AN EXPERIMENTAL STUDY ON THE LATERAL NATURAL FREQUENCY OF BANDSAW BLADES

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ABSTRACT

Two solutions for predicting the lateral natural frequency of bandsaw blades are experimentally verified for saw velocity, span length, and tension stress ("strain"). Both solutions are shown to be accurate provided that the effective span length of the guide system, which has been shown for pressure guides to exceed the distance between facing guide edges, is taken into account.

It was found that wheel rotation can cause resonance vibration in the backside section of the sawblade, where the blade extending from wheel to wheel is unsupported. This resonance vibration, however, can be avoided by installing a guide midway between the wheels.

Keywords: Saw velocity, span length, tension stress, pressure guides.

INTRODUCTION

In recent years, bandsaws have gained increasing popularity in the sawmilling industry, primarily because of the introduction of the so-called thin-kerf, high-strain sawing technique. This technique, when properly applied, results in considerable kerf savings, improved accuracy, and reduced surface roughness of cut lumber. However, at the present time these benefits are not fully exploited, mainly because of problems encountered in operating thin-kerf bandsaws.

The most serious of these problems are saw vibrations, decreased feed speeds (due to the smaller gullet area of thin-kerf saws), and inadequate saw maintenance. It has been found, for instance, that both the unusually high degree of gullet cracking and weld failures observed with thin saw blades and the poor cutting accuracy sometimes found are largely attributable to either excessive vibration or to poor blade preparation. Naturally, it is of the utmost interest to both the sawmilling industry and bandsaw manufacturers that these problems be investigated with the objectives, first, of reducing the vibration of bandsaw blades; second, of increasing feed speed by increasing the velocity of saws (provided the critical buckling load is not exceeded) and, third, of improving saw preparation methods, in particular, saw tensioning.

Excessive or large-amplitude vibrations always occur under conditions that allow a blade to oscillate at or near its natural frequencies. Naturally, such conditions have to be avoided. To determine conditions necessary to avoid this resonance is a complex task. It requires precise knowledge of the influence of a great number of variables on the resonant excitation frequencies.

In recent studies, much emphasis has been placed on lateral vibrations as they are affected by various parameters. Mathematical procedures have been developed to predict resonance vibrations. Mote (1965a) in his first paper on band vibration presented two simple and accurate solutions for calculating the lateral natural frequencies of an axially moving, simply supported band. One of the solutions—the flexible band solution—provides a lower bound to the exact natural frequency and the other—the Galerkin solution— an upper bound. The flexible

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band solution is particularly accurate because the high band tension is the dominant restoring force. The parameters for which these solutions account are band tension and velocity, span length and wheel support. Later, Mote (1965b) extended the solutions by including the effect of periodic, axial, band tension variations, which may result for instance from wheel eccentricities, and in a third study Mote (1968) considered also bending stiffness and periodic, in-plane, edge loading. Anderson (1974) analyzed the same problem and derived a solution that accounts for band tension, velocity, bending stiffness and span length. This solution uses the Galerkin procedure and the finite element analysis for solving the equation of motion. The finite element approach allows for other boundary conditions than simply supported ones to be analyzed.

Another contribution to this subject was made by Soler (1968), who analyzed both lateral and torsional vibrations and showed that point loading couples these motions, with the result that the lowest resonance frequencies can be reduced when an edge load close to the static buckling load is applied. The effect of tensioning and wheel tilting, two parameters to which sawblades of large production bandmills are subjected, was studied experimentally by Kirbach and Bonac (1978). Tensioning was found to affect the torsional but not the lateral vibrations; whereas tilting was shown to affect both types of vibration, but only at very large tilting angles.

Some manufacturers of large bandmills attempted to use solutions presented above for predicting the lateral natural frequencies of large bandsaw blades. They claimed that their experimental data were not always in good agreement with the theoretical data.

