INNOVATIONS IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN

UNIVERSITY OF FORESTRY

Article

EXPERIMENTAL DETERMINATION OF CUTTING FORCE FOR THE DESIGN OF HEAVY-DUTY CIRCULAR SAW SHAFTS

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ABSTRACT

The paper presents results from experimental measurements of the cutting force for beech (*Fagus sylvatica* L.), performed using a heavy-duty circular sawing machine – *Kallion Konepaja Oy* (Finland). The study was carried out in the production workshop of *Pentop* Ltd., Varshets, Bulgaria. The results were obtained under unfavourable operating conditions of the machine – feed speed v_f ranging from 10 to 70 m·min^{-1} and cutting height H_c ranging from 240 to 320 mm. By measuring the power consumption of the electric motor driving the cutting mechanism during both loaded and idle operation, the cutting power was determined, and from it the average cutting force was calculated. The measured maximum force varied between 225 and 650 N. When the cutting force exceeded 450 N, the required power exceeded the rated power of the motor. The results of the study can be applied in the strength and stiffness analysis of shafts of heavy-duty circular sawing machines.

Keywords: *circular sawing machine; beech (Fagus sylvatica L.); wood cutting; cutting force; cutting power*

INTRODUCTION

Compared to bandsaws, circular saws have more limited use in the primary cutting of large-sized wood materials – logs, beams, prisms, etc. [1,2]. The main reasons for this are their disadvantages: a wider kerf, which reduces quantitative yield, more difficult cutting-tool maintenance, and higher cutting-power requirements, etc. Despite these limitations, when processing logs with smaller diameters, circular saws are often the preferred solution.

In its raw state, wood has approximately one-tenth the density of most structural steels (e.g. C45 steel). This implies that the cutting forces during its processing are significantly lower. However, in circular saw machines for cutting logs, the conditions are significantly more severe, which leads to a significant increase in forces. As is known, the diameter of the saw blade in heavy types of circular saws can exceed 1 500 mm [3], which allows the cutting height to reach more than 600 mm. In combination with the significant thickness of the tool and the possibility of cutting hard or frozen wood, these conditions can substantially increase the cutting forces.

Citation: Atanasov, V.; Kovatchev, G. Experimental Determination of Cutting Force for the Design of Heavy-Duty Circular Saw Shafts. INNO 2026, 15 (1). https://doi.org/10.66211/inno.2026.1.395

The classical approach to shaft design begins with an approximate determination of the minimum diameter and the preparation of a design sketch on which the machine elements are arranged. The sketch shows the distances between the elements and between the supports, as well as the acting radial and axial forces, bending and torsional moments. The shaft is considered as a beam, supported on two hinged supports and loaded with external forces and moments, resolved into two mutually perpendicular planes – horizontal and vertical [4].

With the development of computer technology, the process of designing and calculating shafts has become significantly easier. Moreover, in many cases it is possible to use freely available or free software products, without the need to purchase expensive licenses for specialized engineering software [5].

Most models for the preliminary determination of cutting force are based on the concept of specific cutting resistance, which is usually taken into account by a coefficient depending on the wood species. This approach simplifies the description of the cutting process and inevitably leads to some idealization, which can be the cause of deviations between the calculated and measured values. This approach originates in the classical Soviet literature [6,7]. In recent years, more detailed models for determining cutting force have been developed based on specific cutting resistance, Ernst–Merchant theory and elastoplastic fracture mechanics [8–11]. Other studies have examined the influence of feed speed, cutting tool geometry, wood species and number of teeth on cutting force and power in circular saws, using the classical relationships between power, force, speed and torque [12–14]. In addition, numerous studies have been published on the power and energy parameters of wood processing with other types of machines and technologies [15–18]. However, there is a lack of research in the scientific literature aimed at determining these parameters for heavy-duty circular saw machines for primary wood cutting.

The cutting force is one of the primary loads acting on the shafts of circular saw machines, and its magnitude is of crucial importance for their strength and deformation analysis. This is precisely what determines the purpose of the present study: to experimentally determine the cutting force under adverse operating conditions, with the obtained results serving as a basis for the design of shafts for heavy-duty circular saw machines.

