TWO-DRUM WINDER RUN SIMULATION MODEL

by

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ABSTRACT

A dynamic, analytical model for winder run simulations is presented. The model consists of elastic drums, deformable paper rolls and a rigid rider roll beam. A paper roll nip flexibility model is derived and a profound influence of paper roll properties on winder dynamics is demonstrated. The origin of winder vibrations due to specific vibrating paper grades is explained in detail. Winder drum design aspects against vibrations are studied. Finally, some practical measures to reduce winder vibrations are presented.

NOMENCLATURE

a	paper roll semi-contact width in Hertz contact theory
C	damping matrix
c_1	paper roll-front drum nip damping coefficient
c_2	paper roll-rear drum nip damping coefficient
d	paper roll diameter
d_0	core diameter
d_1	front drum outer diameter
E	modulus of elasticity of the paper roll
F	sum of the excitation forces
f_1	lowest eigenfrequency of the drum
$\hat{f}_1$	lowest eigenfrequency of the bare drum
K	stiffness matrix
k	paper roll spring constant
k_1	paper roll-front drum nip spring constant

k_2	paper roll-rear drum nip spring constant
L	bearing to bearing length of the drums
M	mass matrix
m_i	mass of i'th paper roll
N	number of rolls in a set
P	distributed compressive load in the roll-drum nip
P_c	layer to layer pressure in the paper roll
$p(\phi)$	contact pressure distribution in the roll-drum nip
v	web speed = winder running speed
X/F	frequency response function of the drum
x	displacement vector of the generalized coordinates
(x_i, y_i)	coordinates of the center of mass of the i'th roll
δ	deformation of the paper roll
θ_i	rotational angle of the i'th roll around x-axis
λ_i	rotational angle of the i'th roll around y-axis
μ	coefficient of friction between paper layers
ρ	paper roll density
ρA	mass per length of the drum tube
τ	average paper web thickness

INTRODUCTION

The increasing speed of paper machines sets new capacity demands on paper machine winders. Various dynamic phenomena, such as drum vibrations and roll throw-outs, often limit the winder speed. The goal of the present paper is to show how analytical winding dynamics is used in winder design to reduce harmful vibrations.

Due to oscillating loads caused by rotating drums and paper rolls (figure 1), dynamic aspects must be accounted for in winder dimensioning. Undoubtedly, the leading principle is that the frequencies of the major excitation sources should not hit any resonance frequencies of the system. Excitation by solid rotating members such as drums, guide rolls and rider rolls can be eliminated relatively simply by improved balancing and manufacturing accuracy. Also, the outer diameter of the drums should be large enough to decrease the rotational frequencies and to increase the system natural frequencies. The real challenge comes, however, from the properties of the wound paper rolls. The uninterrupted mass flow from the unwind to the winding section gives rise to a continuous change of the structural properties, e.g., natural frequencies, of the winder. Experimental studies have also shown that paper rolls can act as severe excitation sources during winding.

Accordingly, the essential features in winder dynamics are:

- rotational and excitation frequencies of the paper rolls
- growth of the roll mass
- change of the winding geometry due to increasing roll diameter
- paper roll flexibility in the winding nips
- damping characteristics of the paper rolls

The outline of the paper is as follows. Firstly, a model accounting for paper roll flexibility in the nip is developed. This "roll spring model" provides a straightforward inclusion of the paper roll into the winder structure. Secondly, a roll-drum vibration model comprising elastic drums, deformable paper rolls and a rigid rider roll beam is presented. An example of winder natural frequencies as a function of the roll diameter is given. Then, the concept of a vibrating paper grade is elaborated. Finally, some practical measures to reduce winder vibrations are presented.

