cdot\left(\frac{1}{\rho_2}-\frac{1}{\rho_2'}\right)\cos 2\theta'}}{\frac{1}{\rho_1}+\frac{1}{\rho_1'}+\frac{1}{\rho_2}+\frac{1}{\rho_2'}}, \tag{2}$$

where θ' – angle between the main planes of the curvature of the bodies in which ρ_1 and ρ'_1 lie. In this case, angle θ' is 90°.

The maximum compressive stresses in the centre of the contact plane are determined by the formula:

$$\sigma_{\max} = 1.5\frac{F}{\pi ab}. \tag{3}$$

Thus, using a variative method, it is possible, to select such geometric parameters of the end surface of the driven semi-clutch half of the safety clutch that, with at the given contact forces and appropriate materials, make it possible to ensure a condition under which the maximum stresses do not exceed the permissible ones.

To determine the maximum normal stresses arising in the engagement of the socket and a rounded cylindrical finger, the concave profile of the socket (Fig. 2) is considered; it has a curved surface in two planes. That is, we obtain $\rho_1 = \rho'_1 = r$ – the radius of curvature of a cylindrical pin, $\rho_2 = R_x$ – the conditional radius of curvature in the plane of action of the force F, $\rho'_2 = R$ – the radius of the hole.

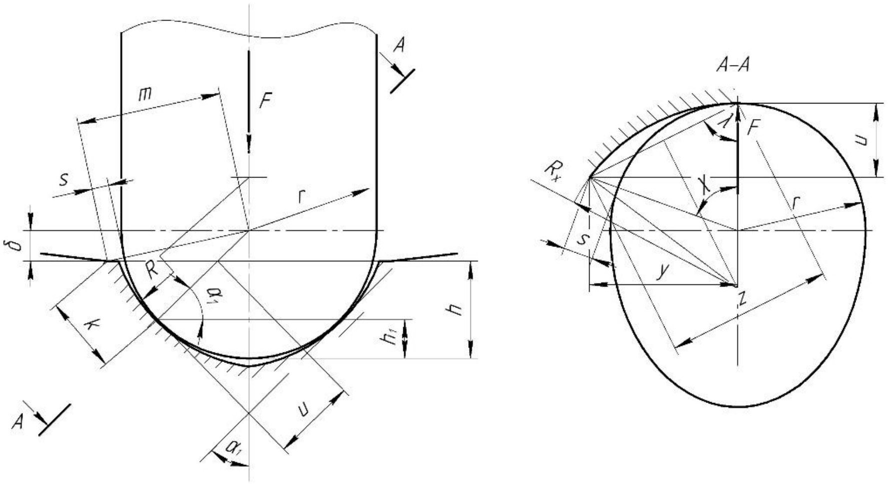

Fig. 2. **Calculation scheme for determining the radius, describing the radius R_x of the hole in the plane of action of the force F**

Therefore, equality (2) will look like this:

$$\cos\psi = \frac{r\left(R - R_x\right)}{2R_xR - r\left(R + R_x\right)}. \tag{4}$$

To determine the values of $\cos\psi$, it is necessary to establish the value of the radius R_x, which describes the conditional radius of the hole in the plane of action of the force F. To do this, using the calculation scheme shown in Fig. 2, after appropriate substitutions and mathematical transformations, the value of the conditional radius of curvature R_x, which determines the radius of the hole with a concave surface in the plane of action of the normal force F, is calculated from the equation:

$$R_x = \frac{\sqrt{\left((r+s)\sin\left(\arccos\left(\dfrac{R\delta}{(r+s)\left((R-r)\sin\alpha_1 - h + r\right)}\right)\right)\right)^2 + \left(r - \dfrac{R\delta}{(R-r)\sin\alpha_1 - h + r}\right)^2}}{2\cos\left(\operatorname{arctg}\left(\dfrac{(r+s)\sin\left(\arccos\left(\dfrac{R\delta}{(r+s)\left((R-r)\sin\alpha_1 - h + r\right)}\right)\right)}{r - \dfrac{R\delta}{(R-r)\sin\alpha_1 - h + r}}\right)\right)}, \tag{5}$$

where α_1 – inclination angle of the hole;
h – depth of the hole;

δ – gap between the semi-clutches– gap size, determined by the formula:
s – size of the gap, determined by the formula:

$$s=\sqrt{r^2+\left(\frac{\frac{r}{R}\left(\left(R-r\right)\sin\alpha_1-h+r\right)-\delta}{\cos\left(\arcsin\left(\frac{\left(R-r\right)\sin\alpha_1-h+r}{R}\right)\right)}\right)^2}-r\,. \tag{6}$$

Using formulas (1), (5) and (6), the tabular values for determination α and β, equation (3) for determination of the maximum contact stresses in the centre of the contact plane in the hole-finger engagement will have the following form:

$$\sigma_H=\frac{1{,}5F}{\pi\alpha\beta\sqrt[3]{\left(\frac{3FrR_x(1-\mu^2)}{E\left(R_x-r\right)}\right)^2}}\le\left[\sigma\right]_H\,, \tag{7}$$

where F – contact force of the engagement elements relative to the magnitude of the torque;
μ – Poisson's ratio;
E – elasticity modulus;
α, β – coefficients, determined from the tables of material strength.

