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SOLVING THE PROBLEM OF REDUCING THE INFLUENCE OF LATERAL FORCE ON THE SAW BLADE STABILITY

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The accuracy of the thickness of lumber is one of the most important indicators of sawing. It is inextricably linked with saw stabilization in the plane of the greatest stiffness. The aim of the study is to eliminate the influence of lateral force on the saw blade and the thickness of the resulting lumber. The issue of eliminating the influence of lateral force in frame sawing and sawing on band saws belongs to the constructive decision in combination with the analytical one according to classical methods. Therefore, the most important issue in the development of a new machine is to identify the presence of a huge range of frequencies of natural and parametric oscillations of saw blades. Previously, these frequencies could not be analytically found to the full extent and, respectively, the tuning out the machine operating frequencies of the possible oscillation frequencies of saw blades could not be carried out. Due to the complexity and the science intensity of the problem solving, it is not conceivable without modern numerical methods of calculation. Among them are the finite element method, modern software of NX and ANSYS, as well as other original programs. One of such methods, which allow to reduce the influence of lateral force, is determination of stability of the plane form of bending by the Euler's method. The technical solution presented by a fundamentally new saw block with a circular translational motion of the blades reduce dramatically the impact of lateral force on the accuracy of sawing in conjunction with a number of other advantages. At the same time, the issue of ensuring the dynamic stability of the blades both when sawing and at idling speed is solved. It is necessary to point out that with circular translational motion the tooth side cutting edges are under alternating load when scraping over the cut surface. Therefore, the tooth cutting element is a subject of increased strength requirements. The angles of their sharpening were adjusted in order to preserve the integrity of the corners of the teeth tips. The possibility of strengthening the teeth lateral cutting edges of saw blades made of steels of different grades was investigated. The reasons of wear and corrosion, both the elements of the saw module and its operating part (blade teeth) were studied and it was decided to supply the teeth with a hard alloy of the stellite type as the most optimal. However, this provision requires additional targeted field tests. Preliminary calculations showed that the daily productivity of a machine with circular translational motion of the blades (model M2005) in comparison with saw frames increases by 2–4 times; in comparison with band saw equipment of any class by 3–6 times; and in comparison with the circular saw equipment (for small and medium enterprises) by 2–4 times. Analyzing the design scheme and the dynamics of the saw modules, it is possible to find a number of advantages of the multi-saw unit presented as part of the machine. The simplicity and reliability of the design allows us to hope for high functional characteristics. Among those we should highlight the following: increasing the accuracy of sawn products due to the rigidity of short blades, increasing the productivity, improving the quality of treated surfaces, as well as reducing the energy consumption, relatively light weight and dynamic balance of the main units with increased mobility of equipment and the absence of a massive foundation.

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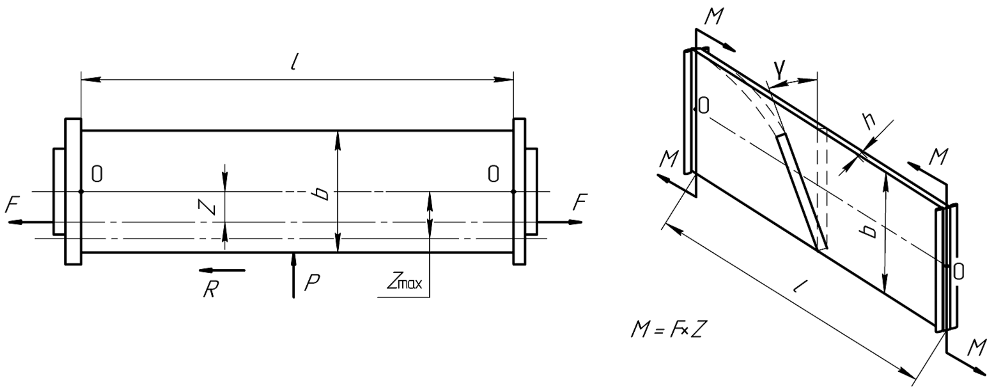
Keywords: saw module, saw block, oscillation frequency, dynamic stability, lateral force, thickness variation.