The main objective of this study was to experimentally check the accuracy of solutions developed by Anderson (1974) and Mote (1965a) for the three parameters of saw velocity, span length, and tension stress. Mote's solution examined in this study is known as the flexible band solution. It is of the form:

$$\omega = \frac{b}{2l}\left(\frac{R_S}{\rho A}\right)^{1/2}\left(1 - \theta\frac{\rho A c^2}{R_S}\right)\left(1 + \eta\frac{\rho A c^2}{R_S}\right)^{-1/2}$$

where: A = cross-sectional area of the band
b = vibration mode (1, 2, 3 . . .)
c = band velocity
l = span length
R_S = band tension
η = a measure of the wheel support stiffness ($0 \leq \eta \leq 1$). $\eta = 1$ when the wheel support is rigid, and $\eta = 0$ when the support is flexible. (The wheel stiffness of all production bandmills equipped with air or hydraulic strain systems is close or equal to 1.)
ω = natural frequency (Hz)
ρ = mass density.

This solution is easy to use and all calculations can be done by hand. Anderson's solution, on the other hand, is rather complex. It involves finite element analysis and requires considerable computer calculations. Because of the complexity of Anderson's solution, it is not presented here. Those interested may refer to Anderson (1974).

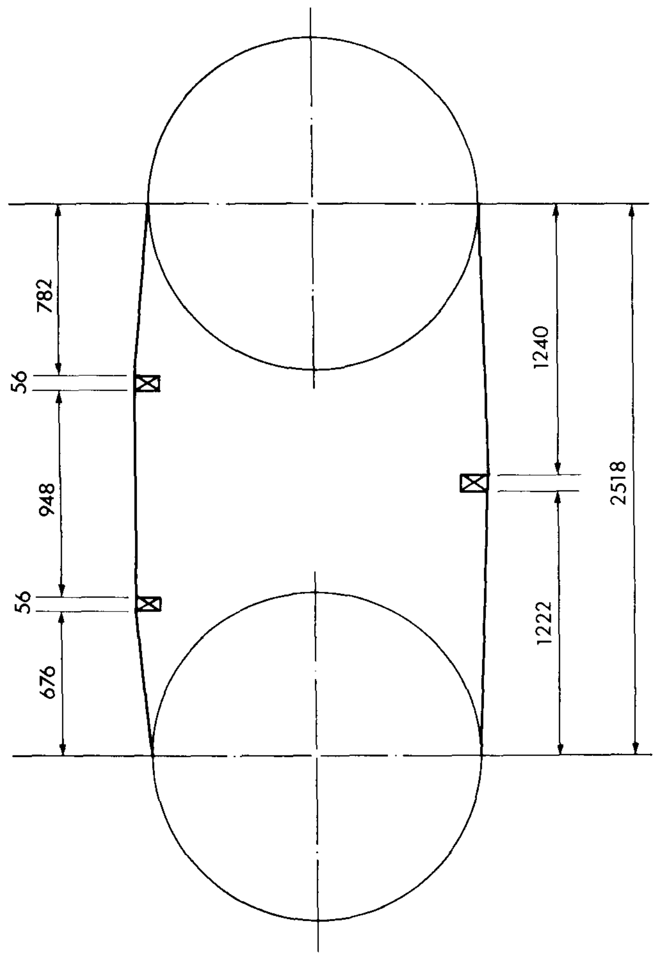

FIG. 1. Span length and position of front and back pressure guides in one of the experiments (dimensions are in millimeters).

As the experimental data of the present study showed some discrepancy between experimental and theoretical data, the objective of the study was extended to investigate the effective span length of pressure guides. It was considered to be a likely cause for the difference between observed and predicted natural frequencies.

Resonance vibration was studied under static and dynamic conditions, but because vibration-related problems (gullet cracks and weld failures) occur almost exclusively under idling conditions, it was decided to limit the experimental work to the noncutting situation.

