

JUSTIFICATION OF THE DESIGN OF WHEELED HARVESTERS FOR TIMBER HARVESTING ON SLOPES

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Abstract: This article addresses issues related to the stability of harvesters against overturning during timber harvesting on slopes. The objective of the research is to provide a theoretical foundation for the maximum slope angle at which a harvester can operate. The most common harvester design configurations are examined. The position of the machine's center of mass is justified. Mathematical relationships are presented, enabling the assessment of harvester stability during the machine design stage. Recommendations for improving the machine's design and operation are provided.

KEYWORDS: harvester, timber harvesting on slopes, stability against overturning, limiting slope angle, center of mass, technological equipment.

1 Introduction

One of the pressing challenges in the logging industry is increasing the level of mechanization for slope harvesting operations. Currently, serially produced machinery, due to its design, is often not suited for work on slopes with gradients exceeding 15–20° and requires the integration of additional technological equipment, such as traction winches, outriggers, and others, into its design [1–6]. Among all types of logging machinery existing today, the harvester is the most promising from the perspective of operation on slopes. A distinctive feature of this machine is its low center of mass, which, as experimental studies show, allows for timber harvesting on slopes with gradients of approximately 25° [7–11].

Despite the significance of this problem for the forest industry, research dedicated to the operation of harvesters on slopes is often experimental [7–11] or focuses on the environmental aspects of the interaction between forest machine wheels or tracks with soils [12–15]. Meanwhile, theoretical research that would justify the possibility of operating harvesters on slopes from the standpoint of stability against overturning receives insufficient attention.

The aim of the research is to develop mathematical relationships that enable a comprehensive approach to determining harvester stability against overturning during operation on slopes at the machine design stage, as well as to identify the influence of design and operational parameters on harvester stability.

2 Materials and Methods

The study consists of five sequential sections. A brief overview of the focus and objectives of each section follows.

Section 3.1 presents a review of the most common harvester design variants that can serve as a baseline for justifying the configuration of a harvester adapted for timber harvesting on slopes. Methods of information gathering and analysis were employed in writing this section. The information presented is based on literature sources [1–17]. The objective of this section is to justify the harvester design for slope operations, which will be used in subsequent sections.

Section 3.2 provides the rationale for the center of mass position of the harvester design proposed in Section 3.1. This section is based on methods of theoretical mechanics, along with information gathering and analysis. A calculation schematic was developed and a coordinate system introduced to determine the harvester's center of mass position (Fig. 2). To solve this task, the harvester was divided into its main structural components, each with its own center of mass relative to the selected coordinate system. The center of mass positions for individual harvester components were determined by searching information in relevant databases [16, 17] and on the websites of forestry machinery and component manufacturers [18–21]. The objective of this section is to determine the center of mass position of the considered harvester design for conducting stability calculations against overturning.

Section 3.3 presents an analysis of the harvester's longitudinal stability during steady-state motion up a slope (Fig. 3). The section was written using methods of theoretical mechanics and forest machine design theory. The objective of this section is to develop a mathematical model for determining the limiting slope angle, followed by calculating its value for the harvester design under consideration. The calculation is performed for different values of main arm outreach and manipulator swing angle.

Section 3.4 presents an analysis of the harvester's lateral stability, accounting for the manipulator swing angle (Fig. 4). The section was written using methods of theoretical mechanics and forest machine design theory. The objective of this section is to develop a mathematical model for determining the influence of the manipulator swing angle on the limiting slope angle value. The calculation results are presented in the section as a corresponding graphical relationship (Fig. 6). The section also presents the relationship between the harvester's lateral stability and track width (Fig. 7).

Section 3.5 presents an analysis of the harvester's stability during motion at an angle to the slope (Fig. 8). The section was written using methods of theoretical mechanics and forest machine design theory. The objective of this section is to develop a mathematical model for determining the limiting slope angle value as a function of the harvester's orientation angle. The calculation results are presented in the section as a corresponding graphical relationship (Fig. 9).

3 Results and Discussion

3.1 Review of wheeled harvester layout variants

Let us consider the most common harvester layout configurations (Fig. 1) and identify the most promising ones from the perspective of potential operation on slopes.

Based on the presented variants, several fundamental differences in harvester layouts can be distinguished.

1. Based on the type of operator cabin placement, harvesters can be classified into three groups: the cabin is rigidly fixed to the front frame module, together with the power unit compartment (Fig. 1, *c, e*); the cabin is mounted on the slewing platform, occupying a central position in the machine's structure (Fig. 1, *a, d*); the cabin is installed on a platform equipped with a leveling system in the central part of the machine (Fig. 1, *b*). The third layout example is the most preferable for operation on slopes. This choice is based on the fact that, due to the

presence of the leveling system, the operator can maintain the cabin in a horizontal position relative to the slope, which ensures the necessary field of view of the work zone and, consequently, positively impacts machine productivity. Furthermore, tilting the operator's cabin towards the slope contributes to increased machine stability by shifting its center of mass. This solution is especially effective when not only the cabin but also the processing equipment is mounted on the slewing platform [23].

