DAMPING CHARACTERISTICS OF BAND SAW PRESSURE GUIDES

by

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SUMMARY

The objective of this study was to compare damping efficiency of block and roller pressure guides.

The frequency spectrum of a band saw blade in contact with various guide designs was measured. All guides tested were found to dampen blade resonant frequencies efficiently. The blade section located on the guide vibrated harmonically due to wheel rotation and blade convolution. It was determined that substitution of block guides with roller guides was feasible. Improved roller guide design parameters were identified.

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OBJECTIVE

The purpose of this study was to investigate damping efficiency of currently used pressure guides and a roller guide prototype. The information generated could lead to the design of more efficient band saw guides.

INTRODUCTION

Guides are an indispensible element of efficient band saw design. Theoretical and experimental research has shown that reduced blade span increases the effective blade stiffness and improves sawing accuracy. However, the blade-guide interaction is not well understood and guide performance criteria have not been refined. Previous studies have shown that increasing the clearance between blade and guide increases lateral blade vibration (Breznjak and Moen, 1972; Pahlitzsch and Puttkammer, 1976). However, the effect of clearance can be interpreted as equivalent to a change in the blade span. Pressure guides displace the blade-wheel line and effectively reduce the span.

Restraining blade pivotal movement at the end of the span by the guides influences the stiffness of the blade in the cutting section. Theoretical demonstration of two limiting conditions is possible through use of simple and clamped supports in an axially moving beam or plate model. Using the finite element method, the natural frequencies of the first few models of vibration for the relatively large spans have been found (Anderson, 1974) to increase by eight percent when clamped, rather than simple supports, were modelled; the improvement of shorter spans was not examined.

Isolation of the blade in the cutting region from the remainder of the system appears to be another important, albeit controversial, criterion for guide efficiency. On one hand, isolation is desirable since even small excitations outside the cutting region can be amplified in the band response between the guides (Miranker, 1960; Mote, 1968). On the other hand, it could be advantageous to maintain access to the cutting span from the noncutting span in order to control its vibration. The mechanism of transfer of vibration energy between the two spans could be either axial or pivotal excitation.

If small clearance, clamped support, and span isolation are to be achieved, face or line contact between blade and guide must be maintained in a dynamic situation.

MATERIALS AND METHODS

Experiments were carried out in the laboratory on a production band mill with the following parameters:

