The role of analytical winding dynamics in winder design

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ABSTRACT

Winder design is guided by the principle that major excitation sources should not hit any of the system's resonance frequencies. Excitation of fixed rotating members (drums, guide rolls, rider rolls) can be eliminated through better balancing and greater manufacturing accuracy. However, the wound paper rolls remain a variable and important source of excitation. This report emphasizes the need to include this source of excitation in winder design. Among the features to be considered in winder design are the rotation and excitation frequencies of the paper roll, growth of roll mass, changes in winding geometry as roll diameter increases, roll flexibility in winding nips, and damping characteristics of the paper rolls. These features are reviewed along with a proposed mathematical approach that includes the paper rolls in the winder structural model by modeling the nip contacts as distributed springs. Practical techniques for reducing winder vibration are also presented.

Application:

Explains how paper properties can affect the roll's performance on the winder and describes techniques for reducing winder vibration.

THE INCREASING SPEED OF PAPER machines sets new capacity demands on paper machine winders. Various dynamic phenomena, such as drum vibrations and roll throwouts, often limit the winder speed. This paper shows how analytical winding dynamics is used in winder design to reduce winder vibrations.

Winders are subject to oscillating loads caused by rotating drums and paper rolls. Thus winders, such as the two-drum model illustrated in **Fig. 1**, must be dynamically dimensioned. Undoubtedly, the leading principle in dynamic dimensioning is that the frequency of the major excitation sources should not hit any of the system's resonance frequencies. Excitation of fixed rotating members such as drums, guide rolls, and rider rolls can be eliminated relatively simply through better balancing and greater manufacturing accuracy. It also helps to use rolls with a large outer diameter, since this decreases the rotation frequency and increases the natural frequency.

The real challenge is dealing with the properties of the paper roll and the excitations it generates. The mass flow of paper from the unwind to the winding section leads to a continuous change of the structural properties—and thus the natural frequencies—of the winder. Experimental studies have also shown that paper rolls can act as severe excitation sources during winding.

Thus the essential features to be considered in the study of winder dynamics are:

- The rotation and excitation frequencies of the paper rolls
- Growth of the roll mass
- The changing winder geometry with increasing roll diameter
- The flexibility of the paper roll in winder nips
- The damping characteristics of the paper rolls.

This paper provides an overview of some of these features and presents examples of practical tools for reducing winder vibration. The discussion is divided into four key topics:

1. **Spring model for roll nip.** A theory describing the flexibility of the paper roll in the nip is derived. This "spring model" makes it possible to incorporate the paper roll into the winder structure.
2. **Roll–drum vibration model.** This model consists of elastic drums, deformable paper rolls, and a rider-roll beam. An example of paper-roll-diameter-dependent winder natural frequencies is given.
3. **Vibrating paper grades.** The origin of the so-called vibrating-paper-grade winder vibration is explained.
4. **Measures to reduce vibration.** The discussion ends with some practical measures to reduce winder vibration.

