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Vibrational and acoustical experiments on logs of spruce

D. Ouis

Summary This paper presents the results of some vibro-acoustical experiments conducted on logs of spruce. The aim from this study was in the beginning to study the possibilities of investigating acoustically the presence of decay in the stems of standing trees, and for assessing the strength of wood in the shape of logs. First, a brief review is made for the different experimental techniques used in studying the response of mechanical systems in general with emphasis on an efficient technique used for evaluating the impulse response of vibrating systems. Then a literature survey on the effects of decay on the strength and on the damping properties of wood are presented with some practical general implications regarding decay inspection of wood composites. Lastly some experiments using vibrations and sound which were conducted on two specimens of wood logs, one sound and one decayed, are presented with some discussions regarding the implementations of these methods for the quality grading and defect detection in wood logs and standing trees.

Introduction

Wood has played a role in the history of mankind such that man has never thought of staying away from its use. For this reason, historians have never talked about a "wood age" in the life of humanity. For the cave man, wood was used for heating and for protection from wild animals and upon leaving his cave, the primitive man relied once more on wood to use it as his first building material.

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D. Ouis
Department of Engineering Acoustics,
Lund Institute of Technology,
S-221 00 Lund, Sweden

The author is Research Assistant at the Department of Engineering Acoustics, Lund Institute of Technology, P.O. Box 118, S-221 00, Lund, SWEDEN. This work has been accomplished at the Department of Applied Electronics, Signal Processing Group, LIT where the author was in a short stay. Sincere thanks are directed to Professor Göran Salomonsson for arranging the financial support to this project and to Professor Sven G. Lindblad for freeing access to the anechoic room and to all the necessary acoustical instrumentation for the well going of the experiments. The practical help of Mr Karl-Ola Lundberg is also gratefully acknowledged

This function is still fulfilled, although to a lower grade, by wood in men's modern society. Even more important, there is nowadays a trend towards encouraging people for habitation in wood made houses and this has in turn boosted research activities in several branches having connection to the building sector. This situation is supported by three main facts which are firstly the relative abundance of wood as a material, then its lesser degrading effect on health as compared to some other materials and lastly that improved techniques are constantly elaborated in both the prevention and the control of eventual fire hazards in wooden dwellings. The use of wood is not restricted to building elements only but extends to other industrial areas as well. Furniture manufacturing for instance is depending almost a 100% on forest products and this applies as well to the non-negligible industry of paper. The comparatively lower price and easier transportation mode of wooden utility poles have made of them an uncomparable substitute to their concrete or metallic counterparts in supporting electric power supply or telephonic cables. These are in short some of the domains where wood is a main provider either as a raw, a semi-finished or finished product, and one can already realize that this natural renewable material stands for an important share in the domestic investment of many countries worldwide.

Like any other material, wood may have some defects in it and these are in general classified under two categories. The first category includes all malformations in the tree trunk like knots, splits and cross grain resulting from the natural adaptive development of the tree, and the other category concerns those defects caused by external agents and which degrade principally the strength of the material. The results of the work presented here are concerned with this latter kind of defects because wood as the material of building bearing parts is sought primarily for its strength. In contrast, the other kind of defects is most often bare detectable and these make sometimes a set of various aesthetic features for furniture elements. Due to natural growth patterns, such variability which concerns almost all properties of wood, is not only restricted from one tree to the next one, but even within the same tree depending on the height and radial position in the stem.

Experimental determination of the response of vibrating systems

The various classical experimental methods generally used in producing or detecting stress waves and vibrations in elastic elements and structures may be found in Appendix C of (Graff 1975). The earliest measurement procedures in mechanics were using methods employing analogue equipments and one can find in Chapter I of (Cremer and Heckl 1988) a comprehensive presentation of the principles behind the various measurement methods and the instrumentation for the generation and sensing of structure-borne sound. Vibrational and acoustical measurement methods are broadly divided into two categories. First, the frequency domain measurements are mainly made to localize the resonance frequencies of vibrating objects. They are accomplished through driving the object under test with a continuous sinusoidal signal through a transducer, and by sweeping slowly the signal frequency. The response is sensed by a gauge and may sometimes be presented on paper by a plotter or any other printing device. This method, sometimes also called resonance method, is also used to determine the damping coefficients of vibrating structures through studying the width and height of the resonance curves. This is illustrated in Fig. 1 for the case of a simple Single-Degree-of-Freedom (SDOF) system. The other measurement method is used for time domain measurements and is best suited for determining the im-

