COMPREHENSIVE MODELLING OF A CALENDER STACK FOR PREDICTION OF OFFSETS

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ABSTRACT

One method to limit calender barring is to vary the offsets of the rolls. The goal is get the caliper travelling from nip to nip interfere destructively reducing the vibration. To predict the ideal offsets, a dynamic model of the vibrating stack is required. Previous models used point masses, connected by linear springs and proportional damping. This research uses the finite element method to add flexible rolls, non-proportional damping and complex vibration modes.

INTRODUCTION

The conventional calender stack is used to improve the finish of the paper after the drying section by pressing the paper in successive nips. At times the calender can start to vibrate for no apparent reason. This vibration, known as calender barring, causes further caliper variations [1]. The consequences include reduced paper quality, corrugated rolls[2] and increased machine down time due to roll maintenance.

One cause of calender barring is regenerative tendency [1], [3], [4], [5], [6] in which vibrating rolls cause caliper variations, which then excite subsequent nips. By varying the offsets (vertical alignment) of the rolls [7], [8], [9], the wrap length can be changed and thus influences the time required for a disturbance to travel from one nip to the next, which changes the phase relationship between barred paper and the vibrating roll. Traditionally, these offsets have been chosen by experienced operators. Prior efforts have aimed at a computational solution. However, the previous model did not account for flexible rolls, as suggested by Parker and Epton [10], or complex mode shapes.

METHODS

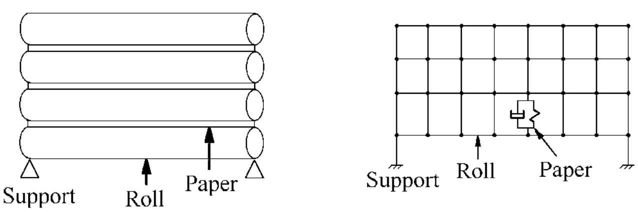
The focus of this research is to add flexible rolls, as in [2], and damping that is not proportional to the spring stiffness to the model. In order to accommodate flexible beam elements, a finite element solution was used. A complex eigen solver is required to solve the second order differential equation which yields complex natural frequencies, modal amplitudes and phase angles.

Finite Element Application

By considering the rolls as thin Euler-Bernoulli beams, a two noded, four-degree of freedom beam element is used to represent the calender rolls. From the dimensions and masses of each roll, the stiffness and mass element matrices are generated.

To obtain the spring stiffness and damping of the paper, the natural frequencies of the stack are measured using modal analysis. This information is used with the roll masses to calculate the spring constants. The method used was developed by Dr. Stu Shelley at the University of Cincinnati and models the stack as a series of point masses connected by springs. These spring constants represent the total stiffness of the web and can be divided by the trim to get stiffness per unit length. The spring constants and damping coefficients are distributed over the nodes of the mesh using a consistent loading scheme to more accurately represent the distributed nature of the paper.

Fig. 1. Calender Stack and Finite Element Model



It is assumed that the calender stack is supported by the king roll. The support for the king roll is represented as a spring and damper in parallel whose spring constant is calculated from the masses and measured frequencies as above.

Finite Element Solution

Once the system is discretized, the equations of motion are written as

$$M\{\ddot{x}\} + C\{\dot{x}\} + K\{x\} = \{0\} \tag{1}$$

As described by Tse et al [11], the equation is transformed into the form

$$\{\dot{y}\} - [H]\{y\} = \{0\} \tag{2}$$

$$\{y\} = \begin{Bmatrix} \dot{x} \\ x \end{Bmatrix} \tag{3}$$

$$-[H] = \begin{bmatrix} -M^{-1}C & -M^{-1}K \\ I_n & 0 \end{bmatrix} \tag{4}$$

By solving (4) using Hessenberg transformations and QR triangularizations as described in [12],[13] and [14], n pairs of complex natural frequencies are obtained where

$$n = 2 \cdot \left(n_{rolls} \cdot n_{nodes\cdot per\cdot roll}\right) \tag{5}$$

Examining equations (1) and (3) a reverse transformation is implemented to get

$$\left[M\omega_n^{\,2}+C\omega_n+K\right]\{x\}=\{0\} \tag{6}$$

Equation (6) is a linear algebraic problem yielding a complex solution due to the complex frequencies. These complex mode shapes are used in the offset optimisation program.

Offset Optimisation

Before the offsets can be selected, it is necessary to quantify how good a given offset combination is. This is performed by calculating the work done on the rolls adjacent to subsequent nips by the caliper variations. The work done is termed the regenerative tendency.

