Winder vibration can reduce operating efficiency and increase maintenance costs

David R. Roisum

*Specific vibration modes can be improved by altering the system's stiffness and mass or by reducing excitation from rotating rolls.**

All winders vibrate during operation. Though winder vibration is often innocuous, it can at times be serious enough to impair the efficiency of the winding operation. Excessive rewound roll vibration can cause production losses by limiting the speed of operation to avoid the risk of throwing a set. Additionally, slitter vibration or sheet flutter can damage the salable product by causing rough slitter cuts and web breaks. Vibration can also reduce machine component life by increasing dynamic loads on clutches, coupling, and bearings (*1-3*). For these reasons and others, vibration can be a costly problem that often demands diagnosis and remedy.

Winders are driven to vibrate in a complex manner by all of the rotating elements in hundreds of different modes. However, most of these modes will not be significant. Therefore, effective diagnosis of winder vibration must begin with a determination of which of these many modes is causing a problem. We must also identify which of the dozen rotating elements in the winder is behind the problem. In identifying the vibration, we must quantify the frequency, amplitude, and shape of the problem modes.

Once the problem is identified, proposals to improve the situation can be made. Such a proposal will entail a combination of (a) reducing the driving force, (b) changing the mass or stiffness of the system, and (c) increasing the damping. Since it is often difficult to make significant changes to the winding system without undue expense or without compromising functionality, it is important to thoroughly understand the problem and its parameters before making structural modifications.

Vibration basics

The simplest vibrating system is shown in **Fig. 1** for a single spring and a single mass. In this system, the mass oscillates periodically at a single natural frequency about its static rest position. The frequency, determined by the system's spring rate and mass, can be measured by a number of different types of analyzers. The amplitude of vibration can be given as a displacement, velocity, or acceleration quantity. Amplitudes can be specified as peak values or as peak-to-peak values.

If we were to excite a system near one of its natural frequencies, the amplitudes may increase to unacceptably large values, as shown in **Fig. 2** (*4*). The magnification of vibration increases as the ratio of the driving frequency to the natural frequency approaches unity. It also increases with decreasing values of damping. Winders have little damping, as do most other steel structures, which means that operating near resonance can cause severe vibration problems.

A slightly more complex system of spring and mass distribution for a simply supported beam is shown in **Fig. 3**, which shows how mode shapes for a beam element are usually represented. The beam's static shape is represented by the bold line segment between supports, and the dotted line indicates the dynamically deflected shape at one extreme of vibration. Only one arbitrarily selected extreme of oscillation is shown, for simplicity's sake, and the other extreme would be a mirror image through the static position. Also, the dynamic movement is magnified so that its character can be easily discerned.

Rolls on stands

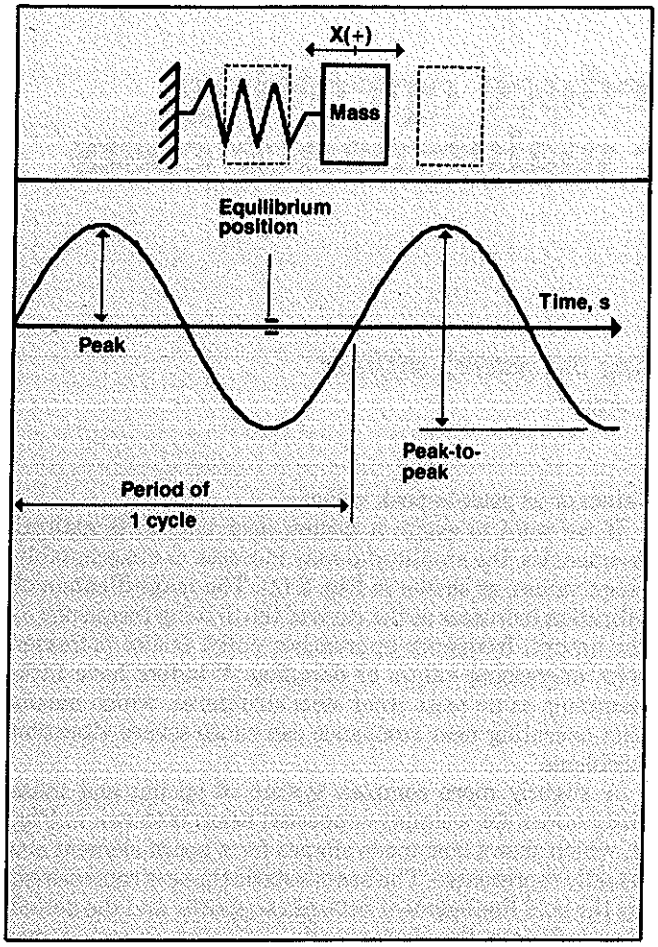
A roll or beam on a stand is a common dynamic system found on winders, and its dynamics can be used to describe the behavior of unwinds, guide rolls, spreader rolls, and other winder elements. As illustrated by the model in **Fig. 4**, mode shapes occur in pairs. The pair with the lowest possible frequency is characterized by a half-sine shape of the dynamically deflected roll (Modes A and C), followed by the pair of modes characterized by an "S"-shaped roll (Modes B and D). Within each mode pair, the mode of lowest frequency is characterized by motion in the direction of the greatest stand flexibility (Bending Modes A and B), followed by motion in the direction of greatest stand stiffness (Axial Modes C and D).

Unfortunately, one can seldom apply textbook formulas to accurately calculate the natural frequencies of even these simplest of dynamic systems. Modes C and D can be calculated from the simply supported beam formula provided that the journals are relatively short and stout and

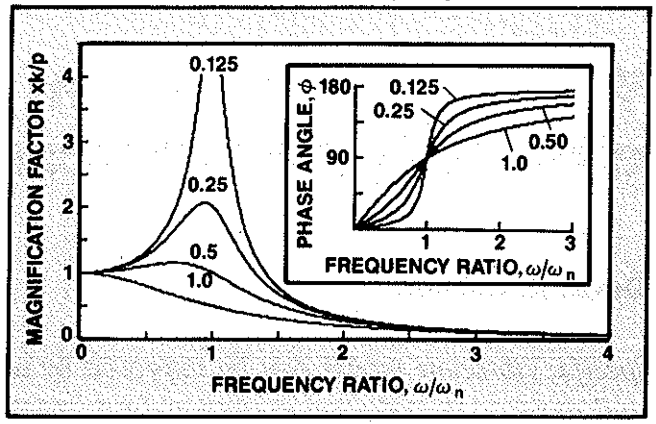
Roisum is manager of winding development, Beloit Corp., Research Center, 1165 Prairie Hill Rd., Rockton, Ill. 61072.