SPRING MODEL FOR ROLL NIP

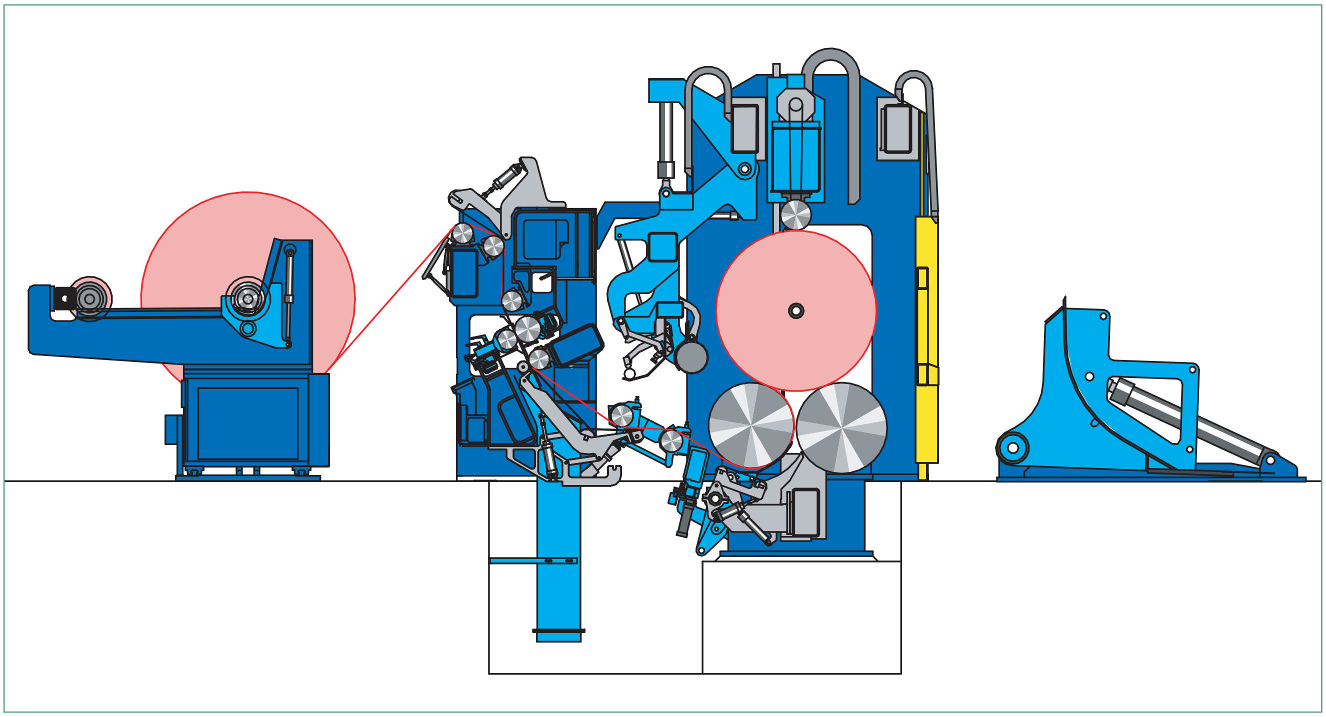
Consider the deformation of a long, linear-elastic, isotropic, and homogeneous cylinder (paper roll) caused by a rigid cylinder (drum), as shown in **Fig. 2**. The upper cylinder of diameter d is supported by the lower cylinder of diameter d_1 and loaded by a distributed line load P from above. This load gives rise to the Hertzian-contact pressure distribution $p(\phi)$ and displacement δ of the cylinder center. It is known *(1, 2)* that in the state of plane strain, the displacement is

$$\delta = P[(1-\nu^2)/(\pi E)][2 \ln(2d/a)-1] \quad (1)$$

where a is the semicontact width, which is calculated from the Hertz formula

$$a = [(2d^*P)/(\pi E^*)]^{1/2} \quad (2)$$

where



1. Two-drum winder

$$d^* = [(1/d)+(1/d_1)]^{-1} \qquad (3)$$

and

$$E^* \approx E/(1-\nu^2) \qquad (4)$$

The constants E and ν are, respectively, the modulus of elasticity and Poisson's ratio of the upper cylinder. Note, however, that when the upper cylinder is a wound paper roll, these quantities depend on the radial stress distribution inside the roll *(3)*. The radial stress can be approximated by the sum of the stress resulting from the Hertzian-contact pressure *(1)* and the winding stress *(4, 5)*. Because the former stress decreases toward the roll center while the latter increases, a simplifying assumption of constant total radial stress distribution is made. If the dependence of E on the radial stress were taken into account more accurately, an analytical expression such as Eq. 1 could not be obtained.

The contact load P is shown as a function of the nip deformation δ in **Fig. 3** for paper rolls of d = 0.4 m and d = 1.2 m. Values for the other relevant parameters are d_1 = 0.85 m, ν = 0.3, and E = 9.8 MPa.