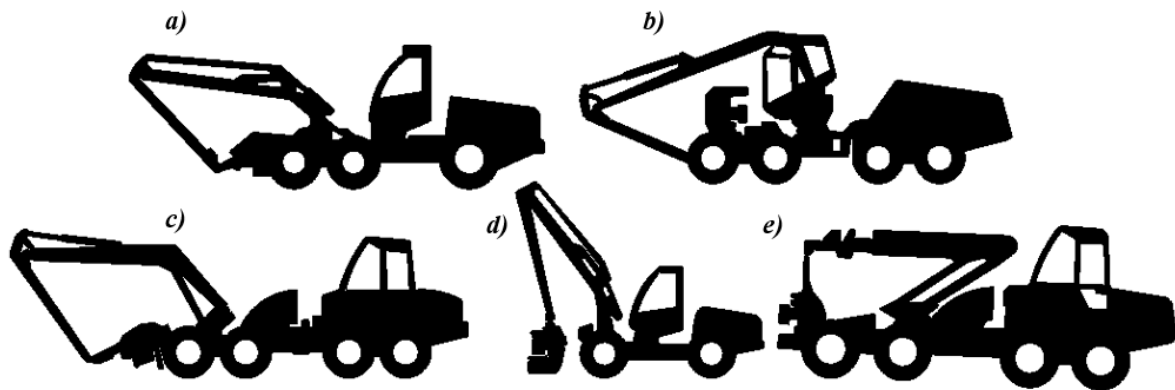


Fig. 1 Layout variants of serially produced wheeled harvesters

2. Based on the wheel formula, harvesters can be classified into three groups: 4x4 (Fig. 1, *d*), 6x6 (Fig. 1, *a*) and 8x8 (Fig. 1, *b*, *c*, *e*). When operating on a slope, a very important factor is the area of the support surface and the traction of the running gear with the soil. Therefore, a more preferable option is a harvester with an 8x8 wheel formula. To increase the traction properties of the running gear with the soil, removable tracks can be installed on the wheels of the tandem axles.

3. Based on the methods of placing the processing equipment, harvesters can be classified into two groups: processing equipment is separately placed on the rear frame module of the machine (Fig. 1, *a*, *c*, *d*, *e*) and processing equipment is mounted together with the operator's cabin (Fig. 1, *b*). From the perspective of machine operation on slopes, both options are acceptable. It should be noted that the manipulator column should be equipped with a tilt function, which allows for shifting the center of mass of the processing equipment towards the slope and contributes to increasing machine stability.

4. Based on the layout-kinematic schemes of the manipulators used, harvesters can be classified into two groups: articulated manipulators with a telescopic extension (Fig. 1, *a*, *b*, *c*, *d*) and telescopic-type manipulators (Fig. 1, *e*). The choice of manipulator type largely depends on the mass of trees being processed at the logging site and the type of cutting operations. For smaller tree masses or during thinning operations, preference can be given to a more compact telescopic-type manipulator. For harvesting larger trees (which is generally typical of trees growing on slopes), preference should be given to an articulated manipulator, as it is capable of withstanding higher bending and torsional moments [22].

As a basic harvester layout adapted for timber harvesting on slopes, preference should be given to a harvester equipped with a platform leveling system. Placement of the operator's cabin is recommended in the central part of the machine, with the processing equipment on the rear frame module. The manipulator column should be equipped with a tilt function. The machine's manipulator should be articulated with a telescopic extension. The harvester is recommended to be equipped with a running gear with an 8x8 wheel formula. The harvester layout proposed based on this review is considered in the subsequent subsections.

3.2 Justification of the harvester's center of mass position

To assess both the longitudinal and lateral stability of a harvester, the most significant factor is the position of the machine's center of mass. Let us justify an approximate position of the harvester's center of mass that can be used at the machine design stage. The center of mass of the harvester can be determined by knowing the positions of the centers of mass of its main structural elements (Fig. 2).

1. Center of mass of the machine's running gear (*A*). The running gear consists of two tandem axles, eight wheels, and an articulated frame. The mass of one tandem axle, for example, from NAF, depending on the model, is approximately 1500...3000 kg [18], and its center of mass is located at the center of the unit, at the differential mounting point. The mass of a harvester wheel, depending on its diameter, can be 200...800 kg [17, 19-21]. The mass of the articulated frame, depending on the harvester model, can be approximately 2000...3000 kg [19-21], and at the design stage, the position of the frame's center of mass can be assumed to be at its geometric center.

2. Center of mass of the operator's cabin (*B*). The mass of a harvester operator's cabin can be approximately 1000...2000 kg [19-21]. At the design stage, the position of the cabin's center of mass can be placed in its lower part, at a distance equal to 1/3 of the cabin's height.

3. Center of mass of the power unit compartment (*C*). The power unit compartment consists of an internal combustion engine, hydraulic pumps, tanks for fuel and hydraulic fluid, and the metal structure of the compartment itself. The total mass of the power unit compartment can be approximately 3000...5000 kg [16, 17, 19-21]. At the design stage, the position of the power unit compartment's center of mass can be assumed to be at its geometric center.

4. Center of mass of the manipulator main arm (*D*). The mass of a harvester manipulator main arm is approximately 800...900 kg, and at the design stage, the position of its center of mass can be placed at a distance equal to 1/3 of the arm's length [17, 22].