*Much of the material in this paper can be found in *The Winder Vibration Book*, written by D. R. Roisum and published by Beloit Corp. in 1985.

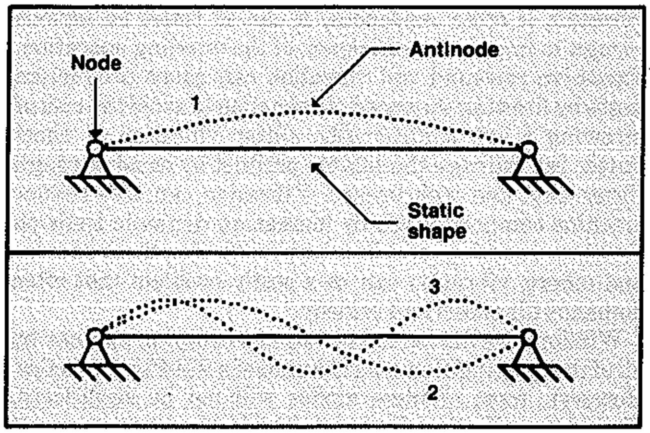
1. In a simple system, the mass oscillates at a single natural frequency.



2. In an excited system, the magnification of vibration increases as the ratio approaches 1.0, where the ratio is that of the frequency caused by excitation to the natural frequency.



3. Oscillations in a more complex system.



provided that the foundation is relatively stiff (*5*). However, the calculation for Mode A, which is often the most important mode, is a complicated expression that does not always model real systems well. As a consequence, nearly all winder vibration problems must be diagnosed with more sophisticated tools such as Fast Fourier Transform (FFT) vibration analyzers and finite element analysis.

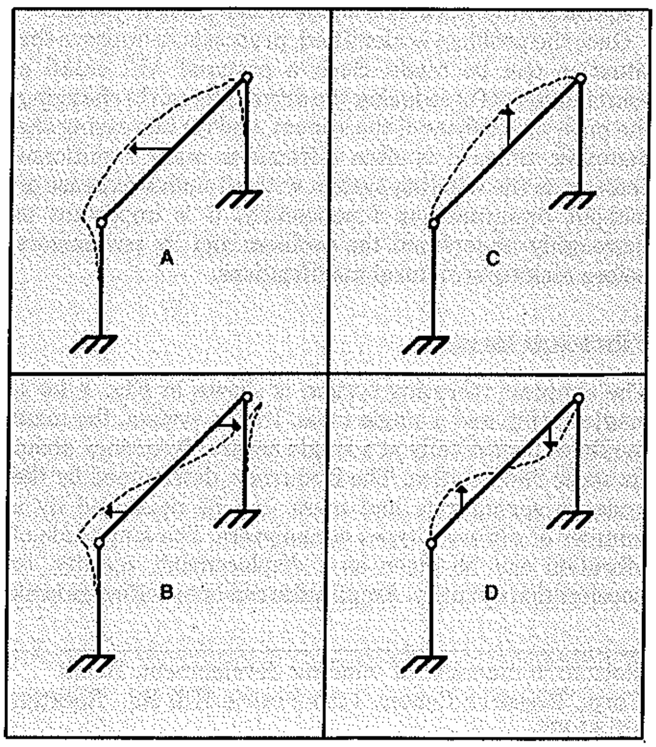
Roll imbalance

Roll imbalances produce most vibrations. A roll imbalance produces a once-per-revolution sinusoidal disturbance that can excite a resonance if the roll frequency equals a system natural frequency. To calculate the imbalance force, we need to know either (a) the roll eccentricity, which is the distance between the center of gravity of the roll and its axis of rotation, or (b) its static imbalance. Knowing either of these quantities will allow us to calculate the amplitude of the disturbing force caused by the roll imbalance.

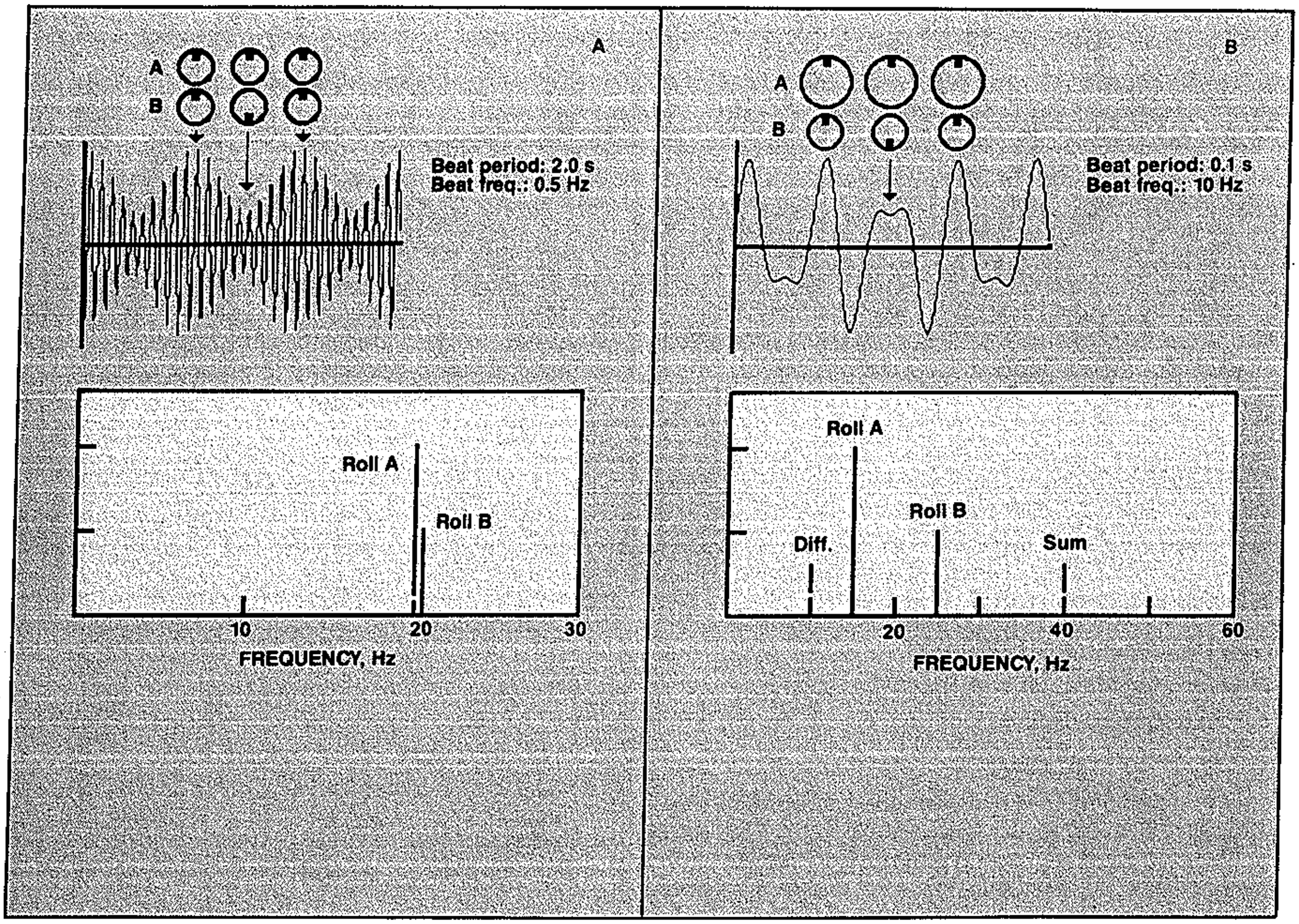
Unfortunately, neither eccentricity nor imbalance is easy to measure. Instead most shops balance rolls to dynamic total indicated runout (TIR), which is an indication of imbalance severity but is not directly related to eccentricity. TIR specifications on high-speed winder rolls are a few thousandths of an inch.