The final step to include the paper roll in the winder structural model involves calculating the spring constant k for the paper roll surface. Using Eq. 1, we obtain

$$k = dP/d\delta$$
$$= (\pi E)/\{2(1-\nu^2)[\ln(2d/a) - 1]\} \qquad (5)$$

Figure 4 shows a plot of the nip spring constant k for a fine-paper roll as a function of P for rolls of d = 0.4 m, 1.2 m, and 1.8 m with E = 9.8 MPa. It can be seen that small-diameter rolls have higher spring constants than large-diameter rolls.

ROLL–DRUM VIBRATION MODEL

In the previous section, we developed a relation between external nip load and roll deformation (constitutive relation). With the aid of this relation, we can now include the paper rolls in the winder model by modeling the nip contacts as distributed springs, with spring constants calculated using Eq. 5. A side view of the model is shown in **Fig. 5**. The roll–drum vibration model consists of N deformable paper rolls, rotating elastic drums, and a rigid rider-roll beam supported by two equal springs. The roll–drum interaction is modeled by a distributed spring and viscous damper in the nip with spring and damping coefficients k_1, c_1 and k_2, c_2 in the front and rear drum nips, respectively. Each roll has four degrees of freedom—the displacement of the center of mass of the roll (x_i,y_i) and the rotation angles θ_i and λ_i around the positive x- and y-axes, respectively. The rider-roll beam has two degrees of freedom—the vertical displacements of the beam end.

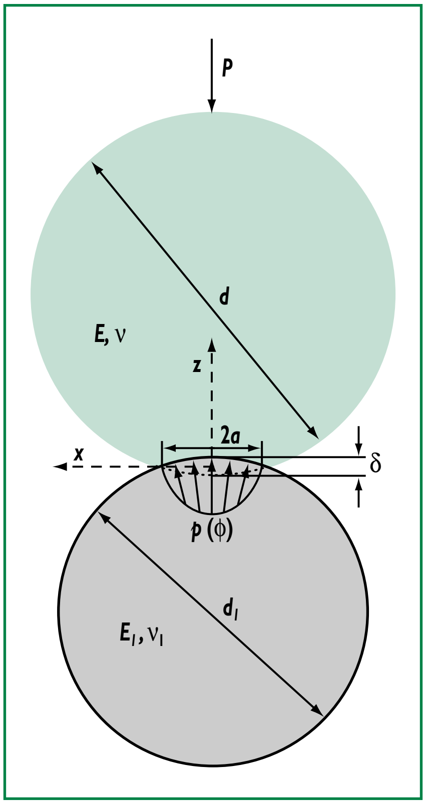
To derive complete equations of motion for the winder, we still need an expression for the roll mass per unit length (m) as a function of time. This is easily obtained from the continuity equation *(6)*, and the result is

$$m(t) = \{0.25\ \pi\rho\ [d^2(t)-d_0^2]\} + m_0 \qquad (6)$$

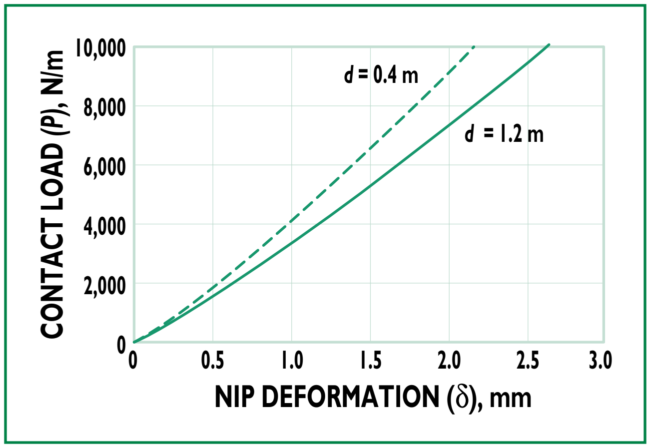
where

ρ = density of the paper
d_0 = diameter of the core
m_0 = mass per unit length of the core