5. Center of mass of the manipulator secondary arm with telescopic extension (*E*). The mass of a harvester manipulator secondary arm is approximately 700...800 kg, and at the design stage, the position of its center of mass can be placed at a distance equal to 1/3 of the arm's length [17, 22].

6. Center of mass of the harvester head (*F*). The mass of a harvester head can vary significantly depending on the model and manufacturer and is approximately 800...2600 kg [16, 17, 19-21]. At the design stage, the position of the harvester head's center of mass can be assumed to be at its geometric center.

7. Center of mass of the manipulator column (*G*). The mass of the manipulator column, depending on the model, is approximately 300...500 kg [17, 19-22]. At the design stage, the position of the manipulator column's center of mass can be assumed to be in the central part of the column, in the area where the hydraulic cylinders of the rack-and-pinion slewing mechanism are mounted.

To determine the center of mass of the harvester (*H*), Fig. 2 shows the approximate positions of the centers of mass of all main structural elements of the machine. An arbitrary origin point of the coordinate system was set at a distance from the machine, and perpendiculars to the axes were drawn from the points corresponding to the centers of mass of each structural element of the harvester. As a result, coordinates X_i , Y_i and Z_i corresponding to a specific structural element were obtained. The coordinates of the harvester's center of mass position were calculated using the following dependencies:

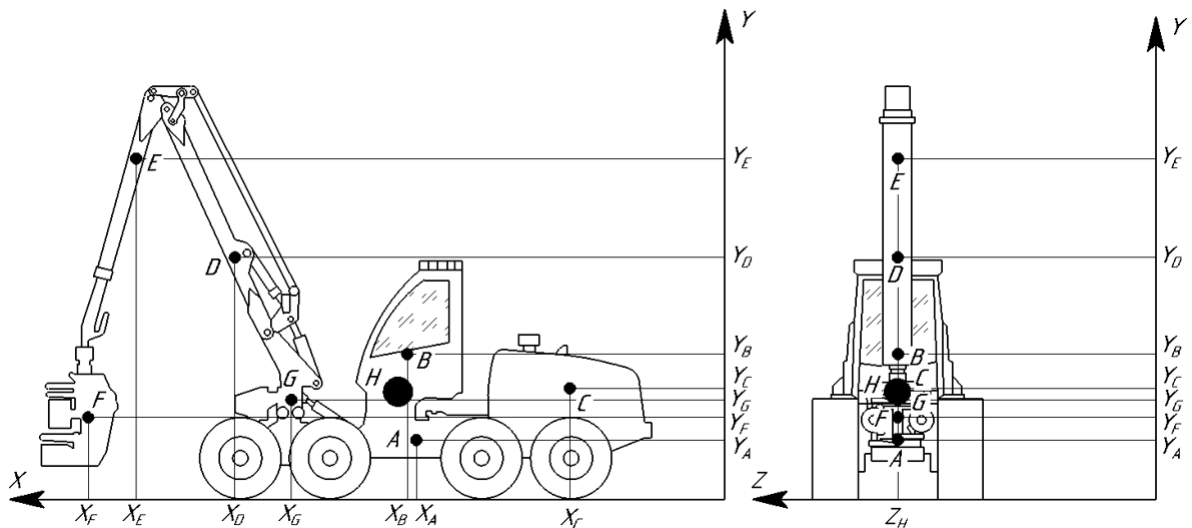


Fig. 2 Calculated scheme for determining the position of the harvester's center of mass

$$\begin{aligned}
 Y_H &= \frac{m_A \cdot Y_A + m_B \cdot Y_B + m_C \cdot Y_C + m_D \cdot Y_D + m_E \cdot Y_E + m_F \cdot Y_F + m_G \cdot Y_G}{m_H}; \\
 X_H &= \frac{m_A \cdot X_A + m_B \cdot X_B + m_C \cdot X_C + m_D \cdot X_D + m_E \cdot X_E + m_F \cdot X_F + m_G \cdot X_G}{m_H}; \\
 Z_H &= \frac{m_A \cdot Z_A + m_B \cdot Z_B + m_C \cdot Z_C + m_D \cdot Z_D + m_E \cdot Z_E + m_F \cdot Z_F + m_G \cdot Z_G}{m_H},
 \end{aligned} \quad (1)$$

where Y_H, X_H, Z_H – coordinates of the harvester's center of mass position, m; m_H – mass of the harvester, kg.

Table 1 presents the adopted values of the coordinates and masses of the main structural elements, as well as the calculation results of the harvester's center of mass coordinates.

Table 1. Initial data and calculation results of the harvester's center of mass position coordinates

Conventional designation of the structural element	Mass, kg	Y, m	X, m	Z, m
A	6600	0,92	4,76	3,98
B	1500	2,24	4,90	3,98
C	4100	1,71	2,39	3,98
D	850	3,73	7,55	3,98
E	750	5,26	9,08	3,98
F	1500	1,26	9,82	3,98
G	400	1,72	6,68	3,98
H	15700	1,66	5,04	3,98

The calculation results for determining the location of the center of mass of the considered harvester layout showed that the machine has a relatively low-positioned center of mass, located in the central part of the machine, in the area of the operator's cabin slewing platform. This position of the center of mass contributes to increased harvester stability during operation on slopes and makes it a more preferable option compared to tracked harvesters and feller-bunchers, whose center of mass is significantly higher.

