Finite Element Resonance Analysis of the Complex Structure of a Crosscut Saw Machine

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Abstract

A crosscut saw machine must be tuned to eliminate abnormal sawing marks or skewing during rapid wood processing. Once a wooden board is placed in the feed port, it is clamped and stabilized by a roller. However, the vibration amplification of the structure still causes the relative position of the blade to change and leads to the problem of saw marks on the wood surface. This article optimizes a saw machine by modal analysis based on the finite element method. The redesigned machine was compared to the original for natural frequency and mode shape. The analysis results revealed that at 31.43 Hz, stress reached the maximum value of 383.24 MPa on the frame of an alternating current motor. The mode shape showed significant deformation of the roller frame. By applying ribs on the chassis frame, vertical bending and torsion were reduced. The frequency of the sixth mode of the original machine was 43.9 Hz, which increased to 52.9 Hz after the redesign. The results showed that this was due to the addition of the ribs. A clamping roller was able to mitigate the vibration through the $+y$ and z directions. The natural frequency of the modality was significantly improved through rib-enforced design. The structure of the improved design exhibited improvement compared to the original machine.

Advanced manufacturing involves the use of technology to improve production. Many developments are taking place in conventional manufacturing, resulting in an upgrade toward high-speed machining. Speed and quality are critical factors in manufacturing. Wood sawing machines were invented 50 years ago, and many of these machines have been used successfully in various fields. With the demands of rapid manufacturing, this maturing industry is gradually facing the challenge of how to avoid quality degradation during rapid production. Saw machine developers have received responses from wood processors downstream who want to reduce the occurrence of defects. After an induction process, the researcher determined the cause of the problem to be excessive machine vibration. This article explores the flaws caused by vibration during high-speed cutting and hopes to reduce this possibility through analysis.

From its cultivation from the forest to its presentation as a final product, a saw is an indispensable tool. In recent literature, however, there is less information about the design of wood processing equipment. As smaller trees are used for lumber, the improvement of saw machines is becoming increasingly crucial. The characteristics of juvenile wood are ideal for saw machines: low density, extensive longitudinal shrinking and swelling, and low strength (Maeglin 1987). However, there is considerable tension concentrated in the juvenile zone. Growth stresses may cause warping, shrinking, and swelling (McLauchlan 1972). Quality control requires the sawing mechanism to be tuned for the elimination of pitfalls such as deformation, skewing, and vibration.

Sawing and Vibration

One study investigated the sound pressure level of circular saw blades for three types of blade bodies (Beljo-Lučić and Goglia 2001). The amount of noise in the machine increased sharply as the motor's rotational frequencies approached 3,900 rpm. Rubber damping rings with a diameter of 80 mm and thickness of 0.3 mm were placed between the saw blade and the collars. The saws with copper corks did not emit a whistling noise. A guide

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assembly could be mounted on each side of the blade (Jacobson 1962). Typically, the guide block was disengaged, and the saw blade was in a smooth, vibration-free running condition. The guide block was useful for preventing stalled operation due to pneumatic pressure. Reciprocating power saws were challenging to operate because of their kicking action and the tendency of the blade to vibrate and bounce. A double-bladed design (Kirbach and Sykes 1989) provided a power-operated saw with a cutting element in the form of a spaced pair of parallel toothed blades. They utilized an intermediate toothed raking to reduce the drawbacks of the saw's kicking and vibration.

Excessive vibration can potentially induce Raynaud's disease (Davis 1978). Moreover, some vibration patterns do not depend on saw thickness (Cudworth 1960). Leander (1937) improved saw design by increasing the saw's capacity and reducing vibration. The cutting edge of the blade was divided into equal cutting parts separated by notches with separate teeth beveled on alternate sides of the blade. The main obstacles of circular saw design are (1) vibration, (2) heating effects, (3) improper tensioning, (4) inaccurate saw filing, and (5) improper feed rates. These factors cause a saw to wander during cutting and produce rough-sawn surfaces. Mote (1964) stated that ''tensioning places the rim of a saw in hoop tension and is an attempt to control the natural frequencies of oscillation.'' The installation of pressure rollers gives a more symmetrical tension pattern. For overhead feed systems, leading-edge guides provide support even beyond the gullets (Wiedenbeck and Araman 1995). The selection of guide type depends primarily on the ease of installation and maintenance.