SPRING MODEL FOR THE ROLL NIP

Let's consider the deformation of an isotropic paper roll when pressed against a rigid drum as shown in figure 2. The roll of diameter d is supported by the drum of diameter d_1 and loaded by a distributed vertical line load P. This load gives rise to the contact pressure distribution $p(\phi)$ and the displacement δ of the roll centre. Assuming a constant modulus of elasticity for the paper roll one obtains [1]

$$\delta = P\frac{(1-\nu^2)}{\pi E}[2\ln(2d/a)-1], \qquad (1)$$

where the semi-contact-width a is calculated from the Hertz formula

$$a = \sqrt{\frac{2d^* P}{\pi E^*}} \qquad (2)$$

with

$$d^* = \left(\frac{1}{d}+\frac{1}{d_1}\right)^{-1}, \qquad (3)$$

$$E^* = \frac{E}{(1-\nu^2)}. \qquad (4)$$

Here E is the modulus of elasticity and ν Poisson's ratio of the paper roll. In general, these quantities are known to depend on the radial stress distribution inside the roll. The radial stress may be approximated by the sum of the stress due to the Hertzian contact pressure and the winding stress [2,3]. As the former stress decreases towards the center, whereas the latter increases, a simplifying assumption of constant total radial stress distribution is made. If the dependence of E on the radial stress is taken into account more accurately, an analytical expression, such as equation (1), can hardly be obtained. The compressive load P as a function of the nip deformation δ is shown in figure 3 for $d = 0.4$ and 1.2 m. The other parameter values used are $d_1 = 0.85$ m, $\nu = 0.3$ and $E = 9.8$ MPa.

As a final step in inclusion of the paper roll in the winder model we calculate the equivalent spring constant for the paper roll. Utilizing equation (1) we obtain

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

The nip spring constant k for a fine paper roll as a function of P for $d = 0.4$, 1.2 and 1.8 m and $E = 9.8$ MPa is shown in figure 4. It is interesting to note that the smaller the roll the larger the spring constant.

ROLL DRUM VIBRATION MODEL

The roll-drum vibration model consists of N paper rolls, rotating elastic drums and a rigid rider roll beam supported by two equal springs (see figure 5). 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 horizontal and vertical displacements x_i and y_i of the center of mass, and the angles of rotation θ_i and λ_i around the positive x- and y-axes, respectively. The rider roll beam possesses two degrees of freedom - the vertical displacements v_3 and v_4 of the beam ends.

To derive the equations of motion we still need an expression for the roll masses m_i as a function of time. This can readily be shown to be [3]

$$m_i = \frac{1}{4}\pi\rho\left[d^2(t) - d_0^2\right]L_i + m_{0i}, \tag{6}$$

where

$$d(t) = \sqrt{d_0^2 + \frac{4\tau}{\pi}\int_0^t v\,dt}\ , \tag{7}$$

ρ is the density of the paper, τ the average web thickness, d_0 and m_{0i} the diameter and mass of the i'th core, L_i the width of the i'th roll and v the web speed.

The equations of motion written in matrix form are

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

where the generalized displacement vector x contains both the translational and rotational coordinates of the rolls, the drum deformations and the rider roll beam end displacements. Note that the mass, damping and stiffness matrixes $M(t)$, $C(t)$ and $K(t)$ and the driving force $\boldsymbol{F}(t)$ are explicitly time dependent due to roll growth during winding. For this reason the natural frequencies 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 figure 6 as a function of the roll diameter.

It can be seen that some eigenfrequencies stay relatively constant during winding

[1] Here we consider momentarily or "frozen" modes with M, C and K calculated at each time instant.

while others decrease monotonously. The former correspond to drum bending modes and the latter to various roll modes.
It can be shown that for isotropic drum bearings the eigenfrequency f_1 of the lowest drum bending mode may be approximated by

$$f_1 = \sqrt{\hat{f}_1^2 + \frac{k}{\rho A}}\,, \tag{9}$$

where $\hat{f}_1$ is the natural frequency of the bare drum (without paper rolls) and ρA the mass per length of the drum tube. Equation (9) implies that the effect of the paper rolls is to increase the lowest drum eigenfrequency.