It should be noted that the position of the harvester's center of mass is not a constant value and largely depends on the manipulator's outreach and rotation angle. In the subsequent analysis

of harvester stability on slopes, the center of mass position will be considered at minimum and maximum outreach, as well as with the manipulator rotated at an angle of 90°.

3.3 Analysis of the longitudinal stability of the harvester

Let us consider the harvester's stability against overturning in the case of its steady longitudinal motion up a slope (Fig. 3).

Overturning of the machine will occur under the condition that the reaction force acting on the front support of the harvester becomes zero ($R_A = 0$). The moment equation relative to the overturning point B is as follows:

$$R_A \cdot l + G_H l_H \cos \alpha - G_H h_H \sin \alpha - M_r = 0, \quad (2)$$

where l – distance between the support reactions acting on the front and rear parts of the harvester, m, G_H – gravitational force of the harvester, N, l_H – longitudinal distance between the harvester's center of mass and the overturning point, m, h_H – vertical distance between the harvester's center of mass and the overturning point, m, α – slope angle, deg., M_r – torque on the harvester wheels, N·m.

The torque on the harvester wheels can be determined by the formula:

$$M_r = P_0 \cdot r_d, \quad (3)$$

where P_0 – traction force, N, r_d – dynamic wheel radius, m.

The traction force can be determined by the formula:

$$P_0 = G_H f \cos \alpha - G_H \sin \alpha, \quad (4)$$

where f – coefficient accounting for the motion resistance of the harvester.

Transforming expression (2) with consideration of dependencies (3) and (4), we obtain an expression for determining the limiting slope angle:

$$\alpha = \tan^{-1} \frac{l_H - f r_d}{h_H + r_d}. \quad (5)$$

Table 2 presents the initial data and the results of calculating the limiting value of the slope angle during the harvester's longitudinal movement up the slope.

The stability analysis of the harvester was conducted for four calculation cases:

- manipulator is positioned along the slope (at an 0° angle) at minimum outreach $L_{min} = 3,5$ m;
- manipulator is positioned along the slope (at an 0° angle) at maximum outreach $L_{min} = 9,0$ m;
- manipulator positioned across the slope (at an 90° angle) at minimum outreach $L_{min} = 3,5$ m;
- manipulator positioned across the slope (at an 90° angle) at maximum outreach $L_{min} = 9,0$ m.

Analysis of the calculation results shows that the harvester possesses sufficiently high longitudinal stability against overturning. On average, the limiting slope angle on which the machine can operate is approximately 37°. The most critical case is when the machine moves with the manipulator at minimum outreach, positioned across the slope; here, the limiting angle is constrained to a value of 30°.