Introduction

Issues of improving the accuracy in frame sawing and sawing on band saws are considered in the works [8–10] of native and foreign experts. In this case, the required accuracy is inextricably linked with ensuring the stability of the blade (saw) in the plane of its greatest stiffness. The principal design scheme for ensuring the blade stability is presented in Fig. 1.



a *b*

Fig. 1. The design scheme of the blade stabilization: *a* – the scheme of accepted design; *b* – the scheme of stability loss of the blade under the influence of the moment of force bend to the blade's ends (*F* – the force of the blade tension; *R* – the cutting force; *P* – the force opposing the feed motion of the blade along the predetermined path; *Z* – the eccentricity of the blade tension; Z_{max} – the maximum possible eccentricity of tension; *b* – blade width; *h* – blade thickness; γ – the angle of the blade exit from its plane; *M* – the force moment of the eccentric tension; O–O – the median line of the blade

The main reasons of lateral forces during sawing in the plane of the least stiffness may be (Fig. 2): out-of-squareness of the front face of the teeth to the saw blade, asymmetry of teeth broadening, installation inaccuracy of saws in horizontal and vertical positions, movement inaccuracy of saws, inaccuracy in feeding the sawing material, structural features of wood, and others [14, 15, 17, 18].

As a result of errors in preparation, installation, movement of saws and feed of the sawing material between the resultant horizontal force P_r and the feed direction at a speed of V_p there is a possible angle of meeting θ. In the presence of the meeting angle θ, the resultant of the horizontal force P_r is decomposed into the force acting on the saw in the plane of its greatest stiffness ($P = P_r \cos\theta$) and the lateral force ($Q = P_r \sin\theta$).

Lateral force acts on the saw in the plane of its least stiffness and has a decisive influence on the accuracy of sawing.

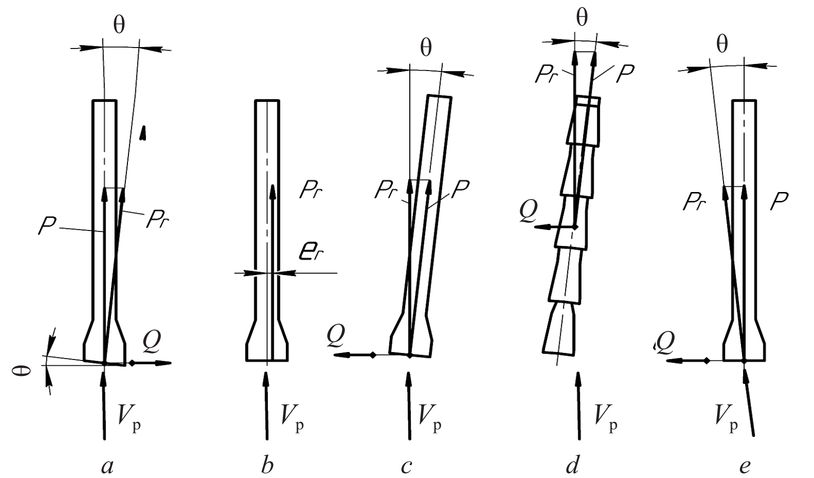


Fig. 2. Reasons for occurrence of lateral forces during sawing: *a* – out-of-squareness of the front face of teeth to the saw blade; *b* – asymmetry of teeth broadening; *c* – installation inaccuracy of the saws in the horizontal position; *d* – installation inaccuracy of the saws in the vertical position and movement inaccuracy of the saws; *e* – inaccuracy in feeding the sawing material

Reducing the influence of lateral force on the accuracy of sawing by classical methods, in frame sawing and band sawing machines, is practically impossible [20]. It should be noted that the stretching line of the saw with a force F is characterized by an eccentricity Z, which is always less than Z_{max}. Calculations using the finite element method (FEM) [4] show that in real operating sawing devices the value of $Z_{max} \leq 0.2b$, otherwise the back edge of the saw loses stability [3], which leads to saw wandering and increased (unallowable) thickness variation of lumber with simultaneous overheating of the tool.