VIBRATING PAPER GRADES

It is known in practice that rolls of certain paper grades can act as vibration sources for the winder. These so called vibrating paper grades include, for example, fine papers, vellum and liner board.

An essential prerequisite for this self-excited vibration is that the wounded rolls are able to retain deformations caused by external load. According to Daly [4], this property is related to paper friction and inter-layer pressure in the roll. Rolls having high layer to layer pressure P_c and high friction coefficient μ will retain the relative displacements of the paper layers caused by a nip load spike, because the high friction force μP_c prevents the layers from sliding relative to each other to restore the deformation. In order to produce a permanent deformation, the applied force must exceed a certain limit. In practice, this means that permanent roll deformations will take place only when the roll diameter is larger than 800 - 1000 mm, when the nip load due to roll mass is sufficiently high.

All winders are engineered so that the lowest drum eigenfrequency is well beyond the rotational frequency of any considerable excitation source. However, when the paper rolls have the above mentioned deformation characteristics, an integer multiple of the roll rotational frequency tends to excite the lowest bending modes of the drums. The 1st - 4th multiples of the roll rotational frequency, along with the lowest drum bending eigenfrequencies, during a typical fine paper run are shown in figure 7. A potential danger of vibration occurs when a roll harmonic crosses a drum eigenfrequency. Usually, crossings below 1 m roll diameter values will not cause irreversible deformations on the rolls; therefore, no severe vibrations are observed in that range. In this example the most likely vibration areas are located around d = 1.19 and 1.32 m, where the third roll harmonic hits consecutively the lowest front and rear drum eigenfrequencies.

Since the rolls are never perfectly round, there will always be some excitation at the lower harmonics of the roll rotational frequency (generally the smaller the harmonic the larger the excitation amplitude). No matter how small the initial disturbance is, when the excitation occurs at or close to the drum eigenfrequency, the drum oscillations will increase and then in turn this drum vibration, being synchronous to the roll harmonic, will 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 resonance, the other will be in a state of forced vibrations. As the eigenfrequencies of the drums are usually quite close, the other drum vibrates at a high amplitude, too.

MEASURES TO REDUCE VIBRATION

Because the problematic drum vibrations usually occur at the 1st drum resonance frequency, it is sufficient to reduce the drum response at that resonance. In the case of a harmonically driven single degree of freedom oscillator the response at resonance is

$$\max\frac{X}{F}=\frac{\sqrt{m}}{c\sqrt{k}}=\frac{1}{c\omega}, \tag{10}$$

where X and F are the amplitudes of the particle displacement and external force, respectively, m is the particle mass, k the spring stiffness, c the viscous damping coefficient and ω the angular eigenfrequency. It can be seen that the response can be reduced by increasing the damping coefficient and the system eigenfrequency. In the case of a winder the situation is basically the same, although 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
- drum width

All these parameters, except roll damping, can be estimated reliably. The roll damping mechanisms are not well understood and no theory for its evaluation is available. Some estimates for roll damping can be obtained by measuring the eigenfrequencies and damping values for the whole winder and adjusting the corresponding calculated values to the measured values by a proper choice of roll damping.

Variation of max X/F as a function of drum diameter for high and intermediate roll damping is shown in figure 8 for five different material damping (η-)values of the drum tube. The drum width is 10.0 m and tube wall thickness 50 mm. It can be concluded that, for high roll damping, the effect of the drum diameter increase on the resonance response is relatively small, whereas for intermediate roll damping, the diameter increase will be an effective method to decrease the drum response. This demonstrates clearly how paper roll properties have an effect on winder dynamics and must be taken into account in winder design.

To avoid vibration, there are also methods which are not based on structural changes of the winder. The basic idea is to avoid running at resonance. This can be accomplished by

- changing the eigenfrequency
- changing the running speed during the winding

An example of the first case is shown in figure 9. The drum eigenfrequency can be switched between two values by altering the bearing stiffness. The run starts at the lower stiffness level. When the second harmonic of the roll is about to hit the drum eigenfrequency, the bearing stiffness is suddenly increased to avoid the roll excitation. Then the bearing stiffness is switched to the lower value again and the procedure is repeated at every roll harmonic.