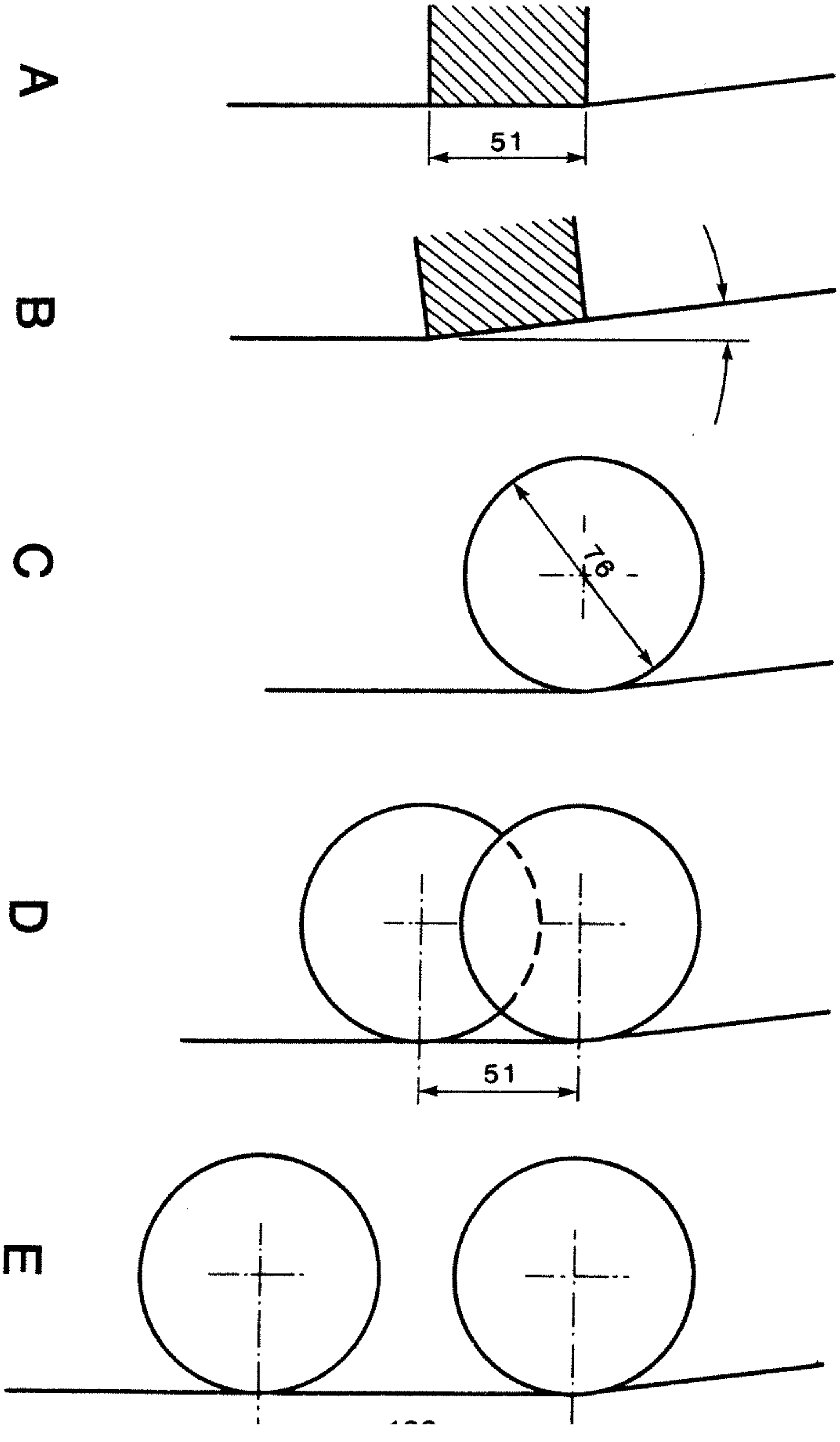
Wheel diameter	1,524 mm
Blade axial velocity	53 m/s
Blade thickness	1.65 mm
Blade width	264 mm
Blade static tension	36,200 N

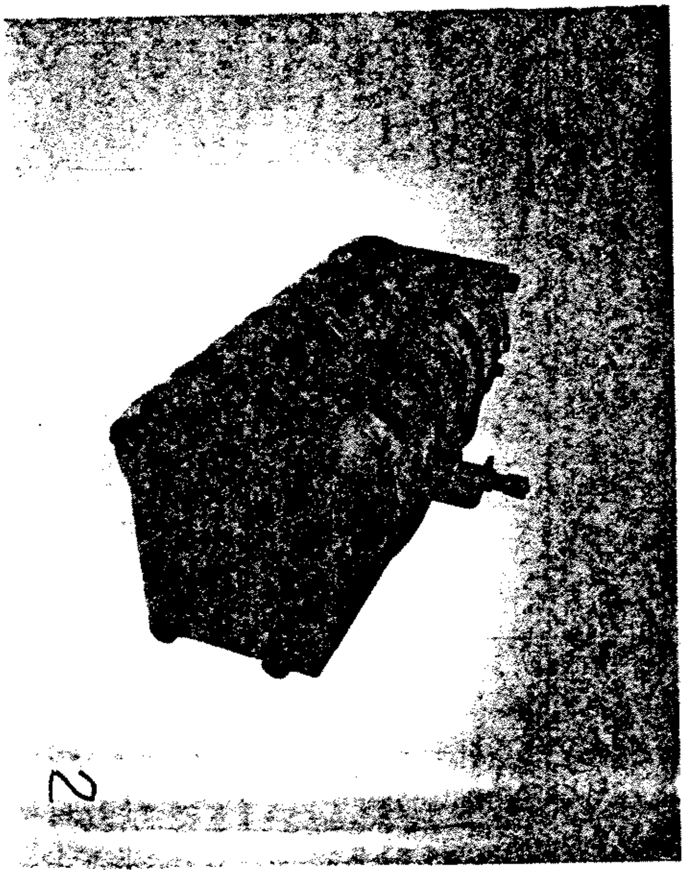
Three types of pressure guides were studied; one was a standard flat face block made from wear resistant composite material. The face was either aligned parallel to the blade-wheel line as in Figure 1A, or inclined as shown in Figure 1B. The other two guides were a one-shaft roller guide, shown in Figure 1C, and a two shaft guide, Figures 1D and 1E). The rollers on each shaft were spaced to allow them to intermesh (Figure 2). The two-shaft roller guide was specially designed to allow alignment of the bottom shaft using pneumatic actuators, applying a force of 83 N each, at the ends of the bottom shaft. After alignment the shaft ends were locked.

