Dynamic analysis of paper machine foundations

Sauren Guha-Majumdar and Makram A. Khoury

Today's faster, wider paper machines have made dynamic analysis of machine foundations a routine part of the design process. This has created an urgent need for industrywide guidelines covering the use of this design tool.

The rotating rolls in today's high-speed fine paper machines can initiate dynamic excitation within a frequency range of 2–20 Hz. Dynamic and static analyses of the machine and its support structure are essential if the machine is to run smoothly, thereby reducing the number of shutdowns, the amount of off-quality production, and maintenance costs. The advent of fast, high-capacity, cost-efficient computers has made it possible to analyze the interaction between complex machines and their support structures.

A fine paper machine generally comprises the following sections: wire, press, dryer, size press, calender and reel, and winder. The machine foundation includes the structural elements below the machine sole plates: sill beams, cross beams, columns, walls, and the building foundation, including the soil or pile support. In this paper, the combined machine and foundation is defined as a system.

The machine manufacturer and the consulting engineer responsible for designing the machine's foundation must thoroughly understand each other's analyses and work closely to construct a structurally sound system at the lowest cost. This article outlines the information requirements for dynamic analysis of a paper machine and its support structure. The report also provides a framework for analyzing and designing paper machine foundations.

Dynamic analysis

Effect of machine speed and width

Efforts to increase productivity have resulted in faster paper machines, with speeds increasing from 2500 ft/min to 5000 ft/min over the last 30 years (*1–5*). The higher machine speeds have made the vibration level an increasingly critical factor in analyzing machine operation. The sinusoidal harmonic excitation forces generated by a rotating roll is proportional to the square of the roll's rotational frequency. Thus the unbalanced dynamic force of a roll at 5000 ft/min is 1.56 times that of the same roll at 4000 ft/min and four times that of the roll at 2500 ft/min.

Guha-Majumdar, senior engineer, and Khoury, structural engineer, are affiliated with Brown and Root, Inc., 4100 Clinton Dr., Houston, TX 770020-6299.

Machine widths also have increased to accommodate demands for higher productivity. Fine paper machines are being manufactured with widths exceeding 400 in. Wider machines imply heavier machine components and sill beams. Since system frequency is inversely proportional to the square root of the mass, heavier components will result in a lower system frequency

Increases in machine speed and width have made it difficult to achieve a high-tuned system, i.e., a system whose first natural frequency is higher than the excitation of the highest roll frequency at the highest machine speed (for a roll of significant mass). Consequently, the machine–foundation system will, in many cases, have to be designed to operate under resonant conditions.

Data requirements

Machine manufacturers develop technical data, including vibration criteria and an analytical model, for each machine section. The consulting engineer must have access to this information in order to design an effective and efficient foundation for the paper machine.

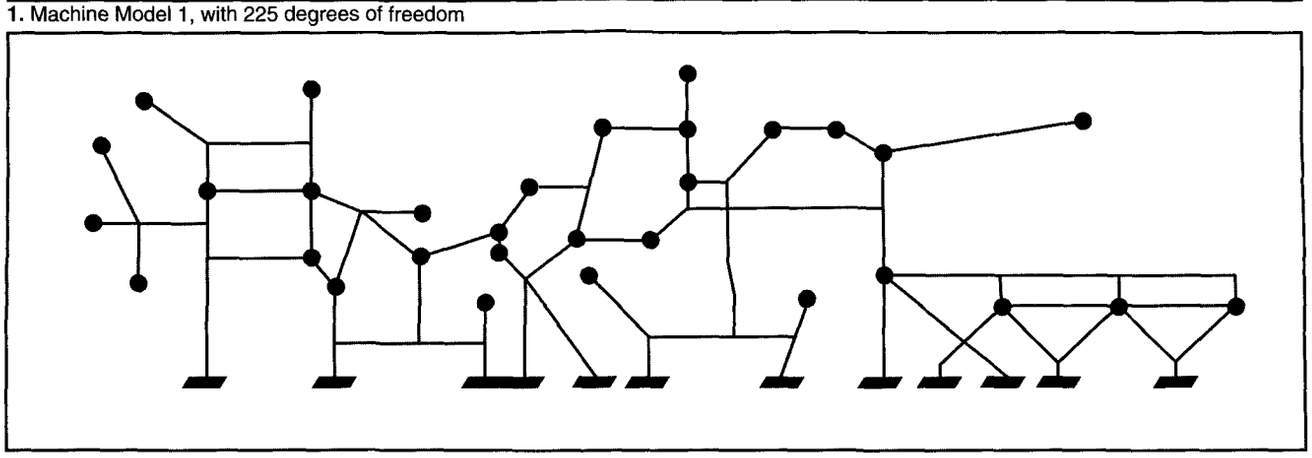
Analytical model. The manufacturer develops an analytical model for dynamic analysis to simulate the properties of the machine components. **Figure 1** is a simplified two-dimensional analytical model of the front and back sides of a machine section. The model consists of lumped masses at the joints connected by beam or truss elements that represent parts of the machine section. The dynamic models prepared by machine manufacturers are similar to this. The model's accuracy directly affects the results of the dynamic analysis of the system.