An example of the second case is shown in figure 10, where the data is obtained from actual measurements. The lowest drum eigenfrequency of this winder is 29 Hz. The 1st - 4th roll harmonics are shown in figure 10 (a) and the peak to peak displacement of the front drum center in figure 10 (b); in the upper panel the running speed is held constant between the acceleration and deceleration stages, whereas in the lower one the resonance is passed quickly by decreasing the running speed when the 3rd and 4th roll harmonics are about to hit the drum eigenfrequency. The maximum runout is decreased from 1.9 to 0.75 mm when this type of vibration control is used. The speed control is automated and the winder capacity is noticeably increased, because the winder can be operated at higher running speeds.

Methods based on changes in the roll structural properties (e.g. roll hardness) have not been successful in shifting the drum eigenfrequency. These include sudden changes of the rider roll load and the tangential force difference of the drums. The reason for the failure lies in the fact that the value of the roll spring constant has a minor effect on the lowest eigenfrequency of the drum.

One of the most efficient methods to avoid vibrations is to increase the drum stiffness ($\sim 1/L^4$) by reducing the drum span. In practice, this is accomplished by a sectional drum with intermediate supports. The support(s) can relatively easily be installed below the front drum while this is not so easy for the rear drum due to paper web beneath.

SUMMARY

It has been demonstrated how the mass, elasticity and damping properties of the paper rolls should be taken into account in winder dimensioning. Because of the explicit time dependence of the winder structure, simulations of the entire winding process should be carried out during the design stage. Finally, some examples of vibration reduction for vibrating paper grades are presented.

REFERENCES

1. Johnson, K. L., Contact Mechanics, Cambridge University Press, 1984.

2. Hakiel, Z., "Nonlinear model for wound roll stresses," Tappi Finishing and Converting Conference 1986, pp. 9-15.

3. Jorkama, M. I., "On the winder vibration analysis," Licentiate's Thesis, 1996, Helsinki University of Technology.

4. Daly, D., "How Paper Rolls on a Winder Generate Vibration and Bouncing," Paper Trade Journal 151: 48 ,1967.

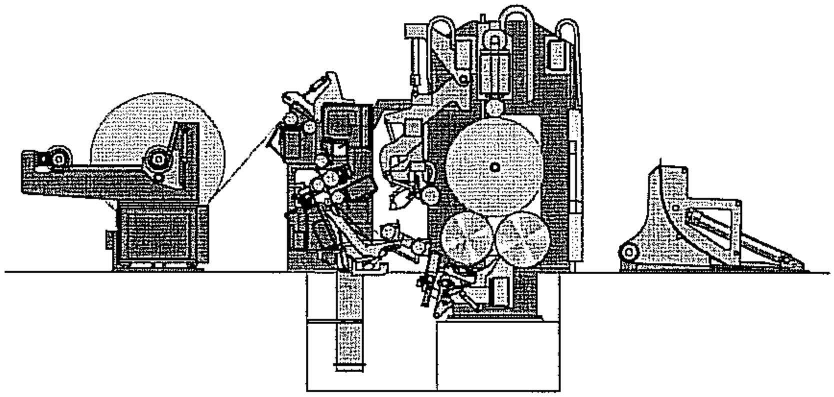

Figure 1. Two-drum winder.