However, high accuracy with simultaneous increase of productivity of sawing by group (battery) method can be achieved under the condition when $Z > 0.5b + h_z$ (where h_z is the height of the tooth of the saw blade). In this case, the lateral force, as perturbing, is beyond the point of application of the resultant force P_r. In other words, the line of the tensile force of the blade must be in front of its teeth, which corresponds to the line of application of force, advancing the cutting blade along a given trajectory at a speed of V_p. Thus, the condition of the working movement of the band saw blades is provided in the negative feedback mode with the guarantee of the condition of its necessary stability, which is the purpose of the study and further calculation.

Since the 1950s some developers and inventors of the sawmill equipment have found solutions of the above issue in the creation of the saw block (Fig. 3) with circular translational motion of saw blades [1, 2, 5–7, 16, 19]. The issue of development, manufacture and testing of the mentioned equipment has quite long history and engineers and woodworkers realizing the prospects of such a design tried to recreate this in practice. As it turned out, this is not a simple design scheme [11–13]. Attempts to create it ended in failure without the necessary scientific support. It should be said that the problem of transmission of circular translational motion in the mechanism of such saw block (Fig. 3) with the help of elements with unidirectional stiffness (saw blades) does not have a strict mathematical solution.

Objects and methods of research

Therefore, the most important, in the development of the saw block of the new machine, was the determination of a significant spectrum of frequencies of natural and parametric oscillations of the saw blades. Previously, these frequencies could not be analytically found to the full extent and, respectively, the tuning out the machine operating frequencies of the possible oscillation frequencies of saw blades could not be carried out. It must be reiterated that, owing to the novelty, complexity of tasks and scientific intensity of the problem, its solution is impossible without applying the most modern numerical calculation methods. Among them – FEM, modern software products of NX and ANSYS, as well as the original software.

One of such methods, which allowed to reduce the influence of lateral force, is determination of stability of the plane form of bending by the Euler’s method.

The solution of this urgent problem, taking into account the above, was carried out by the scientists and designers of the Bauman Moscow State Technical University (Moscow, Russia) in the development of prototypes of a fundamentally new multi-saw block with circular translational motion of the saw blades [3].

Results and discussion

On the schematic design of a multi-saw block.

The presented technical solution (Fig. 3) covered by patents for the invention [5] and successfully passed out the preliminary international patent examination according to the Patent Cooperation Treaty (PCT) system (PCT/RU, 99/00102 from 08.06.2000).