VERIFICATION OF ANDERSON'S SOLUTION AND MOTE'S FLEXIBLE-BAND SOLUTION FOR BAND VELOCITY, TENSION STRESS AND SPAN LENGTH

A newly built production bandmill with a wheel diameter of 1550 mm, driven by a 50-hp variable-speed motor, was used in this experiment. The wheels of the

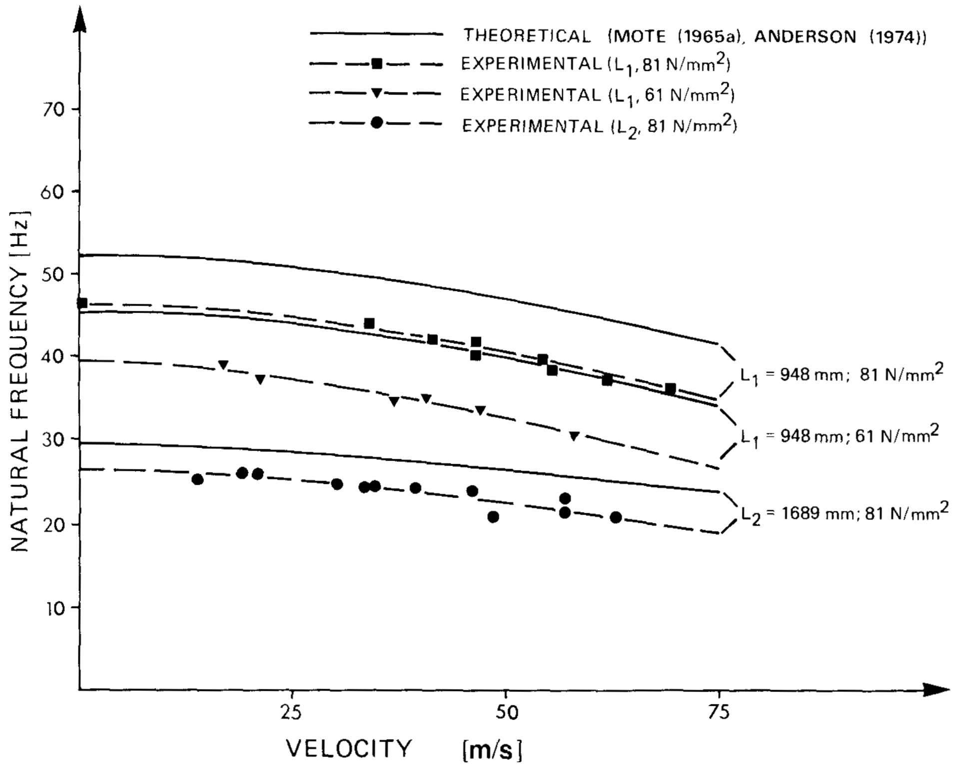

FIG. 2. The effect of band velocity, span length, and tension stress on the lateral fundamental frequency of the bandsaw blade. The wheel support stiffness was assumed to be $\eta = 0$.

mill were statically balanced prior to assembly and dynamically balanced on the mill. The bandmill was equipped with a hydraulic strain system and 56-mm-long pressure guides. As shown in Fig. 1, two guides were mounted on the front of the bandmill and, in a later phase of the experiment, one on the backside. The manometer type of "strain" indicator was used to determine or set desired tension-stress levels. The accuracy of this meter was checked with strain gauges and it was found to vary considerably with tension-stress level. This required that all tension-stress data taken from the "strain" indicator had to be corrected.

Two identical single-cutting bandsaw blades were examined. They measured 235 mm in width and 1.4 mm in thickness, had a pitch of 51 mm, and were tensioned by the manufacturer for optimum cutting performance.

The effect of saw velocity was studied in the range from 0 to close to 75 m/sec.; the effect of tension stress at two stress levels, 61 and 81N/mm^2; and the effect of span length at three lengths, 948, 1240, 1689 and 2518 mm. The shortest span was formed between two pressure guides (between the nearest faces on the guides), the medium spans between the front upper pressure guide and a wheel, and the longest between the two wheels.

Two different techniques were used to obtain data. In one technique, the natural frequencies were determined from resonance peaks of self-excited resonance observed while the blade was accelerated and decelerated between 0 and approximately 75 m/sec.