Let us consider the engagement of a groove and a pin; the contact plane in this case has a flat surface in one plane and a spherical concave surface in the other. That is, we obtain $\rho_1 = \rho'_1 = r$, $\rho_2 = r$ – the radius of the groove, $\rho'_{2\to\infty}$.

Consequently, equality (2) will be equal to $\cos\psi = 1$. Therefore, using tabular data, it was determined that the coefficients α and β are equal to 1; $\psi = 90^o$, therefore, it was established that the coefficients are equal to 1.

So, equality (3) for determination of the contact stresses in this case will have the form:

$$\sigma_H=\frac{1{,}5F}{\pi\sqrt[3]{\left(\frac{3Fr(1-\mu^2)}{E}\right)^2}}\le\left[\sigma\right]_H\,. \tag{8}$$

In order to establish the adequacy of theoretical calculations, determination of the values of the contact stresses in the engagement elements of the safety clutch of the screw conveyor, using the Solid Works Premium 2012 software, there was designed a model of the safety clutch. In addition, the following parameter values were adopted: the ball radius r = 12 mm; conditional radius of curvature of the hole R_x = 13.2 mm; elasticity modulus $E = 2\cdot10^{11}$ Pa; Poisson's ratio μ = 0.3. During the investigations the forces in the contact of the engagement elements F gradually changed with a certain step from 100 to 1000 N, which made it possible to determine the contact stresses σ_H and obtain a diagram of their distribution.

Results and discussion

To determine how the maximum stresses, arising in the contact zones as a result of interaction of the hole and the pin, will depend on the conditional radius R_x of curvature as well as the force of contact interaction. Fig. 3 shows graphical dependences of the conditional radius R_x of curvature on the radius of curvature of the finger r, the radius R of the hole and the size of the gap δ between the semi-clutches, i.e. $R_x = f(r, R, \delta)$.

It has been established that an increase in the radius R entails an increase in the value R_x by 9%; with an increase in r from 10 to 14 mm – R_x increases by 30%; as the gap δ increases from 0.5 to 4.5 mm, – R_x decreases by 5%. Consequently, the parameter r has the greatest influence on the change in the value R_x. When analysing the stresses σ_H, the value R_x was changed in the R_x = 11.0…15.4 mm range.

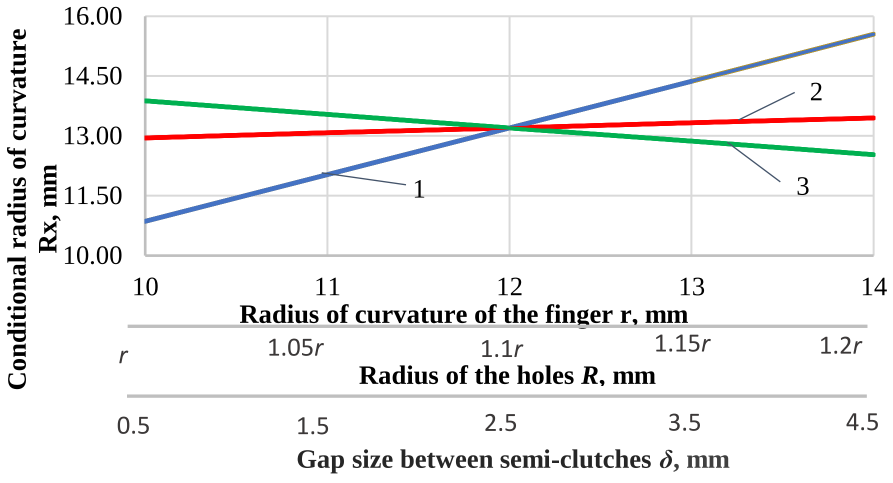

Fig. 3. **Graphs of changes in the conditional radius R_x of curvature depending on:** 1 – radius of urvature of the finger r; 2 – radius of the holes R; 3 – gap size δ between semi-clutches

To determine the maximum contact stresses in the centre of the engagement contact plane the hole – pin, in Fig. 4 there are shown graphs of changes of the stresses σ_H versus the conventional radius of curvature R_x.