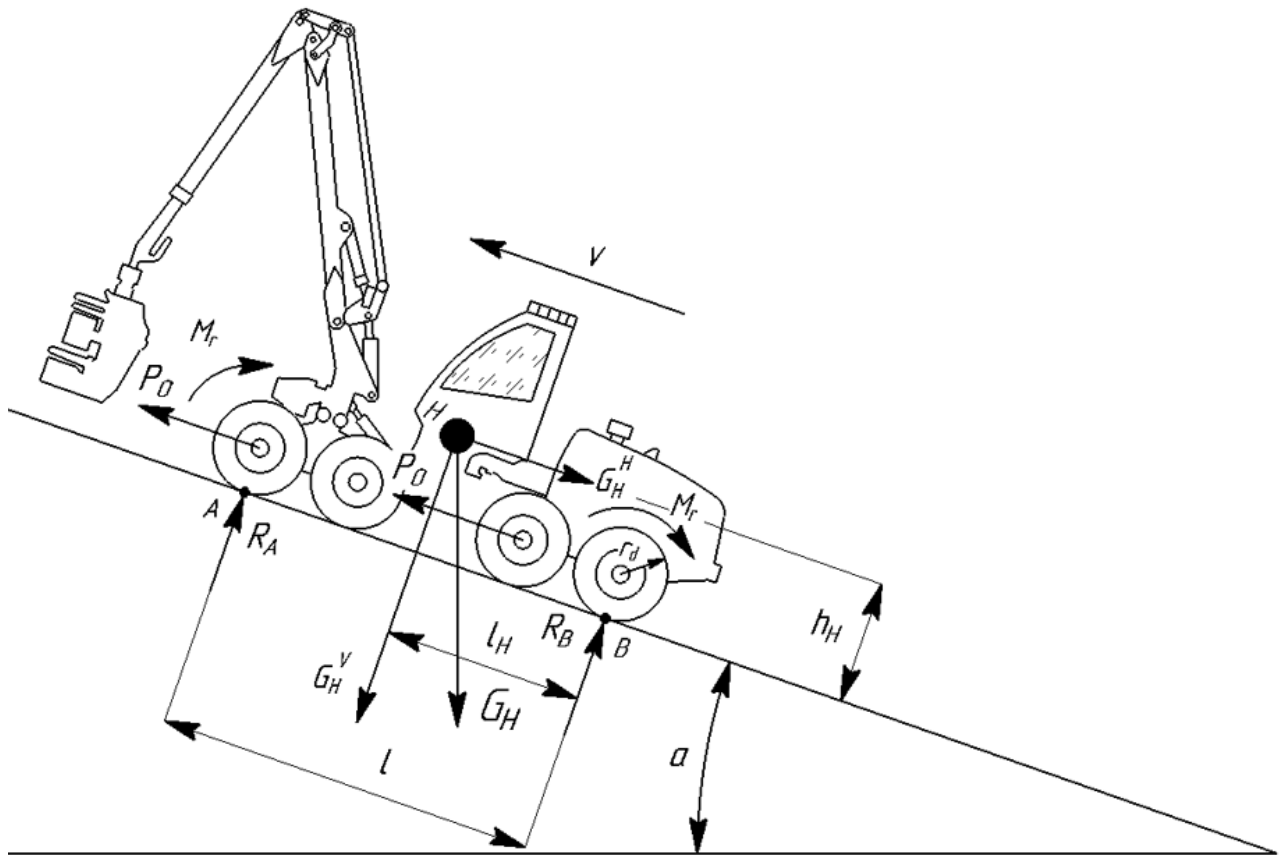


Fig. 3 Calculated scheme for determining the longitudinal stability of a harvester

Table 2. Initial data and calculation results of the limiting slope angle value during the machine's longitudinal movement up a slope, considering the machine manipulator's outreach and rotation angle.

Manipulator rotation angle, deg.	Manipulator outreach, m	l_H , m	h_H , m	f	r_d , m	α , deg.
0	3,5	1,98	1,66	0,12	0,75	38
0	9,0	2,74	1,49	0,12	0,75	49
90	3,5	1,52	1,66	0,12	0,75	30
90	9,0	1,55	1,49	0,12	0,75	33

3.4 Analysis of the lateral stability of the harvester

Let us analyze the stability of the harvester against overturning during its tilt. Consider two calculation schemes. In the first case, the harvester's manipulator is positioned in front of the machine (Fig. 4, *a*); in the second case, the manipulator is positioned at a 90° angle and directed towards the base of the slope (Fig. 4, *b*).

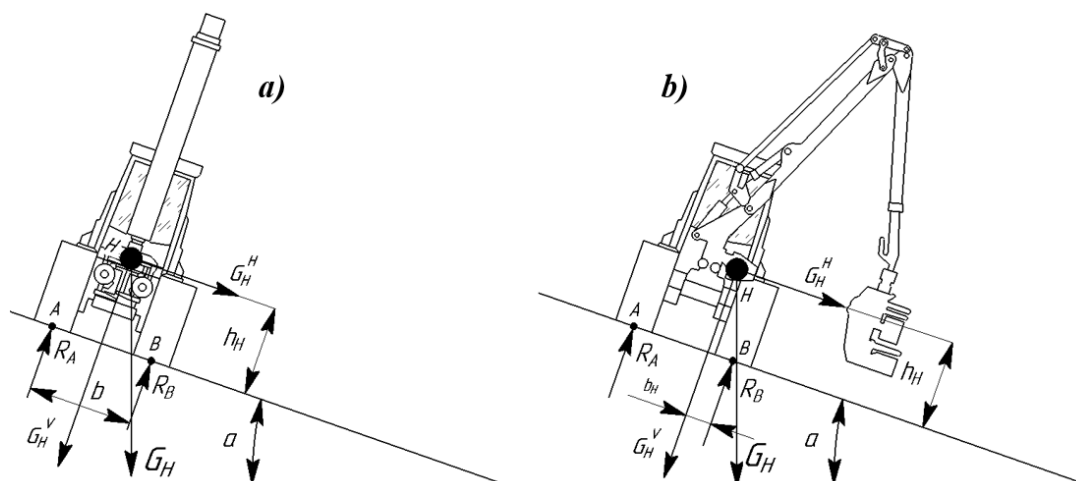


Fig. 4 Calculated scheme for determining the lateral stability of the harvester: *a* – manipulator positioned in front of the machine, at an angle of 0°; *b* – manipulator rotated 90° towards the base of the slope.

Overturning of the machine will occur under the condition that the reaction force becomes zero ($R_A = 0$). The moment equation relative to the overturning point B is as follows:

$$R_A \cdot b - G_H b_H \cos \alpha + G_H h_H \sin \alpha = 0, \quad (6)$$

where b – the track width, m; b_H – the distance between the harvester's center of mass and the overturning point along the length, m

Transforming expression (6), we obtain the dependency for determining the limiting slope angle during harvester tilt:

$$\alpha = \tan^{-1} \frac{b_H}{h_H}. \quad (7)$$

It should be noted that the lateral stability of a harvester on slopes significantly depends on the rotation angle of the processing equipment, as this causes a shift in the machine's center of mass. Let us derive a mathematical model describing the lateral stability of a harvester depending on the manipulator's rotation angle. When forming the model, we will adopt the following assumption: the manipulator's outreach remains constant during its rotation.

The shift in the harvester's center of mass occurs due to the rotation of the moving elements of the processing equipment—the manipulator main arm and secondary arm, and the harvester head. We introduce an auxiliary coordinate system $X'Y'Z'$, passing through the initial point of the harvester's center of mass position (Fig. 5).

