WOUND ROLL GENERATED UNSTABLE VIBRATION ON A TWO-DRUM WINDER

by

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ABSTRACT

Nip contact between the paper roll, winding drum and rider roll or some other nip roller may cause that the wound roll is deformed into a convex polygon. This deformation process is accompanied with a strong vibration. The conditions under which this phenomenon occurs depend very much on the web properties. For example, in the paper industry some bulky grades with a high layer-to-layer coefficient of friction are known to be prone to this unstable vibration.

In this paper a simple wind-up model of a two-drum winder, capable of capturing quite comprehensively this phenomenon, is developed. The pattern formation is modelled via viscoelastic surface deformation. This results in a system of linear delay differential equations. Performing Laplace transformation to the system equations enables to study the stability of the system as a function of the web properties, nip drum stiffness, wind-up geometry and damping. The model parameters related to the viscoelastic surface deformation are measured experimentally for several paper grades.

The paper is concluded by studying the system stability in a certain resonance condition. It is demonstrated that the system can be stabilised by changing the structural parameters of the winder.

NOMENCLATURE

A	system matrix in the Laplace-domain
c_{ix}, c_{iy}, i=1,2,3	viscous damping coefficients of the rear drum, front drum and rider roll in the horizontal and vertical directions, respectively
k_{ix}, k_{iy}, i=1,2,3	spring coefficients of the rear drum, front drum and rider roll in the horizontal and vertical directions, respectively

$e_i(t)$, i=1,2,3	deviation of the wound roll's shape from circular evaluated at the rear drum, front drum and rider roll nips, respectively
f_i	ith harmonic of the wound roll rotation frequency
f_{ni}	ith natural frequency of the system
$k_{1\beta}$, $k_{2\beta}$	spring constants of the delayed and instant recovery elements of the wound roll at the rear drum nip, respectively
$k_{1\alpha}$, $k_{2\alpha}$	spring constants of the delayed and instant recovery elements of the wound roll at the front drum nip, respectively
$k_{1\phi}$, $k_{2\phi}$	spring constants of the delayed and instant recovery elements of the wound roll at the rider roll nip, respectively
$c_{1\beta}$, $c_{1\alpha}$ and $c_{1\phi}$	viscous damping coefficients of the delayed recovery elements at the rear drum, front drum and rider roll nips, respectively
m_0	core mass per unit length
m_1, m_2, m_3	the masses of the rear drum, front drum and rider roll, respectively
s	the Laplace-variable
s_i	the complex roots of equation det**A** = 0 (i = 1,2,3,...)
t	time variable
T_1	travel time from the front drum nip to the rear drum nip
T_2	travel time from the rider roll nip to the front drum nip
T_3	travel time from the rear drum nip to the rider roll nip
u_i, i=12,3	deformation of the delayed recovery element at the rear drum, front drum and rider roll nip, respectively
x, y	the horizontal and vertical displacements of the wound roll
x_i, y_i (i =1,2,3)	the horizontal and vertical displacements of the front drum, rear drum and rider roll, respectively
X	column vector of the state variables in the Laplace-domain
z_β, z_α, z_ϕ	internal translational degrees-of-freedom of the wound roll at the rear drum, front drum and rider roll nips, respectively
w_β, w_α, w_ϕ	delay terms in the system equations at the rear drum, front drum and rider roll nips, respectively
τ_1, τ_2, τ_3	the relaxation times of the wound roll at the rear drum, front drum and rider roll nips, respectively

INTRODUCTION

This paper extends the single drum winder dynamic stability study presented at the previous IWEB [5] to two-drum winding. The background information is presented in that paper in detail and will not be repeated here in full. The novelty in this paper is the influence of the interplay between the winding drums and the rider roll on the system stability.