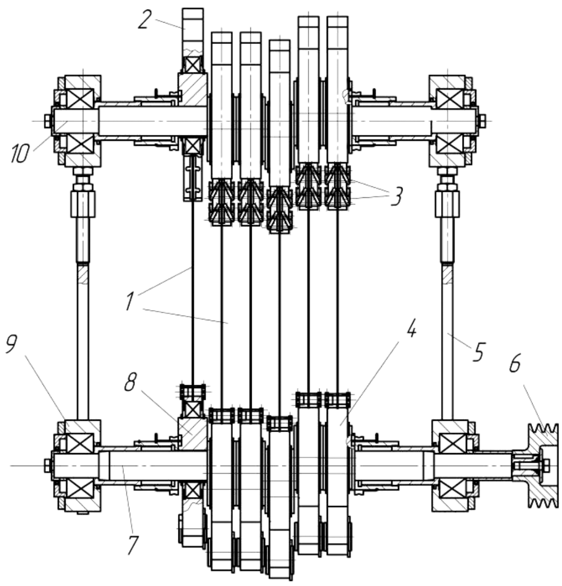
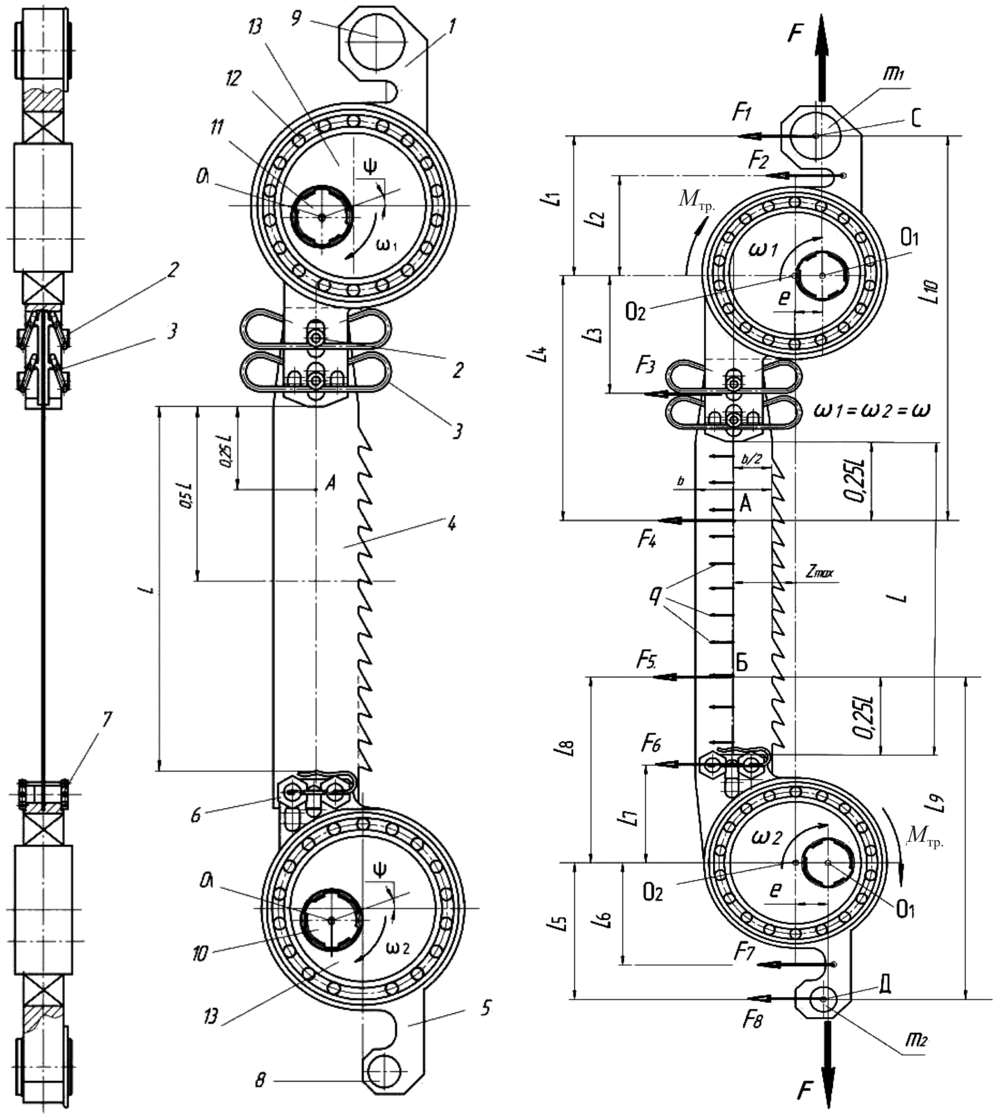


Fig. 3. Saw block with circular translational motion of the blades: *1* – saw blades; *2* – the upper hinge joint with the elements of movable fixation of the saw blade and correction mass; *3* – elastic elements; *4* – the lower hinge joint with the clamping elements of the saw blades and correction mass; *5* – side structure, *6* – pulley of the lower shaft of the saw block; *7* – lower shaft; *8* – top (eccentric); *9* – bearing supports of the shafts; *10* – upper shaft