To demonstrate this point, the fundamental natural frequency of a system was determined using two different models:

- Model 1—A model of a machine section with 225 degrees of freedom and 85 members representing front and back sides. This is the model illustrated in Fig. 1.
- Model 2—A single-degree-of-freedom model of the same machine section. This model is illustrated in **Fig. 2**. The single-degree-of-freedom model has an equivalent mass, center of gravity, and stiffness, producing an equivalent fundamental natural frequency of the machine in the machine direction.

Both models were attached to the same foundation. Model 1 resulted in a fundamental natural frequency 20% higher than

1. Machine Model 1, with 225 degrees of freedom



Model 2. Thus Model 2 would produce (a) a conservative and expensive foundation for a high-tuned system or (b) an unsafe foundation for a low-tuned system, i.e., a system whose first natural frequency is lower than the excitation of the lowest roll frequency at the lowest machine speed.

In a force–response analysis, the results from Model 2 would have been unreliable. Although this difference in frequency was obtained for a particular case, a simplified single degree of freedom representing a machine section will generally yield a lower frequency. Model 1 detected coupled modes that would be missed in a model with a single degree of freedom. Given these results, it is imperative that machine manufacturers provide a detailed representative model of each machine section.

Damping ratios. Damping dissipates the energy in a vibrating system by suppressing vibratory motion. Energy is often dissipated in the form of heat loss. The dissipation of vibratory energy reduces the amplitude of vibration and makes it possible to operate a system even if it is in resonance.

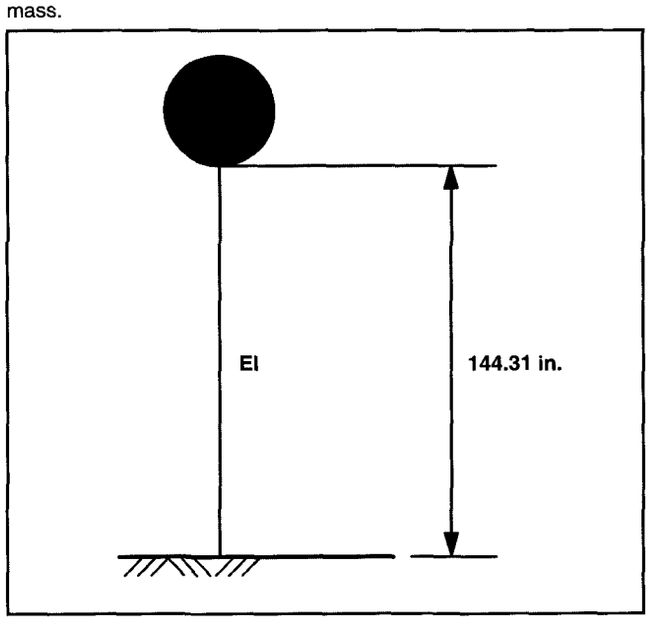
The damping ratio is the actual resistance in damped harmonic motion to that necessary to produce critical damping. In order to perform a force–response analysis, it is necessary to know the damping ratio of the system or its components (*5*). In the structural analysis, damping is in the form of dry friction and hysteresis loss, which are not well understood and therefore are approximated. Measurement of vibration amplitudes of operating machines will help determine the damping ratios of general paper machine components and systems. Such information will be helpful for future designs.

The manufacturer provides the consulting engineer with the damping ratio for the machine. The effect of damping is illustrated in **Fig. 3** (*6*), where the dynamic magnification factors are substantially higher for frequency ratios in the range of 0.75–1.25, especially where system damping ratios are low. The magnification factor, or dynamic load amplification factor, increases by a factor of up to 50 for a damping ratio of 1%.

Vibration criteria. Vibration analysis is done in the three principal axes of each machine section. Vibration criteria differ for each of the following situations:

- High-tuned or low-tuned system
- System in resonance.