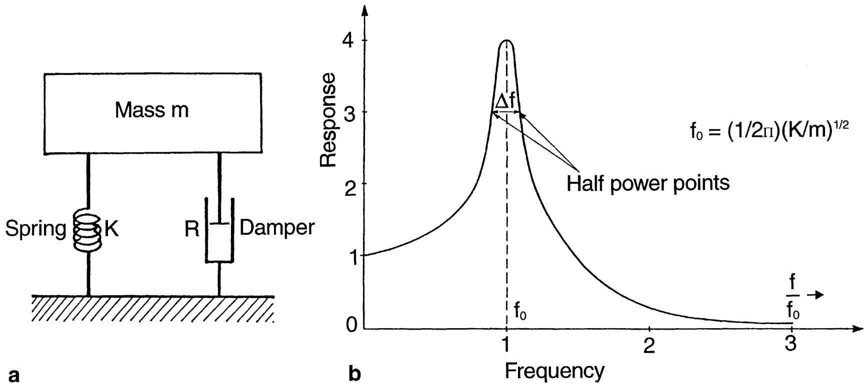


Fig. 1a,b. Example of a Single-degree-of-freedom (SDOF) system in vibration. **a** System, **b** Typical frequency response

pulse response of a system (the response of the system when the excitation is made of an extremely short signal), and consequently the strength and delay of the different reflections occurring in the system. The impulse response of a system is in a way its signature and most, if not all, of the important quantities characterizing the system may actually be processed from its impulse response. In this case, the response is often visualized on the screen of an oscilloscope. The input signal can be a short wavetrain concentrated around a specific frequency (e.g. a so-called Gauss-tone) for dispersion and mode-coupling investigations, or a single short broad-band pulse (rectangular or half-cosine) for reflections determination. Theoretically, this latter case would be related to the frequency domain method through a simple Fourier transform, and the shorter is the signal, the more distinctly the reflections are separated on the time scale. Ideally, a short pulse, and in the limit a Dirac delta pulse being vanishingly short, infinitely intense but containing a finite value of energy would be very broad-band in its frequency containt. Unfortunately, commercial pulse generators are impossible to fulfil such a performance, and continuous signal generators are rather limited in the frequency range of the signals they deliver.

These different measurement methods of limited flexibility necessitated rather long times to carry out average measurements, and extreme care and patience to avoid repeating them. In the various engineering branches dealing with wave propagation and vibrations, experimentators and theoriticians in quest of verifying their theoretical predictions were thus in need of more efficient measuring equipment. This had to wait until around the year of 1965 with the upcoming of the Fast Fourier Transform (FFT) algorithm which enables transforming time-data into the frequency domain or vice-versa in an efficient manner. This event followed by the rapid developments of the Very Large-Scale Integration (VLSI) in the semi-conductor technology and that of the digital techniques in filtering and averaging marked a decisive turn in the history of experimental research when these factors introduced in the market moderately-priced FFT-based measurement systems. Earlier spectra measurements based on cross-correlation calculations became very rapid and the old methods using random noise excitations became more elegant and diversified. For impulse response measurements, a powerful measuring scheme of widespread use nowadays in room- and structural acoustics is based on Maximum Length Sequences (MLS) a kind of pseudorandom test signal which has in a way similar properties to random signals though it is deterministic. This method which is briefly presented here is described in detail in (Kuttruff 1991).

Let a stationary signal s(t) be applied to a linear system for which we are seeking the impulse response g(t), Fig. 2.