The central link of the saw block is a dynamically balanced saw module shown in Fig. 4*a*. Dimensional and mass characteristics of all six saw modules of the saw block are identical. Correction masses *8* and *9* placed in the hinge joints *1* and *5* minimize the negative bending moments in the saw blade *4* arising from the rotation of the shafts *10* and *11* of the saw block. Fastening of a blade (blades) is carried out by the fasteners *6* and *7*, and also by the fingers *2* and the elastic elements *3*. In this case, up to 3 and more blades can be simultaneously installed in each saw module. The corresponding angular arrangement of the saw modules, relative to each other on the shafts *10* and *11* of the saw block (Fig. 3, positions *7* and *10*) provides dynamic balancing of inertial forces of the whole saw block. Thus, the force influencing the bearing supports of the shafts *9* and the side structures *5* (Fig. 3) arises only from the weight of the saw block and the tension forces (*F*) of the blades.



a *b*

Fig. 4. The saw module of the saw block with circular translational motion of the blades: *a* – angular position of the saw module with the parameter Ψ (angle of rotation of the tops in relation to the point O_1 of the shafts of the saw module); *b* – the scheme of the acting forces in the saw module when it rotates with a frequency ω; *1* – upper hinge; *2* – finger; *3* – elastic element; *4* – blade; *5* – lower hinge; *6* and *7* – fasteners; *8* – correction mass of the lower hinge joint; *9* – correction mass of the upper hinge joint; *10* – lower shaft; *11* – upper shaft; *12* – bearing; *13* – lower top and upper top (eccentrics) (*q* – distributed load of inertial forces of the blade)

The main role in ensuring the dynamic stability of the blades as part of the saw module is performed by the correction masses of the upper and lower hinges. During the rotational movement of the saw module, the inertial forces of the correction masses F_1 and F_8 (Fig. 4*b*) relative to the points of rotation of the hinge joints O_2 balance the inertial forces F_4 and F_5 arising in the saw blade, which ensures the preservation of the geometry of the saw blade. In this case, the points of the reduced masses of the hinges with the elements of fasteners of the saw blades without correction masses are also at points O_2 (O_2 – the axis of rotation of the bearings *12*.

We consider the kinematics and dynamics of the saw module in order to calculate and ensure the working motion of the blades of saw modules in the negative feedback mode.

The following conditions are accepted. The lower and upper eccentrics (tops) *13* with the O_2 centres are rigidly connected with the lower and upper shafts. The blade is rigidly connected with the fastening nodes of the upper and lower hinges. The length of the blade is represented by the value L with the mass m_p. The distance from the centre O_2 to the eccentrics of shafts (point O_1) is equal to the eccentricity of rotation of the saw modules e, where O_1 is the axis of rotation of the saw block.

When considering the kinematics of the system (saw module), the hypothesis of small deformation is accepted, and the considered saw module is considered absolutely rigid. The lower eccentric *13* rotates around the axis of the lower shaft with a constant angular velocity ω. The movement from the lower shaft is transmitted directly through the blade to the upper shaft *11*, at the angles of rotation of the eccentrics within $90^o < \Psi > 270^o$, i.e. within the angles of working motion when cutting. The saw module performs a so-called circular translational motion, in which all its points move along similar trajectories with equal speeds and accelerations. The inertial forces q of the blade are represented by two components given at the points A and B.

To ensure the required stiffness, the blade is pre-stretched by the force F applied by the shafts with eccentricity e in relation to the centres of rotation of the eccentrics, as well as eccentricity Z_{max}, applied by the eccentrics in relation to the central axis of the blade (Fig. 4*b*). Thus, in the extreme positions of the saw module, during its rotation, the values of the eccentricities of the blade tension Z change. At $\Psi = 0^o$, the total eccentricity $Z = Z_{max} - e$, and at $\Psi = 180^o$, the total eccentricity $Z = Z_{max} + e$.