The most common self-excited winder vibration categories are [1]:

a) Vibration during the initial acceleration

This vibration mode occurs when the winder speed is accelerated from zero speed to full running speed. Typically, this vibration state develops fast at the very end of the acceleration stage. Typical paper grades for which this type of vibration can occur are rough and bulky grades with high COF - such as book papers. However, also some thin, coated and calendered paper grades can vibrate in this mode. Because of this vibration, the roll edges can become uneven and the web can brake. Although not confirmed, the vibration mode relating to this type of vibration is believed to be the one where the wound roll has the largest amplitude (wound roll eigenmode). Since the rolls are still quite light when the vibration takes place, the corresponding natural frequency is quite high, typically 40 – 150 Hz.

b) Roll bouncing, resulting in eccentricity
This is clearly the most serious vibration problem for two-drum winders nowadays. Typically, the grades experiencing this problem are easily wound up to the roll diameter 500 – 700 mm but then, little by little, start increasingly to develop eccentricity. The paper grades with a tendency to the above-mentioned vibration include DIP newsprint and bag paper. This vibration mode occurs always at the roll rotation frequency and is hence not accompanied with audible sound. On a two-drum winder the rolls are seen to bounce in a more or less irregular pattern from drum to drum. Due to the roll eccentricity, also the core chucks are vibrating heavily. The mechanism and mode for this type of vibration is quite complex, involving interplay of the adjacent rolls due to the edge contact and frictional forces.

c) Wound roll excited drum resonance vibration
This vibration occurs generally at steady running speed when a multiple of the wound roll rotation frequency matches or is in the vicinity of the drum natural frequency. Depending on the running speed and the value of the natural frequency, the multiple number of the rotation frequency can be 2, 3, 4 or 5. Paper grades vibrating in this mode include uncoated fine paper and sackkraft.

One of the earliest papers where the essential features of this self-excited vibration mechanism were explained was written by Daly [1]. Without any modeling, Daly explained the vibration phenomena using a washboard road analogy. Later Möhle *et al.* [2] studied two-drum winder vibrations and developed a simple one-degree-of-freedom mathematical model based on experimental observations. In their model, the generation of the wavy roll was implemented as a purely plastic residual deformation developed one revolution earlier and reentering the nip. Although the model was simple, they could nicely explain the unstable regions occurring at certain roll rotation frequencies. A more comprehensive two-drum winder model including all interacting structural elements of the wind up was presented by Jorkama [3]. Various eigenmodes of the wind up were presented and requirements for damping the vibrations were studied. However, the self-enforcing development of the wavy roll surface was omitted. Sueoka *et al.* [4] have presented an analogous calender rubber roll-covering polygonalization model. In their model, the development of the roll surface deformation pattern is based on a viscoelastic model of the behavior of the rubber cover. Their model results in a constant coefficient, linear, time delay ordinary differential equation system which stability is extensively studied. Jorkama and von Hertzen [5] studied a similar problem as in this paper but only for single nip winding. Even with that model it was possible to explain the basic nature of the above mentioned winder vibration categories.

The present paper follows the outline of Ref. [5]. The model presented in this paper enables the study of the influence of the interplay between the winding drums and the rider roll on the system stability. Due to the complexity of the system, the analysis is restricted to the study of a certain resonance condition, which falls in to the category c) above. The framework laid in this paper provides a variety of possibilities for further analysis. For example, the influence of the stiffness and damping properties of the winder structure would be valuable for the winder manufacturer to know.

THEORY

Consider the wind up model of Figure 1 consisting of the rear drum, front drum, rider roll and the paper roll. The drums and the rider roll are modeled as two degree of freedom systems with co-ordinates shown in the figure. The covers of the drums and rider roll are undeformable but the paper roll has additional internal degrees of freedom along the nip lines enabling the roll surface to become deformed. The deviation of the paper roll's shape from circular at the rear drum, front drum and rider roll nips (precisely just before entering the nip) is denoted by $e_1(t)$, $e_2(t)$ and $e_3(t)$, respectively. In addition, the paper roll has horizontal and vertical translational degrees-of-freedom, denoted by x and y, respectively.