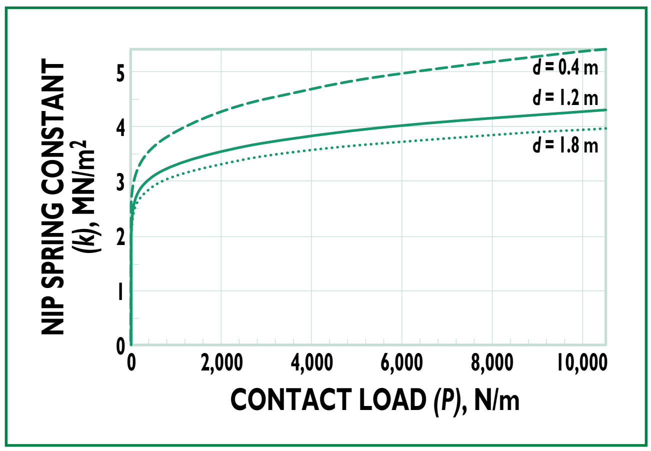
and where



2. *Nip deformation caused by the drum nip*



3. Contact load* P *as a function of nip deformation for two different roll diameters.



4. Nip spring constant* k *as a function of the compressing load* P *for three different roll diameters

$$d(t) = \{d_0^2 + [(4\tau/\pi)\int_0^t V dt]\}^{1/2} \qquad (7)$$

where

τ = average web thickness
V = web speed

When the equations of motion are derived, one obtains a set of second-order differential equations of the form

$$M(t)\ddot{x} + C(t)\dot{x} + K(t)x = F(t) \qquad (8)$$

where x is the displacement vector, containing translation and rotation coordinates of each roll, drum deformations, and rider-roll-beam end displacements. $M(t)$, $C(t)$, and $K(t)$ are time-dependent mass, damping, and stiffness matrixes, respectively, and $F(t)$ is the sum of the excitation forces.

Because the stiffness and mass properties of the winder change in time, it is obvious that the natural frequencies (resonances) of the winder also change in time.[1] As an example, the eigenfrequencies of a 9.1-m-wide winder with five rolls of width 1.68 m are shown in **Fig. 6** as a function of the roll diameter. Note that some frequencies stay relatively constant during winding, while others decrease monotonically. The mode shapes corresponding to the former are drum bending modes, while those corresponding to the latter are roll modes.

It can be shown that in the case of isotropic drum bearings, the eigenfrequency f_1 of the lowest drum bending mode is approximately

$$f_1 = \{\hat{f}_1^2 + [k/(\rho A)]\}^{1/2} \qquad (9)$$

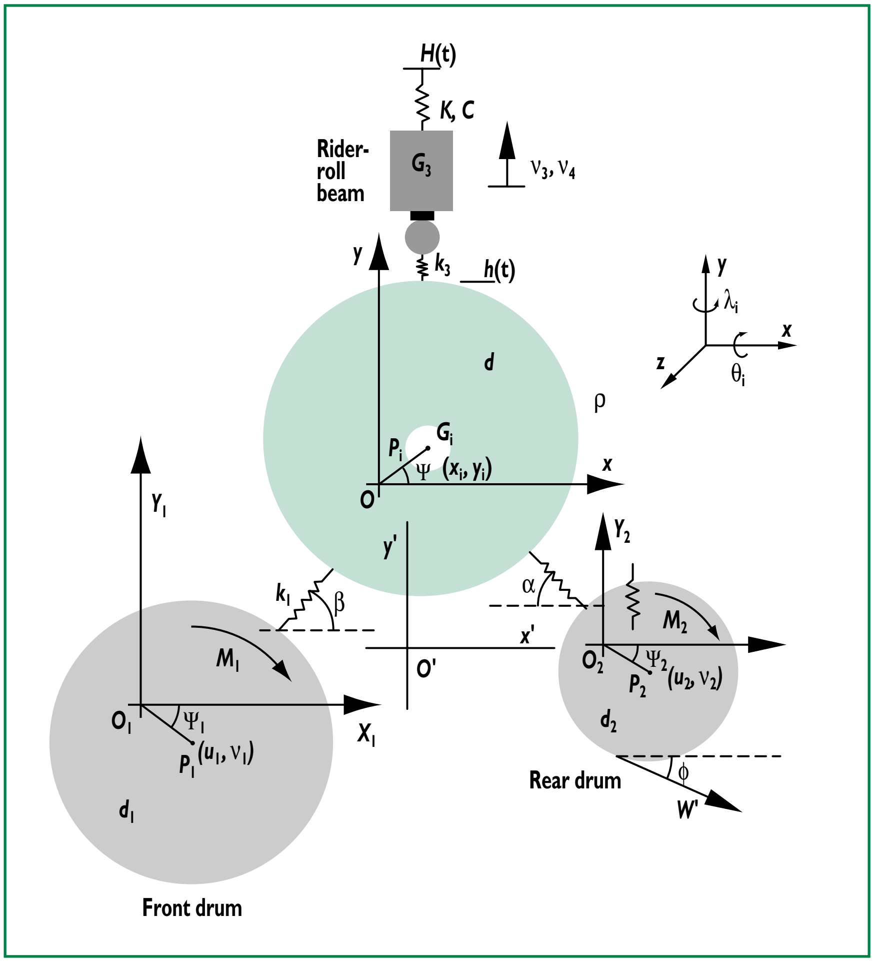
where $\hat{f}_1$ is the natural frequency of the bare drum (without paper rolls) and ρA the mass per unit length of the drum tube. From this formula, it can be seen that paper rolls increase the lowest natural frequency of the drums.