Analytical solution.

Since the bending moment $M_F = F \times Z$ acts on the blade, the left-most position of the saw module at $\Psi = 180^o$ is adopted, when considering the conditions of maintaining the flat shape of the blade (when the value of M_F is maximum).

In order to determine the limit values of the saw module total eccentricity Z limited by the loss of stability of the blade under the influence of moments of inertia of all its parts and the tension force F, the analysis of the dependence of Z on the factors stabilizing and ensuring the stability of the blade.

In accordance with the condition of maintaining the flat shape of the saw blade, which is under the influence of extracentral stretching [1], the maximum (critical) moment of M_{cr} is determined:

$$M_{cr} = \pi/\ell \, (EJGJ_t)^{1/2}, \tag{1}$$

where E – the modulus of elasticity, with $G = E/2(1 + \mu)$, $\mu = 0.3$; $EJ = E[h^3 b \frac{1}{12}]$ – the stiffness of the blade to bend in a direction perpendicular to the plane of action of external moments; $GJ_t = [E/2(1+\mu)]\ \beta h^3 b$ – torsional stiffness, $\beta = 0.33$ – a function of the ratio b/h, b – the width of the blade, h – the thickness of the blade; $J_t = \beta h^3$.

Then:

$$M_{cr} = \pi/\ell\ \{E[h^3 b/12]\ [E/2(1+\mu)]\beta h^3 b\}^{1/2}. \quad (2)$$

For a blade (plate) with the pinched ends, the preservation of the geometry can be provided under the following condition $2M_{cr} > \Sigma M_{pm}$, where ΣM_{pm} – the sum of the moments of the acting inertial forces in the saw module when it rotates with an angular velocity ω.

Let's consider the interaction of forces and moments for the upper part of the saw module (Fig. 4*b*), limited to half the free length of the blade (0.5L). For the upper part of the saw module the total bending moment of the blade is determined by the dependence:

$$M = M_{bcm} - M_{trv} - M_{bcmp} + M_F, \quad (3)$$

where M_{bcm} – moment of inertia of the upper corrective mass, $M_{bcm} = m_1 e\omega^2 L_1$; m_1 – the value of the upper correction mass, which provides balancing ½ of the upper part of the free length of the saw blade L; e – eccentricity of circular translational motion of the saw module; ω – rotation frequency of the saw module; L_1 – value of the shoulder of the acting force of inertia of the upper correction mass m_1 to the centre of the bearing of the upper hinge joint (point O_2); M_{trv} – the resulting moment of friction from the tension force of the blade and the inertial forces of the upper hinge joint changing in direction during rotational motion, $M_{trv} = (m_{wh} e\omega^2 d_p/2 + F)\, f_{tr}$; m_{wh} – the mass of the upper hinge joint taking into account the masses of the nodes of the movable fastening of the blade and the correction mass m_1; $m_1 = (0.5 m_п e\omega^2 L_4)/L_1$ – from the stabilization condition ½ of the free length of the blade; L_4 – value of the shoulder of the acting inertia force of the upper half of the mass m_p to the centre of the bearing of the upper hinge joint (point O_2); d_p – the diameter of the bearing; F – load on the bearing from the blade tension force; f_{tr} – reduced bearing friction coefficient; M_{bcmp} – moment of inertia forces is given top mass free length of the blade, stabilized by the correction mass m_1, $M_{bcmp} = 0.5 m_p e\omega^2 L_4$; m_p – the mass of the free length of the blade; M_F – the moment of tension of the blade, $M_F = F(Z_{max} + e)$; Z_{max} – maximum eccentricity of the blade tension, which varies in time and depends on the spatial (angular) location of the saw module.

It should be noted that the centre of mass of the upper hinge, taking into account the fastening unit of the movable fixation of the blade, is located in the centre of the bearing of the upper hinge joint (point O_2).