Ensuring safety during wood processing is essential. Increased cutting speed and security are key factors in saw machine design. Precise control of blade positioning can prevent defects in the wood surface caused by sawing. Weselyk (2004) invented an efficient method of straightening the face and edge surfaces of wood stocking which the inner edge is straightened; the workpieces may then simultaneously rip. No lumber-length–based differences exist in the milling throughput rates of like-sized pieces during operation (Wiedenbeck and Araman 1995).

Crosscut saw system

A cutoff saw system (Fig. 1) features a numerical controller and workpiece feed system that allows for rapid processing. The operation usually provides a production rate that is three to five times higher than that of manual cutoff saws and uses less wood material. A defect blowing system permits rapid defect removal. The maximum feed speed is 80 m/min, which is the equivalent of 80 cuts per minute. Several cutting modes are available, including fixed length, sequential cutting, and finger joint cutting. The motor runs through the chain to achieve a spindle speed of 2,880 rpm.

The crosscut table saw features a base with a stand leg and brace. A blade-elevating wheel adjusts the saw to the position of the material to be cut. The front edge and side of the saw are usually equipped with a safety guide to avoid accidental touching. An adjustable extension table, rip fence head, and fence guide bar smoothly push the wooden strip. The servo motor drives the wood on the track according to the preset cutting size. This step requires precise positioning control to avoid swerving. Multiple rollers push the wood to be cut at its front and back ends. Linear actuators use

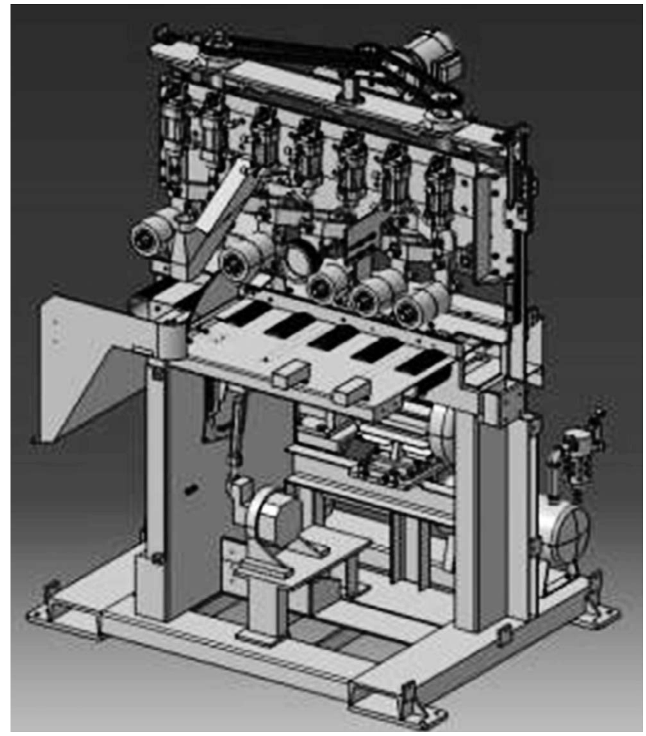

Figure 1.—The detailed structure of a cutoff saw.

compressed air to shift the log downward. The press wheel and crawler clamp (Kline et al. 1992) fix the position of the log sawed by the blade.

Problems with the saw machine

External forces might compromise the performance of the saw blade and shift the shaft; external shock causes short-term displacement. This situation is prone to occur during rapid cutting. The load disappears when the wood is cut, then a sudden increase in resistance occurs the moment the tool touches the piece of wood. This constant load change also increases the likelihood of saw blade vibration as the cutting rate increases. Steel deformation occurs according to the stress–strain curve diagram of steel. Engineers can employ appropriate reinforcement structures to increase the strength of the overall structure and thereby avoid vibration caused by external stimuli. The vibration amplitude increases due to the accumulation of absorbed energy. The amplitude of the object is as follows (Tell and Kopmaz 2006):

$$a = \frac{a_0}{1 - \left(\frac{f}{fn}\right)^2} \tag{1}$$

where a is the magnitude of the object, a_0 is the amplitude of the forced vibration, f is the forced vibration frequency, and f_n is the natural frequency of the structure. A two-spring system was analyzed using harmonic balance techniques (Timoshenko and Young 1990). The natural frequency of a machine is related to structural design. If the natural frequency is free from external excitation, one can avoid vibration amplification and resonances (Steinberg 2000). In a system with two or more degrees of freedom, the vibration mode of one degree often influences the other mode. A coupled mode can occur in translation, rotation, or combinations of the two.