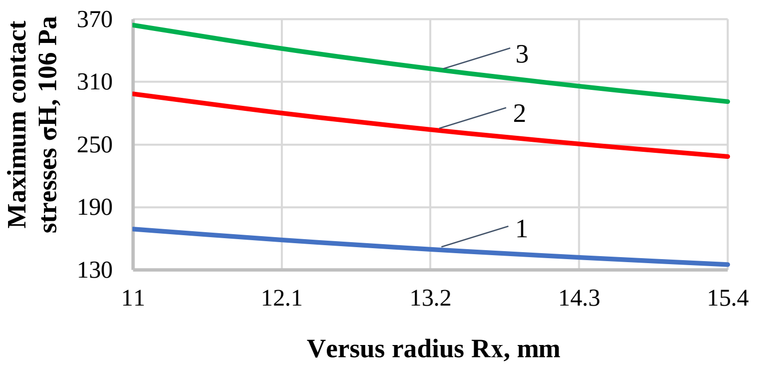

Fig. 4. **Graphs of changes in the maximum contact stresses σ_H versus the radius R_x:** $1 - F = 100$ N; $2 - F = 550$ N; $3 - F = 1000$ N

It has been established that an increase in the value R_x leads to a decrease in the stresses σ_H by 20...25%. With an increase in the strength F 10 times, σ_H increases by 52...57%.

To determine the maximum contact stresses in the centre of the contact plane the groove-pin, in Fig. 5 there are shown graphs of changes of the stresses σ_H depending on the radius of curvature of the pin r.

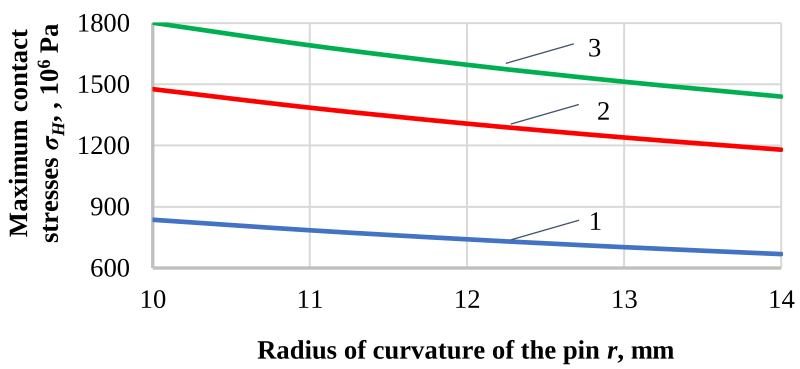

Fig. 5. **Graphs of changes in maximum contact stresses σ_H versus the radius r of curvature of the pin:** $1 - F = 100$ N; $2 - F = 550$ N; $3 - F = 1000$ N

It has been established that an increase in the value of r leads to a decrease in the stress σ_H by 20...25%. With an increase in the strength F 10 times, it increases by 53...56%.

In order to establish the adequacy of theoretical calculations for determination of the values of contact stresses in the engagement elements of the safety clutch of the screw conveyor using the SolidWorks Premium 2012 software, diagrams of the distribution of the normal contact stresses in the semi-clutches with a contact force of the engagement elements F = 100 N are presented, which are shown in Fig. 6.