The response-signal s′(t) collected by a sensing detector (a microphone for a room or an accelerometer for a vibrating structure) at the other end of the measurement chain may then be expressed as a convolution integral according to:

$$s'(t) = \int_{-\infty}^{+\infty} s(t')g(t - t')\,dt' \tag{1}$$

where t is time. In present days, analogue signals are digitized by Analogue to Digital Converters (ADC) and continuous integrals of the Fourier type are approximated by discrete summations (Discrete Fourier Transform, DFT), all along with many other processing stages in a single instrument. The cross-correlation function of the excitation signal s and the received signal s′ is then given by:

$$\varphi_{ss'}(\tau) = \lim_{T_0 \to \infty} \frac{1}{T_0} \int_{-T_0/2}^{+T_0/2} s(t)\,dt \int_{-\infty}^{+\infty} g(t')s(t + \tau - t')\,dt' \tag{2}$$

which gives after interchanging the orders of integrations:

$$\begin{aligned} \varphi_{ss'}(\tau) &= \int_{-\infty}^{+\infty} g(t') \left[\lim_{T_0 \to \infty} \frac{1}{T_0} \int_{-T_0/2}^{T_0/2} s(t)s(t + \tau - t')\,dt \right] dt' \\ &= \int_{-\infty}^{+\infty} g(t')\varphi_{ss}(\tau - t')\,dt' \end{aligned} \tag{3}$$

From this last result, one can see that the cross-correlation $\varphi_{ss'}(\tau)$ is equal to the sought impulse response $g(\tau)$ if the auto-correlation function $\varphi_{ss}(\tau)$ of the input signal s is a delta function or at least approximates it. A signal with an auto-correlation function concentrated at zero and which therefore can be used for this measurement is the random noise. Actually, to avoid the cumbersome and rela-

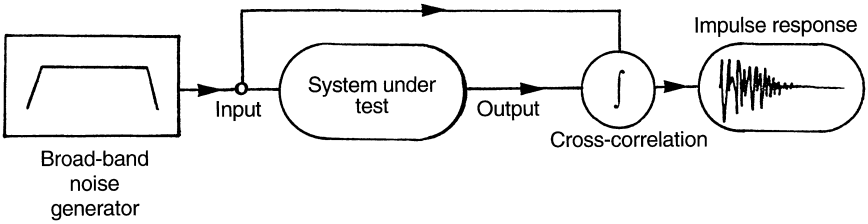


Fig. 2. Illustration of a measurement method based on the cross-correlation principle

tively time-taking operation of the integration, the cross-correlation function of two signals is evaluated by means of Fourier transforms. First, the signals undergo Fourier transforms (rapidly done through the FFT) which are then multiplied with each other to give a cross-spectral density function and lastly, this latter goes through an inverse Fourier transform (IFFT) giving thus the cross-correlation function (Newland 1986).

Physical systems encountered in daily life are obviously much more complex than the SDOF ones and their analysis is not so that simple as illustrated in Fig. 1. Strictly speaking, all mechanical systems are continuous. The degree of freedom of a SDOF system is defined as the elastic motion of the single mass in the system and this concept is easily generalized to Multiple-Degree-of-Freedom (MDOF) systems if one succeeds in describing the extended system as the build-up of many small simple ones connected to each other by some coupling mechanisms. This idea is not new in itself, and actually, one of the earliest reports on such discretization dates back to Lagrange in 1788.

For studying the motion of systems such as beams and plates, the degrees of freedom may for instance be taken as the displacement amplitudes of selected points in the structure. The motion analysis of the whole system can thus be satisfactorily described only by taking into account a large enough number of the constituant simple elements. When this is possible, the study of a vibrating system subdivided into n elements, which, incidently, is called a lumped-parameter system, leads to solving a system of an equally large number of equations each describing the behaviour of each element in connection to the rest of the elements by some coupling characteristics. The constitutive elements are usually the combinations of ideal dimensionless masses and massless springs, dashpots and resistances. Based on this discussion, it is thus not surprising to find in the literature the analogy between a complex system like a plate or a room to that of a combination of an infinite number of resonators each resonating at its own resonance frequency and decaying with its own damping characteristic.

In Fig. 3 one sees the frequency response of a relatively simple mechanical system having three vibrating masses, hence a three DOF system, and coupled to each other by some springs.