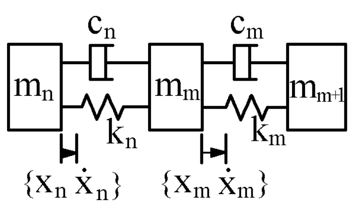


Fig. 2. Parameters for Nip Pair n & m

Calculation of Work Done

Considering a point mass spring system as in Figure 2, the force in the springs and dampers can be calculated for a given set of small displacements and small velocities. Multiplying the force by the velocity, the feedback power is obtained. Integrating over one period of the disturbances yields the work done. Mathematically, this is written as

$$W=\int_0^T F(t)\,\dot{x}_m(t)\,dt \tag{7}$$

$$F=k_n x_n(t)+c_n\,\dot{x}_n(t)-k_m x_m(t)-c_m\,\dot{x}_m(t) \tag{8}$$

This gives the work done for any nip pair *n* and *m*. The final expression, shown in Appendix A, contains the maximum amplitudes and phase angles of the rolls. The complex modal amplitudes and phase angles are substituted into these variables in order to obtain the work associated with a particular mode.

The time required for the sheet to travel between the nip pair in question is also present in the solution. When the offsets are varied, the paper wrap length between nips changes and phase between the caliper variation and the roll velocity changes which results in a change in the work done for that nip pair.

Total Regenerative Tendency

The expression for the work done requires a given set of offsets, one vibration mode, one machine speed and a single nip pair. To obtain the total regenerative tendency, the expression must be summed over all the nip pairs and all the modes. Since the expression is derived for a lump mass system, it is necessary to treat each vertical arrangement of nodes in the finite element mesh as a separate mass-spring system. When of these slices of the calender stack are added together, the total regenerative tendency is obtained for the whole stack and for all the natural frequencies as

$$R=\sum_{slc=1}^{n_{node}}\sum_{mde=1}^{n_{mode}}\left(\sum_{i=1}^{n_{roll}-2}\sum_{j=i+1}^{n_{roll}-1}\left(\left|W_{ij,mde,slc}\right|+W_{ij,mde,slc}\right)\right) \tag{9}$$

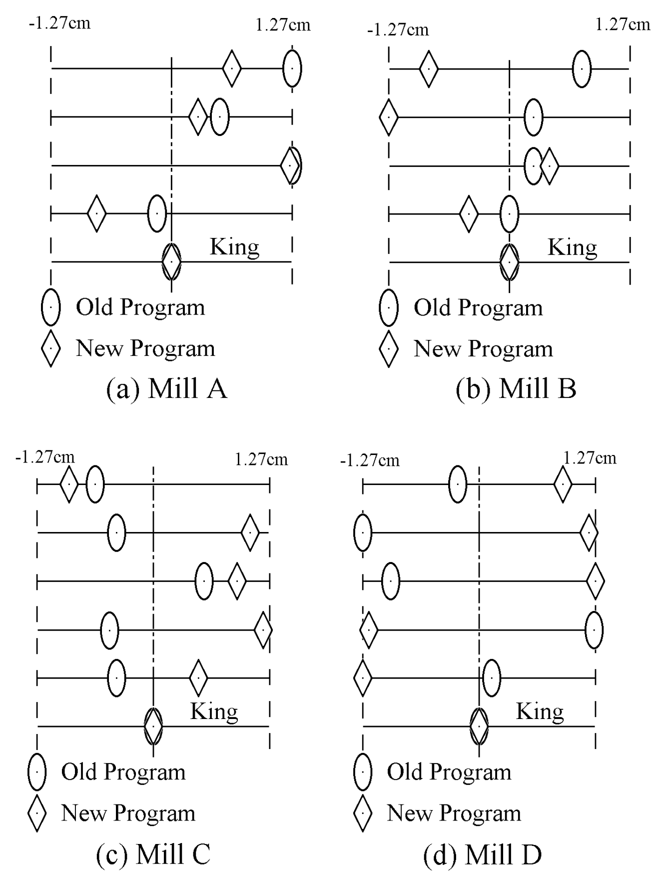
Only positive work is considered in equation (9). Previous work with the old model discovered a closer correlation to empirical data when this form is used. Since this research uses the previous program as comparison, the same general form of the equation is retained.

For each different set of offsets, this calculation must be repeated. The resulting total regenerative tendencies are compared for the various offset combinations and the offset selection with the minimum is returned as the optimal setting.

RESULTS

Data files were obtained for various machines. The machine characteristics, including masses, roll dimensions and spring constants were used in the new program's input files. The natural frequencies and mode shapes were calculated for both systems and compared. Given the identical offset limits and iteration resolution, each program determined the optimal offset setting for each file.

Fig. 3. Offset Predictions for Various Mills



As shown in Figure 3, offsets predicted by the current program did not correlate with the old program. However, the old program attained limited success which means that different results would be desired.

Some aspects of the new program differ greatly from the old program. One significant difference is the number of modes available to determine the regenerative tendency. With the old program, a five roll stack gave five mode shapes. Now with five rolls, one can obtain up to 100 modes. The question arises, which modes are important? For these comparisons, the first twenty modes were included in the calculation. Future research could include determining the frequency response function and then weighting each mode based on the relative magnitude.