There are three sources of excitation in this system: the AC motor continues to drive the circular saw, the servo motor operates during the movement of the wood chips, and the air compressors generate vibrations when the gas pressure is insufficient. The motor transmits rotational kinetic energy. The frame structure and support frame are similar to a cascading spring system. The total modal characteristics must prevent an increase in amplitude. The heavy motor is mounted on the lower side bracket and treated as a concentrated load in the middle of a beam. The Rayleigh method is suitable for determining the resonant frequency of a uniform beam with a concentrated load:

$$fn = \pi/2\left[EIg/L^3\left(wL^3 + 2W\right)\right]^{1/2} \tag{2}$$

where E is the modulus of elasticity, I is the moment of inertia, L is the beam length, w is weight, and W is the weight of the concentrated load. In reality, the pressures do not locate at a single point, and the support structure for the saw is not precisely a cantilever beam. These complex force transmission paths and individual modalities can be identified using finite element analysis.

Materials and Methods

The analysis of a cutoff saw is a complicated task. This study investigated the vibration mode and resonant frequency of the structure that is excited by the motor. A dynamic finite element model was built to identify the vibrating mode of the machine. Each vibrating mode was associated with a specific frequency. Figure 2 shows the flowchart of the dynamic analysis performed using the ANSYS computer-aided analysis package (Ansys Inc., Canonsburg, Pennsylvania).

The spindle becomes shortened or elongated due to vibrations, thus affecting the position of the saw blade. External forces perpendicular to the direction of the shaft can cause the axis to bend. Existing stainless-steel models were reviewed (Arrayago and Gardner 2015) during material selection. Sufficient frame stiffness reduced deformation during device operation.

Simplified model

Eckelman and Rabiej (1985) applied the finite element method (FEM) to investigate the design parameters of wood products. FEM has also been used to investigate how springs in a sofa seat foundation apply force to the face of the sofa's front rail (Zhang et al. 2000). The effects of tenon fit and mortise-and-tenon joints were investigated experimentally and via FEM three-dimensional modeling. The tensile loads were found to increase significantly as tenon fit increased. FEM was verified as a prediction tool (Hu and Guan 2018).

Hu et al. (2018) also provided a way to predict the long-term compression creep behavior of wood by FEM. The influence of a nonridged joist end support on deflection was also investigated (Chui et al. 2004), indicating that natural frequency is sensitive to support stiffness. Models of a diesel engine crankshaft were created (Deshbhrater and Suple 2012) to analyze vibration mode and deformation. The rigidity of an automotive chassis was analyzed in terms of its shock and vibration resistance for improving its payload (Irshad and Krishna 2014).

Another study applied a finite element program to simulate the forging process (Coupez et al. 1991). An automatic mesh generation for complex geometries enabled the simulation of complex geometries. Pellicane and Davalos-Sotelo (1993) proposed a mesh technique for predicting the load-deformation behavior of a bolted joint in wood. The mesh superposition technique enabled a three-dimensional system to be idealized in two dimensions while modeling the combined loading of bolted joints structures.

The computational aspects of Galerkin's approximation were studied (Roop 2006) using continuous piecewise polynomial basis functions on triangular elements. Another study (Martínez et al. 2015) investigated the combination of optimal structural properties and shape appearance but encountered difficulty during computation (Starý et al. 2014). Calculation of rotating ring deformations compared the simulation time as related to a modeling approach. The influences of stretcher positions on the mechanical properties of chairs were investigated through experimental and FEMs (Hu et al. 2018).

The main steps of establishing the finite element model were the following:

1. Create a three-dimensional CAD model and select features that were important for analysis.
2. Define material properties: modulus of elasticity and Poisson ratio of parts.