Determination of dimensional and weight parameters of the saw module ensuring the preservation of the geometry of the blade was carried out under the condition of equilibrium of the moments of forces for the upper part of the saw module:

$$M_{cr} = F(Z + e) - M_{bcm} - M_{tr}. \quad (4)$$

At the same time, the values of the friction moments of the upper and lower hinge joints differ due to the difference in the values of their masses. For the purpose of simplification of calculations it is offered to consider in calculations M_{tr} value as

$$M_{tr} = [(m_{wh} - m_{nsh})g^{-1} 4\pi^2 e n^2_{кор}\, d_p/2 + F] f_{tr}, \quad (5)$$

where m_{nsh} – the mass of the lower hinge joint taking into account the masses of the blade attachment units and the correction mass m_2; $n_{кор}$ – acceptable speed (2 500 min^{-1}), previously entered into the formula for determining M_{cr}, $n_{кор} = 41.66$ s^{-1}.

It is necessary to take into account the limitation of rotation speeds of saw modules of 3 000 min^{-1}, since the maximum operating frequency of the selected bearings with grease does not exceed 3000 min^{-1} (bearing No. 92124, GOST 3478–79, durability 15 000–20 000 hours).

Based on the condition:

$$F(Z_{max} + e) - (M_{trn} - M_{trb}) = M_{cr}, \text{ and } (M_{trn} - M_{trb}) = \Delta M_{tr},$$

then,

$$Z_{max} = (M_{cr} - Fe + \Delta M_{tr})/F, \quad (6)$$

which corresponds to:

$$Z_{max} = (M_{cr} + \Delta M_{tr})/F - e.$$

The calculations determining the dependence of the maximum eccentricity of the saw blade tension on the value of its tension force are carried out. The results of the calculations of the adopted design of the saw module are presented in the table.

Values of the maximum eccentricity of the saw blade tension depending on the value of its tension force

L	Z_{max} at various values of F, N						
	500	550	600	700	750	800	900
0.25 m	0.100	0.091	0.081	0.060	0.059	0.054	0.045
0.35 m	0.066	0.056	0.050	0.039	0.034	0,030	0.024

The values m_1 and m_2, as well as L_9 and L_{10} can be variable, and their interdependencies are determined by arithmetic operations from the equilibrium condition of the dynamic system (saw module, Fig. 4*b*), where m_2 – the value of the lower correction mass of stabilization ½ of the free part of the saw blade.

Since $L_8 0.5 m_p = L_5 0.5 m_2$, and $L_4 0.5\, m_p = L_1 0.5 m_p$, it is possible to determine the value of the correcting mass of the lower part of the saw module as:

$$m_2 = m_1 L_1 L_8 (L_5\, L_4)^{-1}.$$

The analysis of the results of the calculations for ensuring the elimination of the influence of the lateral force on the saw blade with the condition of the working movement of the band saw blades in the negative feedback mode showed for this design of saw modules the necessity of the following conditions:

the tension force of the blade is provided within F = 500...800 N at the free length of the blade L = 0.25 m, the width of the supporting structure of the saw b = 80 mm, the thickness S = 1.4...1.5 mm and the height of the teeth h_z =15 mm.

the tension force of the blade with the free length of the blade L = 0.35 m should be in the range of 500...550 N.

the rigidity and stability of the blades allows to use of blades made of band.

Conclusion

As it was shown by preliminary calculations, the increase in the daily productivity of the machine with a circular translational motion of the blades (model M2005) in comparison with saw frames increases by 2–4 times; in comparison with band saw equipment of any class by 3–6 times; in comparison with the circular saw equipment (for small and medium enterprises) by 2–4 times.

Analyzing the design scheme and the dynamics of the saw modules, it is possible to note a number of advantages of the multi-saw unit presented as part of the machine. The simplicity and reliability of the design allows us to hope for high functional characteristics, among which we should highlight the following: increasing the accuracy of sawn products due to the rigidity of short blades, increasing productivity, improving the quality of the treated surfaces, as well as reducing energy consumption, relatively light weight and dynamic balance of the main units with increased mobility of equipment and the absence of a massive foundation.