The rotation of each moving element of the processing equipment around the Y' axis is carried out along the following radii: main arm – X'_D , secondary arm – X'_E , and harvester head – X'_F . The projections of these radii onto the Z' axis are determined by the following dependencies:

$$Z'_D = X'_D \cdot \sin \varphi; \quad Z'_E = X'_E \cdot \sin \varphi; \quad Z'_F = X'_F \cdot \sin \varphi, \quad (8)$$

where φ – manipulator rotation angle, deg.

The position of the center of mass of the processing equipment relative to the original XYZ coordinate system is determined by the expression:

$$Z_J = Z_0 - \frac{(m_D \cdot X'_D) \sin \varphi + (m_E \cdot X'_E) \sin \varphi + (m_F \cdot X'_F) \sin \varphi}{m_D + m_E + m_F}, \quad (9)$$

where Z_0 – initial coordinate of the manipulator's center of mass position, m.

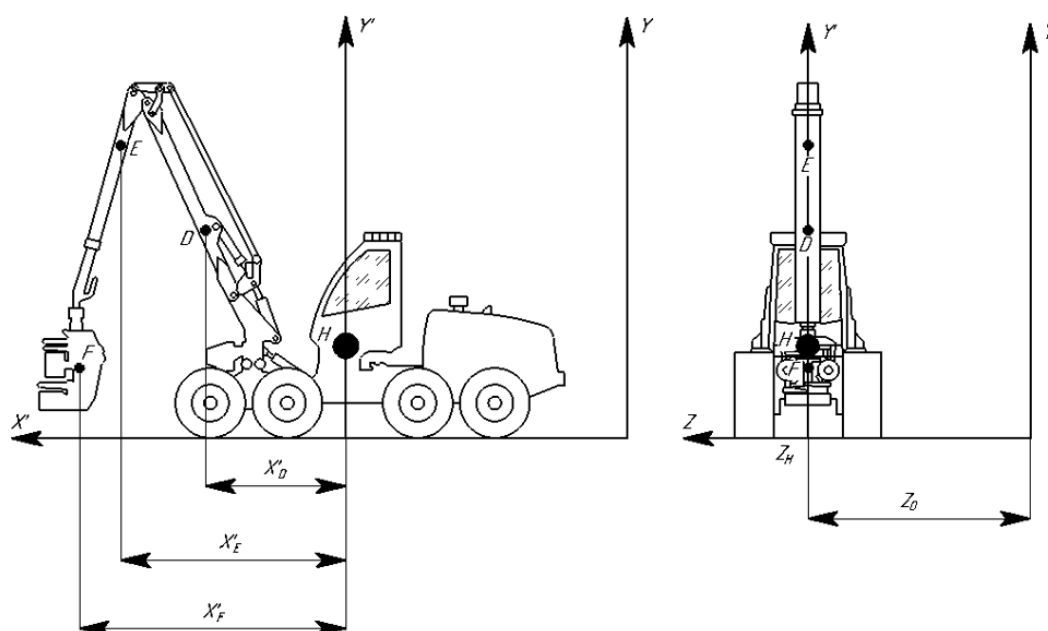


Fig. 5 Calculated scheme for determining the shift in the harvester's center of mass position depending on the manipulator rotation angle

Then, the position of the harvester's center of mass along the Z axis, accounting for the manipulator's rotation angle, will be determined by the dependency:

$$Z_{H(\varphi)} = \frac{Z_J \cdot (m_D + m_E + m_F) + Z_0 \cdot (m_A + m_B + m_C + m_G)}{m_H}. \quad (10)$$

Using dependencies (7), (9), and (10), a graph of the relationship between the limiting slope angle α and the manipulator rotation angle φ was plotted (Fig. 6).

Analysis of the results shows that the lateral stability of the harvester is significantly lower than its longitudinal stability and largely depends on the manipulator's rotation angle. For example, when the manipulator is rotated 90° , machine overturning will occur on a slope with an angle of approximately 5° . Improving the lateral stability of the harvester can be achieved, for instance, by increasing the track width and using tires with a larger width.

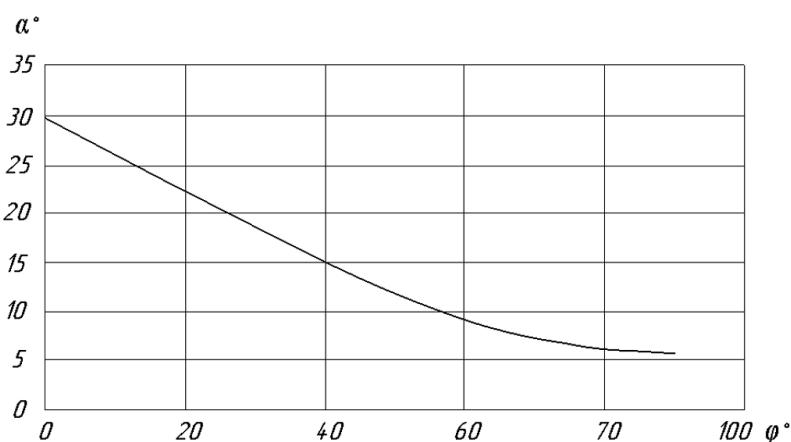


Fig. 6 Graph of the relationship between the limiting slope angle α and the manipulator rotation angle φ

Figure 7 shows a graph of the relationship between the limiting slope angle and the harvester's track width ($\varphi = 90^\circ$).