For dynamic analysis however, we need to put a value on eccentricity or imbalance to describe the forcing function. Since eccentricity is difficult to measure, we can use a

4. Mode shapes occur in pairs. Modes A and B are in the machine direction, whereas Modes C and D are in the vertical direction.



5. Beating of rolls of similar diameter (left) and of different diameters (right).



conservative estimate of ¼ of the static runout. With this estimate, we can calculate the static imbalance:

$$\text{static imbalance (in.}\cdot\text{lb)} = w\, e$$

and

$$\text{force (lb)} = 0.0004\, w\, e\, v^2/D^2 \qquad (1)$$

where

w = weight of roll, lb

e = eccentricity (in.) = TIR/4

v = speed, ft/min

D = o.d. of roll, in.

The imbalance force increases with the square of the speed, increases proportionally with eccentricity, and decreases with the square of the roll diameter. This means that small-diameter, high-speed winder rolls can be the most difficult to balance.

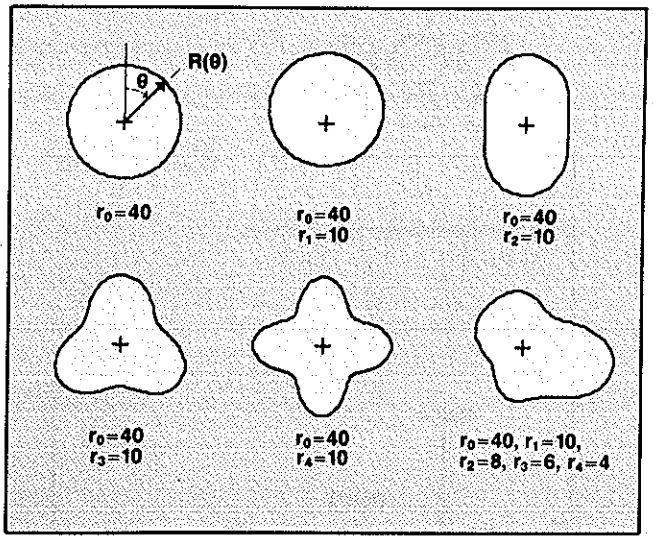
The "half-critical" of a roll (often called the "first critical") is a phenomenon that occurs in a roll whose cross section does not have perfect radial symmetry. Half criticals exist but are seldom a real problem because they are often barely detectable with instruments. Rolls supplied today are bored and turned precisely enough to nearly eliminate the problem of half-criticals.

Roll beating occurs whenever two or more rolls are rotating within dynamic vicinity of each other. Beating is caused when the imbalances of the rolls align so that the forces are in phase, resulting in a high vibration amplitude. After a time, the roll imbalances will drift out of phase, such that forces oppose each other and result in a reduced vibration amplitude. The imbalances go in and out of phase in a sinusoidal manner as a result of differences in their rotational speeds. For rolls of the same nominal diameter, the differences in rotational speeds will be small, and the beat frequency will be only a couple of cycles per minute. If the rolls are of different nominal diameter, the beat frequency will be much higher. Beating may be significant because the maximum vibration seen on a structure will occur when the imbalances of adjacent rolls nearly align.

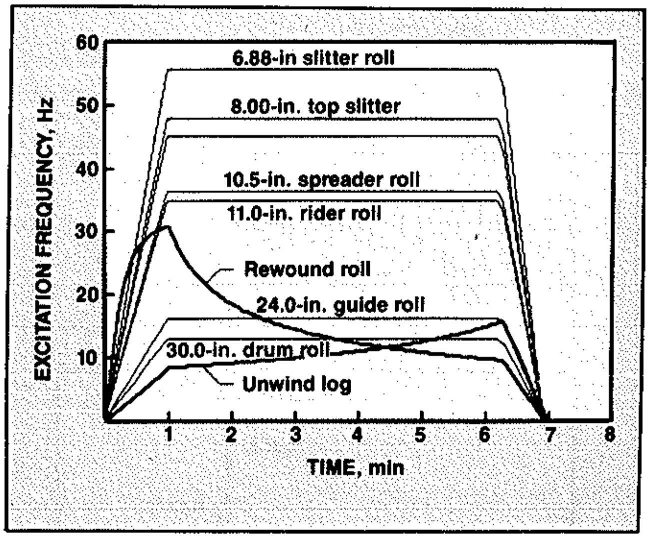
For two rolls of the same nominal diameter, the forcing function experienced and the vibration amplitude vary sinusoidally with a frequency equal to the beat frequency. For example, Roll A turning at 20.0 Hz has an imbalance force of 10 lb. Roll B turning at 20.5 Hz has an imbalance force of 5 lb. As illustrated in **Fig. 5A**, the rolls will beat at 0.5 Hz (20.5 Hz – 20.0 Hz) with a maximum force of 15 lb (10 lb + 5 lb), a mean force of 10 lb, and a minimum force of 5 lb (10 lb – 5 lb).

For two rolls of nominally different diameters, an analyzer will give us the frequencies of both rolls plus the sum and difference frequencies. In **Fig. 5B**, for example, the

6. Elemental components of roll sidedness.



7. Excitation frequencies during the winding cycle.



following frequencies will be detected:

1. Roll A's fundamental of 15 Hz
2. Roll B's fundamental of 25 Hz
3. Sum frequency of 40 Hz
4. Beat frequency of 10 Hz.

As the number of rolls increases, the number of sum and difference frequencies increases dramatically. Fortunately, the amplitude of these frequencies is usually much smaller than the fundamentals and can be neglected. However, vibration problem solving requires us to discriminate the potentially serious fundamental frequencies from the innocuous beat frequencies.