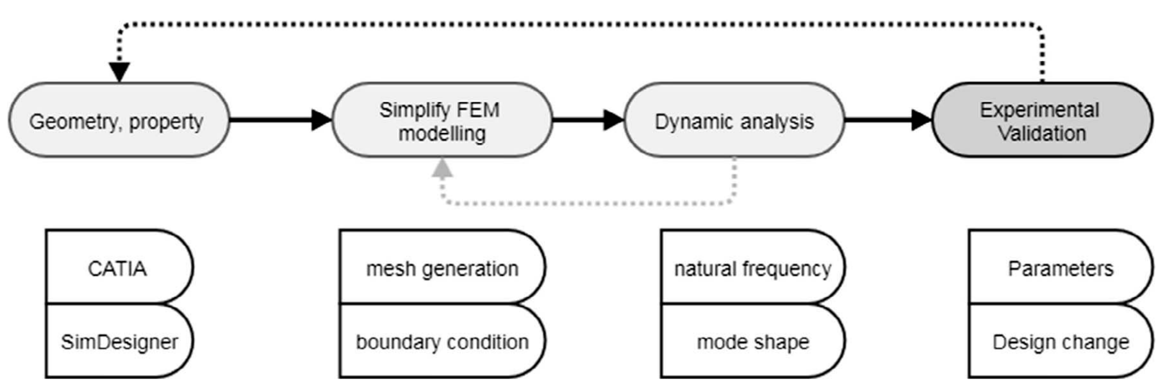

Figure 2.—Flowchart of the dynamic analysis. FEM = finite element method.

3. Divide the geometry into large numbers of small pieces (meshing).
4. Define boundary conditions and apply loads at the part.
5. Solve the equation for the defined material properties and boundary conditions.

The cross saw included a saw table, a guide, a mandrel, and a lifting wheel. A saw blade was attached to the mandrel, and the belt and the pulley were connected to the motor to move the wood on the saw table. Long strips needed to be pushed using the lever roller. If the processed wood was frugal or too wet, it might have caused the wood to pop out; for this reason, the saw blade had a safety cover. We simplified the parts through geometric simplification. Motors were replaced with a lumped mass. Part of the three-dimensional shape was redrawn to exhibit uniformity, as shown in Figure 3, without affecting the modal shape. Rounded corners, holes, and discontinuous deformities were removed to reduce computational difficulty. Fine mesh would have increased the cost of calculation, and round-off errors could have been magnified and reduced the accuracy of the result.

For the construction model of complex structures, we focused on the chassis seat, the main beam, and the components with weight, such as the motor and the pressure bar, which are closely related to the structure. For example, we assumed that the dynamic properties of those parts had not changed when we simplified its shape; similarly, any other changes to the dynamic characteristics of the simplified model were assumed to be insignificant. The boundary conditions included the loads and constraints that represented the effect of the surrounding environment on the model. Redundant supports and excessive restrictions tend to add stiffness to the model. The characteristics of constrained joints and application of boundary conditions compromised the performance of the analysis process.

Redesign of the chassis structure

The mesh was constructed using an ANSYS preprocessor. A gradient mesh generation was applied, and the mesh density increased in the critical area. We assumed that the component was within the linear region. The boundary conditions of the elements were set as follows:

1. The base of the machine was fixed.
2. The axis and the bearing hole were in close contact during rotation regardless of the tolerance.
3. The distance between the components with contact parameters was specified.

Modal analysis is the basis for improving the individual modal and structural resonance frequencies. We used CATIA three-dimensional modeling software to build a complete digital model that was simplified according to the force distribution. We modified the mesh before the modal analysis, using a dense mesh at the stress concentration or significantly deformed areas.

The main setting parameters in the operation steps of the analysis software were as follows. The ''Advanced Size'' function was set as follows: the curvature in the ''Relevance Center'' was set to ''Coarse,'' the minimum mesh size was 5 mm, and the maximum mesh size was 20 mm. The number of meshes and computational complexity was reduced. Figure 4 depicts the mesh of the simplified model. The number of grids was 143,094, and the number of nodes was 313,759. The parameters of steel are density of 8.01 kg/m^3, modulus of elasticity of 200 GPa, and Poisson ratio of 0.25. The features of iron are density of 7.87 kg/m3, modulus of elasticity of 200 GPa, and Poisson ratio of 0.291.