THEORETICAL BACKGROUND

During the interaction between wood and cutting tools, certain forces arise. They are determined by successively decomposing them into radial (F_{Ri}) and tangential (F_{Ci}) components (Figure 1). The total radial (F_R) and tangential (F_C) forces are the sums of the forces acting normal to the tangent to the cutting trajectory and in the direction of the cutting speed vector [19]

$$F_R = F_{Ri} + F_{Ro} - F_{Rc}, \tag{1}$$

where F_R is the radial cutting force, N; F_{Ri} – the radial force acting on the cutting edge, referred to as the cutting edge reaction, N; F_{Ro} – the radial force acting on the flank face of the cutting tool, referred to as the machined surface reaction, N; F_{Rc} – the radial force acting on the rake face of the cutting tool, referred to as the chip reaction, N.

Equation (1) shows that the radial cutting force may be either positive or negative, i.e., it may either press the cutting tool into the workpiece or push it away from the workpiece. When the chip thickness is large, the cutting angle is small, the clearance angle is large, and the cutting edge radius is small, the radial force tends to press the tool into the workpiece. Conversely, when the chip thickness is small, the cutting angle is large, the clearance angle is small, and the cutting edge radius is large, the radial force tends to push the cutting edge away from the workpiece [3].

The tangential cutting force is equal to the sum of

$$F_c = F_{Cn} + F_{Cd} + F_{Cz}, \tag{2}$$

where F_C is the tangential cutting force, N; F_{Cn} – the tangential force acting on the cutting edge, referred to as the penetration force, N; F_{Cd} – the tangential force acting on the flank face of the cutting tool, referred to as the chip deformation resistance force, N; F_{Cz} – the tangential force acting on the rake face of the cutting tool, referred to as the chip reaction, N.

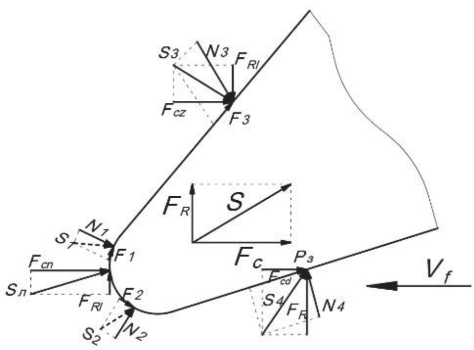

Figure 1. Distribution of forces acting on the cutting edges (Source: Grigorov, 1992)

One of the most common equations for calculating the cutting force is obtained by averaging the instantaneous cutting forces over one revolution of the cutting tool [6, 20]

$$F_c = \frac{P_c}{v_c} = \frac{K_c s_t H_c v_f}{v_c}, \tag{3}$$

where P_c is the cutting power, W; v_c – the cutting speed, m·s^{-1}; K_c – the specific cutting resistance, N·m^{-2}; s_t – the width of cut, m; H_c – the workpiece height, m; v_f – the feed speed, m·s^{-1}.

The literature often, radial force is mentioned as being equal to the cutting force multiplied by a coefficient that takes into account the degree of tool wear

$$F_R = m.F_C, \tag{4}$$

where m is the coefficient that depends on the wear of the cutting edges [21].

In addition to the cutting forces, the shaft is subjected to forces transmitted by the belt drive, as well as to bending and torsional moments. Figure 2 presents the loading scheme used for the strength design of a circular saw shaft [4, 22].

As shown in Figure 2, the shaft is loaded at its two overhung sections by the cutting force and the belt tension force, which act at angles α_1 and α_2 with respect to the horizontal plane (Fig. 2a). However, in addition to the tangential cutting force shown in the figure, the radial cutting force may also be included to obtain a more comprehensive loading model (Figure 3). In this case, both forces should be resolved into their components along the *y*– and *z*– axes, after which the corresponding components are summed