Roll shape

Many vibration problems can be attributed to a paper roll that is not concentric. The vibration of a winding set, particularly, is usually caused by a noncircular, nonconcentric shape in the paper roll. Runouts for rolls vary from less than 0.002 in. for steel rolls to about 0.005 in. for rubber-covered spreader rolls to more than 0.250 in. for a roll of paper. These runouts cause movement similar to a cam shaft moving an engine valve.

To determine dynamic forces caused by nonconcentric rolls, we must be able to accurately describe the shape of a roll. Because the outside diameter of a roll is repetitive in a progression around the surface, its periodic nature can be described with the Fourier series expansion, as follows:

$$r_0 + r_1 \cos(\theta + \theta_1) + r_2 \cos(2\theta + \theta_2) + \ldots R(\theta) = r_0 + \sum_{i=1}^{n} r_i \cos(i\theta + \theta_i) \qquad (2)$$

where

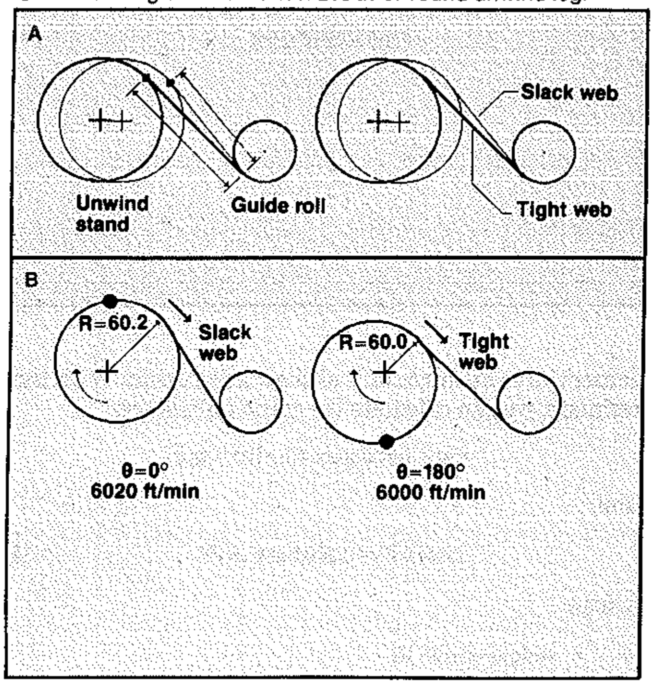
$R(\theta)$ = radius at rotation location θ

r_0 = average roll radius

θ = the rotational location

i = the sidedness number

8. **A:**Vibrating unwind stands. **B:**Out-of-round unwind log.



r_i = the ith contribution to sidedness

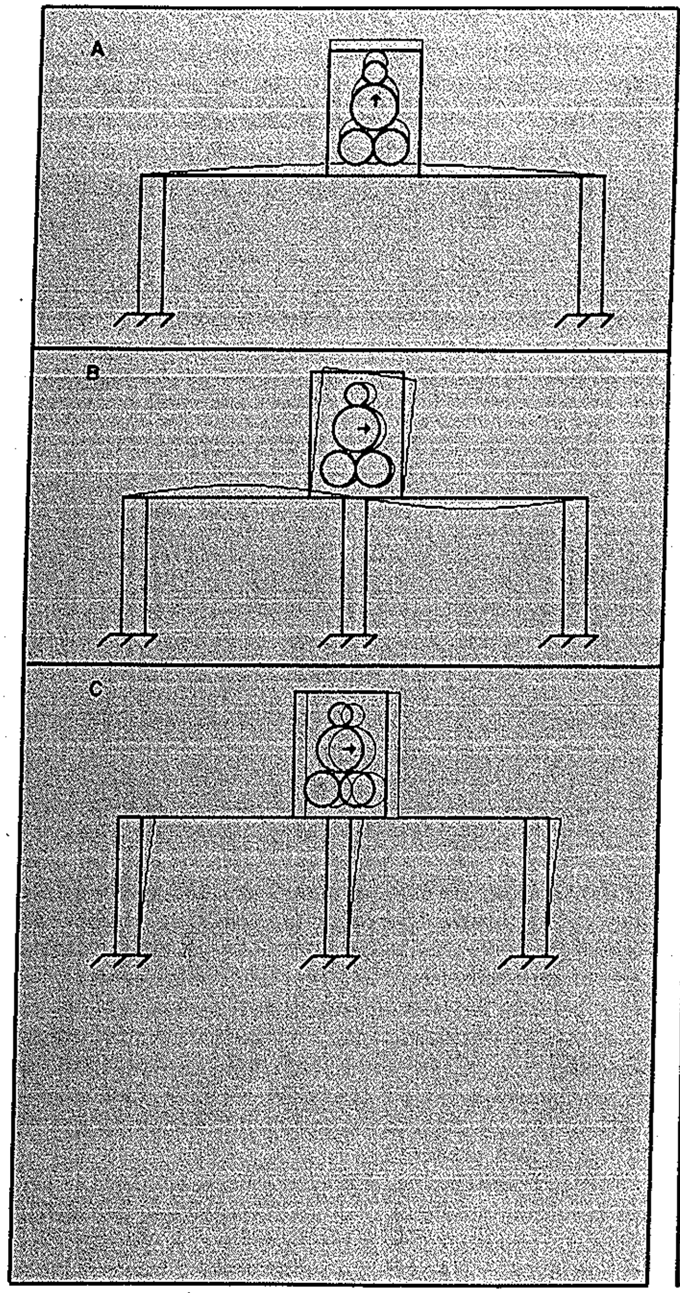
θ_i = the phase relationship of sidedness.

With this expansion, we are able to describe exactly the shape of a roll composed of a superposition of several elemental components of sidedness as illustrated in **Fig. 6.** Roll shape can be measured during winding or unwinding with noncontacting optical displacement transducers and after winding with a roll periphery tracing device called a "Radiavarius."