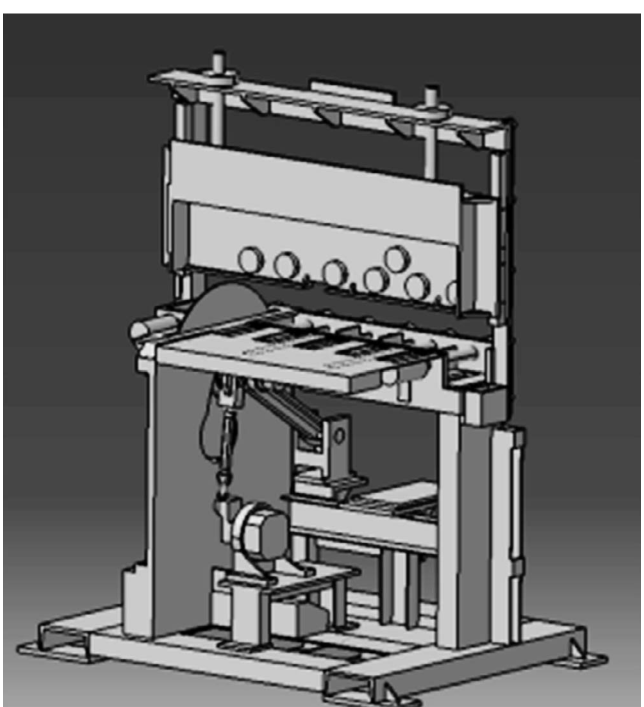

Figure 3.—View of the simplified model.

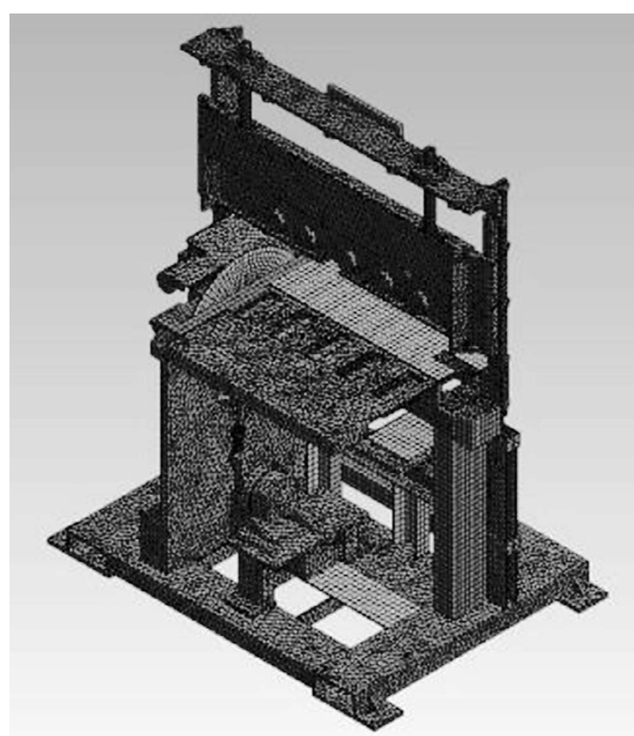

Figure 4.—Finite element mesh of a crosscut saw machine.

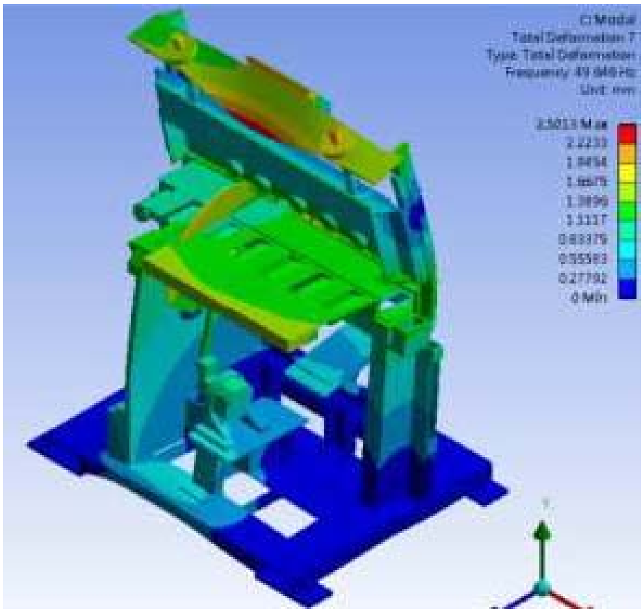

Figure 5.—Mode 7 of the original machine, $f_7 = 49.6$ Hz (related bending of the roller frame).