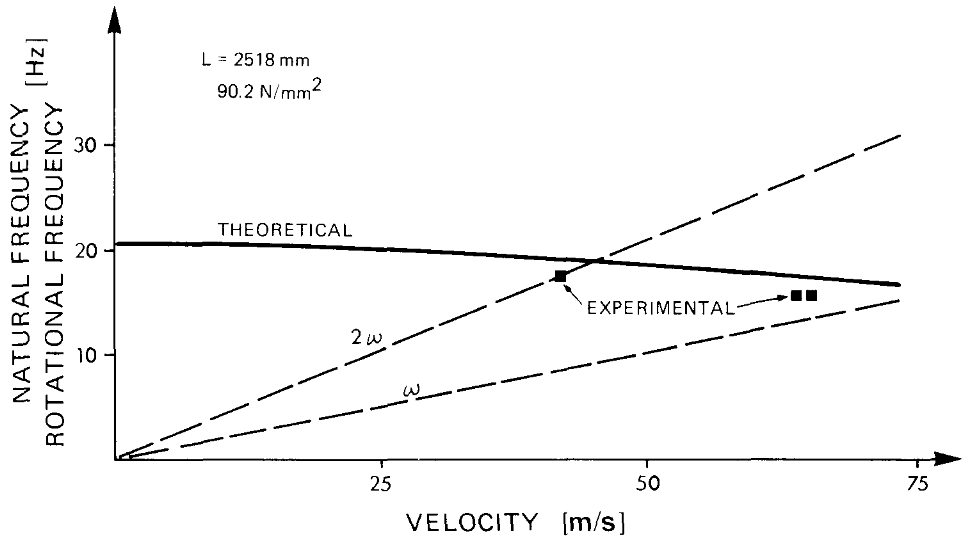

FIG. 3. Lateral fundamental frequency of the wheel-to-wheel span on the backside of the bandmill. The experimental data are the result of self-excited resonance vibration.

In the second technique, an electromagnet was used to excite the blade. It was mounted in the center of the span formed by two guides or one guide and a wheel. The magnetic oscillator was swept through a frequency band ranging from approximately 2 to 200 Hz to excite the blade. Resonance vibrations were picked up with two noncontact eddy-current displacement transducers and recorded with a cathode-ray-tube recorder.

Typical experimental data are presented in Fig. 2. The solid lines represent predicted fundamental frequencies as calculated with Anderson's and Mote's solutions. For the testing conditions used in this experiment, the two solutions differed only slightly. On average, values obtained with Anderson's solution exceeded those of Mote's solution by only ½ Hz for simple supports. For this reason, data of the two solutions are combined in one curve. As mentioned earlier, Anderson's solution, in contrast to Mote's, takes into account the bending stiffness of the sawblade. But the fact that the data of the two solutions differ only slightly shows that this parameter at tension stresses applied in sawmilling has only a very limited effect on the natural frequencies.

Both experimental and theoretical data show the well-known phenomenon—the decrease of the natural frequency with increasing span length and saw speed and the increase with increasing tension (Fig. 2). Since the three parameters are presented in the sawmilling range, it appears that span length exerts the greatest influence on the lateral fundamental frequency when the three parameters are changed within the operational ranges.

As Fig. 2 shows, the experimental data deviate considerably from the theoretical curves. The differences vary between individual data sets and are found to be largest at the higher frequency levels. A maximum difference of 7 Hz is evident, whereas the deviation at the lower frequency level (Fig. 2) amounted on the average to only 3 to 4 Hz.

Deviation of the experimental data from the theoretical would have been even

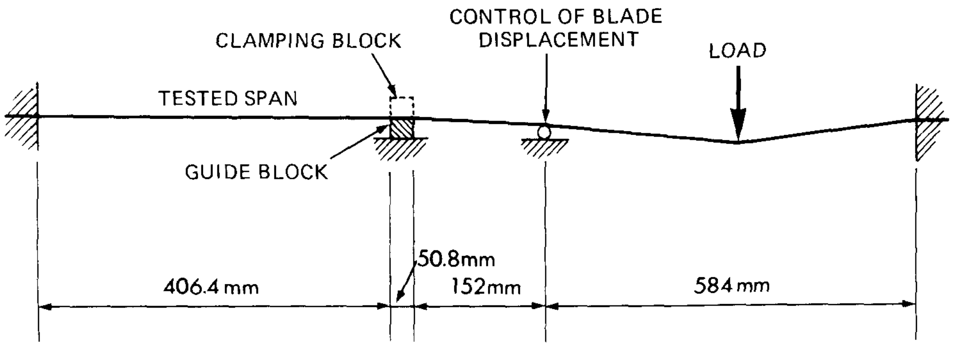

FIG. 4. Schematic of the testing device used for studying the effective span length of pressure guides.