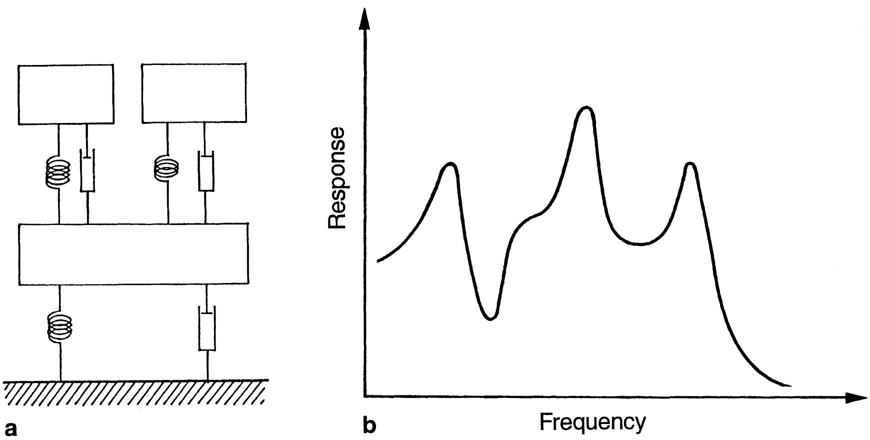


Fig. 3a,b. Example of a three-degree-of-freedom system. **a** Mechanical system, **b** Typical corresponding frequency response

On the other hand and for a bounded system, reflection of waves at the boundaries is of most practical importance inasmuch as this phenomenon is responsible for the existence of sets of frequencies and associated patterns of vibrations specific for a finite system. A bounded system can be driven by a periodic excitation (forced excitation) of any frequency but when left for itself, it would vibrate freely at specific frequencies called characteristic or natural-frequencies or eigenfrequencies. To each eigenfrequency corresponds a specific vibration pattern called characteristic or natural-mode, eigenmode or simply mode. The eigenfrequencies can be measured from the frequency response curve of the system where they exhibit quite marked peaks at the resonance frequencies and having widths inversely proportional to their respective quality factors: it is more difficult to separate visually the resonance frequencies of a well damped vibrating system than those of a less well damped one. The values of the resonance frequencies depend primarily on the size of the system and on the way it is hinged, i.e. the boundary conditions to be satisfied by the equation governing the motion of the system. The mathematical form of this latter is also decisive for the complexity of the expressions of the eigenmodes and their corresponding eigenfrequencies. When the size of the system extends, the frequency distance between two neighbouring resonances diminishes in proportionality to the size of the system and this feature is more pronounced for systems extending in more than one dimension in space. At the extreme case, a system extending to infinity, be it only in a single direction, would vibrate freely at any frequency. It is worth noting at this point that as a universal observation, most of the complex vibrating systems exhibit their strongest responses somewhere at their few lowest modes and experimental investigations have therefore their most interesting applications at relatively low frequencies.

In one dimension, the simplest example of a vibrating continuous system that one can think may be that of the flexible string. If such a string of length l and of mass per unit length ρ is stretched at its both extremities by a tension force F (assuming the direction of F along, let us say, an x-axis), then the eigenmodes for small transverse vibrations (linear treatment) would be proportional to $\sin(n\pi x/l)$, where n is an integer and x the distance from any of the two extremities of the string. The corresponding eigenfrequencies are given by $f_n = nc/2l$, where c is the phase velocity of the wave propagation in the string and is equal to $(F/\rho)^{1/2}$. These relatively simple forms for the eigenmodes and the eigenfrequencies of the vibrating elastic string are found from considering the displacement u of the string's particles which in this case satisfies the well known Helmholtz' wave equation:

$$\frac{\partial^2 u}{\partial x^2} - \frac{1}{c^2}\frac{\partial^2 u}{\partial t^2} = 0 \tag{4}$$

If one takes the case of a beam, things become somehow more complicated. For the first, the motion of the beam cannot be described only by its flexibility i.e. the tension is not the only restoring force in it as in the case for the string. A beam presents some stiffness in its motion: one needs to spend more effort to bend a bar than a flexible string. The stiffness property of a solid beam is put into evidence by the well known fact that when bending a piece of metal for instance, the curve resulting from the bending has two sides; a convex one where the material is stretched and a concave one where the material is squeezed. This property is the basic mechanism for maintaining vibrations in solid elastic structures and is characterized by a bending stiffness which will be defined shortly later. Secondly, for the case of the string one can consider in general only two motion possibilities for its extremities: either free or fixed. For a bar on the other hand, and due to its stiffness, there are more ways of attaching its extremities to the rest of a structure. The four most widely used ideal boundary conditions are schematically represented in Fig. 4:

These cases are summarized in what follows:

a. Simply supported: no transversal deflection but free for rotation
b. Clamped : no translation and no rotation
c. free: free for lateral deflection and free for rotation
d. guided: (in this case) free for transversal deflection but no rotation.