$$F_y = F_{C2} + F_{R2} = F_C cos\alpha_1 + F_R sin\alpha_1, \tag{5}$$

$$F_z = F_{C1} - F_{R1} = F_C sin\alpha_1 - F_R cos\alpha_1. \tag{6}$$

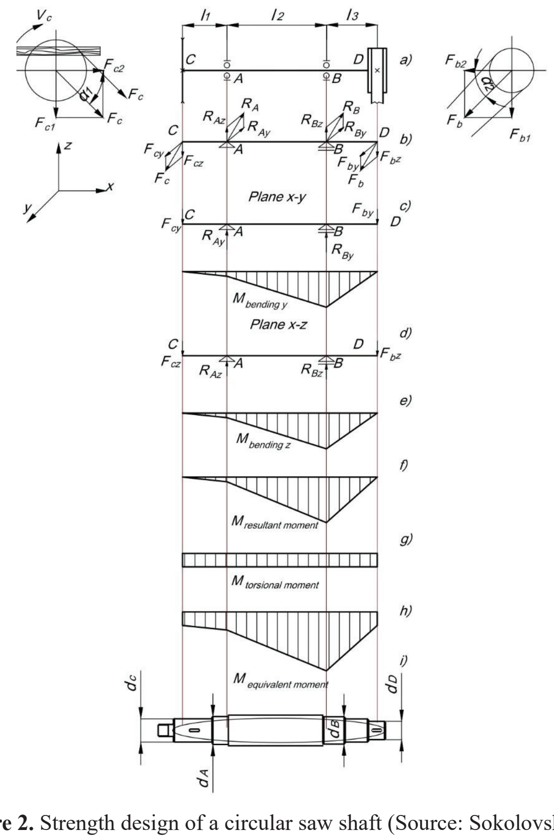

Figure 2. Strength design of a circular saw shaft (Source: Sokolovski, 2014)

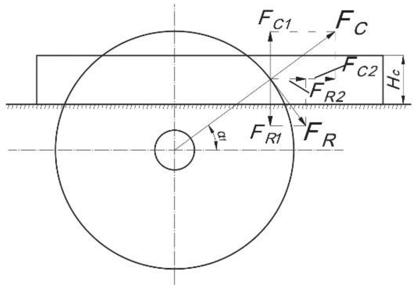

Figure 3. Distribution of cutting force components

Due to the limited length of this paper, the procedure for the strength design of shafts is not presented in detail. A comprehensive description can be found in the specialized literature. Only the main steps are briefly outlined below. After resolving the applied forces into the *xy* and *xz* planes, the support reactions and the bending moments are determined, followed by the calculation of the resultant bending moments (Fig. 2b–f). The torsional moment is determined from the transmitted torque, after which the equivalent moment is calculated at the critical cross-sections *A*, *B*, *C*, and *D* (Fig. 2g, h). Finally, the shaft diameters at cross-sections *C* and *D* are determined on the basis of the transmitted torque, while the equivalent moments at cross-sections *A* and *B* are subsequently evaluated (Fig. 2i).

MATERIALS AND METHODS

The experimental study was conducted at *Pentop Ltd.*, Varshets, Bulgaria, using a heavy-duty circular sawing machine manufactured by *Kallion Konepaja Oy* (Finland). The study evaluated cutting forces under demanding operating conditions representative of industrial practice.

The test specimens were beech (*Fagus sylvatica L.*) flitches with a nominal thickness of 100 mm and a moisture content of approximately 16%. The specimens and the device used to measure wood moisture content are shown in Figure 4.

Figure 4. Test samples

A heavy-duty circular saw machine from *Kallion Konepaja Oy* – Finland was used to conduct the experiments. A schematic of its cutting mechanism is presented in Figure 5. Some of the more important key technical parameters were: power of the electric motor driving the cutting mechanism P_e = 55 000 W, speed of the electric motor $n_e \approx$ 1500 min^{-1}, diameter of the belt pulley mounted on the electric motor shaft D_1 = 250 mm, diameter of the belt pulley mounted on the main shaft of the machine D_2 = 370 mm.