greater if a correction of the effect of saw-tooth weight on the natural frequency had not been made. The theoretical solutions consider only smooth-edged blades. In applying these solutions to a toothed saw blade, the blade is normally treated as smooth-edged with a width measured from the gullet to the backside. In this way, the weight of the teeth, which add to the total weight of the blade, is excluded in the calculations and, therefore, somewhat higher frequency values are obtained. It was found that this upward shift amounted to approximately 1 Hz for the blades used. For this reason, all theoretical calculations considered the "tooth effect" by increasing the unit mass of the blade by 0.4 g/cm^3 to 8.2 g/cm^3 (unit mass of saw steel = 7.8 g/cm^3).

Deviation of experimental from theoretical data is approximately of the magnitude as observed by bandmill manufacturers. Their observations were made with bandmill situations similar to the one used in this study. In general, agreement between experimental and theoretical data for the cutting section of the blade has to be considered rather poor. It is questionable whether the solutions at this stage could be used for practical applications.

All experimental data presented in Fig. 2 were obtained by exciting the blade with an electromagnet, whereas data for the span length of L_3 = 2518 mm (Fig. 3), which is the length of unsupported blade on the backside of the bandmill, were obtained by self-excitation. As can be seen in Fig. 3, every time the wheel rotational speed or its double coincided with the natural frequency of the 2518-mm span, the blade in this section underwent resonance vibration. The vibration was so strong that it affected the whole bandmill structure. Since bandmills in general are accelerated to speed levels in the vicinity of 50 m/sec, the blade with a span length of 2518 mm runs at least through the left resonance shown in Fig. 3. Probably some of the blade failures may relate to this particular resonance.

Furthermore, it is very likely that any vibration of the backside, which occurs because of the great span length at relatively large amplitudes, causes considerable tension-stress fluctuations in the blade. These fluctuations can be expected to be carried over the front side of the mill, where they may enhance vibration in the cutting section of the blade and therefore deteriorate cutting accuracy.

To eliminate the wheel-excited resonance vibration of the back span, a guide was installed in the center position between the two wheels as shown in Fig. 1.

In this way, the resonance frequency of the blade was moved above the rotational frequency of the wheel for the saw-speed range tested. No wheel-excited resonance was observed thereafter. This suggests that, in designing new bandmills, serious consideration should be given to breaking down the span of the backside into at least two approximately equal sections.

THE INFLUENCE OF PRESSURE GUIDES ON THE BLADE NATURAL FREQUENCY

In the following, an attempt will be made to determine the cause of disagreement between the predicted and observed vibrational behavior of a blade. It appears from Figs. 2 and 3 that the two solutions are correct for effect of saw velocity on fundamental natural frequency. As can be seen, the slopes of the theoretical curves and the experimental points are essentially identical. But the data do not show (Fig. 2) how well the solutions describe the effect of the other two parameters, and no conclusions can be derived for the cause of the difference between the theoretical and experimental data.

The bandmill used in the natural-frequency experiment was equipped with pressure guides, a guide type now commonly used on large North American bandmills. This guide type consists of only an inner guide block which pushes the blade approximately 9.5 mm out of its natural path between the two wheels. It is claimed that this guide system outperforms conventional guides because there is a constantly tight contact between the saw blades and the guide block.

Looking at these one-sided guides, one has to question whether the effective span length between two guide blocks is the distance between the edges, which face the other guide block, or is actually longer. If the latter is true, the natural frequency would be lower than calculated for the distance between the two inner edges.