Results and Discussion

Mode shape of the saw machine

The structure resonated as the frequency of the saw structures approached the value of the motor frequency. This amplification caused the position of the saw blade to change, leading to the problem of saw mark formation on the wooden board. According to the modal analysis results, at 31.43 Hz, stress reached the maximum value of 383.24 MPa on the AC motor frame, and at 43.93 Hz, strain reached the maximum amount of 6.542 mm in the main structure. Figure 5 shows mode 7 of the machine; the interference of external excitation was reduced by structural reinforcement.

According to the modal analysis results, the stiffness of the machine structure was insufficient, especially in the main beam, base, servo motor seat, and roller. As indicated in Figure 5, the distortions of the roller frame caused uneven pressure at the roller. Three improvements will be made as follows:

1. Limit the amount of deformation of the axle beam to avoid extreme swing.
2. Add 10 reinforcing ribs near columns on both sides.
3. Add nine pieces of reinforcing ribs to stabilize the base.

Modal analysis results

Modifications were made to the original machine following analysis. Ribs were installed in structurally sensitive areas to enhance their strength (Figure 6). Figures 7 and 8 reveal that the frequency of the improved machine design was favorable overall compared with the original design. The frequency of mode 5 was the closest to the motor frequency, and the deviation limited its resonance. The frequency of mode 5 of the original machine was 37.1 Hz, which improved to 50.2 Hz in the redesigned machine.

Referring to Table 1, the modalities before and after improvement were not one-to-one correspondences. We needed to observe similar modalities to confirm that the two belong to a similar pattern. The authors compared the effect of this mode on the saw blade offset. Since the motor's rotational frequency was 48 Hz, we paid special attention to modes 5 to 8. The torsional deformation of the improved mode 7 was suppressed, so there was no mode (blank) corresponding to the original machine. The reinforcing strips increased the rigidity of the structure so that all the frequencies of the improved modes 5 to 8 were raised above 50 Hz, and the difference in mode shape caused less excitation. The frequencies of the improved modes 6 and 8 were higher than the rotational frequency and deviated from the vibration source, indicating a decrease in the degree of amplification.

The deformation of the original modes 6 and 7 (located at the saw blade) was high. In contrast to the analysis results of the improved model, those of the original mode indicated that when the motor speed of the machine exceeded 3,100 rpm, the structures around the saw blade were affected. When the cut piece of wood was pushed forward, the swing of the saw blade itself left cut marks on the wood. The uneven edges needed subsequent reprocessing (grinding and cutting). The improved mode exhibited no problems related to resonance or wood twill at the current speed (2,880 rpm).

The rib placed on the improved structure was stiffer and had a different orientation, which eliminated the possibility of twisting and distortion. If the base moved without

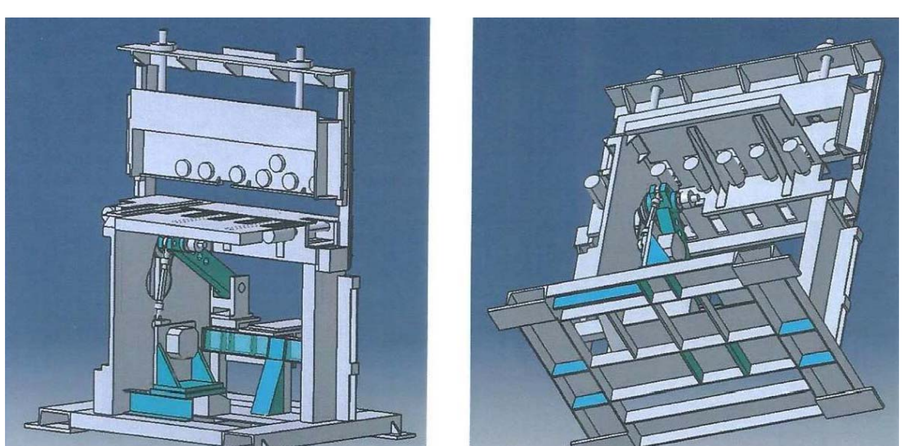

Figure 6.—Structurally reinforced ribs in blue.