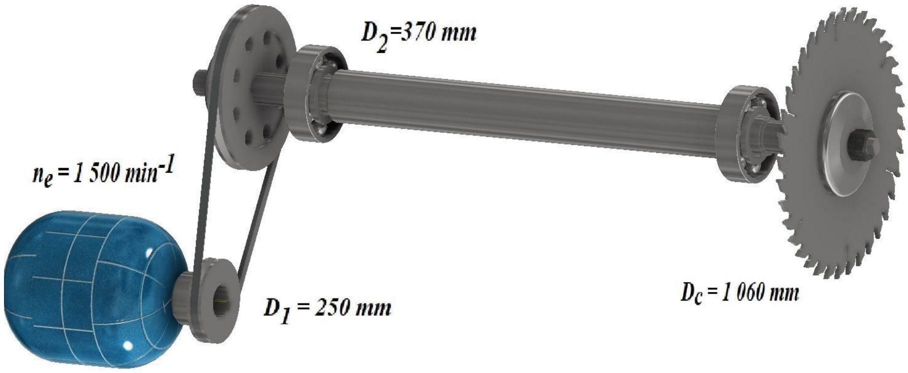

Figure 5. Kinematic diagram of the cutting mechanism of the studied machine

The circular saw blade used in the experiments was manufactured by *HW Kreissägeblätter* and had a diameter (D_c) of 1 060 mm. The blade was equipped with stellite-tipped teeth having a pitch of p = 40 mm and a tooth height of h' = 22 mm. The blade body thickness was s = 4.1 mm, while the kerf width s' was 5.4 mm. The rake angle (γ) was 25°, and the clearance angle (α) was 15°.

$$v_c = \pi D_c n_e \frac{D_1}{D_2}, \tag{7}$$

where D_c is the diameter of the circular saw blade (cutting diameter), m; n_e – the rotational speed of the drive motor, s^{-1}; D_1 – the diameter of the driving pulley, m; D_2 – the diameter of the driven pulley, m.

Feed speed was adjusted according to the motor load and compared with the rated power. When the rated power was exceeded, the feed speed was reduced to prevent overload and possible accidents. Feed speed was adjusted from the machine control panel.

The cutting force was calculated using Equation (3), after first determining the cutting power. The latter was obtained from the electrical power consumed by the drive motor of the cutting mechanism under both idle and cutting conditions. The measurements were performed using a *US301EM* multifunction power meter manufactured by *Unisyst Ltd.* (Bulgaria). The measuring device operates within a voltage range of 0–100 V and a current range of 0–5 A. The measurement accuracies are 0.2% for voltage measurements over a 500:1 dynamic range, 0.2% for current measurements over a 20:1 range, 0.1% for power measurements over a 1000:1 dynamic range, and 0.05% for power factor measurements. The active power was measured, as it represents the electrical power converted into useful mechanical work. The device was connected to the machine's electrical supply through three *CNC*® current transformers (100/5 A) and three voltage transformers (220/100 V) manufactured by *UNITRAF AD.*

The cutting power was calculated by the formula [17]

$$P_c = \left(\frac{P_{work}-P_{idle}}{100}\right).\eta, \tag{8}$$

where P_{idle} is the input power of the cutting mechanism under idle conditions, W; P_{work} – the input power of the cutting mechanism under cutting conditions, W; η – the efficiency coefficient of the cutting mechanism.

The mechanical efficiency of the cutting mechanism was determined as follows

$$\eta = \left(1-\frac{P_{idle}}{P_{work}}\right).100. \tag{9}$$

RESULTS AND DISCUSSION

The cutting speed is required for determining the cutting force. For the investigated machine, the cutting speed $v_c \approx 56$ m·s^{-1} was calculated using Equation (7). The mean density of the ten test specimens was 824 kg·m^{-3}, while their average moisture content was 40.3%. The average ambient temperature at the workplace during the experimental investigations was approximately 20 °C. Therefore, temperature was not expected to have a significant influence on the obtained results.

As previously mentioned, the present study forms part of a broader investigation aimed at determining the power-energetic parameters of a heavy-duty circular saw machine, including cutting power, cutting force, specific cutting resistance, and specific energy consumption. This paper presents only the results obtained under the most unfavorable operating conditions, identified during the preliminary experiments conducted to establish the variation ranges of the investigated factors.

Figure 6 illustrates the effect of feed speed on the cutting force during the simultaneous cutting of wood test specimens, corresponding to a height of cut of 240 mm. The feed speed was varied within the range of 10 to 70 m·min^{-1}. The reported values correspond to the arithmetic mean of 10 experimental measurements performed under each investigated condition.

Since the present study has a practical engineering focus, feed speed was selected as the investigated process parameter. However, it can readily be expressed in terms of feed per tooth

f_z by calculating the rotational speed of the machine shaft n_e from the transmission ratio of the belt drive (i.e., the ratio of the diameters of the driving and driven pulleys – D_2/D_1) and the number of saw teeth z, which in the present case was 60.

Preliminary experiments were also conducted at a height of cut H_z of 320 mm. However, under these conditions the drive motor was subjected to severe overload, even at feed speeds v_f of only 20-25 m·min^{-1}. Consequently, only the results obtained at a height of cut of H_c=240 mm are presented in this paper. The cutting force was determined by first calculating the cutting power using Equations (8) and (9). The resulting power values were then divided by the cutting speed according to the well-established relationship between power, force, and velocity (Equation 3).