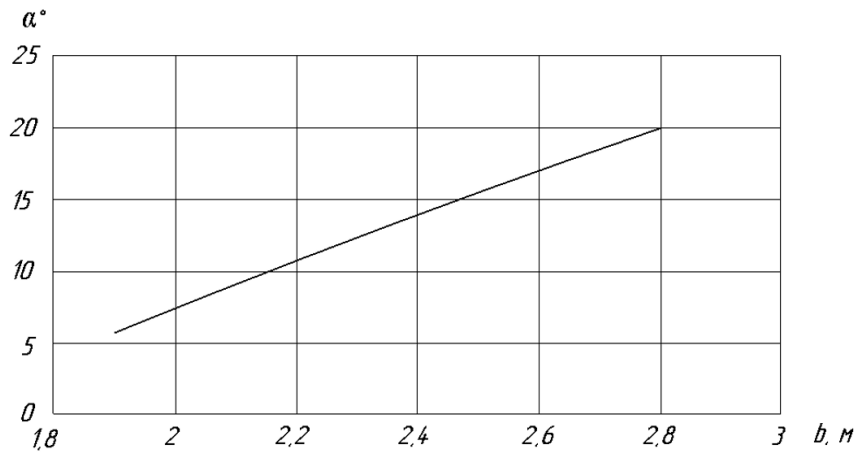


Fig. 7 Graph of the relationship between the limiting slope angle α and the track width b

As can be seen from the graph, increasing the track width significantly contributes to improving lateral stability. For example, by increasing the harvester's track width to 2,8 m, the machine can operate on slopes with an angle of approximately 20 degrees. In general, every increase in track width by 0,5 m contributes to an average increase in the limiting slope angle of 1,6 degrees.

3.5 Analysis of harvester stability during movement at an angle to the slope

In practice, the movement of a harvester on a slope cannot be unambiguously classified as purely longitudinal or lateral. Typically, the machine moves at a certain angle β relative to the slope [24] (Fig. 8).

In the considered case, the limiting slope angle on which the harvester can operate during its steady motion is characterized by the expression:

$$\alpha = \tan^{-1} \frac{\sqrt{l_1^2 + l_2^2} \cos \left[90 - \left(\beta + \sin^{-1} \frac{l_1}{\sqrt{l_1^2 + l_2^2}} \right) \right] - f r_d}{h_H + r_d} \quad (11)$$

where l_1 – longitudinal distance between the harvester's center of mass and the overturning point, m; l_2 – lateral distance between the harvester's center of mass and the overturning point, m; β – angle between the direction of machine movement and the base of the slope, deg.

Figure 9 presents a graph of the relationship between the limiting slope angle and the harvester's heading angle (processing equipment positioned in front of the machine ($\varphi = 0^\circ$), manipulator at minimum outreach).

Analysis of the results shows that moving the harvester at an angle to the slope contributes to increased stability against overturning. For example, when the harvester moves strictly across the slope ($\beta = 0^\circ$), the limiting slope angle is constrained to a value just below 20° . However, when the machine turns to an angle $\beta = 20^\circ$, the harvester is already capable of operating on slopes with an inclination of approximately 30° . The most preferable option is moving the harvester at an angle $\beta = 65^\circ$; in this case, the maximum theoretical value of the limiting slope angle is achieved, which is approximately 40° . A further increase in the machine's heading angle leads to some decrease in stability, down to a limiting slope angle of 38° , which corresponds to the case of the harvester's longitudinal movement up the slope (Table 2).