To test the effective span length of a pressure guide, a testing apparatus simulating the frontside of a bandmill was built. A schematic drawing of the apparatus is shown in Fig. 4. Basically, a smooth-edge blade section 105 mm wide and 0.9 mm thick was mounted over a 51-mm-long block in the same way as a blade is mounted over pressure guide blocks. The ends of the blade were tightly clamped between steel blocks. The tension stress was induced simply by placing weights on the blade as shown in Fig. 4. Two stress levels were applied, 55.0 N/mm^2 and 72.4 N/mm^2. Stress measurements were made with two strain gauges mounted on the testing section of the blade, one close to each of the two edges.

An adjustable roller-type support was mounted underneath the blade on the side of load application, to control displacement of the blade. In this way, the displacement could be set to the level applied in practice.

A frequency generator and a loudspeaker were used to excite the blade with frequencies from 2 to 200 Hz. Data recording and analysis were carried out following the same procedure described earlier under the main study.

After the fundamental frequency of the pressure-guide-mounted blade was measured, a matching block was tightly clamped onto the pressure guide block as shown in Fig. 4. Then the fundamental frequency of the clamped blade was determined.

The measured and calculated natural frequencies of the pressure-guide-mounted blade were found to differ in the same way as observed on the bandmill. But

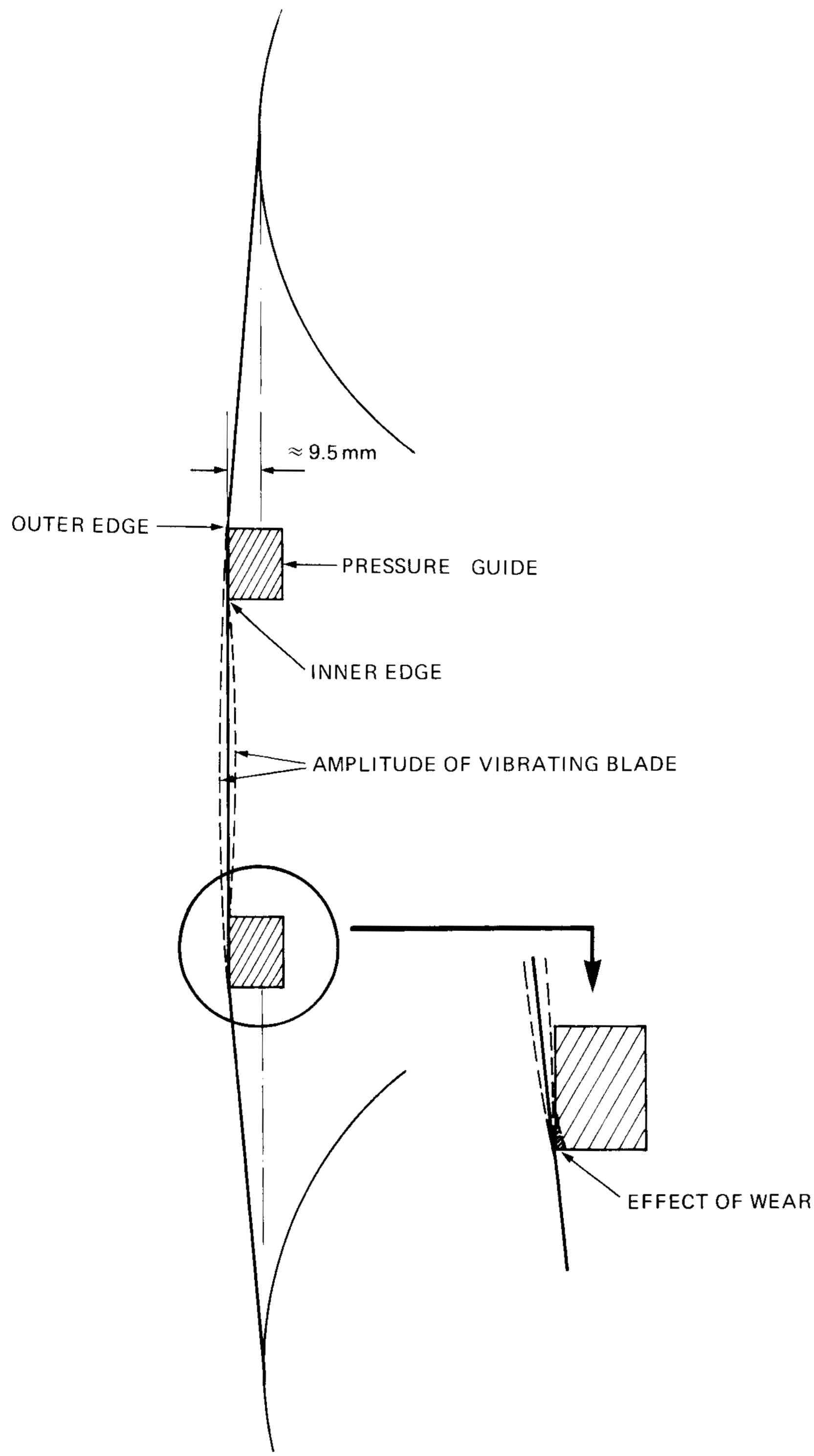

FIG. 5. Effective span length in a pressure guide system.

after the blade was clamped, the natural frequency was found to have increased by 5 Hz. This proves that the guide system is the major cause of the discrepancy observed between theoretical and experimental data. If the effect of the guides is taken into account in Fig. 2, the difference is reduced to approximately 2 Hz.

The shifting of the natural frequency by clamping indicates that the effective span length of the pressure-guide system is not the distance between the inner edges of the guide blocks (the edges facing the other guide), but the distance between regions close to the outer edges, as shown in Fig. 5. With freshly prepared guide blocks, the actual span length can be expected to be extremely close to or even at the outer edges of the blocks; but with progressive wear of the edges, the span length will decrease. This suggests that the natural frequency will increase.