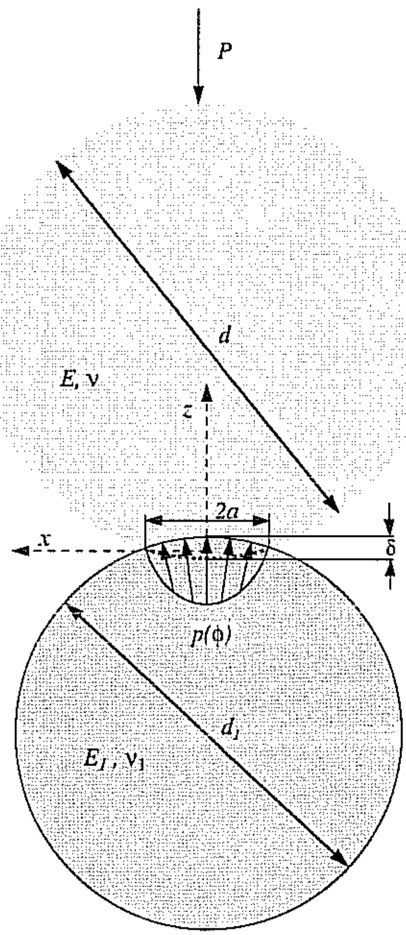

Figure 2. Nip deformation due to the compressive load P.

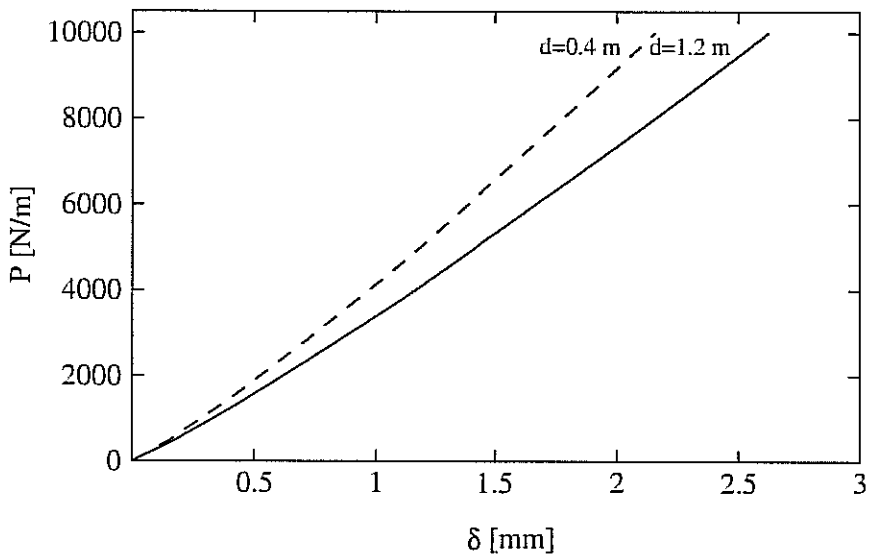

Figure 3. Relation between compressive load P and roll displacement δ.

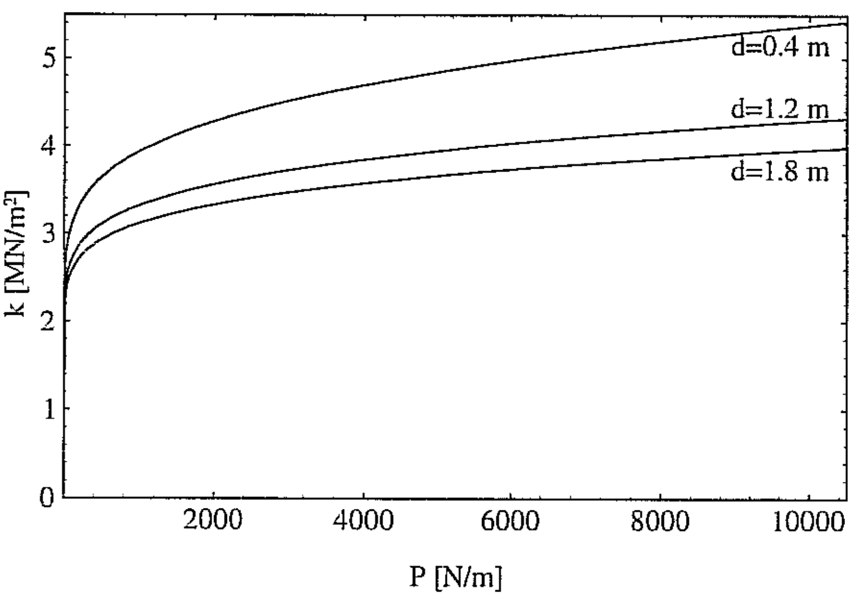

Figure 4. Nip spring constant k as a function of the compressive load P.