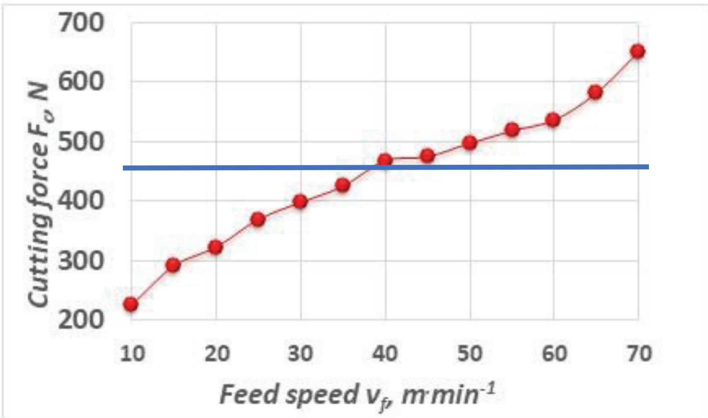

Figure 6. Influence of feed speed on the cutting force during cutting of beech wood with a heavy-duty circular saw machine

As shown in Figure 6, the cutting force can reach considerable values. At the beginning of the curve, corresponding to a feed speed of approximately 10 m·min^{-1}, the cutting force is about 225 N. As the feed speed increases to 70 m·min^{-1}, the cutting force exceeds 650 N, corresponding to a cutting power of approximately 35 kW.

In previous studies conducted by the authors on the cutting power of various woodworking machines, it was found that the idle power consumption is approximately 30% of the rated power of the drive motor [15,16]. In the present study, the drive motor of the cutting mechanism has a rated power of 55 kW. The experimental results showed that when the cutting power reached approximately 25 kW, the motor operated at its rated power, whereas any further increase resulted in motor overload. Although the overload observed during the present experiments was only temporary, prolonged operation under such conditions would significantly increase the risk of motor failure. In Figure 6, the threshold corresponding to the rated motor power is indicated by a solid blue line. This threshold corresponds to a cutting force of approximately 450 N. Consequently, this value was adopted in the design example presented below.

After determining the average tangential cutting force, assumed in this study to be 450 N, the radial cutting force can be calculated using Equation (4). As reported in the literature, the tooth wear coefficient for circular saw blades generally ranges from 0.5 to 2 [21]. In the present study, a value of m=1 was assumed.

The obtained results may be compared with reported cutting forces during the sawing of hardwoods using mobile horizontal band saw machines. Under such conditions, the cutting force rarely exceeds 50 N [15]. However, such a comparison should be interpreted with caution because the operating conditions differ considerably. For example, in mobile horizontal band saws the feed speed rarely exceeds 10 $m \cdot min^{-1}$, particularly at a height of cut of 240 mm. A further comparison can be made with studies conducted on a *Dura* (Türkiye) vertical log band saw, which represents a heavier class of machine. Even under these conditions, the cutting force rarely exceeds 70 N during log sawing [17]. It should be noted, however, that the experiments in that study were performed using Scots pine (*Pinus sylvestris L.*) and poplar *(Populus alba L.)* specimens.

As shown in Figure 3, the tangential and radial cutting force components (F_c and F_r), acting perpendicular to and opposite the feed direction, can be readily determined as a function of the contact angle. These force components can subsequently be applied in commercial CAD/CAE software packages, such as *Autodesk Inventor* or *SolidWorks*, to perform stress and deformation analyses of existing shafts or to support the design of new ones. As an example, assuming a contact angle of α_1=20°, the calculated force components acting in the y- and z-directions were Fy=577 N and Fz = -269 N, respectively. Based on the rated power of the drive motor (55 kW) and its rotational speed (n_e=1 500 min^{-1}), the transmitted torque and the corresponding forces generated by the belt drive were determined. The resulting belt force components were F_{Ry}=1 980 N and F_{Rz}= 1 980 N (assuming a representative angle of α_2 = 45°). Figure 7 (A and B) presents von Mises stress and displacement induced by these loads in a representative circular saw shaft manufactured from tool steel, which is equivalent to C45 steel (AISI 1045 Steel). The maximum equivalent stress reached 191.9 MPa, while the maximum displacement was 137.2 μm. This means that the selected example shaft is subjected to significant loads.