The fact that the effective span length is close to the distance between the two outer edges can be attributed to the limited bending capability of the saw blade. It is very likely that the bending radius is too small for the blade to undergo complete bending around the outer edges of the guide blocks. The result is that a small clearance is formed between guide face and saw blade over most of the guide face. A schematic drawing of this phenomenon is shown in Fig. 5. The small clearance probably remains essentially the same as long as the guide face is not entirely worn, but it may largely disappear when a wear stage that rounds the whole guide face is reached.

CONCLUSIONS

1. Anderson's solution and Mote's flexible-band solution accurately predict the lateral resonance frequency of a bandsaw blade provided that the effective span length of the guide system is known. The effective span length of new or slightly worn pressure guides exceeds the distance between the nearest faces of the two guides, causing the blade to vibrate at lower frequencies than predicted from the guide-to-guide distance.
2. The frequency of the wheel rotation and the fundamental natural frequency of the saw blade on the backside of the bandmill, where no guide support exists, often coincide. This results in excessive (large amplitude) vibration. Mounting of a backguide halfway between the wheels eliminates this problem, which suggests that production bandmills should also be equipped with backguides.
3. Special care should be exercised in using tension stress values from "strain" indicators for theoretical calculations. The use of more accurate stress-measuring devices is recommended.

REFERENCES

ANDERSON, D. L. 1974. Natural frequency of lateral vibrations of a multiple span moving bandsaw. Report prepared for the Western Forest Products Laboratory, Fisheries and Environment Canada, Vancouver, B.C. 18 pp.

KIRBACH, E. AND T. BONAC. 1978. The effect of tensioning and wheel tilting on the torsional and lateral fundamental frequencies of bandsaw blades. Wood and Fiber. 9(4):245–251.

MOTE, C. D. 1965a. Some dynamic characteristics of band saws. For. Prod. J. 15(1):37–41.

———. 1965b. A study of band saw vibrations. J. Franklin Inst. 279(6):431–444.

———. 1968. Dynamic stability of an axially moving band. J. Franklin Inst. 285(5):329–346.

SOLER, A. I. 1968. Vibrations and stability of a moving band. J. Franklin Inst. 286(4):295–307.