The guides were only studied at the upper location of the cutting span. The bottom guide, standard block/parallel face, remained unchanged throughout the experiment. All the guides were accurately aligned with a 9.5 mm offset from the wheels. The span length, between the inner edges of the guides, was kept constant at 792 mm, except for the one-shaft roller guide where the span was 843 mm, and the two-shaft roller guide (E) which had a span of 741 mm.

The vibration spectrum of the idling blade was investigated at 10 locations on the guide. Figure 3 shows the locations of the measuring probes and the probe holder is shown in Figure 4. The vibration spectrum was derived from Fourier transform of blade transverse displacement data. The vibration spectrum obtained on the guide was compared with the spectrum of the blade section above the guide and the spectrum of the unsupported part of the blade at location 6 (Figure 3), both obtained in idling; and with the spectrum of the cutting section of the blade obtained in both the stationary and dynamic condition through transfer function measurement.

The measuring procedure followed the sequence of guide changes from C to D, E, B and A. Measurements were taken after 15 minutes of warm-up period, except for the last guide which was run for eight hours and measurements taken every two hours. Measurements of face wear were also taken on guide A after eight hours of idling.


A
51
B
C
76
D
51
E



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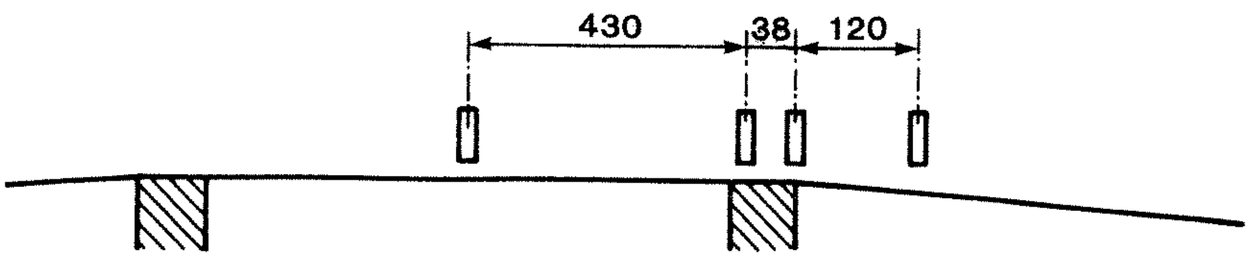
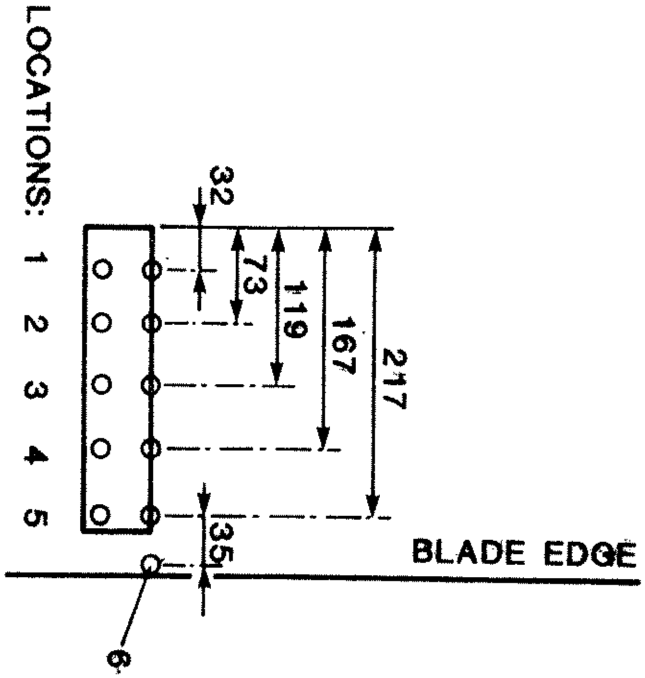