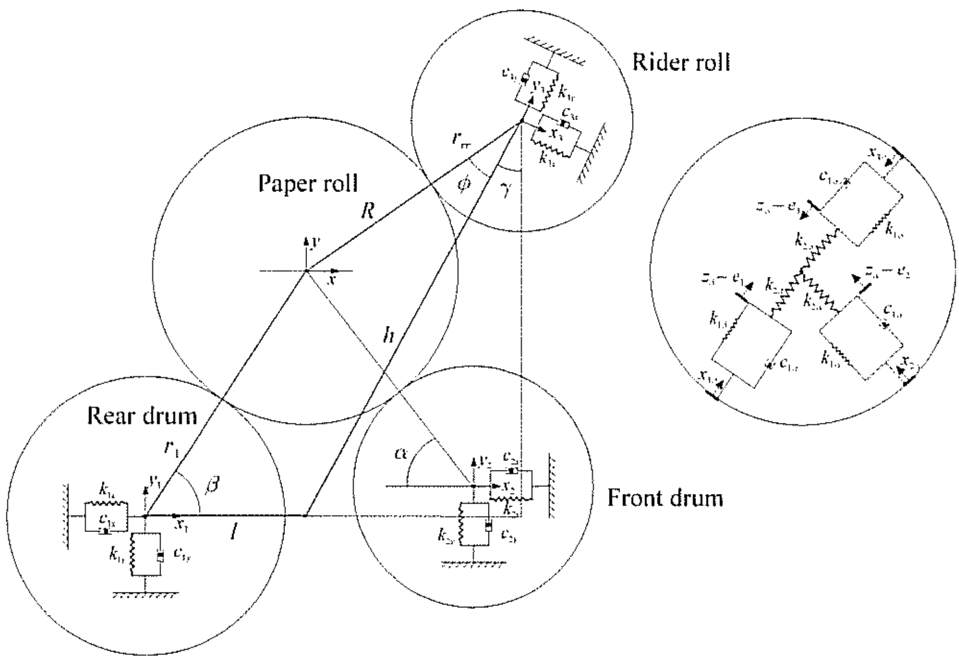

Figure 1. Winder model geometry and co-ordinates, roll internal degrees of freedom depicted right.

The equations of motion for the rear drum are

$$
\begin{aligned}
&m_1\ddot{x}_1 + c_{1x}\dot{x}_1 + k_{1x}x_1 - k_{1\beta}\cos\beta\left[z_\beta - x_{1\beta} - e_1(t)\right] - c_{1\beta}\cos\beta\left[\dot{z}_\beta - \dot{x}_{1\beta} - \dot{e}_1(t)\right] = 0\ , \\
&m_1\ddot{y}_1 + c_{1y}\dot{y}_1 + k_{1y}x_1 - k_{1\beta}\sin\beta\left[z_\beta - x_{1\beta} - e_1(t)\right] - c_{1\beta}\sin\beta\left[\dot{z}_\beta - \dot{x}_{1\beta} - \dot{e}_1(t)\right] = 0\ , \quad (1) \\
&k_{2\beta}\left(x\cos\beta + y\sin\beta - z_\beta\right) - k_{1\beta}\left[z_\beta - x_{1\beta} - e_1(t)\right] - c_{1\beta}\left[\dot{z}_\beta - \dot{x}_{1\beta} - \dot{e}_1(t)\right] = 0\ ,
\end{aligned}
$$

the equations of motion for the front drum are

$$m_2\ddot{x}_2 + c_{2x}\dot{x}_2 + k_{2x}x_2 + k_{1\alpha}\cos\alpha\left[z_\alpha - x_{2\alpha} - e_2(t)\right] + c_{1\alpha}\cos\alpha\left[\dot{z}_\alpha - \dot{x}_{2\alpha} - \dot{e}_2(t)\right] = 0 ,$$
$$m_2\ddot{y}_2 + c_{2y}\dot{y}_2 + k_{2y}y_2 - k_{1\alpha}\sin\alpha\left[z_\alpha - x_{2\alpha} - e_2(t)\right] - c_{1\alpha}\sin\alpha\left[\dot{z}_\alpha - \dot{x}_{2\alpha} - \dot{e}_2(t)\right] = 0 , \quad (2)$$
$$k_{2\alpha}\left(-x\cos\alpha + y\sin\alpha - z_\alpha\right) - k_{1\alpha}\left[z_\alpha - x_{2\alpha} - e_2(t)\right] - c_{1\alpha}\left[\dot{z}_\alpha - \dot{x}_{2\alpha} - \dot{e}_2(t)\right] = 0 ,$$