VIBRATING PAPER GRADES

Certain paper grades can act as vibration excitation sources when the winder is run. Such paper grades are commonly called vibrating paper grades. The most common vibrating paper grades are fine papers, vellum, and linerboard.

In general, an essential prerequisite for this self-excited paper-roll vibration is that the wound rolls tend to retain deformations caused by external load. As studied previously by Daly *(7)*, this roll characteristic is typically related to the paper friction properties and interlayer pressure within the roll. When a load on the roll surface is relaxed, the friction force prevents the layers from sliding relative to each other and eliminating the deformation. In practice, during winding, only the nip-load oscillations will be capable of producing plastic deformations on the roll surface. Rolls with a high layer-to-layer pressure (P_C) and high-friction coefficient (μ)—and thus a high sliding-friction μP_C value—will retain the relative

[1]In a strict sense, the concept of natural frequency is not defined for this system with time-dependent coefficient matrixes. Thus what we consider here are momentary or "frozen" modes, with *M*, *C*, and *K* calculated at each time instant for fixed *t*.



5. Roll–drum vibration model

displacements of the outer paper layers caused by a nip-load spike. To produce a permanent deformation, the applied force must exceed a certain limit. This means that plastic roll deformations will take place only when the roll diameter is larger than 800–1000 mm, when the roll mass is sufficiently high to produce a nip load with the required force.

All winders are engineered such that the lowest drum eigenfrequency is well beyond the rotation frequency of any considerable excitation source. However, when a winder is running paper rolls that retain deformations, the integer multiples of the roll rotation frequency tend to excite the first bending modes of the drums. **Figure 7** shows the 1st–4th multiples of the roll rotation frequency, together with the lowest nip-direction drum-bending-mode eigenfrequencies, during a typical fine-paper winder run. Potential vibration areas exist where a roll harmonic crosses a drum resonance frequency. Generally, the crossings below 1-m roll diameter will not cause irreversible deformations on the rolls and, hence, there will not be any severe vibrations in that range. The most likely vibration area in this example is located around d = 1.19 m and d = 1.32 m, where the 3rd roll harmonic hits consecutively the front and rear drum resonances.

Because the rolls are never perfectly round, there will always be some excitation at the lower harmonics of the roll. (Generally, the smaller the harmonic, the bigger is the excitation). No matter how small the initial disturbance is—giving excitation at or close to the drum resonance frequency—the drum oscillation at that frequency will increase. In turn, this drum vibration, being synchronous to the roll harmonic, will then increase the roll deformation corresponding to that harmonic, and so on. If the front and rear drum eigenfrequencies are unequal, and one drum is vibrating at its resonance, the other will be in a state of forced vibration caused by the deformed roll. Because the eigenfrequencies of the drums are usually quite close to each other, the nonresonating drum vibrates at high amplitude because of its close resonance.