430
38
120


LOCATIONS: 1 2 3 4 5
32
73
119
167
217
35
6
BLADE EDGE


4

RESULTS AND DISCUSSION

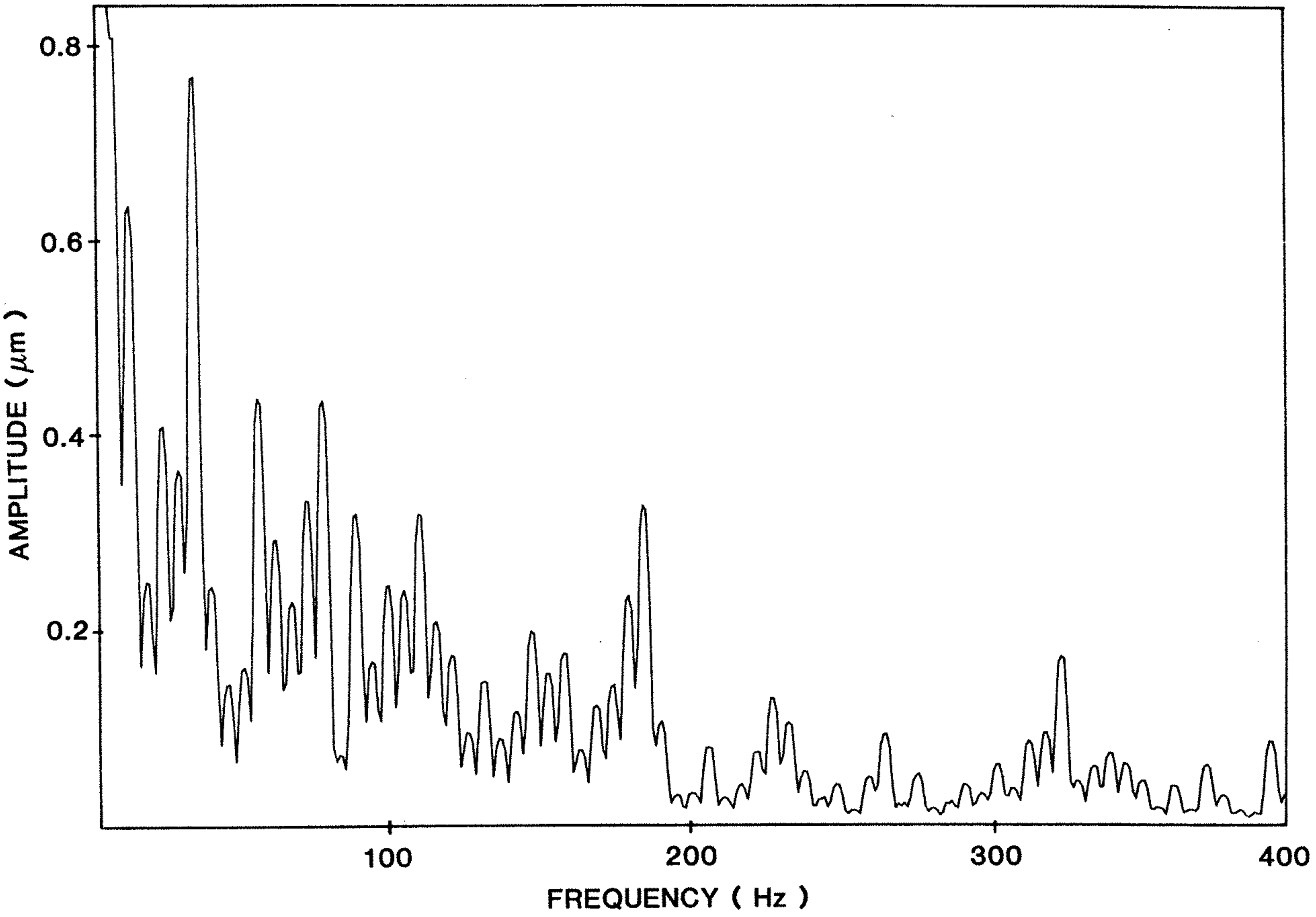
Figure 5 shows a typical frequency spectrum, in the 0 to 400 Hz range, obtained at one location of the guide face. Tables 1 and 2 show the eight largest amplitudes, in the 0 to 100 Hz range, measured from these spectra, for locations 1 and 2 respectively. Results for the remaining three locations are similar and are not shown.

Examination of frequency spectra, such as the one shown in Figure 5, reveals that all major amplitudes occur at frequency multiples of 5.45 and 10.83 Hz. These two frequencies correspond to the frequencies of blade convolution and wheel rotation, respectively. Harmonic vibration is also evident in the spectrum of the blade section above the guide, shown in Figure 6.