Experimentally, the case b is the most difficult to realize while the case c is the easiest one. There are many other ways for attaching a beam from its ends or for supporting it along the whole of its length (see for instance Graff 1973, p. 154) or (Timoshenko et al. 1974) but the four types described above are the simplest ones used for theoretical calculations. It follows from this preliminary discussion that the equation of motion for the particles of a beam must be more complicated than that of a string. The often quoted Euler-Bernoulli theory provides the simplest form of the equation for describing the bending motion of a beam (or a plate). For the free vibrations of the undamped beam, this equation has the following form:

$$EI\frac{\partial^4 u}{\partial x^4} + \rho\frac{\partial^2 u}{\partial t^2} = 0 \qquad (5)$$

where E is the modulus of elasticity or Young's modulus of the material of the beam and ρ its mass per unit length. I is the beam's moment of inertia and depends on the geometry of its cross section which for instance for a rectangular beam with sides b and h (bending direction) is given by:

$$I = \frac{bh^3}{12} \qquad (6)$$

One usually uses in vibration problems of beams the bending stiffness B = EI which was introduced earlier and which is a measure of the beam's resistance to

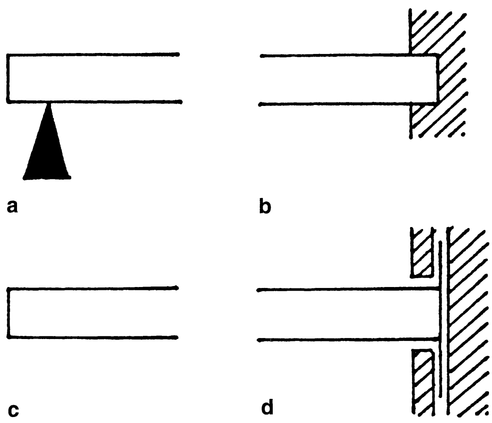


Fig. 4a–d. The four ideal boundary conditions of most use for a transversally vibrating beam

bending. At first sight, one notices the fourth order of the space partial derivation of the field variable as compared to the second order one in the usual wave equation (equation 4). This has important consequences on the behaviour of the solutions of the equation. By choosing a harmonic solution with a circular frequency ω to equation 6, one deduces the frequency equation from which the phase velocity of the wave is expressed as:

$$c = \sqrt[4]{\frac{B}{\rho}}\sqrt{\omega} \tag{7}$$

from which one recognizes the frequency dependence of the wave velocity; waves with different frequencies spread with different velocities downward the beam and an initial disturbance, a wave paquet, distords as it moves on the beam. This phenomenon is called dispersion in analogy with the dispersion of white light into its fundamental components as it spreads through the material of a glass prism. Next, by choosing a set of boundary conditions for the general solutions of the beam equation, one can calculate the eigenfrequencies for the beam and their corresponding eigenmodes. For a bar of length l which is simply supported at its ends, the eigenfrequencies are given by (Cremer and Heckl 1988):

$$f_n = \frac{\pi}{2}\sqrt{\frac{B}{m}}\frac{n^2}{l^2} \tag{8}$$

and we can see that the discrete eigenfrequencies are not in proportionality to the integer numbers as for the case of the string but to their squares: the overtones are not simple harmonics of the fundamental tone, hence the non-musical sound emitted by a struck bar as compared to a string.

Although the whole study of sound is a study of vibrations, it has somehow become an agreed upon matter within the acoustical community to well differentiate between sound and vibration. The word sound has become mainly devoted to encompass all the phenomena of perturbation propagation in materials, mainly liquid or gaseous, whereas vibration is in a way merely confined to the motion of material structures, mostly solid, which could under favourable circumstances generate sound waves. One thus often says that a plate which vibrates generates sound, and in this case, the separation area between sound and vibration becomes in an abstract form just the radiating surface of the plate.