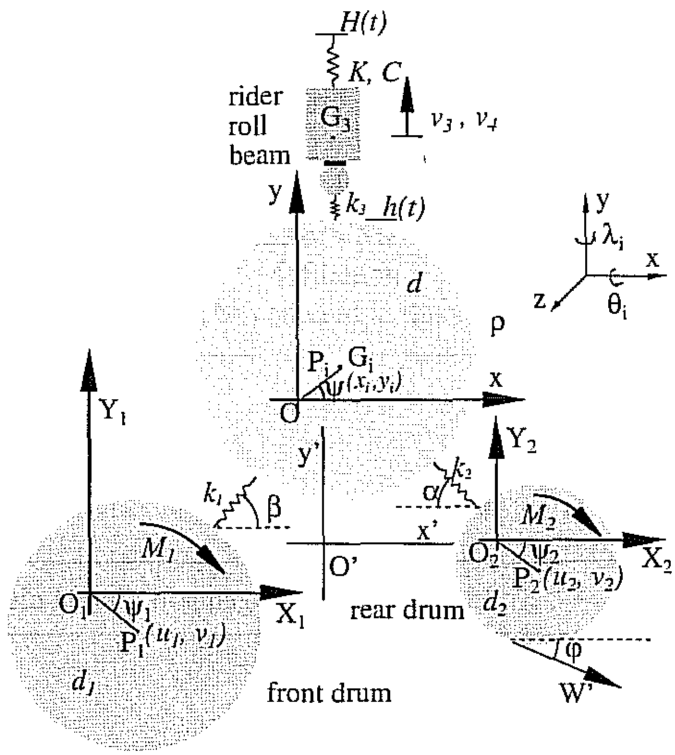

Figure 5. Roll-drum vibration model.

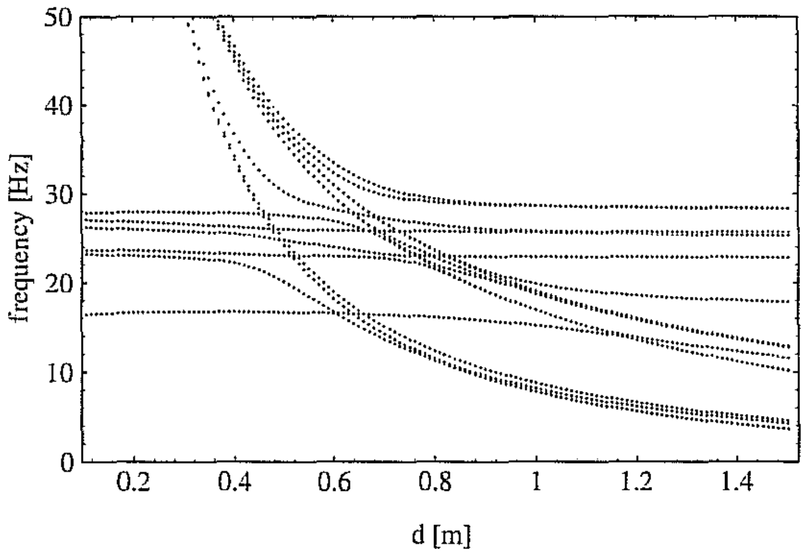

Figure 6. Eigenfrequencies of a winder with five equal rolls as a function of the roll diameter.