Natural frequencies of the blade section above the guide appear to be around 56 Hz in the lateral mode and 77 Hz in the torsional (Figure 6). Frequencies of the equivalent modes of vibration in the blade cutting section (shown in Table 3) are affected by the guide type. Transfer function measurements of the same section in the dynamic condition indicate that lateral frequency is reduced by about 6 Hz and torsional by about 7 - 9 Hz compared to the static condition. However, none of the natural frequencies can be identified in the frequency spectrum when the blade touches the guide.

Contact pressure between guide face and blade, at the upper probe, for inclined guide B is very small; the increase in amplitude in the 54 - 55 Hz range measured with the upper probe on the guide (Tables 1 and 2) suggests the influence of the blade's upper section which has a natural frequency in this frequency range. Similarly, for the one-shaft roller guide C the blade is not supported at the lower probe and an increase in amplitude takes place at frequencies close to the fundamental frequency of the blade cutting section. The large amplitude (5.7 μm) at a blade torsional frequency of 87 Hz, identified at location 6 (shown in Figure 7) had no effect on the blade amplitude at adjacent location 5 (where at frequency 87 Hz the amplitude was only 0.18 μm). The absence of blade natural frequencies on the guide face indicates that blade damping by the guide is very efficient. Harmonic oscilations due to wheel rotation are probably excited by vibration of the guide itself.

Comparison of Tables 1 and 2 showed that frequency spectra were affected by the location of the probes, however, differences between various guide types are not very large. After eight hours of idling, a reduction in amplitude was observed at 11 Hz for standard guide A. This is probably caused by wear of the guide face (shown in Figure 8). The two-shaft roller guide D exhibited similar frequency spectra to guide A. Amplitudes were only significantly reduced when roller guide E, with large distance between shafts, was used. The absence of vibration at 217 Hz, the rotational frequency of the rollers, was noted in spectra obtained with all roller guides. Substitution of roller guides for conventional guides is therefore possible. A comparison of measurements taken by the two probes at the



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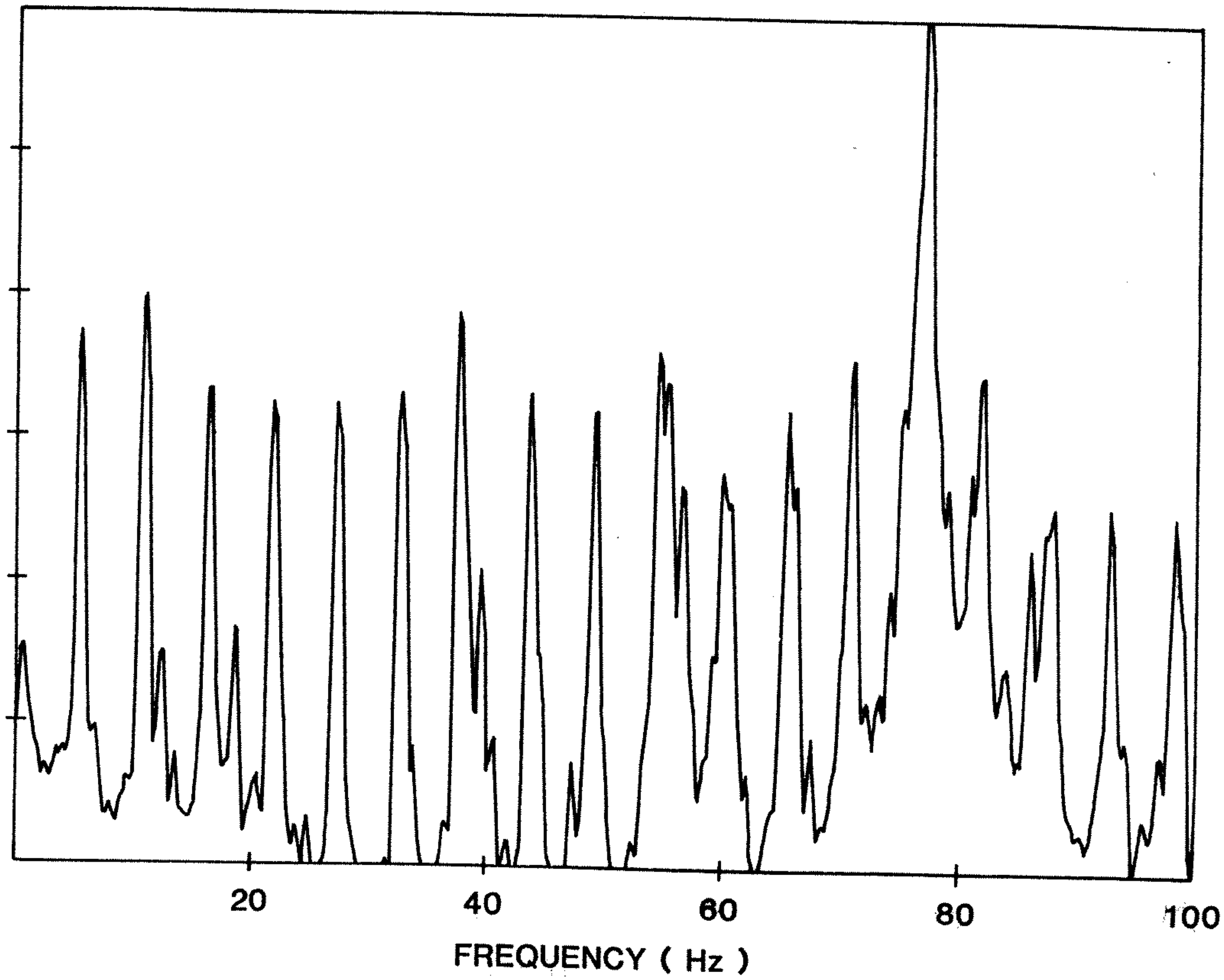