Non-destructive detection of decay in wood by vibrations

Vibrational methods are commonly used in engineering for assessing the strength of materials and for checking the good serviceability of structures. Moreover, the driving of a structure at resonance may cause mechanical failure of the structure itself and reduce drastically its life time. As wood is a solid material and wood products enter with a good proportion into the wide assortment of building elements, it would then be natural to adapt the use of these vibrational methods for testing the mechanical behaviour of wood as a raw material or the performance of its composites under various physical conditions. Much research has been conducted with this objective, and the mechanical properties of timber have been reported back to at least the first half of the nineteenth century (Hearmon 1966), much long before the development of ultrasonics and the radiological scanning techniques. Many researchers have in fact appreciably contributed to understand the mechanical behaviour of wood and to quantify its physical characteristics. For instance, the knowledge of the damping properties of wood helps for a better reduction of vibrations and for more isolation of unwanted noise (or oppositely at some advantage for more sound radiation as in the case of musical instruments). However, wood is a much highly anisotropic material which makes its mechanical properties much dependent on the direction of their determination. Moreover, due to its remarkable inhomogeneity, even within the same tree stem, it is quite impossible to derive a general standard testing method for wood, and the diverse published data are often made on the basis of average figures. The most successful experimental results, i.e. those which support strongly the theoretical predictions, have been conducted on small clear specimens because the presence of the least defect, in some instances even a single knot, may result in a wide variation of the value of the sought quantity (Chui 1991; Skatter 1996). Despite that, much progress has been made during the last years in the characterization of wood by non-destructive vibration testing methods. In fact, it seems as if wood research is taking its way back towards the reintegration of vibro-acoustics into the case of testing tools (Bodig and Jayne 1882; Sobue 1986a,b).

For our part of concern, the progress of recent research on the detection of decay in wood by vibration testing will be shortly reviewed. But first, the effects of decay on the various physical properties of wood must be exposed.

Effect of decay on the strength properties of wood

In the living tree, the wood of the trunk has to support the compression loads from the weight of the crown above, and it has to resist the bending forces of the wind. The ability of wood to resist loads is manifested in its strength and this property depends on several factors which include the type of load (tension, compression, shear), its direction and the wood species. Ambient conditions of temperature and moisture content are also important as are past histories of load and temperature (Schniewind 1989, p. 245). The strength of a sample depends also on its size and specially on whether it is a clear piece free of defect or containing defects in the form of knots, splits and the like.

The strength of a material is usually characterized by the values of its Modulus of Elasticity (MOE) and its Modulus of Rupture, (MOR). For small amplitude vibrations, the MOE determines the velocity of propagation of longitudinal waves in the material and under the assumptions of elastic behaviour, the MOE is defined as the slope of the stress-strain curve below the linear proportionality limit. The MOR on the other hand is a measure of the ultimate stress before rupture in a sample of material and is thus a parameter of mechanical failure. Toughness, or the ability of a solid material to withstand shock loading is another quantity for assessing the strength of wood. The values of the MOE for wood are subject to the influence of various factors. The direction in which the MOE is measured is of first importance and the lowest of its values is in the T-direction which is tangential to the growth rings. About a doubling of this value is acceptable in the R-, radial direction but in the L-, longitudinal direction along the axis parallel to the tree stem, the value is about an order of magnitude larger. The moisture content (MC) of wood is also another important factor. MC is defined as the rate in weight of water content to dry wood. This parameter has also a wide range of values depending on wood species and age of timber. For a freshly sawn timber, MC values have been measured in the range 35 to about 300%. A value to be retained in this context is the Fiber Saturation Point (FSP) which is designated as the MC value at which the wood cell cavity has lost all of the free water inside it but the cell walls are still satured with water (Schniewind 1989, p. 143). The value of FSP is around 27% for the majority of the northern timbers and this is a critical value at which changes occur in most of the important physical properties of wood. Above the FSP, MC has almost no effect on the elastic properties of wood, but below this value the MOE increases as MC decreases (Dunlop 1981; Schniewind 1989, p. 77). The velocity of sound decreases for increasing MC (Nanami et al. 1992a) and is influenced by MC even above the FSP (Burmester 1965). The determination of the MOE is also affected by many factors like the geometry of the specimen and the thickness of the stressed area (Kunesh 1968).