MEASURES TO REDUCE VIBRATION

Because problematic drum vibration always occurs at the drum's first resonance frequency, it is sufficient to study measures to reduce the drum response at the resonance. In the case of a single-degree-of-freedom oscillator, any basic vibration textbook shows that the response at the resonance is

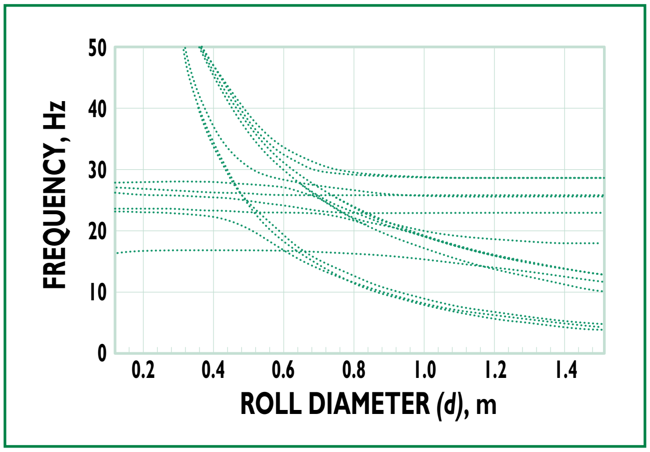
$$\max(X/F) = m^{1/2}/[c(k)^{1/2}] = 1/c\omega \qquad (10)$$

where

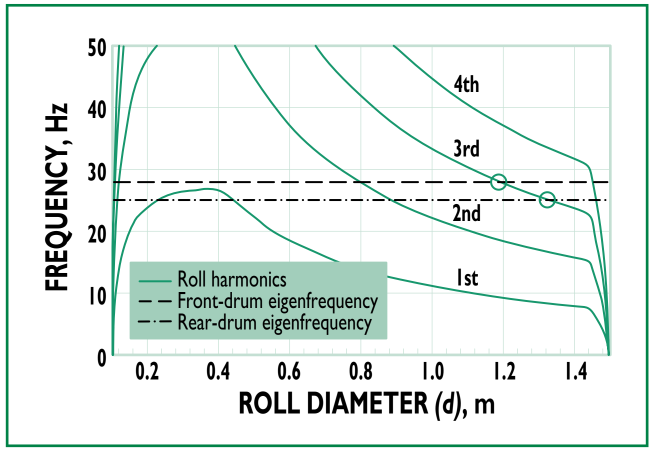
X = amplitude of vibration
F = amplitude of external force
m = mass
k = stiffness
c = viscous damping coefficient
ω = angular natural frequency.

The immediate conclusion is that the response is reduced by increasing the damping and natural frequency. In the case of a winder drum resonating at its first eigenfrequency, the situation is basically the same except that the number of parameters involved is much larger. These parameters are

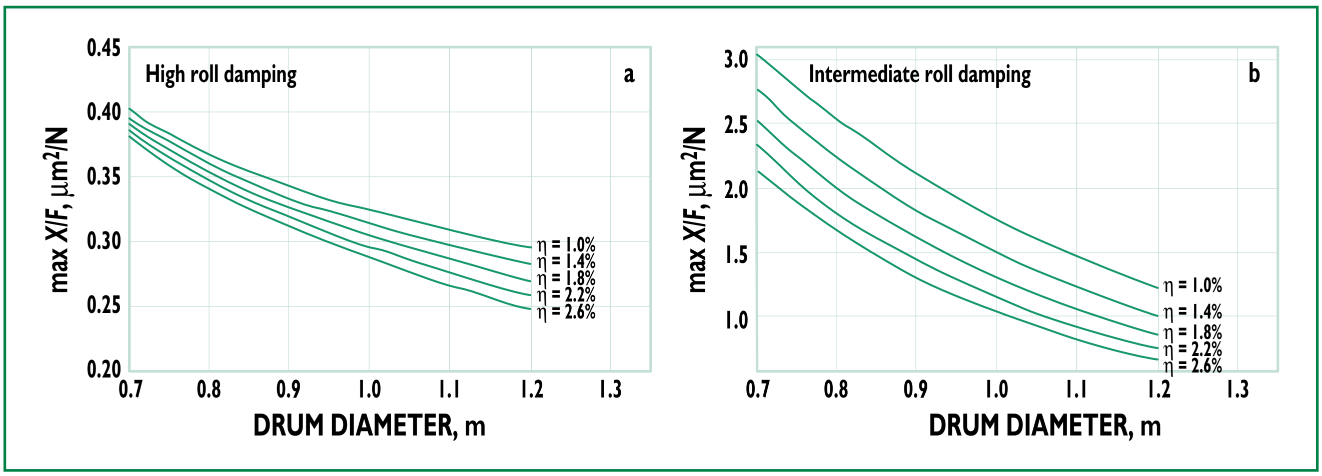
- Paper roll flexibility
- Paper roll damping
- Bearing stiffness
- Bearing damping
- Drum tube dimensions and material properties
- Drum width.