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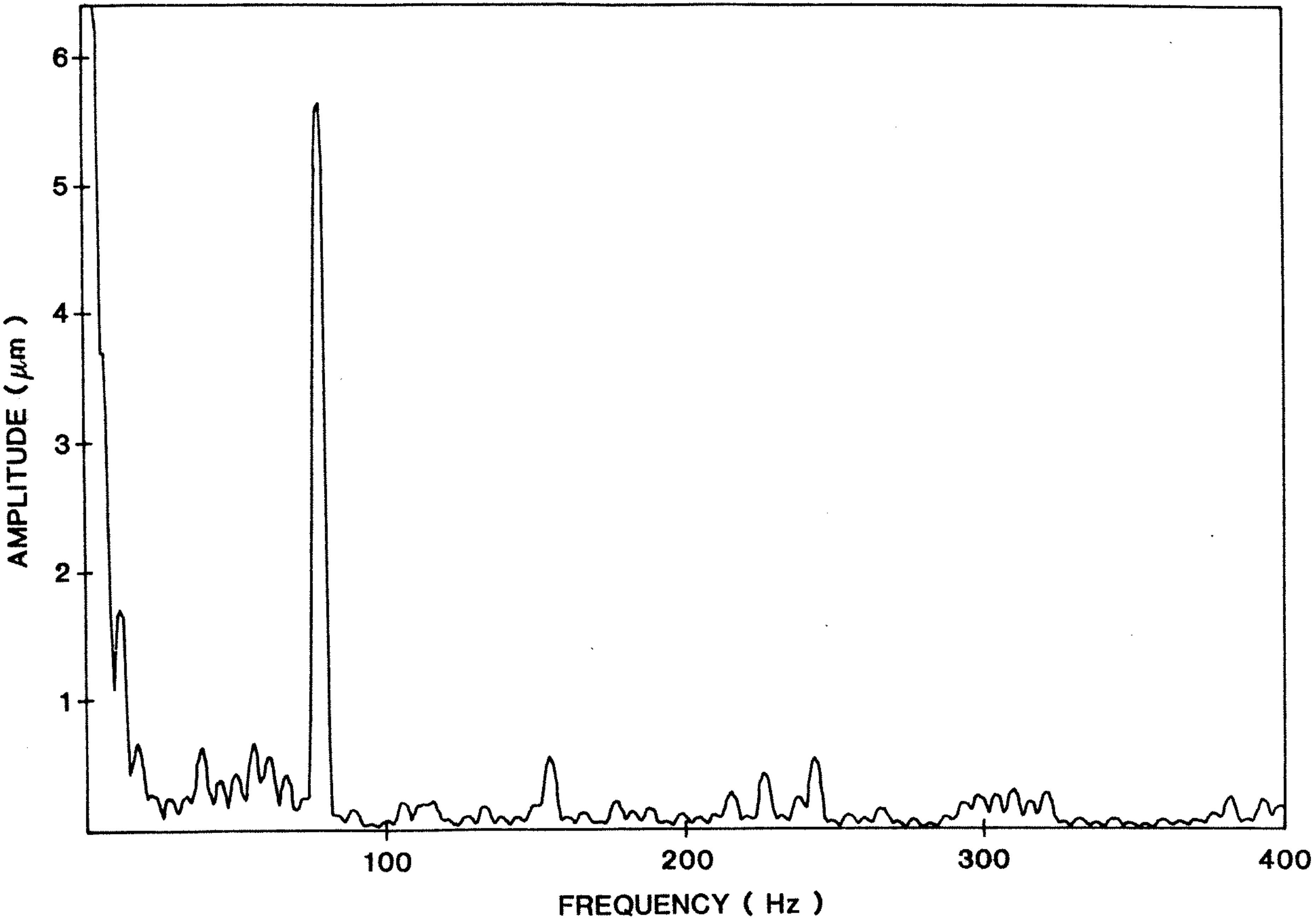
AMPLITUDE (μm)
6
5
4
3
2
1
FREQUENCY (Hz)
100
200
300
400