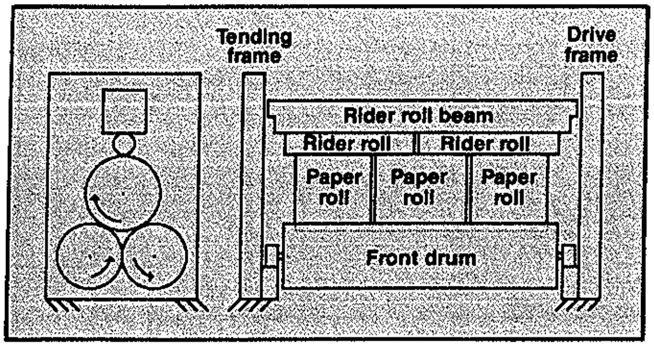
Excitation frequencies

In a simple winding cycle, the winder accelerates, runs at

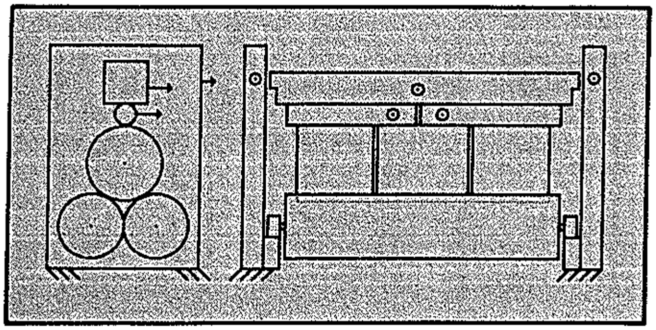
9. Winder vibration showing (A) bending, (B) twisting, and (C) shearing.



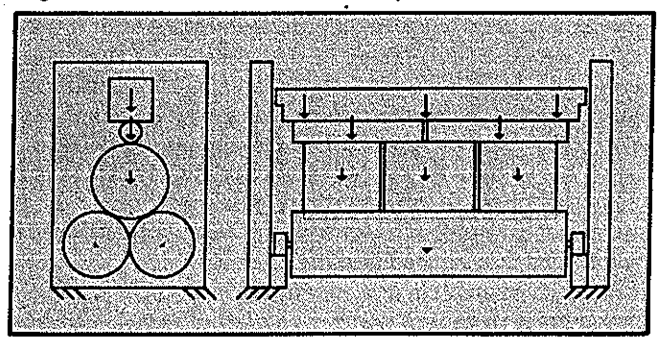
10. Normal operation with rigid body rotation of the rider, wound, and drum rolls about their longitudinal axes. (In the figures that follow, a circle with a dot indicates motion toward, while a circle with an "X" indicates motion away from.)



11. Seldom a problem, the tending- and drive-side framework bends in the machine direction as a cantilever, and the rider roll beam bends in the machine direction as a simply supported beam.



12. One of the worst winding vibration problems, the rider roll beam translates vertically with little bending, and the drums bend slightly as simply supported beams, with the rewound rolls following the vertical motions of the overall system.



constant speed, and decelerates to a stop. In this process, the various rotating elements in the winder will create excitation forces in the frequency range of 0–80 Hz. The largest forces come from the rewind roll and unwind log at frequencies up to 40 Hz, and the smaller aluminum rolls can generate smaller forces up to 80 Hz for an 8500-ft/min winder. Additionally, sided paper rolls and misaligned or loosely mounted steel rolls can generate integral multiples of the fundamental frequencies.

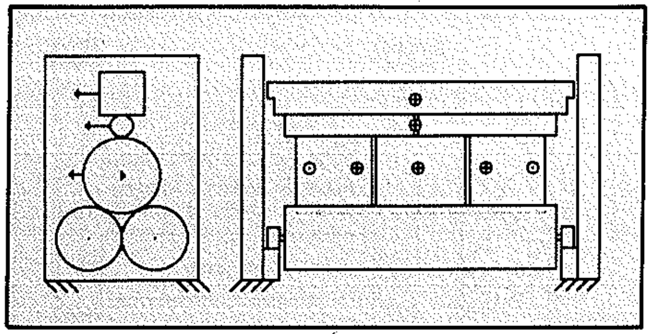
The potential for winder vibration is aggravated by the large range of forcing frequencies of the rewound roll set. As **Fig. 7** shows, the fundamental rotational speed of rolls on a 6000-ft/min winder sweeps a range from 0 Hz up to 60 Hz. Therefore, any of the numerous winder system resonances in this range are eligible for excitation.

Unwinds and sheet flutter

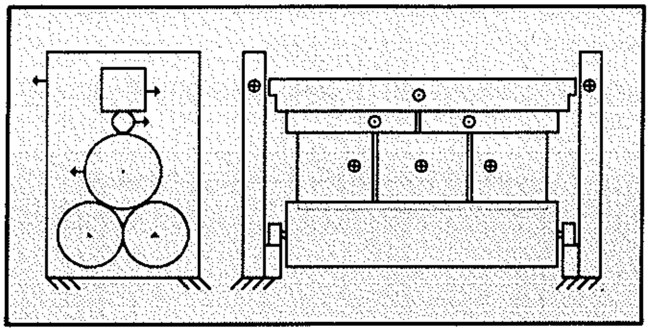
The unwind is the single largest contributor to web instabilities such as sheet flutter and tension fluctuations. Resulting problems often include snapoffs, rough slitter cuts, ragged roll edges, and the need to limit speed. These web instabilities are caused by two principal mechanisms. First, if the unwind stand vibrates, the vibration will be transmitted into the sheet run through oscillation of the unwinding tangent, as shown in **Fig. 8A.** Second, if the unwind log is out-of-round, the surface speed at the unwinding tangent will oscillate, as shown in **Fig. 8B.**

The first requirement of a stable sheet run is to reduce unwind stand vibration. We can reduce the driving force by having a centrically structured unwind log and reel spools

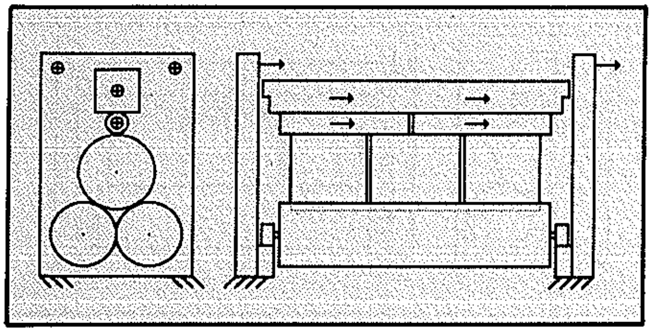
13. Not usually a problem, the rider roll beam bends in the machine direction as a simply supported beam, and motions of the wound rolls take on a shape similar to that of the rider roll except at a smaller magnitude.