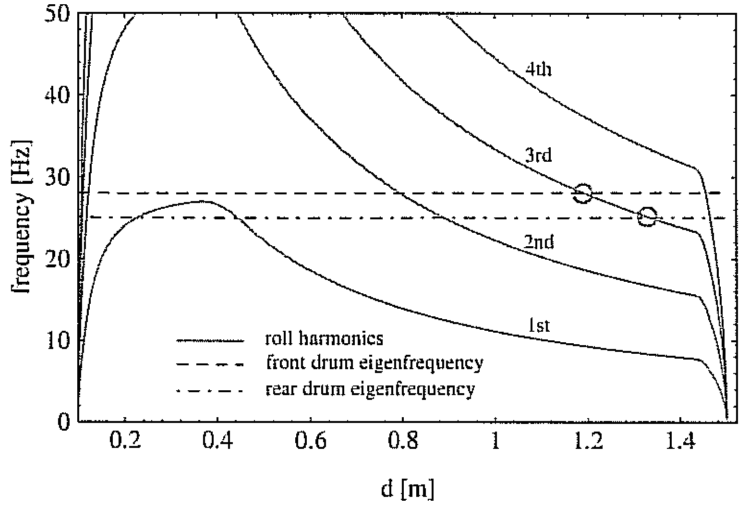

Figure 7. 1st-4th harmonics of the roll rotational frequency and the lowest eigenfrequencies of the drum bending modes.

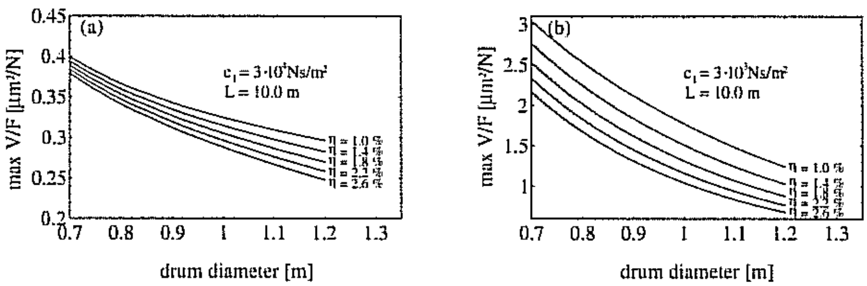

Figure 8. Resonance response max X/F at drum center as a function of drum diameter for (a) $c_1 = 30$ kNs/m^2 and (b) $c_1 = 3$ kNs/m^2.

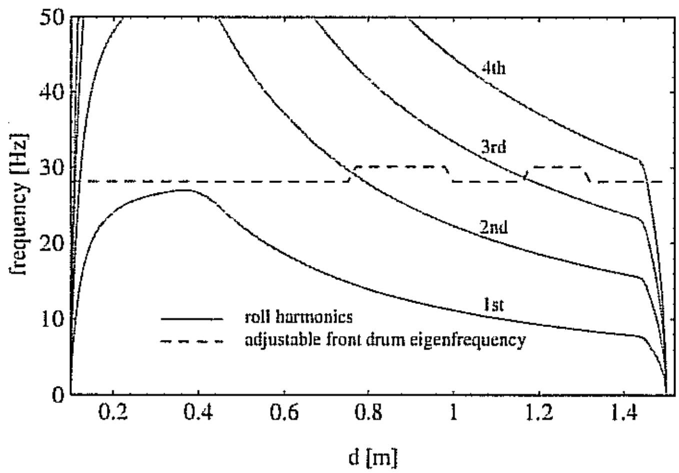

Figure 9. Avoiding winder vibration by changing the eigenfrequency during a set.