the equations of motion of the rider roll are

$$m_3\ddot{x}_3 + c_{3x}\dot{x}_3 + k_{3x}x_3 + k_{1\phi}\sin\phi\left[x_{3\phi} - z_\phi - e_3\right] + c_{1\phi}\sin\phi\left[\dot{x}_{3\phi} - \dot{z}_\phi - \dot{e}_3\right] = 0 ,$$
$$m_3\ddot{y}_3 + c_{3y}\dot{y}_3 + k_{3y}y_3 + k_{1\phi}\cos\phi\left[x_{3\phi} - z_\phi - e_3\right] + c_{1\phi}\cos\phi\left[\dot{x}_{3\phi} - \dot{z}_\phi - \dot{e}_3\right] = 0 , \quad (3)$$
$$k_{1\phi}\left[x_{3\phi} - z_\phi - e_3\right] + c_{1\phi}\left[\dot{x}_{3\phi} - \dot{z}_\phi - \dot{e}_3\right] - k_{2\phi}\left[z_\phi - x\sin(\phi+\gamma) - y\cos(\phi+\gamma)\right] = 0,$$

and, finally, the equations of motion of the paper roll are

$$m\ddot{x} = -k_{1\beta}\cos\beta\left[z_\beta - x_{1\beta} - e_1(t)\right] - c_{1\beta}\cos\beta\left[\dot{z}_\beta - \dot{x}_{1\beta} - \dot{e}_1(t)\right] + k_{1\alpha}\cos\alpha\left[z_\alpha - x_{2\alpha} - e_2(t)\right]$$
$$+c_{1\alpha}\cos\alpha\left[\dot{z}_\alpha - \dot{x}_{2\alpha} - \dot{e}_2(t)\right] - k_{1\phi}\sin\phi\left[x_{3\phi} - z_\phi - e_3(t)\right] - c_{1\phi}\sin\phi\left[\dot{x}_{3\phi} - \dot{z}_\phi - \dot{e}_3(t)\right] = 0 , \quad (4)$$
$$m\ddot{y} = -k_{1\beta}\sin\beta\left[z_\beta - x_{1\beta} - e_1(t)\right] - c_{1\beta}\sin\beta\left[\dot{z}_\beta - \dot{x}_{1\beta} - \dot{e}_1(t)\right] - k_{1\alpha}\sin\alpha\left[z_\alpha - x_{2\alpha} - e_2(t)\right] -$$
$$c_{1\alpha}\sin\alpha\left[\dot{z}_\alpha - \dot{x}_{2\alpha} - \dot{e}_2(t)\right] - k_{1\phi}\cos\phi\left[x_{3\phi} - z_\phi - e_3(t)\right] - c_{1\phi}\cos\phi\left[\dot{x}_{3\phi} - \dot{z}_\phi - \dot{e}_3(t)\right] = 0 .$$

Identically to the treatment of Ref. [5], the e_1, e_2 and e_3 functions will be eliminated from Eqs. (1) - (4) by expressing them as functions of the deformations at the previous nips, i.e.,

$$e_i = -u_i(t - T_i)e^{-t/\tau_i} \ , \ i = 1,2,3, \quad (5)$$

where

$$u_1 = x_{1\beta} - z_\beta, \quad u_2 = x_{2\alpha} - z_\alpha, \quad u_3 = z_\phi - x_{3\phi} ,$$

and (6)

$$\tau_1 = \frac{c_{1\beta}}{k_{1\beta}}, \quad \tau_2 = \frac{c_{1\alpha}}{k_{1\alpha}}, \quad \tau_3 = \frac{c_{1\phi}}{k_{1\phi}} .$$

Above the τ_i's are the characteristic relaxation times of the rear drum, front drum and rider roll nip contacts, respectively, and T_1 is the travel time from the front drum nip to the rear drum nip, T_2 is the travel time from the rider roll nip to the front drum nip and T_3 is the travel time from the rear drum nip to the rider roll nip.