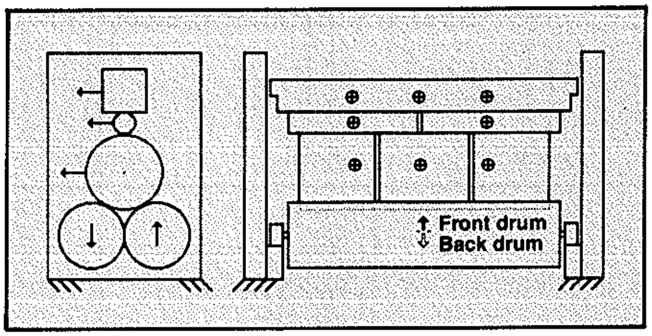
14. Similar to that of Fig. 13 except that frame bending is greater, wound rolls begin rocking from drum to drum with corresponding drum bending. Similar motions occur on the cross shaft, cross ties, and the ejector beam.



15. Not likely to be a problem, the framework bends in the cross-machine direction as a cantilever beam, and the rider roll beam follows the frame motion. But the CD forcing functions are too small to excite.



16. A serious problem mode, the rider roll beam bends as a simply supported beam in the machine direction, the wound rolls rock in unison from one drum to the other, and the drums bend vertically and out of phase.



that are balanced and that have journals which are in good condition. Additionally, the unwind system must be stiff, with the reel spools held securely in rugged stands, and it must be supported by a substantial foundation.

The second requirement of a stable sheet run is to maintain a constant surface speed during a revolution of the unwind log. This means that the log must be round and concentric with the spool axis. If the unwind log is out-of-round, the web surface speed is less at a small radius than when the unwind tangent is at a larger radius. This difference causes strain changes and tension fluctuations that will be transmitted into the sheet run.

To make concentric and round unwind logs at the reel, the log must be well structured using the tension, loading, and centerwind torque parameters. To keep a log concentric from the time it is wound until it is ready for unwinding involves good reel handling practices. The logs should not be set on the floor, bumped against other logs, or loaded roughly into the unwind stand. Also, if the logs are going to sit for any length of time, a slow-speed drive could be used to slowly rotate the log to keep it from sagging. Additional requirements for a stable sheet run include a well tuned tension system and drive and short draw lengths.

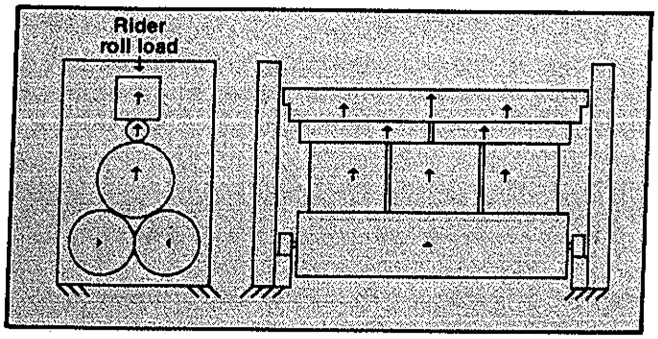
The foundation

The winder foundation is an important element of the dynamic system and must be absolutely solid under all conditions of winder operation. The evaluation of an existing foundation is easy. If foundation vibration can be sensed without instruments (which is about 0.5 mils peak-peak at the base plates), the foundation may be inadequate. However, we would like to develop the ability to predict the necessary foundation design parameters to ensure stable performance. Unfortunately, the traditional criteria for paper industry foundation design based on load capacity or stress has given rise to many winders with inadequate foundations. Winders run at much higher speeds than the paper machines. As a consequence, winder foundations must also be designed based on dynamic considerations as well as on load capacity.

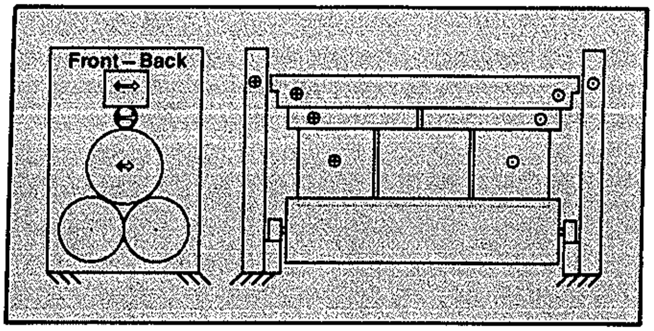
Figure 9 shows several typical foundation designs and the resulting problems. Figure 9A shows a foundation bending when the winder is positioned over an unsupported section of flooring and is excited by vertical forces. Figure 9B shows a better and more typical design, where the winder is supported directly underneath by columns. However, the "S"-shaped twisting of the floor is easily excited by horizontal forces and can also be a problem. Figure 9C shows a foundation which has been isolated by expansion joints, giving rise to shearing. These typical winder foundation geometries are inadequate for vibration-prone grades.

Ideally, the winder foundation would not be excited to resonance by any of the rotating elements. Unfortunately, a foundation that is soft enough to reduce the frequencies of major system resonances to below the rotational frequency of a large paper roll at moderate speeds (about 5 Hz) is too flexible to be practical. Conversely, even an infinitely stiff foundation will not increase the lowest frequency of wide winder system resonances above the rotational frequency of a small paper roll at high speeds

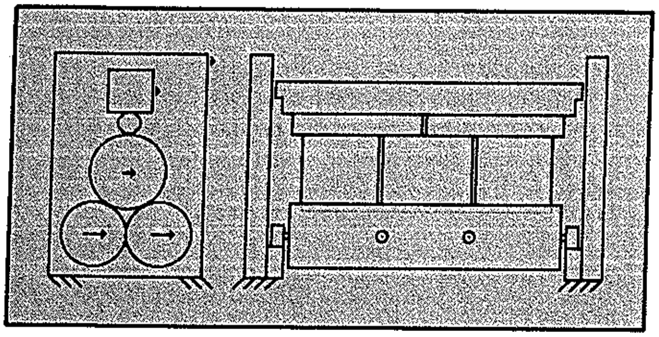
17. A potentially serious mode, the rewound roll and the rider roll assembly move in phase with each other but out of phase with the rider roll loading system. The drum roll bends slightly.



18. Not a potential problem, the front and back framework bend out of phase with each other in the machine direction as cantilevers, and the rider roll and the wound rolls follow. Not easily excited.



19. A serious problem, the drum rolls bend in phase in the machine direction. There are smaller movements on the rider and rewound rolls.



(about 40 Hz). Consequently, winder resonances are inevitable.