The new program still needs to be tested. It is expected that the new configuration will reduce the barring on the stack. This would need to be done on several operating stacks to test for consistency.

CONCLUSION

The offsets predicted by the new model differed greatly from the old model in many cases. The old program gave inconsistent results in reducing barring, it is expected that the new model would yield different results. The new program needs to be tested on several operating calender stacks to verify that its predictions consistently decrease the amount of barring.

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APPENDIX A
Energy Feedback

$$E = -k_n \omega_i \pi \begin{pmatrix} A_{n,i} A_{m,i} \sin\left(\omega_i \cdot t_{nm} + \phi_{m,i} - \phi_{n,i}\right) \\ -A_{n,i} A_{m+1,i} \sin\left(\omega_i \cdot t_{nm} + \phi_{m+1,i} - \phi_{n,i}\right) \\ -A_{n+1,i} A_{m,i} \sin\left(\omega_i \cdot t_{nm} + \phi_{m,i} - \phi_{n+1,i}\right) \\ -A_{n+1,i} A_{m+1,i} \sin\left(\omega_i \cdot t_{nm} + \phi_{m+1,i} - \phi_{n+,i1}\right) \end{pmatrix}$$
$$-c_n \omega_i^{2} \pi \begin{pmatrix} A_{n,i} A_{m,i} \cos\left(\omega_i \cdot t_{nm} + \phi_{m,i} - \phi_{n,i}\right) \\ -A_{n,i} A_{m+1,i} \cos\left(\omega_i \cdot t_{nm} + \phi_{m+1,i} - \phi_{n,i}\right) \\ -A_{n+1,i} A_{m,i} \cos\left(\omega_i \cdot t_{nm} + \phi_{m,i} - \phi_{n+1,i}\right) \\ -A_{n+1,i} A_{m+1,i} \cos\left(\omega_i \cdot t_{nm} + \phi_{m+1,i} - \phi_{n+1,i}\right) \end{pmatrix}$$
$$+c_m \omega_i^{2} \pi \left(A_{m,i}^{2} - 2 A_{m,i} A_{m+1,i} \cos\left(\phi_{m,i} - \phi_{m+1,i}\right) + A_{m+1,i}^{2} \right)$$

where

k_n	is the nth spring constant
c_n	is the nth damping constant
ω_i	is the ith natural frequency
$A_{x,i}$	is the total mode shape amplitude of the xth roll for the ith mode
$\phi_{m,i}$	is the mode shape phase angle for the mth roll for the ith mode
t_{nm}	is the time required for a disturbance traveling from nip n to m.

APPENDIX B
Detailed Results

Table I. Mill A Frequencies

Mode	Frequencies (Hz) New	Old
1	4.252	4.000
2	7.430	127.8
3	60.67	250.8
4	133.2	363.8
5	148.9	448.7
6	165.6	
7	189.6	
8	255.4	
9	259.9	

Table II. Mill B Frequencies

Mode	Frequencies (Hz) New	Old
1	5.291	3.440
2	9.602	156.5
3	38.04	312.5
4	100.6	568.1
5	196.4	936.1
6	229.0	
7	234.5	
8	254.1	
9	291.5	

Table III. Mill C Frequencies

Mode	Frequencies (Hz) New	Old
1	4.956	5.000
2	8.640	89.54
3	77.75	181.4
4	85.21	266.7
5	100.9	350.7

Table IV. Mill D Frequencies

Mode	Frequencies (Hz) New	Old
1	5.218	5.389
2	9.216	84.50
3	50.55	173.4
4	78.78	238.0
5	92.18	298.3

6	138.4	456.5	6	120.0	354.6
7	167.5		7	134.8	
8	186.5		8	159.0	
9	199.4		9	170.3	

Table V. Mill A Offsets

Roll	Offsets (mm)	
	New	Old
5	6.35	12.7
4	2.79	5.08
3	12.4	12.7
Queen	-7.87	-1.52
King	0.00	0.00

Table VI.Mill B Offsets

Roll	Offsets (mm)	
	New	Old
5	-8.47	7.62
4	-12.7	2.54
3	4.23	2.54
Queen	-4.23	0.00
King	0.00	0.00

Table VII.Mill C Offsets

Roll	Offsets (mm)	
	New	Old
6	-9.17	-6.35
5	10.6	-3.97
4	9.17	5.56
3	12.0	-4.76
Queen	4.94	-3.97
King	0.00	0.00

Table VIII.Mill D Offsets

Roll	Offsets (mm)	
	New	Old
6	3.97	-2.33
5	12.0	-12.7
4	12.7	-9.63
3	-12.0	12.6
Queen	-12.7	1.38
King	0.00	0.00