2. Machine Model 2, single-degree-of-freedom system. EI is rigidity of member, and 144.31 in. is the location of center of gravity of machine mass.



High-tuned or low-tuned system. In a high-tuned system, the first natural frequency of the system is higher than the excitation of the highest roll frequency at the highest machine speed (for a roll of significant mass). In a low-tuned system, the first natural frequency of the system is lower than the excitation of the lowest roll frequency at the lowest machine speed. Excitation frequencies are determined using Eq. 1.

$$f_i = 12S_i/60D_i\pi \qquad (1)$$

where

f_i = excitation frequency (f_1 = highest, f_2 = lowest), Hz

D_i = roll diameter (D_1 = smallest, D_2 = largest), in.

S_i = design speed (S_1 = highest, S_2 = lowest), ft/min

Manufacturers commonly recommend the following criteria for the vertical frequency (f_v) and horizontal frequency (f_h) of the system:

$f_v > 2 \cdot f_1$

$f_h > 1.2 \cdot f_1$ or $< 0.7 \cdot f_2$

For a system satisfying the high-tuned criteria, a force–response analysis is generally not required because of the excellent track record of machine performance.

Operating speeds for fine paper machines typically range from 2500 ft/min to 4500 ft/min. The excitation frequency of a 38-in.-diam. roll ranges from 4.19 Hz to 7.55 Hz for this speed range, while the excitation frequency of a 60-in.-diam. roll ranges from 2.65 Hz to 4.78 Hz. In a machine section containing both 38-in. and 60-in. rolls, excitation frequencies can range from 2.65 Hz to 7.55 Hz. Using the design-criteria factors of 0.7 and 1.2 for horizontal frequency, the design range in the machine direction would be 1.86 Hz to 9.06 Hz. Based on our experience, the fundamental natural frequency of a system ranges from 3 Hz to 7 Hz, depending on the machine and foundation properties. Resonant conditions prevail at different production rates with different machine rolls.

System in resonance. In addition to a model, damping ratio, and pseudodynamic loading of each machine section, the following information is needed to perform a force–response analysis of the system:

- Forcing function
- Vibration amplitude
- Phase angle.

1. Forcing function: The design criteria include the unbalanced force for each roll at different machine-speed intervals. The unbalanced forces represent dynamic time-dependent excitation on the machine components that the industry will tolerate, considering the effect on equipment wear and paper quality. The exciting force, F, for an unbalanced rotating mass is given in Eq. 2.

$$F = me\omega^2[\sin(\omega t + \phi)] \quad (2)$$

where

m = unbalanced rotating mass

e = eccentricity of unbalanced mass

ω = angular frequency of the roll

t = time

ϕ = phase angle of the rotating mass

2. Vibration amplitude: For each machine section, this criterion specifies allowable vibration amplitudes at critical locations. Limits for vibration of general rotating machinery are shown in **Fig. 4** (*7, 8*), where the upper line in zone B is the allowable vibration amplitude. Similar vibration criteria are needed for the pulp and paper industry.

3. Phase angle: The phase angle specifies the time relationship between two rolls with the same frequency rotating such that their peak values of the same sign (positive or negative) do not occur simultaneously (*6, 9*). A common statistical approach is to use the square root of the sum of squares of responses of each roll. A conservative approach is to add the absolute response of the two rolls.

Designing the foundation

The structural engineer must satisfy both static and dynamic requirements while resolving layout problems.

Static analysis

The consulting engineer applies the loads provided by the manufacturer at the specified locations. Analysis is performed using commonly available finite-element computer software (*10–13*). The concrete and steel members of the foundation are designed to comply with ACI (*14*) and ASCE (*15*) codes and to meet the manufacturer's deflection criteria (*16*).

Several factors are considered in the analysis and design of the foundations:

- Shear deformation (especially when depth-to-span ratio is small)
- Properties of uncracked concrete sections (when the level of stress is low)
- Clear span—the effective length of the beam—and the use of rigid links at joints where structural members overlap
- Design of sill beams using deep-beam theory.

Isolation of machine sections

Designers have not always routinely performed dynamic analyses of paper machine systems. The need was not as compelling as it is on today's high-speed machines, and analysis was complicated by the widespread practice of attaching machine foundations to the operating floor.