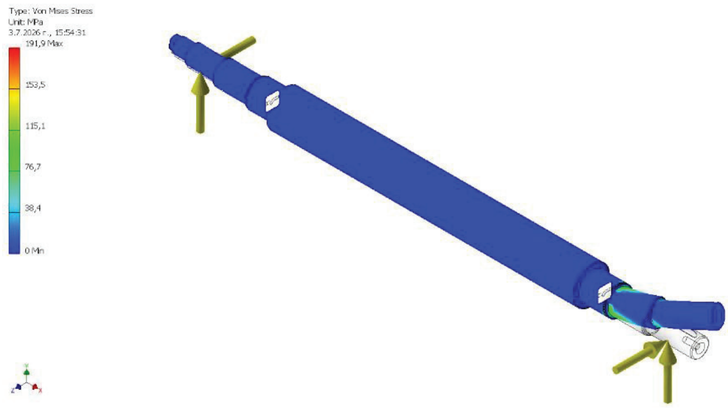

Figure 7 A. von Mises stress distribution in a representative circular saw shaft

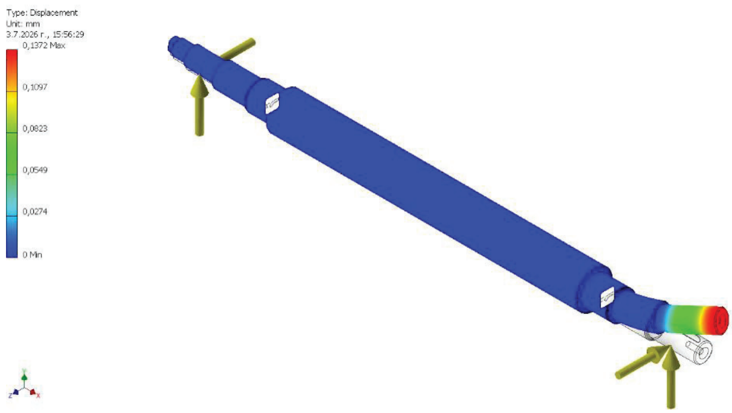

Figure 7 B. Displacement in a representative circular saw shaft

CONCLUSIONS

Based on the results of the present study, the following conclusions can be drawn:

- The maximum cutting force under unfavorable operating conditions for a heavy-duty circular saw machine was experimentally determined. This value provides a practical basis for the stress and deformation design of circular saw shafts.

- The cutting force increases substantially with increasing feed speed and reaches values exceeding 650 N at a height of cut of 240 mm. However, cutting forces above approximately 450 N correspond to operating conditions in which the drive motor exceeds its rated power and therefore occur only for short periods.

- The proposed procedure enables the experimentally determined cutting force to be resolved into its tangential and radial components and subsequently applied in numerical stress and deformation analyses using commercial CAD/CAE software.

- Compared with previously published data for log band saws, heavy-duty circular saw machines generate significantly higher cutting forces, highlighting the necessity of considering these increased loads in shaft design.

AUTHOR CONTRIBUTIONS

Conceptualization, V.A. and G.K.; methodology, V.A. and G.K.; software, V.A.; validation, V.A. and G.K.; formal analysis, V.A.; investigation, V.A. and G.K.; resources, V.A.; data curation, V.A.; writing–original draft preparation, V.A.; writing—review and editing, V.A.; visualization, V.A.; supervision, V.A.; project administration, V.A.; funding acquisition, V.A., G.K. All authors have read and agreed to the published version of the manuscript.

FUNDING

This research received no external funding.

CONFLICTS OF INTEREST

The authors declare no conflicts of interest.

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INNOVATION IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN

1/2026

INNO vol. XV

ISSN 1314-6149

e-ISSN 2367-6663

Indexed with and included in CABI

INNOVATION IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN

Science Journal

Vol. 15/ p. 1–158

Sofia 1/2026

ISSN 1314-6149/e-ISSN 2367-6663

FACULTY OF FOREST INDUSTRY – UNIVERSITY OF FORESTRY – SOFIA

The Scientific Journal is indexed with and included in CABI.

Cover Design: Desislava Angelova

Printed by: INTEL ENTRANCE

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