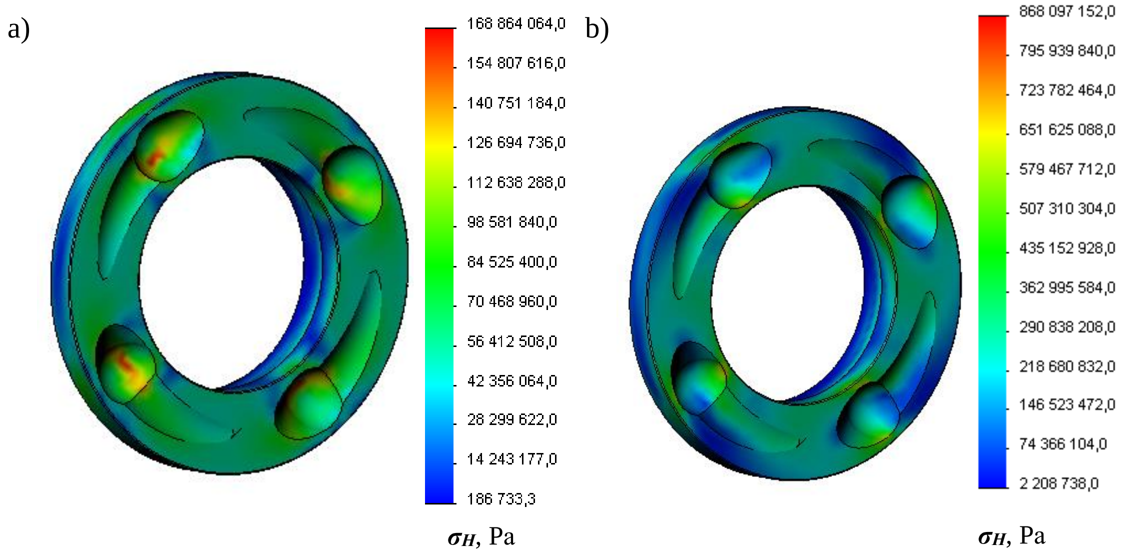

Fig. 6. **Diagrams of distribution of the contact stresses in the engagements of the safety clutch:** a – pin-hole; b – pin-groove

In Fig. 7 there are presented the results of theoretical (solid line) and experimental (dashed line) investigations.

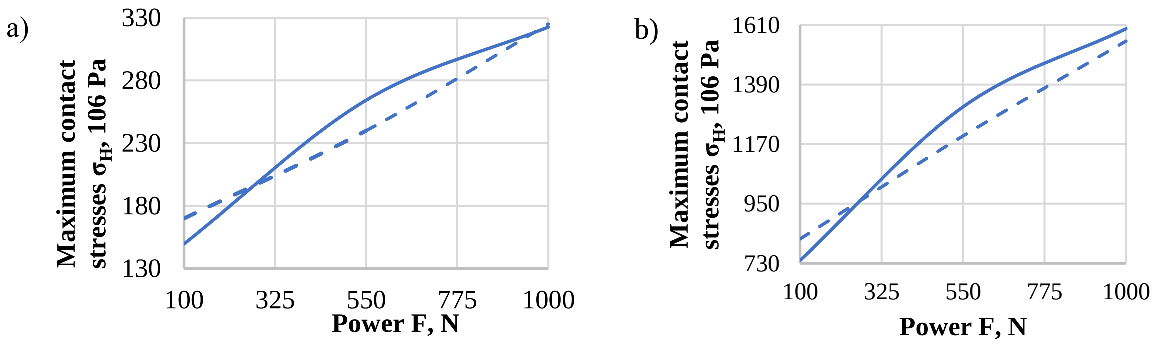

Fig. 7. **Graph of changes in contact stresses depending on the force of the contact elements of engagements:** a – pin-hole; b – pin-groove

From the analysis of the graphical dependencies it is evident that the error between the results of the investigations is in the range of 11.2...18.6% for the pin-hole engagement and in the range of 9.7...17.8% for the pin-groove engagement.

Thus, based on the results of comparative studies, it was established that the analytical dependencies (7) and (8) previously derived to determine the contact stresses arising in the engagement elements of the safety clutch, adequately reflect the real operating processes of the developed protective mechanism. Therefore, using these dependencies, as well as the tabular data of the characteristics of materials regarding the permissible maximum stresses on the contact plane $[\sigma]_H$, it is possible to select a material for the semi-clutch of the safety coupling of the screw conveyor, which will ensure the strength condition for contact stresses.

Conclusions

1. Based on the patent search and analysis of the existing design and technological schemes of the protective devices for screw conveyors, a new design of a screw conveyor safety clutch with time-separated modes of slipping and axial displacement of the screw for automatic restoration of the operating condition in the event of an overload is proposed.
2. Based on a performed theoretical calculation, graphs of changes in the contact stresses that arise in the engagement elements of the safety clutch are constructed, the level of which is the determining indicator that affects the intensity of wear of the contact surfaces.
3. It has been established that an increase in the value of the conditional radius of curvature R_x in the hole-pin engagement leads to a decrease in the stress σ_H by 20...25%. With an increase in the force F 10 times, σ_H increases by 52...57%. It has also been established that an increase in the radius of curvature of the finger r in the groove-pin engagement leads to a decrease in the stress σ_H by 20...25%. With an increase in the force F 10 times, it increases by 53...56%.
4. Using the SolidWorks Premium 2012 software, diagrams of the distribution of contact stresses during operation of the safety clutch were constructed, based on the results of which it was established that the analytical dependencies, presented in the article, can be applied in the engineering design of various standard sizes of this protective mechanism.

Author contributions

Conceptualization, V.B.; methodology, V.B. and O.T.; validation, V.B., O.T,, A.Ab. and M.T.; formal analysis, V.B and A.Al.; investigation, V.B., O.T., V.K. and M.T.; data curation, A.Ab., V.B. and M.T.; writing – original draft preparation, V.B., A.R.; writing – review and editing, V.B., A.Ab. and A.R.; visualization, V.K., M.T.; project administration, V.B.; funding acquisition, A.R. All authors have read and agreed to the published version of the manuscript.

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