It should be said that thanks to the inventions of the scientists of BMSTU (Moscow) and the recommendations of scientists of NArFU (Arkhangelsk), a fundamentally new scheme of positioning of the saw frame blades without the direct participation of the machine operator in the program control mode was developed. The design of the saw block, shown in Fig. 3, allows its modernization by introducing electromechanical drives for provision software control (robot mode) of the sawing process in order to increase the utilization rate of business wood and reduce labor costs.

At present, the design documentation of the letter "O" has been corrected and prepared for the serial production of the basic model of a multi-saw machine with a circular translational motion of the blades.

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РЕШЕНИЕ ЗАДАЧИ СНИЖЕНИЯ ВЛИЯНИЯ БОКОВОЙ СИЛЫ НА УСТОЙЧИВОСТЬ ПИЛЬНОГО ПОЛОТНА

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Точность толщины пиломатериалов является одним из самых важных показателей пиления. Она неразрывно связана с обеспечением устойчивости пилы в плоскости ее наибольшей жесткости. Цель исследования – устранение влияния боковой силы на пильное полотно и разнотолщинность получаемого пиломатериала. Классическими методами, при рамном пилении и пилении на ленточнопильных станках, задача устранения влияния боковой силы относится к конструктивному решению в совокупности с аналитическим. Поэтому важно при разработке нового станка выявить наличие огромного спектра частот собственных и параметрических колебаний пильных полотен. Ранее эти частоты не могли быть найдены аналитически в полном объеме и, соответственно, не могла быть осуществлена отстройка рабочих частот станка от возможных частот колебаний пильных полотен. В силу сложности и наукоемкости решение этой проблемы невозможно без применения современных численных методов расчета. К ним относятся: метод конечных элементов, программные продукты NX и ANSYS и другие оригинальные программы. Одним из методов, который позволяет снизить влияние боковой силы, является определение «устойчивости плоской формы изгиба по Л. Эйлеру». Техническое решение, представленное принципиально новым пильным блоком с круговым поступательным движением полотен, значительно снижает влияние боковой силы на точность пиления в совокупности с рядом иных достоинств. Одновременно с этим решается задача обеспечения динамической устойчивости полотен как при пилении, так и в режиме холостого хода. Необходимо отметить, что при круговом поступательном движении боковые режущие кромки зубьев испытывают знакопеременную нагрузку, скобля по поверхности пропила. Поэтому к прочностным характеристикам режущего элемента зуба предъявляются повышенные требования. Для сохранения целостности уголков кончиков зубьев были скорректированы углы их заострения. Проведено исследование возможности упрочнения боковых режущих кромок зубьев пильных полотен, изготовленных из сталей различных марок. В ходе изучения причин износа и коррозии элементов пильного модуля и его рабочей части (зубьев полотна) принято решение о снабжении зубьев твердым сплавом типа «стеллит». Однако это требует проведения дополнительных натурных испытаний. Предварительные расчеты показали, что суточная производительность станка с круговым поступательным движением полотен (модель М2005) по сравнению с лесопильными рамами увеличивается в 2–4 раза, с ленточнопильным оборудованием любого класса – в 3–6 раз, с круглопильным оборудованием (для малых и средних предприятий) – в 2–4 раза. Анализ конструктивной схемы многопильного блока и динамики движения пильных модулей выявил ряд достоинств. Простота и надежность конструкции позволяет надеяться на высокие функциональные характеристики, среди которых следует особо отметить рост производительности оборудования, повышение точности пилопродукции за счет жесткости коротких полотен, улучшение качества обработанных поверхностей пиломатериалов, снижение энергопотребления, относительно малый вес, динамическую сбалансированность основных узлов при повышенной мобильности оборудования и отсутствии массивного фундамента.

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Ключевые слова: пильный модуль, пильный блок, частота колебаний, динамическая устойчивость, боковая сила, разнотолщинность.

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