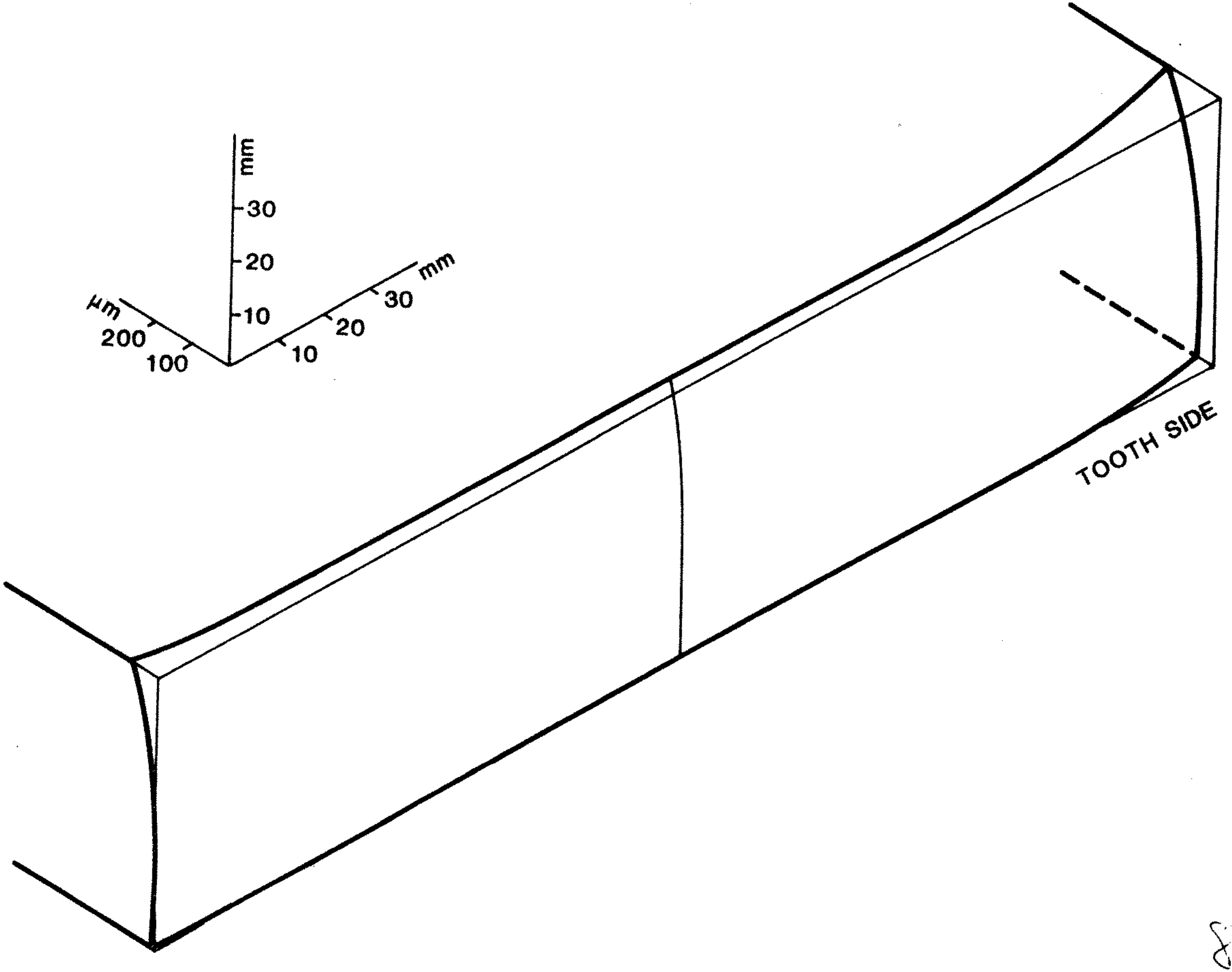

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Table 1

Band Saw Blade Amplitude (in μm) at Location 1 on Various Guide Types (Figures 1 and 3)

Frequency (Hz)	Guide Type					
	A	A (8-hr.)	B	C	D	E
Upper Probe						
6	0.16	.15	.21	.18	.12	.22
11	1.02	.43	.96	1.01	1.06	.18
16-17	.44	.38	.46	.47	.51	.41
22	.26	.17	.27	.32	.36	.16
33	.06	.07	.11	.06	.05	.13
54-55	.08	.04	.95	.06	.07	.03
65	.08	.10	.31	.11	.10	.09
75	.11	.29	.44	.21	.20	.05
Lower Probe						
6	0.17	.21	.31	.30	.18	.31
11	1.42	.63	1.31	1.52	1.48	.29
16	.67	.58	.68	.70	.74	.62
21-22	.44	.25	.43	.46	.52	.28
32-33	.16	.13	.09	.10	.08	.18
54-55	.12	.08	.14	.09	.09	.06
65-66	.14	.17	.32	.25	.15	.08
75	.19	.27	.25	.91	.30	.18

Table 2

Band Saw Blade Amplitude (in μm) at Location 2 on Various Guide Types (Figures 1 and 3)

Frequency (Hz)	Guide Type					
	A	A (8-hr.)	B	C	D	E
Upper Probe						
6	0.77	.61	.70	.60	.59	.41
11	.60	.37	.42	.61	.61	.12
16-17	.24	.27	.18	.27	.25	.39
22	.39	.30	.35	.34	.40	.32
33	.73	.60	.70	.60	.63	.47
54-55	.42	.30	1.31	.34	.39	.24
65	.22	.20	.08	.19	.22	.14
75	.41	.46	.45	.41	.38	.25
Lower Probe						
6	1.15	.92	.97	1.10	1.07	.07
11	.82	.56	.74	.77	.89	.11
16	.33	.42	.36	.33	.40	.67
21-22	.58	.45	.51	.50	.58	.47
32-33	1.12	.89	.90	.69	.90	.38
54-55	.65	.47	.52	.52	.57	.24
65-66	.32	.30	.23	.32	.32	.23
75	.67	.56	.68	1.18	.59	.18

Table 3

Fundamental Natural Frequencies (in Hz) of the Blade Cutting Section in Stationary Condition

	Guide Type					
	A	A (8-hr.)	B	C	D	E
Lateral	66.0	67.5	68.0	62.5	69.5	70.0
Torsional	87.5	88.7	87.0	81.5	86.5	91.5

same traverse location on the guide, show the lower probe to obtain generally larger amplitudes (Tables 1 and 2) for all guides, except guide E. The difference between probes probably indicates a pivotal displacement of the blade on the guide. Therefore, guide E should have the ability to improve the stiffness of the cutting section of the blade. This improvement could not be verified in present experiments since the blade cutting section, associated with this guide type, has un unchangeable span length.

CONCLUSIONS

The following conclusions can be drawn from the results of this study:

1. Band saw pressure guides dampen resonant vibration of the blade section above the guide and the blade cutting section very efficiently. Remaining dominant frequencies of the part of the blade in contact with the guide face are excited by wheel rotation and blade convolution.

2. Pressure guide design affects the damping efficiency. Standard sliding guides perform well when aligned parallel to the blade, and their performance improves with wear of the guide face. Roller guides compare well with sliding guides. The roller guides did not appear adversely affected by roller rotation. It appears that the two-shaft roller guide can provide an advantage particularly if the shafts are relatively wide apart.

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