Employing similar algebraic manipulations as in Ref. [5] the equations of motion of the rear drum become

$$
\begin{aligned}
&m_1\ddot{x}_1 + c_{1x}\dot{x}_1 + \left(k_{1x} + k_{2\beta}\cos^2\beta\right)x_1 + k_{2\beta}\cos\beta\sin\beta y_1 - k_{2\beta}\cos^2\beta x - \\
&k_{2\beta}\cos\beta\sin\beta y - k_{2\beta}\cos\beta\left(x_{1\beta} - z_\beta\right) = 0\ , \\
&m_1\ddot{y}_1 + c_{1y}\dot{x}_1 + \left(k_{1y} + k_{2\beta}\sin^2\beta\right)y_1 + k_{2\beta}\cos\beta\sin\beta x_1 - k_{2\beta}\cos\beta\sin\beta x - \\
&k_{2\beta}\sin^2\beta y - k_{2\beta}\sin\beta\left(x_{1\beta} - z_\beta\right) = 0\ , \\
&k_{1\beta}w_1 + c_{1\beta}\dot{w}_1 + k_{2\beta}\left(x_{1\beta} - z_\beta\right) + k_{2\beta}\cos\beta x + k_{2\beta}\sin\beta y - k_{2\beta}\cos\beta x_1 - \\
&k_{2\beta}\sin\beta y_1 = 0\ ,
\end{aligned} \tag{7}
$$

the front drum equations of motion become

$$
\begin{aligned}
&m_2\ddot{x}_2 + c_{2x}\dot{x}_2 + \left(k_{2x} + k_{2\alpha}\cos^2\alpha\right)x_2 - k_{2\alpha}\cos\alpha\sin\alpha y_2 - k_{2\alpha}\cos^2\alpha x + \\
&k_{2\alpha}\cos\alpha\sin\alpha y + k_{2\alpha}\cos\alpha\left(x_{2\alpha} - z_\alpha\right) = 0\ , \\
&m_2\ddot{y}_2 + c_{2y}\dot{y}_2 + \left(k_{2y} + k_{2\alpha}\sin^2\alpha\right)y_2 - k_{2\alpha}\cos\alpha\sin\alpha x_2 + k_{2\alpha}\cos\alpha\sin\alpha x - \\
&k_{2\alpha}\sin^2\alpha y - k_{2\alpha}\sin\alpha\left(x_{2\alpha} - z_\alpha\right) = 0\ , \\
&k_{1\alpha}w_2 + c_{1\alpha}\dot{w}_2 + k_{2\alpha}\left(x_{2\alpha} - z_\alpha\right) - k_{2\alpha}\cos\alpha x + k_{2\alpha}\sin\alpha y + k_{2\alpha}\cos\alpha x_2 - \\
&k_{2\alpha}\sin\alpha y_2 = 0\ ,
\end{aligned} \tag{8}
$$

the rider roll equations become

$$
\begin{aligned}
&m_3\ddot{x}_3 + c_{3x}\dot{x}_3 + \left(k_{3x} + k_{2\phi}\sin^2\phi\right)x_3 + k_{2\phi}\cos\phi\sin\phi y_3 - k_{2\phi}\sin\phi\sin(\phi+\gamma)x - \\
&k_{2\phi}\sin\phi\cos(\phi+\gamma)y - k_{2\phi}\sin\phi\left(x_{3\phi} - z_\phi\right) = 0\ , \\
&m_3\ddot{y}_3 + c_{3y}\dot{y}_3 + \left(k_{3y} + k_{2\phi}\cos^2\phi\right)y_3 + k_{2\phi}\cos\phi\sin\phi x_3 - k_{2\phi}\cos\phi\sin(\phi+\gamma)x - \\
&k_{2\phi}\cos\phi\cos(\phi+\gamma)y - k_{2\phi}\cos\phi\left(x_{3\phi} - z_\phi\right) = 0\ , \\
&k_{1\phi}w_3 + c_{1\phi}\dot{w}_3 - k_{2\phi}\left(x_{3\phi} - z_\phi\right) - k_{2\phi}\sin(\phi+\gamma)x - k_{2\phi}\cos(\phi+\gamma)y + \\
&k_{2\phi}\sin\phi x_3 + k_{2\phi}\cos\phi y_3 = 0\ ,
\end{aligned} \tag{9}
$$