6. Eigenfrequencies of a winder with five equal-diameter paper rolls as a function of the roll diameter



7. The 1st–4th harmonics of the roll rotation frequency and the resonance frequencies of the drum bending modes



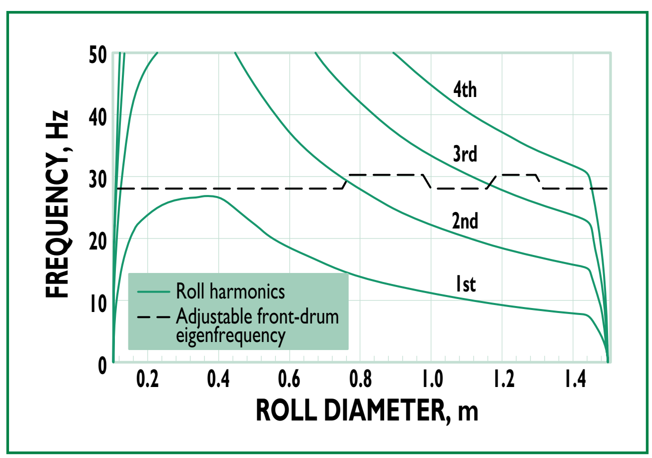
8. Resonance response max X/F at drum center (drum length = 10 m) as a function of drum diameter for (a) high roll-damping coefficient (c_l = 30 kN·s/m²) and (b) intermediate roll-damping coefficient (c_l = 3 kN·s/m²)

All of these affecting quantities can be reliably estimated except for roll damping. The roll damping mechanism is not well understood, and no theory for its evaluation is available. However, some estimates of roll damping can be obtained by measuring the eigenfrequencies and corresponding damping values of the winder and then adjusting the winder model roll damping to meet the measured values.