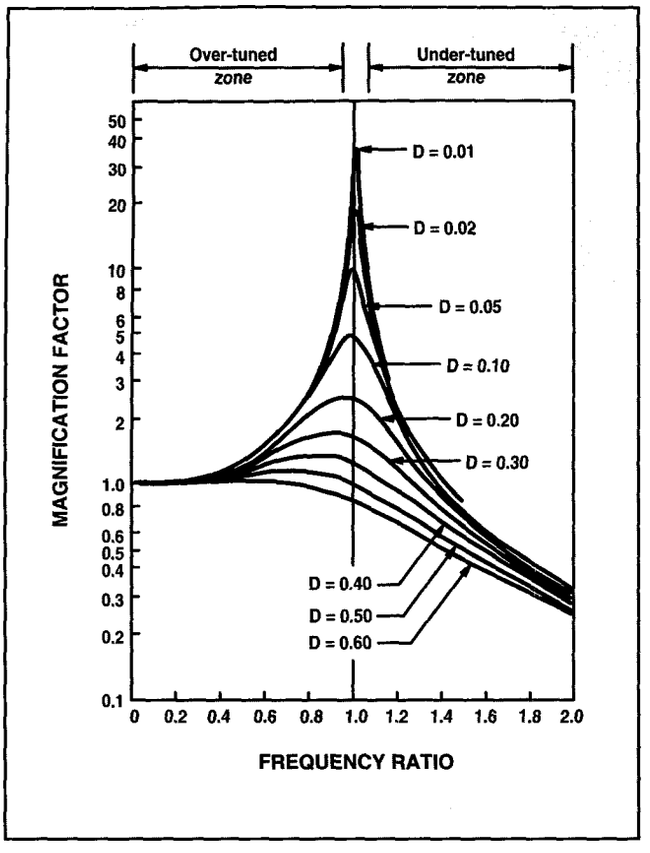
Foundations for modern paper machines are not connected to the operating floor, and the foundation for each machine section is typically isolated from the others. Isolation eliminates transmission of vibration between the building and the machine sections and from one section to another. Isolation of the dryer sections also helps control deflection from thermal expansion (*3*).

Frequency analysis

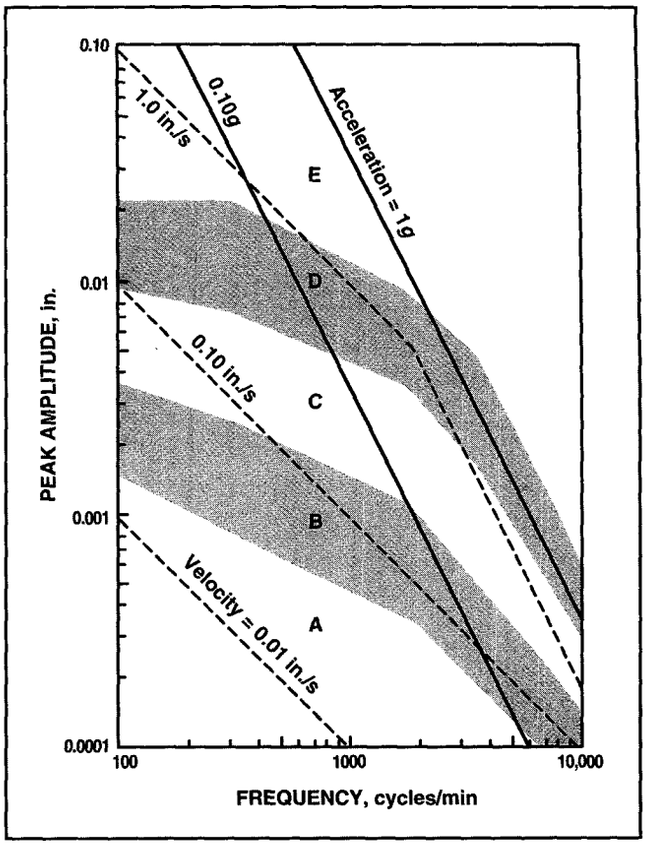
The consulting engineer must satisfy the frequency requirements specified by the machine manufacturer. The first step in this process is to perform a frequency analysis on the combined analytical model of the machine, its support structure, and the foundation piles (or the soil). For a rigid foundation supported on soil, the equivalent spring constants and damping ratios can be obtained from Tables 10-13 and 10-14 of Richart *et al.* (*17*). The spring constants for piles or shear modulus of soil are determined by soil consultants. Methods of interpreting field test are described by Richart *et al.* (*17*).