and, finally, the paper roll equations become

$$
\begin{aligned}
& m\ddot{x}+\left[k_{2\beta}\cos^2\beta+k_{2\alpha}\cos^2\alpha+k_{2\phi}\sin^2(\phi+\gamma)\right]x+\left[k_{2\beta}\cos\beta\sin\beta-k_{2\alpha}\cos\alpha\sin\alpha\right. \\
& \left.+k_{2\phi}\cos(\phi+\gamma)\sin(\phi+\gamma)\right]y-k_{2\beta}\cos^2\beta x_1-k_{2\beta}\cos\beta\sin\beta y_1-k_{2\alpha}\cos^2\alpha x_2 \\
& +k_{2\alpha}\cos\alpha\sin\alpha y_2-k_{2\phi}\sin\phi\sin(\phi+\gamma)x_3-k_{2\phi}\cos\phi\sin(\phi+\gamma)y_3+ \\
& k_{2\beta}\cos\beta\left(x_{1\beta}-z_\beta\right)-k_{2\alpha}\cos\alpha\left(x_{2\alpha}-z_\alpha\right)+k_{2\phi}\sin(\phi+\gamma)\left(x_{3\phi}-z_\phi\right)=0\ , \\
& m\ddot{y}+\left[k_{2\beta}\sin^2\beta+k_{2\alpha}\sin^2\alpha+k_{2\phi}\cos^2(\phi+\gamma)\right]y+\left[k_{2\beta}\cos\beta\sin\beta-k_{2\alpha}\cos\alpha\sin\alpha\right. \\
& \left.+k_{2\phi}\cos(\phi+\gamma)\sin(\phi+\gamma)\right]x-k_{2\beta}\cos\beta\sin\beta x_1-k_{2\beta}\sin^2\beta y_1-k_{2\alpha}\sin^2\alpha y_2 \\
& +k_{2\alpha}\cos\alpha\sin\alpha x_2-k_{2\phi}\sin\phi\cos(\phi+\gamma)x_3-k_{2\phi}\cos\phi\cos(\phi+\gamma)y_3+ \\
& k_{2\beta}\sin\beta\left(x_{1\beta}-z_\beta\right)+k_{2\alpha}\sin\alpha\left(x_{2\alpha}-z_\alpha\right)+k_{2\phi}\cos(\phi+\gamma)\left(x_{3\phi}-z_\phi\right)=0\ .
\end{aligned}
\tag{10}
$$

Above the driving terms are

$$
\begin{aligned}
w_1 &= u_1(t)-u_1(t-T_1)e^{-t/\tau_1}\ , \\
w_2 &= u_2(t)-u_2(t-T_2)e^{-t/\tau_2}\ , \\
w_3 &= u_3(t)-u_3(t-T_3)e^{-t/\tau_3}\ .
\end{aligned}
\tag{11}
$$

It should be noted that in the analysis the relaxation times are taken as independent parameters since with the current constitutive model it is not possible to describe simultaneously the nip damping and the dent recovery on the roll surface.

In order to analyze the stability of the system, the Laplace transformation is performed for Eqs. (7) - (10). The travel times T_i are treated as constants when the Laplace transformation is done. During most of the winding cycle, this is a good assumption since the angular frequency of the roll does not change much during one roll revolution. When all the initial values are set to zero, the Laplace transformation yields the system equation

$$
\mathbf{AX}=\mathbf{0}\ , \tag{12}
$$

where

$$
\begin{aligned}
\mathbf{X} &= (X_1,Y_1,X_{1\beta}-Z_\beta,X_2,Y_2,X_{2\alpha}-Z_\alpha,X_3,Y_3,X_{3\phi}-Z_\phi,X,Y)'\ , \\
\mathbf{0} &= (0,0,0,0,0,0,0,0,0)'\ .
\end{aligned}
\tag{13}
$$

The system matrix $\mathbf{A}$ is obtained in a straightforward manner from the equations of motion (7) - (10) and is not given here. The stability of the system is studied by inspecting the roots of the characteristic equation

$$
\det\mathbf{A}=0\ . \tag{14}
$$

Since there exist time lags, the number of characteristic roots is infinite. The system is stable if all the real parts of the roots are negative. On the other hand, the system is unstable if one or more of the real parts are positive. In Ref. [4] it is shown that the number of roots with positive real part is finite.