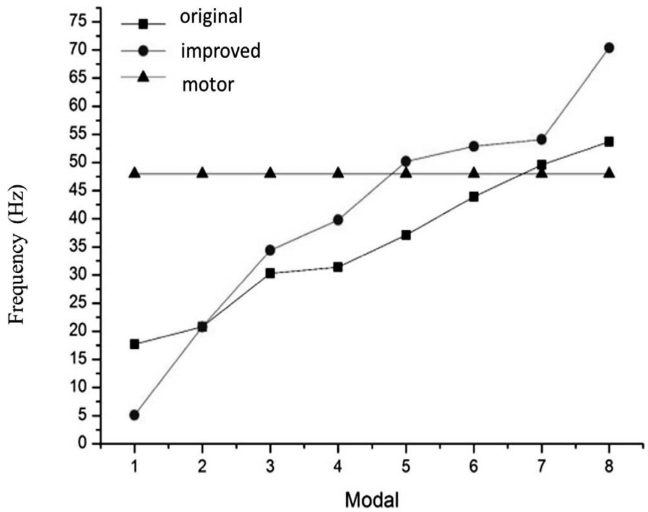

Figure 7.—Redesigned structure and related modal frequency.

twisting, then the translational mode did not couple with the rotational mode. The mode shape showed that for an applied load placed at the center of gravity along the y axis, the motor moved along the y axis without rotation. The improved modes differed from the original, particularly modes 6 and 7, for which the frequency was significantly improved.

The linear actuator featured a plurality pushing roller; the rectangular frame was placed in a relatively high position. This design affected the structural rigidity; the square frame was bent with hinged ends and subjected to a lateral load. The bent horizontal frame rotated at a specific angle and caused the vertical structure to turn in a manner dependent on the external force. When the rectangular frame bent, the degree of bending deflection was higher than that of other deformation types. When the reinforced rib was applied to the horizontal frame, it limited vertical bending and increased the strength of the frame. The L-shaped bar was laterally extended and formed a higher stiffness structure. Therefore, a clamping roller could withstand the vibration through the $+y$ and z directions.

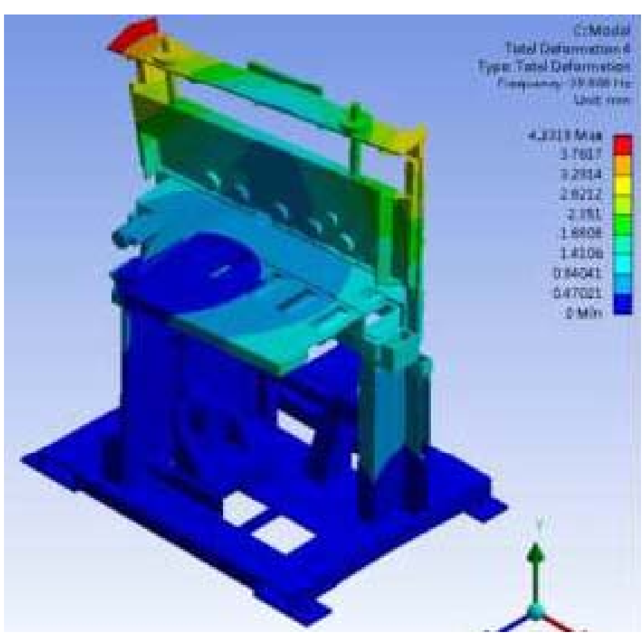

Figure 8.—Modal frequency of redesigned mode 4 is 39.8 Hz.

Table 1.—Summary of frequencies (Hz) in modal analysis.

Mode	Original frequency (Hz)	Redesign frequency (Hz)	Note
1	17.7	20.8	
2	20.8	—	No corresponding modality
3	30.3	34.4	
4	31.4	39.8	
5	37.1	50.2	
6	43.9	52.9	
7	49.6	—	No corresponding modality
8	53.7	70.4	

Conclusions

In this study, crosscut circular saws were used to perform sequential cutting with a vertical compressing clamp roller that stabilized the wooden board. Blade vibration can seriously influence the surface quality of wood panels. An unbalanced AC motor causes periodic deformation of the crosscut saw. If the cascade structure is vibrating, the swinging movement will affect the surface quality of the wood. An optimal sawing machine design based on FEM was investigated. Modal analysis was conducted, and the frequencies of the redesigned machine and the original were compared with natural frequency and the mode shape. The results show that the redesign worked due to the addition of ribs, which proved that the finite element was able to optimize the structure of the saw machine. By changing the stiffness of the chassis, the improved mode exhibited no problems related to wood log twill at the speed of 2,880 rpm.

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