A three-dimensional analytical model is recommended in cases where geometry and mass distribution are asymmetrical. If the fundamental frequency of the system is much higher than 1.2 times the highest excitation frequency, cost can be reduced by reducing the stiffness of the structure. If frequency criteria are not met, a force–response analysis is performed.

3. Vibration magnification factor as a function of frequency ratio (excitation frequency/natural frequency) at various levels of damping (*D*)



4. Peak horizontal vibration amplitude (measured at the bearing) as a function of rotational frequency. Upper limit of zone B is the maximum allowable amplitude for general rotating machinery.



Force–response analysis

Dynamic loading and the steady-state response of each machine section is harmonic and sinusoidal. The displacements, velocities, or accelerations at or near resonance are obtained from a force–response analysis. The equation of dynamic equilibrium solved in a harmonic-response analysis is given in Eq. 3:

$$\{F\} = [M]\{D''\} + [C]\{D'\} + [K]\{D\} \quad (3)$$

where

$\{F\}$ = load amplitude vector

$[M]$ = mass matrix

$[C]$ = damping matrix

$[K]$ = stiffness matrix

$\{D''\}$ = acceleration vector

$\{D'\}$ = velocity vector

$\{D\}$ = displacement vector

Mathematically, the equation represents a series of second-order differential equations. Some commonly used computer programs for performing force–response analysis are STRUDL, NASTRAN, ANSYS, and SAP (*10–13*). System response is determined over the frequency range that corresponds to the operating range of the roll under consideration. Separate analyses are performed for each exciting roll, and the response is measured at the required locations. Since the analysis is linear, the method of superposition is used to determine the final response.

In finite-element analysis, a damping matrix is created to solve the response of the time-related dynamic forcing function. A common method (*9, 18*) is to combine a fraction, α, of the stiffness matrix with a fraction, β, of the mass matrix, shown in Eq. 4.

$$[C] = \alpha[K] + \beta[M] \quad (4)$$

Equation 4 is the Rayleigh or the proportional damping. With this damping matrix, the set of second-order differential equations described are linear, and the mode shapes are decoupled. If $\beta = 0$, the higher modes are lightly damped. If $\alpha = 0$, the higher modes are heavily damped. The term α is the dominating factor in dynamic analysis of machine systems, since the excitation is close to the first few natural frequencies of the system. The stiffness of the system is the most critical factor in determining vibration amplitude.

The damping ratio can be incorporated in the computer analysis by one of the following two methods.

Method 1—Perform force–response analysis of the system using the lowest damping ratio, i.e., the damping ratio of the machine's steel frame and steel support base frame. This damping ratio, 0.5–1%, will yield conservative results. Method 1 is recommended for structures supported with a steel base frame.

Method 2—Same as Method 1, except damping ratios of each system element are specified. Concrete damping is in the range of 3–5%, while soil damping can be as high as 50% (*17*). Method 2 is recommended unless restricted by computer capacity.

Conclusion

The quest for greater productivity has led to the development of faster, wider paper machines, and this trend is likely to continue. Efficient operation of today's high-speed paper machines requires a well-designed foundation that can sustain vibration within a tolerable range. Dynamic considerations tend to determine the sizes of major components of the foundation. Dynamic analysis, in turn, has become an essential part of the design process.

The analysis is elaborate and requires close interaction between the machine manufacturer and the consulting engineer responsible for designing the machine's foundation. This is especially true for machines whose speed and width stimulate roll frequencies that are in resonance with system frequency. A high-tuned system is preferable and provides the safest design, but a force–response analysis is imperative in low-tuned and resonant conditions.

Models of the machine and foundation are required to determine frequencies and vibration amplitudes. An adequate model with sufficient mass points is essential to obtain accurate results. The model can be reduced in size only if the engineer thoroughly understands the system response. Reducing a machine model to a single degree of freedom results in unreliable output.

The recommended practice is to isolate each machine section with a separate foundation and to isolate each of these foundations from the operating floor. Isolation prevents transmission of vibrations from one section to another and between the building and the paper machine. Isolation also makes the size and the cost of the dynamic analysis manageable.

The pulp and paper industry urgently needs standardized criteria to streamline the task of analyzing and designing paper machines and their foundations. A good starting point would be development of guidelines for frequency analysis, force–response analysis, forcing function, vibration amplitude, damping, and phase angle of response.

Studies involving field measurements on operating machines are needed to determine acceptable limits of machine operation. Such studies would best be carried out by committees consisting of papermakers, machine manufacturers, and consulting engineers. The TAPPI committee on paper machine dynamic foundation design would be a good candidate for undertaking this essential task. □

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