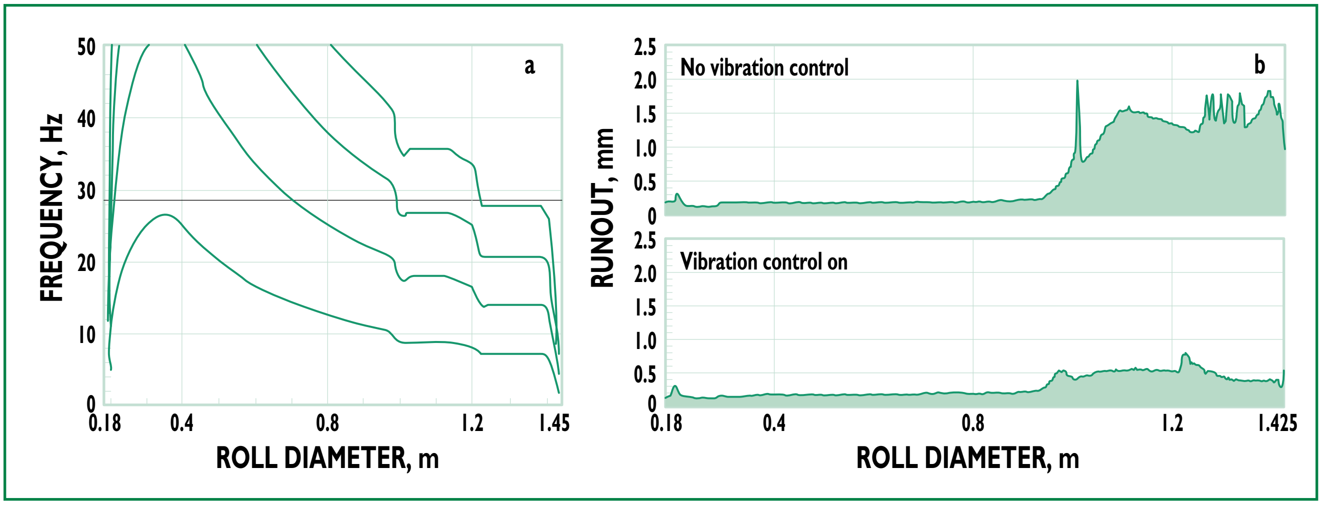
KEYWORDS

Machine speed, winders, dynamics, machine design, vibration, rotation, wound rolls, drums, frequencies, analysis, models, simulation, paper grades, rider rolls.

Figure 8 shows the variation of max X/F as a function of drum diameter for five different drum tube material damping η values for high (**Fig. 8a**) and intermediate (**Fig. 8b**) roll damping. The drum width is L = 10.0 m and drum tube wall thickness = 50 mm in both cases. The results show that with high roll damping, increasing the drum diameter has a relatively small effect on the resonance response. On the other hand, with inter-



9. Winder vibration can be reduced by adjusting the resonance frequency of the drum to avoid coinciding with harmonics of the paper roll during a set



10. (a) 1st–4th harmonics of the roll and the resonance frequency of the drum and (b) front-drum run-out with and without vibration control

mediate roll damping, increasing the drum diameter is an effective method to decrease the drum response. This clearly demonstrates how paper-roll properties influence winder dynamics and must be taken into account in winder design.

There are also techniques for avoiding vibration that are not based on structural changes in winder design. The basic idea behind such techniques is to avoid running the winder at the resonance. This can be accomplished by either (a) changing the eigenfrequency or (b) changing the running speed during the set.

The principle of the first technique is shown in **Fig. 9**. In this example, the drum eigenfrequency is changeable between two values by altering the bearing stiffness. The run starts at the lower stiffness level. Then, when the second harmonic of the roll is getting closer to the resonance, the stiffness is increased to pass the roll excitation. Between the second and third harmonic, the bearing stiffness is again switched to the lower value and then the procedure is repeated at every roll harmonic.

The second method is demonstrated in **Fig. 10**, where the data are obtained from actual measurements. The winder in question has a lowest drum eigenfrequency at 29 Hz. **Figure 10a** shows the 1st–4th roll harmonics, while **Fig. 10b** shows the peak-to-peak displacement of the front drum center. The upper panel in Fig. 10b corresponds to a case where the running speed is held constant between acceleration and deceleration. The lower panel depicts a case where the resonance is passed quickly by decreasing the running speed when the third and fourth roll harmonics are about to hit the drum eigenfrequency. The maximum runout is decreased from 1.9 mm to 0.75 mm when this type of vibration control is used. The speed control is automated, and it increases capacity noticeably because the winder can be operated at higher running speeds.

Methods based on changing the roll structural properties— such as roll spring constant and hardness—to shift the eigenfrequency have not been successful. These include sudden changes of the rider-roll load and tangential force difference of the drums. The reason for this is that the effect of the roll spring constant on the first eigenfrequency of the drum is quite small.

The most efficient method is to increase the drum stiffness by reducing the drum span (drum tube stiffness is $\sim 1/L^4$). This is accomplished by a sectional drum with intermediate supports. While this can be implemented relatively easily on the front drum, the rear drum—with the paper web passing beneath it—cannot be supported from below.

SUMMARY

The mass, elasticity, and damping properties of the paper roll must be taken into account in winder design. Because the winder structure changes over time as the roll builds, full-scale simulations of the entire winding process should be carried out during the design stage. **TJ**

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