RESULTS

Characteristic of winder dynamics is that both the excitation and natural frequencies change with time. The change of the excitation frequency stems from the constantly increasing roll diameter and changing running speed, which consists of acceleration, steady running speed and deceleration. The natural frequencies change because the mass of the paper roll increases during winding. According to experimental winder vibration studies [3], it is known that in severe vibration cases the excitation source is almost exclusively the wound roll. Hence, it is instructive to start with determining the situations when the multiples of the roll rotation frequency hit the resonances. The calculated results in this section are for a system without a rider roll. The system parameters used in the calculations are determined experimentally. Drum parameters are determined from modal tests of the drums and paper roll parameters from relaxation tests of paper samples. The measured relaxation times vary from 1 to 4 seconds.

Figure 2 shows the evolution of the natural frequencies of the winder during a winding cycle. The system parameters used in the calculation are shown in **Table 1**. From the figure it can be seen, that some of the frequencies remain relative constant as the diameter grows, while some other natural frequencies change considerably with the roll diameter. The former ones correspond to natural modes where the paper roll movement is relatively small, whereas the latter ones are modes with large paper roll relative movement. It should be noted that during the evolution of time, this character can change considerably. In other words, the mode shapes are also changing with roll diameter.

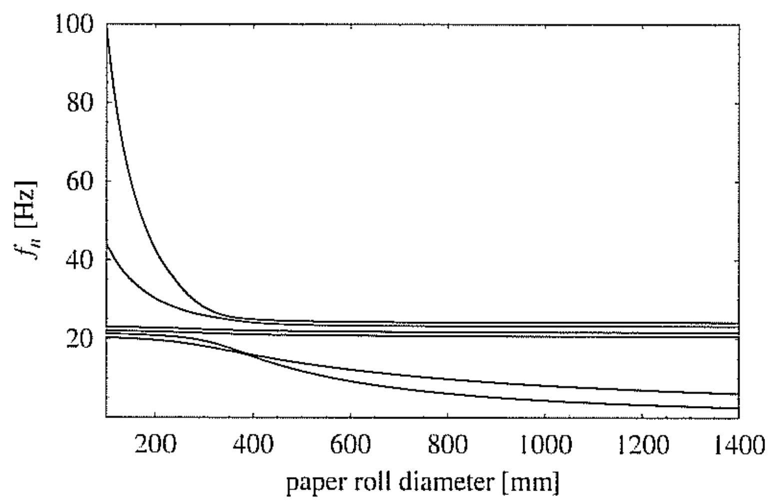

Figure 2. Evolution of the natural frequencies of the system.

Figure 3 depicts the resonance condition when the 3rd harmonic of the roll rotation frequency excites the vertical natural frequency of the drums. The left panel shows the evolution of the natural frequency and the 3rd harmonic during the winding cycle. These curves cross at approximately 1250 mm roll diameter. The right panel shows the evolution of the real part of the corresponding root of the system stability equation (14) with low and intermediate drum damping. When the damping is low, the system comes unstable in the vicinity of the resonance, whereas for intermediate damping the system is stable. This supports what has been observed in practice for "wound roll excited drum resonance vibration" cases, i.e., the drum damping is an important parameter influencing the winder sensitivity to category c) vibration.

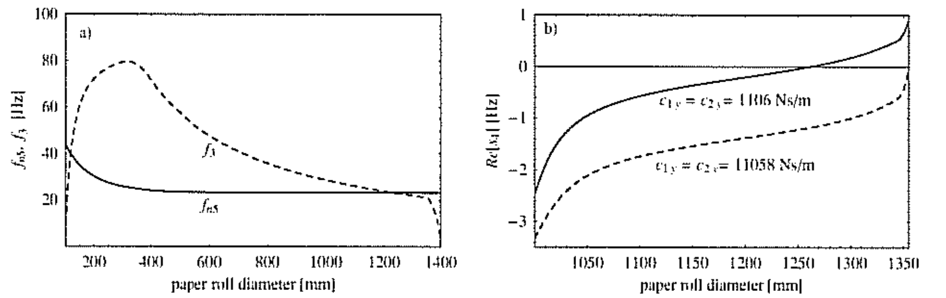

Figure 3. a) Evolution of the vertical natural frequency of the drums and the 3 harmonic of the roll rotation frequency during the winding cycle and b) evolution of the real part of the corresponding characteristic root for two drum vertical direction damping cases.