Decay results from the attack of tracheid cell walls and the binding lignin by fungi. Laboratory testing of the chemical decomposition of conifers by fungi has revealed that brown rot fungi are more destructive than white rot fungi (Kirk and Highley 1973). Blue stain for instance has been found to have a negligible effect on either MOE or MOR of spruce (Glos 1989; Pratt 1979). Loss in strength which is expressed as a percentage value of the comparable value for sound or undecayed wood has been proposed for use in laboratory evaluation of decay severity (Toole 1971). Loss in strength of decayed wood often occurs before significant loss in weight and a significant reduction in material density has in general not been noticed during the decay process (Pratt 1979). Brown rot is by far more degrading to the mechanical properties of spruce and white pine than white rot even in its early stages when decay is still invisible. In conifers, the decomposition course is also strongly influenced by the latewood portion and the density of annual rings (Bariska et al. 1983). The MOE is more affected by decay than the MOR. The general relationship between strength and the effect of decay shows an initial rapid loss of strength in the early stages followed by a gradual decrease in the rate at higher weight losses (Kim et al. 1994). Toughness is more sensitive to the early stages of decay than the MOE (Kim et al. 1994; Wilcox 1978). The consequences of the early stages of decay on strength loss of wood have been reported as an average loss in toughness of around 50% for only 1% weight loss (Wilcox 1978). Measurements of strength in radial compression (force perpendicular to axis of tree) on samples of Douglas-fir have showed that incipiently decayed wood sustains an average 3% loss in MOR for 11% in MOE (Pratt 1979) and similar measurements confirmed the vulnerability of sapwood to fungal activity and its faster degradation as compared to that of heartwood (Smith and Graham 1983). These results support also earlier similar observations on the correlation between strength loss in compression and the extent of decay (Breeze and Nilberg 1971).

Effect of decay on the damping properties of wood

The damping capacity of a material under vibration is determined by its ability to decrease the amplitude of vibrations when left for itself. Wood has also some damping properties and these are in turn affected by many factors. Decay, because it changes the physical structure of wood has also noticeable effects on its damping properties. Generally speaking, decay increases damping (Dunlop 1983), and brown rot has a more noticeable effect than white rot on particularly coniferous woods (Bariska et al. 1983). Damping depends also on the direction along which it is determined. In transverse direction, radially to the tree stem, damping may be as high as three times than damping in the longitudinal direction, i.e. along the grain. This has sometimes made measurements in the longitudinal direction on wooden poles and standing trees more useful and attractive than in the transverse direction (Fukada 1950; Dunlop 1981). By knowing the resonance properties of wooden elements, it is often possible to detect their possible hosting to decay through exploring the widening of the frequency response peaks at the resonances. These general observations ought also to be interpreted in connection with the fact that damping in wood is a property depending on frequency, temperature and MC, varying from species to species (Fukada 1950) and affected by the lateral position and height in the tree trunk (Ono 1983). Unfortunately, up to this date there is not a significant research on a more quantitative correlation between the different stages of decay and the relative amount of increase in damping, especially if one knows that damping and the MOE go together in measurements of the sound radiation by impacted wood (Tanaka et al. 1986).

Characterization of wood and its composites by vibrations: Review of selected literature

For a variety of reasons and purposes non-destructive testing of wood is more desirable than the use of destructive methods. As for the reasons, these can be for instance the non-possibility of removing a sample from the suspected element for laboratory testing because of its bearing function or its valuability. The purposes of non-destructive evaluation of wood, or indeed of any material, are also numerous. These can be shortly ranked in order of importance in: the rapidity of carrying a test (this may be of particular concern to the early phases of the manufacture of wood products), the repeatability and reliability of its results, the handiness of its equipment and its cost. Earlier practices of testing of wood by vibrational methods have focused mainly on the determination of its acoustical properties for the making of musical instruments. As the building industry cannot stay out of the need of timber, a classification of its elastic and strength properties became a necessity and the physical properties of wood related to its strength are nowadays tabulated alongside with those of concrete and steel. More recently, as the housing market is turning towards the more erection of wooden light-weight dwellings, high demands are required for the realization of a less noisy indoors environment. This in turn requires the knowledge of the damping properties of wood and an improvement of this property with a proper design of its products.