How then do we minimize vibration through foundation design if resonance is inevitable? First, we can make the foundation stiff to reduce the number of excitable resonances. Second, we can make the foundation massive to reduce the amplitude of vibration of the remaining inevitable resonances. These two criteria indicate that a well designed foundation is one where the winder is supported by two continuous walls located directly under the base plates and which span from the unwind to the front drum. These walls are relatively inexpensive and ensure that the kind of bending, twisting, and shearing illustrated in Fig. 9 can't easily be excited. Additionally, the mass and stiffness of the continuous wall support ensure a minimum number of winder resonances along with minimum severity.

Framework

Framework design also has a major influence on vibration magnitudes. From a simplified point of view, an ideal framework is just rigid enough so that dynamic deflections are small at all roll bearing housings or other key locations. Larger deflections can be tolerated on noncritical elements such as cross ties and roll ejector beams without compromising the efficiency or productivity of the winder.

Winder frameworks are structurally complex and vary considerably between installations, making analysis and evaluation difficult. However there are a number of considerations that apply to most machines. Just as with foundations, it is not possible to eliminate all resonances. The framework should be simple, stiff, and massive.

Clearances must be kept to an absolute minimum on sliding bearings such as those found on the core chucks and rider roll sides, because vibration amplitudes almost always exceed the clearances. Minimum clearance is also necessary to eliminate the vicious circle where clearances increase vibration, which further increases clearances, and so on until a machine that once ran smooth becomes a maintenance problem.

Finally, platelike behavior must be avoided at all joints and attachment points. If a bearing or beam is attached to an unsupported thin plate area of a framework, the joint will easily bend. This reduction in system stiffness reduces natural frequencies and increases vibration amplitudes.

Slitters

Slitter vibration should be minimized to increase cut quality, slitter blade life, and operational efficiency. Symptoms of excessive slitter vibration include rings on the paper rolls, frequent web breaks, and short life between blade grindings. The source of slitter vibration can be either externally generated by the winding set or internally generated by the slitter blades themselves.

Externally generated vibration caused by a rough winding set can be transmitted through an insubstantial foundation to the slitter section. Slitter beams are easy to excite because they often have resonances in the typical rotational frequency range of the winding paper rolls. Reducing externally generated forces calls for balancing of all winder rolls. Any remaining forcing effects can be minimized by isolating the slitter section from the winder and increasing the mass and stiffness of the foundation. Additionally, the strategic bracing of slitter beams at or near antinodes and an increase in the beam's mass will reduce the amplitude of vibration.

Internally generated vibration caused by the runout of the blades and bands causes a displacement forcing function equal in magnitude to the sum of the runouts of the top and bottom blades. Consequently, slitter blades and bands must be ground to tight runout tolerances. Additionally, slitter mountings must be stiff and clamped to the linear ways without clearance or play.

Winding set vibration

Winding set vibration is often a severe problem that limits

Vibration terminology

Amplitude is the magnitude or extent of oscillation of a body from one extreme to its neutral position (peak) or to its opposite extreme (peak to peak). Amplitude is a measure of the severity of vibration and can be expressed in three interrelated quantities:

- **Displacement** is the amplitude of the position changes measured in thousandths of an inch or mils (0.001 in. = 1 mil).
- **Velocity** is the amplitude of the speed changes measured in units of in./s.
- **Acceleration** is the amplitude of the time rate of change of velocity measured in g force ($1\ g = 386$ in./s^2).

Units are selected for vibration analysis on the basis of the type of vibration equipment used, the type of analysis, and personal preferences. Acceleration, though used only occasionally, is most useful when dynamic forces are to be determined. Velocity is the most common unit of measure because it gives a direct indication of severity, which is independent of frequency. Displacement yields information on the extent of motion. Most vibration analyzers are capable of displaying any of the three units of measure.

Antinodes are positions of maximum movement on a structure when excited at a particular natural frequency.

Boundary conditions are constraints applied to points on a structure that can be either fixed (no movement) or free (no external restraint). Boundary conditions for a point must be specified for each of the six kinds of displacements: three translations and three rotations. Boundary conditions have a tremendous effect on natural frequencies. For this reason, the designer should make efforts to maximize fixation at supports. Additionally, play and clearance in a structure, often neglected in analysis, can have a deadly effect on vibration by reducing the natural frequencies and increasing the severity of vibration. Amplitudes are nearly guaranteed to exceed any clearances already present.

Damping (or dampening) is the resistance to motion which dissipates vibratory energy. Damping is the mechanism that keeps machinery from literally flying apart when it is operating at resonance. There are a number of different types of damping, but the most common is viscous damping, in which the resistant force is proportional to the speed. As Fig. 2 shows, vibration would be infinite at resonance without damping. Damping can be measured by observing the rate of vibration decay after the structure is struck. Most metal structures are lightly damped.

Forced vibration occurs when a structure is driven by a periodically varying force such as an unbalanced roll, and the roll will continue to oscillate as long as the varying force remains. Forced vibration caused by rotating elements is the most common cause of winder vibration problems.

Forcing function is a broad term describing the cause of vibration in terms of amplitude, frequency, position, and direction. A forcing function can be displacement disturbance such as a rotating cam shaft moving a valve or a forced disturbance such as a roll imbalance. In many cases, winder vibration is caused by a rotating roll imbalance, with the energy being supplied by the electric drive motors.

Free vibration occurs when a body continues to oscillate at its natural frequencies with no outside forces to drive it, like a bell after it is struck or a guitar string after it is plucked, and the vibration will die after a time as a result of damping. The "thunk" or "rap" test utilizes free vibrations to determine natural frequencies.

Frequency is the rate at which a body moves from one extreme to the other and back again and is commonly measured in Hertz (or cycles per second) or in radians/s. ($1\ \text{Hz} = 2\pi = $ radians/s) The usual frequency range of interest for winder mechanical vibration is 5–70 Hz and for noise is 20–20,000 Hz.

Mode shape (or Eigenvector) is the dynamic shape of a structure at a particular natural frequency. Each

speed and causes eccentrically wound rolls. The source of the vibration is almost inevitably the out-of-roundness of the paper roll. Though this runout may typically be only 0.020 in., it is greater than the runouts of the drums or rider roll. It is the cam action of the out-of-round paper roll running on the winder drums that can generate tons of excitation force.

The problem is extremely dependent on grade. Rough or bulky grades such as kraft, book, tissue, and carbonizing papers result in vibration amplitudes that are considerably greater than those that occur with other grades such as newsprint and linerboard. These grades have high paper-paper friction coefficients that lock in deformations to the paper roll's surface caused by inevitable disturbances (*6*).

The extremely large frequency range means that many winder resonances are inevitable at some time during the winding cycle. Extensive finite element studies have indicated that winders of production width, regardless of configuration, framework, foundation, or manufacturer, will have several resonances less than 30 Hz. Changes in design can reduce the number of resonances and their severity, but they are inevitable because the paper roll itself is structurally soft compared with the other winder machine elements.