Table 1.

Parameter values used in the calculations

Parameter	Notation	Value
Paper thickness	τ	100 μm
Acceleration	a	0.5 m/s^2
Deceleration	b	0.5 m/s^2
Steady state running speed	v_0	30 m/s
Rounding time of the running speed curve	t_p	12 s
Core diameter	d_0	0.1 m
Final roll diameter	d_e	1.40 m
Web width	L	8 m
Paper density	ρ	750 kg/m^3
Rear drum mass	m_1	4000 kg
Front drum mass	m_2	4000 kg
Rear drum horizontal stiffness	k_{1x}	73.0 MN/m
Rear drum vertical stiffness	k_{1y}	83.5 MN/m
Front drum horizontal stiffness	k_{2x}	66.3 MN/m
Front drum vertical stiffness	k_{1y}	76.5 MN/m
Rear drum horizontal damping coeff.	c_{1x}	16.21 kNs/m
Rear drum vertical damping coeff.	c_{1y}	11.06 kNs/m
Front drum horizontal damping coeff.	c_{2x}	16.21 kNs/m
Rear drum vertical damping coeff.	c_{2y}	11.06 kNs/m
Mass of the core/length	m_0	5 kg/m

Stiffness coefficient of the viscoelement, rear drum nip	$k_{1\beta}$	100 MN/m
Stiffness coefficient of the viscoelement, front drum nip	$k_{1\beta}$	100 MN/m
Damping coefficient of the viscoelement, rear drum nip	$c_{1\beta}$	10 kNs/m
Damping coefficient of the viscoelement, front drum nip	$c_{1\alpha}$	10 kNs/m
Stiffness coefficient of the roll, rear drum nip	$k_{2\beta}$	10 MN/m
Stiffness coefficient of the roll, front drum nip	$k_{2\alpha}$	10 MN/m
Relaxation time of the viscoelement, rear drum nip	τ_1	1 s
Relaxation time of the viscoelement, front drum nip	τ_2	1 s

CONCLUSIONS

With the presented model, the most common winder vibration cases can be represented. The importance of adequate damping of the winding drums was demonstrated by studying the resonance condition where the 3rd harmonic of the paper roll excited the drum vertical natural frequency. This unstability zone is identified as "wound roll excited drum resonance vibration" which is well known in practice.

There are several directions for further analysis. Among winder manufacturers the influence of the wind up geometry and the eigenproperties of the winding drums has been speculated for years. This model enables the possibility to study that theoretically. Another, very challenging approach, would be to improve the constitutive description of the wound roll.

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Name & Affiliation	**Question**
Keith Good Oklahoma State University	Thank you Marko, excellent talk. I enjoyed it. During your discussion you didn't say much about what the impact of how the winding operating conditions were set up. For instance, more torque to the rear drum would increase wound roll pressures and would increase the radial modulus of the wound roll. I would assume an increase in modulus would affect the frequencies. Do you study optimal winding profiles or how winding operating parameters in fact affect your dynamic work?
Name & Affiliation	**Answer**
Marko Jorkama Metso Paper	Very good question. I think that the situations are quite rare where you can help the vibration problems by changing the winding parameters. However, there are some cases. Usually this kind of vibration which was described can be helped by winding softer rolls. I'm unsure why it is so. Maybe the dampening of the roll is increasing. Or then the layer to layer movement is more easily restored because the internal pressure is lower. The natural frequencies are not affected that much with the winding parameters because there are some other requirements for the roll structure also.
Name & Affiliation	**Question**
Dan Carlson 3M	Can you clarify on the one plot where you showed the 3rd overtone and 5th overtones were coinciding? Why is the frequency for f_3 falling with increased diameter? Why is there a peak there?
Name & Affiliation	**Answer**
Marko Jorkama Metso Paper	This line is the natural frequency, but this line is the rotation frequency. When we accelerate the winder then the frequency is increasing despite the fact that the roller's diameter is increasing. Now when we reach the steady running speed then the frequency is dropping.