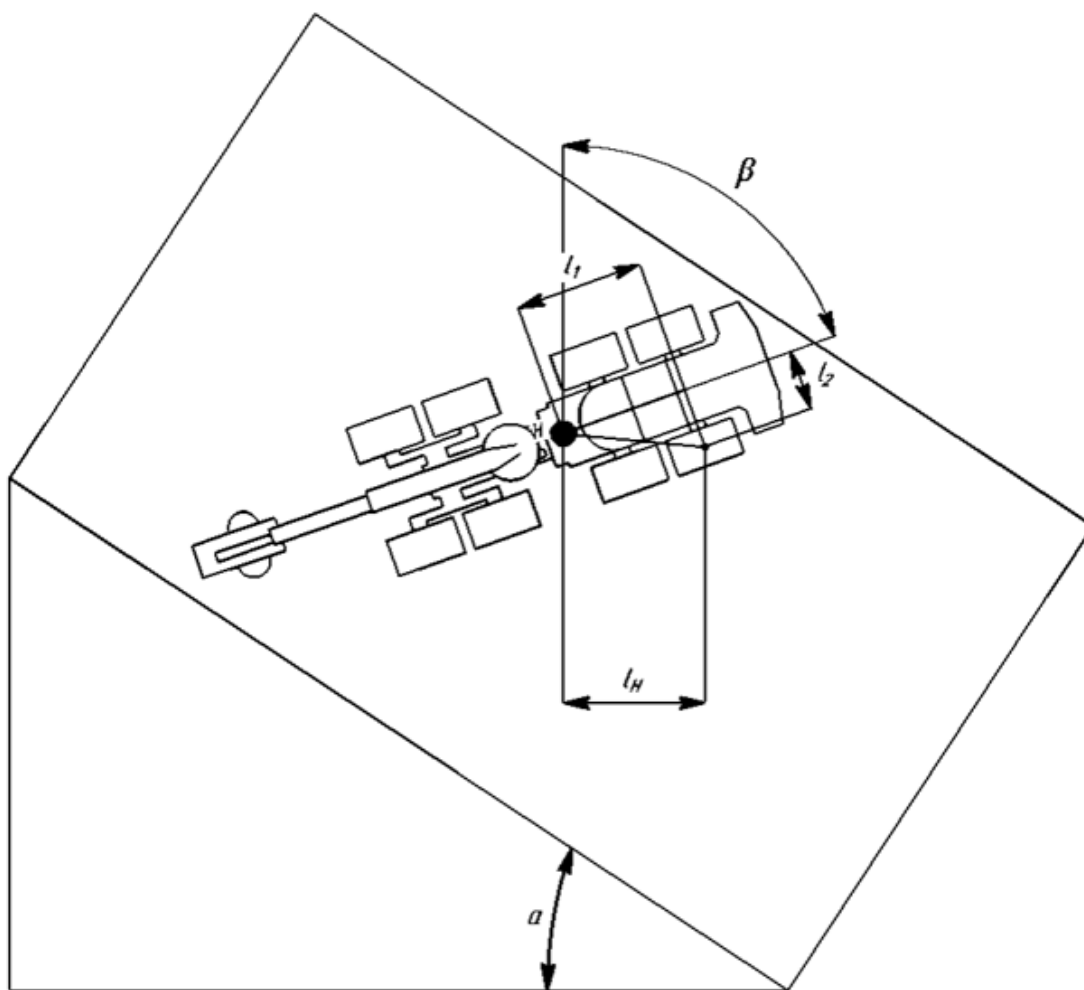


Fig. 8 Calculated scheme for determining the stability of a harvester during its movement at an angle to the slope

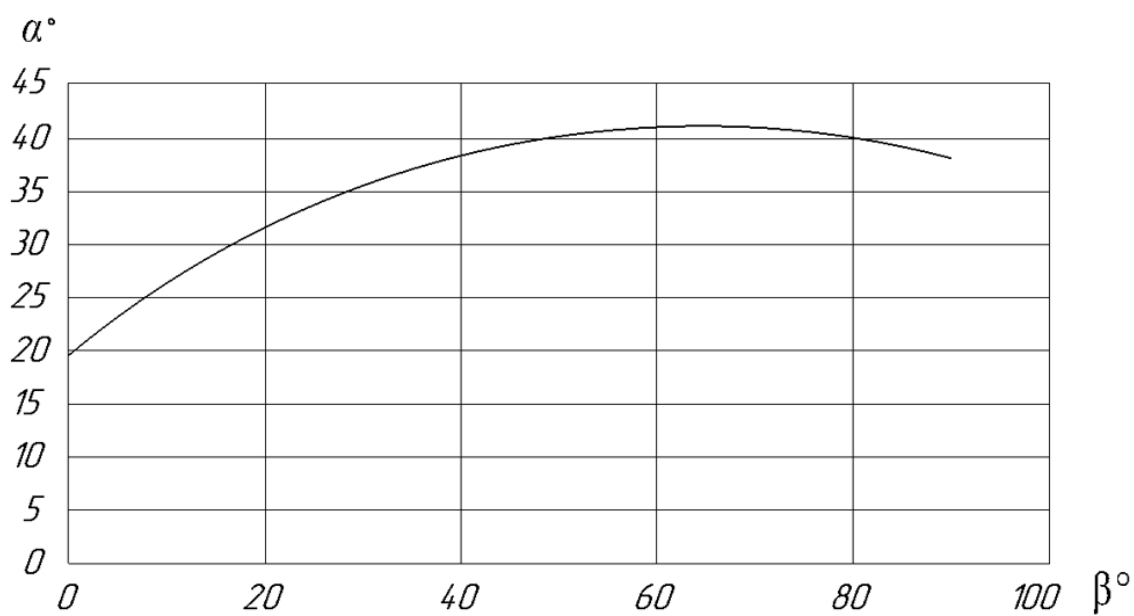


Fig. 9 Graph of the relationship between the limiting slope angle α and the harvester's heading angle β

CONCLUSION

The analysis of harvester stability during operation on slopes presented in the article demonstrates that the currently recommended value for the limiting slope angle is quite conservative. The results of the analysis indicate that, provided operators adhere to certain conditions during harvester operation and given specific machine design features, timber harvesting can be safely conducted on slopes with angles of approximately 35°. The mathematical models presented in the study will facilitate a comprehensive assessment of harvester stability against overturning during both longitudinal and lateral machine movement on a slope, as well as during movement at an angle to the slope. To enhance harvester stability during slope operation, it is recommended to equip machines with manipulators featuring limited outreach and a restricted rotation angle (approximately 45°). In the undercarriage design, preference should be given to tandem axles with a width of at least 2.5 m. During harvester operation, strictly lateral movement across the slope should be avoided, favoring movement at an angle instead.

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