Fortunately, not all resonances are easily excitable. For resonance to be severe, the forcing function must be large enough, oriented in the direction of that mode, and located near an antinode. Since most of the force generated by the paper roll is directed in the vertical and machine directions, cross-direction modes are seldom a problem.

The results of finite element analysis on a 300-in.-wide, two-drum winder, with 30-in.-diameter drums are prenatural frequency has its own mode shape determined in a complex manner by the mass and stiffness distribution of the system. Mode shape information is invaluable because it tells us how we may increase the natural frequencies for a particular mode. This may be done by removing mass at or near antinodes and by increasing stiffness on components that experience large relative dynamic deflections.

Natural frequency (or Eigenvalue) is the frequency at which a structure tends to vibrate. Examples are the pitch (frequency) at which a bell rings and a guitar string that vibrates when it is struck. Any structure possesses an infinite number of natural frequencies, each with its own characteristic mode shape. However, as a practical matter, usually only the lowest half dozen modes or frequencies are of significance.

Nodes are the points on a structure which do not move when excited at a particular natural frequency.

Resonance is a condition in which the frequency of the forcing function nearly equals one of the natural frequencies of a structure. Resonance can often be a serious problem because the amplitude or severity can be magnified as much as 10 times when approaching this critical condition, which is often called the "critical speed." The severity depends upon many factors:

1. The frequency ratio is equal to the frequency of the forcing function divided by the natural frequency. The closer this ratio gets to unity, the worse the vibration becomes.
2. The severity increases somewhat linearly with increasing magnitude of the forcing function.
3. Severity increases with decreasing damping.

sented in **Figs. 10-19** (*7*). The relative amplitudes of vibration of the areas of major movement are shown as arrows. Several innocuous modes involving cross ties and ejector beams have been omitted for brevity. The most serious modes are the vertical and rocking modes, which are easily excited by the winding set. Unfortunately, there is presently no known solution to eliminating these vibration modes.

Vibration measurement

Field measurements are the only reliable means of quantifying winder vibration amplitudes, which are a measure of the severity of a problem. Measurements of frequency can usually determine which rotating element is responsible for a specific vibration mode. Additionally, field measurements are needed to verify mode shapes derived from finite element modeling (FEM) and to adjust FEM models.

Serious field studies require a FFT vibration analyzer. This instrument is composed of a pickup and electronics to process the vibration wave forms. The pickup is usually a magnetically affixed accelerometer. An accelerometer usually reads vibration in one direction only, so it must often be reoriented to map out all motion at a single point. The FFT converts the signal from the pickup of amplitude vs. time into a plot on a video display of amplitude vs. frequency. Auxiliary equipment often includes tape recorders for storing data, plotters for displaying data, and shakers to excite a structure.

Before the equipment is set up, a background study is required to help narrow the search. Observations from winder operators can be invaluable in determining where the vibration manifests itself and under what conditions. Though they may lack technical training, the operators live with the winder daily and usually know more about the problem than anyone else. Besides consulting with the operators, one should make a survey of all safely accessible bearing housings and the framework to determine vibration severity by touch.

There are three typical approaches for field measurements. The first is a "sweep test," in which the operators run the winder for a full set under normal conditions. The FFT is set in the peak mode to determine the severest amplitudes, frequencies, and locations of vibrations. The second is a static "rap test" to confirm resonant frequencies. The third is the "resonance" test, in which the operator runs at resonance for as long as possible. The frequency of vibration will be an integral multiple of the offending roll's rotational frequency. The runout and balance of that roll should be checked.

In some cases, it is possible to do mode shape analysis with a single pickup if the vibration is consistent. This analysis is carried out by measuring the vibration amplitude along the structure at the offending frequency and then plotting the scaled readings on the scaled isometric sketch of the winder.

Finite element

FEM is a powerful numerical technique used to solve simultaneously the many differential equations needed to describe a physical system. The analysis requires highly technical and experienced modelers, large computers, expensive software, and usually a considerable amount of time. Often, the results need confirmation by actual measurements.

However, the advantages are numerous. The FEM output can include static and dynamic deflections, mode shapes, natural frequencies, forces, and stresses at any location in the structure. Post-processors can pictorially display any of the outputs to give a more intuitive understanding of the problem than computer printouts. Once the model is generated, potential solutions can be tested on computer. This evaluation of several alternative solutions can often be a quicker and less expensive approach than modifying the machine itself.

Evaluation of vibration problems

The first step in evaluating a winder vibration problem is to carefully define the detrimental effects in either reduced productivity or increased operating costs. It is important to determine whether excessive vibration is a major parameter or if it is merely a minor contributor to the problem. High vibration in itself does not necessarily imply a problem, and the real problem often lies elsewhere.

The next step is to specifically identify which of the numerous modes is causing the problem. This can be done with field measurements of vibration amplitude after narrowing the search by gathering observations from the operator and making a survey by touch. Once the mode shape has been identified, it can be altered by increasing the stiffness of structural elements which undergo large movements. Gussets strategically connected from near the antinode of the moving element to a nearby solid member can increase the effective stiffness.

The last step is to identify which specific rotating element is causing the problem. If that roll is straightened, stiffened, or balanced, the problem will be reduced. Also, if the path of transmission of the vibration from the rotating element to the excited element is interrupted or stiffened, the problem will be reduced.

Conclusions

Winder vibration is a complicated phenomenon that often requires considerable effort to diagnose. However, with the proper tools and approach, one can usually identify the specific problem modes and their causes. Once offending modes have been identified, these can often be improved by altering the system's stiffness and mass or by excitation from rotating rolls. Though winder vibration analysis can be difficult, the advantages are a smoother and more efficient finishing room operation.

Literature cited

1. Hines, G., Predicting failures with vibration analysis, Instruments and Control Systems, 1978.
2. McLain, D. A., Pulp & Paper 55(2):118 (1981).
3. Thorp, B., PIMA 65(7):27 (1984).
4. Timoshenko, S., Vibration problems in engineering, Wiley, New York, 1974.
5. Blevins, R., Formulas for natural frequency and mode shape, Van Nostrand Reinhold Co., New York, 1979.
6. Daly, D., Paper Trade J. 151(50):48 (1967).
7. Parssinen, R., A finite element study of the Super-L Winder to reduce vibration, Beloit Corp., 1985.

Received for review June 4, 1987.

Accepted June 18, 1987.

Presented at the 1987 Finishing and Converting Conference Proceedings.