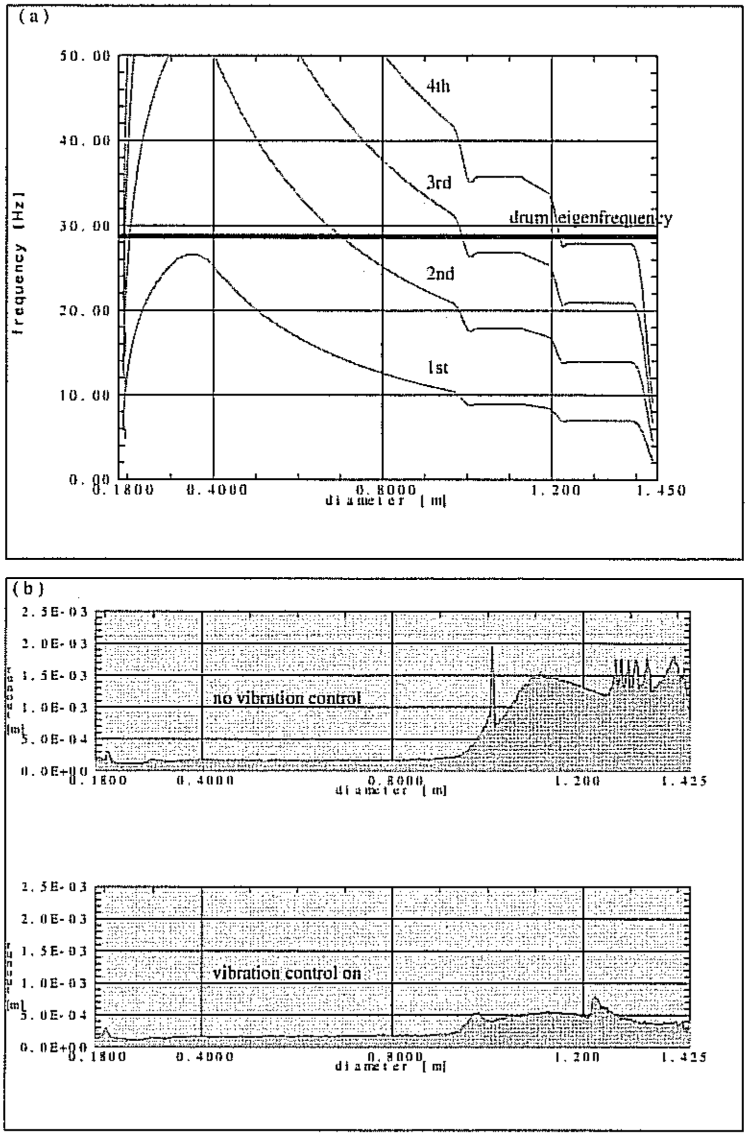

Figure 10. (a) 1st-4th harmonics of the roll and the lowest eigenfrequency of the drum (b) front drum run-out with and without vibration control.

Question - One of your first slides was a vibration signature and you specifically had your sensor mounted on the front drum bearing on the back side, was that just a typical signature, or was that the location that was the most critical for your investigation?

Answer - When we have that vibration, the paper creating the vibration, it doesn't actually matter where you measure vibration, every place is vibrating in the same manner.

Question - So then it wasn't in the front drum that it was not the most critical.

Answer - No

Question - Do you have any physical justification for using non-frictional contact solution rather than mathematical difficulty that was justified for or is not important.

Answer - Yes.

Question - Does it justify the means?

Answer - It is probably the weakest part of this model. But we have to have a very simple formula for this dynamic model because we can simulate it for 20 minutes at 200 Hz. The computational time sets some limits for incorporating winding model.

Question - Many modes of vibration may be termed as rocking modes. What happened to vertical modes taught by predecessors. Is it because of the usual unequal diameter drums or is there some other reason for that?

Answer - They are all vertical roll modes in this model.

Question - Can you tell us how much speed decrease is required to get through critical speeds? And roughly what the impact is on production rate.

Answer - 200 meters per minute typically. The capacity is actually increased, because the winder can now be operated at higher speeds. By avoiding Resonance Zones.

Question - What are the inputs of your vibration control? Are they measured values or calculated values?

Answer - We haven't discussed any control issues in this model yet. Running speed is just a given function.

Question - - In my experience, when you start changing speeds in the winder you also introduce structural changes in the roll do you have a way for compensating for that?

Answer - The vibrating paper grades are fine papers and vellum which are effected by the drop of running speed. That is why we can use this method.