Most of the vibrational measurements on wood are made on relatively small samples. This restriction on the size of the test specimen is also a requirement from many standards. The test specimen has to be as free as possible from growth defects like knots and cross-grain, unless the measurement is for the investigation of these very defects (Chui 1991; Divós and Sugiyama 1993; Gerhards 1982; Glos 1989; Schad et al. 1996). The anisotropy of wood should always be borne in mind because all the elastic parameters characterizing it depend on the measurement direction of their determination. If L, R, T design respectively the longitudinal, radial and tangential directions in the tree stem, it is always much easier to obtain samples in the LR or the LT planes than in the RT plane. Some authors have quite recently discussed the pros-and-cons of the inspection and characterization of wood by the diverse vibrational methods (Chui 1989; Chui and Smith 1990; Glos 1989; Haines et al. 1996).

The testing methods have also been given a share of cause in explaining the discrepancies observed sometimes in the various reported data on a same quantity. Recently, many questions has arisen again around the effect of the size of the wooden test specimen on the determination of the elastic parameters. As an example, the determination of the value of the modulus of elasticity, MOE, is made from measurements on beams and depends on whether the experiment is of a static or of a dynamic type. This has to be taken into consideration because usually static values of MOE are sensibly smaller than the dynamic ones (Burmester 1965). This change in elastic properties with frequency is a fact that is more or less true for almost all materials and wood is not an exception. The dynamical experimental procedure is often opted for because of its rapidity and again, the evaluation of the MOE in this case depends on whether the beam is set into flexural or longitudinal vibrations. Experimental results show lower values of MOE for larger specimens and higher values from longitudinal measurements as compared to those from bending results. These variations in the measured data have also been attributed partly to the rather viscoelasticity (time dependent deformation) of wood (Bodig and Jayne 1982, p. 185; Dunlop 1978; Haines et al. 1996) which supports observations using verly low frequencies (Becker 1980).

Although wood is not a perfectly elastic material (Bodig and Jayne 1982), much of the published material on the characterization of this material supports to a relatively high degree of confidence the incorporation of wood into the category of elastic materials. Already from the beginning, researchers in wood technology were aware of using the most efficient theoretical tools in predicting the wood properties they were in quest of. One of the pioneers in this field is Hearmon who through his long carrier has provided the scientific community with valuable information regarding the use of the bending wave theory with its various improvements to experiments involving wooden beams (Hearmon 1958; 1966). In his earlier publications he could prove that the simple beam theory of Euler should include the effects of rotatory inertia (Rayleigh's correction) and shear (Timoshenko correction) in order to match the experimental results to the theoretical calculations. Later contributions by other workers were mainly devoted to the determination of the MOE of wood through measurements on beams with different sizes. Calculations using FEM have given more evidence to the fact that improved theories give more accurate estimates of the eigenfrequencies and the eigenmodes for the different vibration modes of beams (Ohlson and Perstorper 1992).

Measurements were not restricted to only the MOE, although the knowledge of its value could help estimating other quantities of interest. Burmetser (1965) using short pulses at a single frequency measured the propagation velocity in different wood species and the influence of the morphological factors of wood on its value. The velocity of propagation can also be measured from reading on the screen of an oscilloscope the time separation between the traces of a stress wave at positions with known space interval (Marra et al. 1966). Knowing then the density of the material, the value of the MOE is a matter of a very simple calculation. The use of stress waves has also been used in finding the correlation between MC and MOE and by way to elaborate a tool for a possible sorting of green wood (Ross and Pellerin 1988).

Damping is another important property in the study of viscoelastic materials. Unfortunately, for wood, research is in need of more completeness in this field for a better understanding of the internal damping mechanisms and a more efficient use of the material in construction. Fukada after extensive studies on 30 different kinds of woods by vibrations in the middle audio frequency range found that the MOE shows hardly any variation while damping exhibited different behaviour for wood from broad-leaf trees than that from needle-leaf trees (Fukada 1950). Becker has limited his studies on MC and temperature effects to a single variety of hardwood at very low frequencies (Becker 1980).

During the last years, research is gaining a little more ground in determining the elastic properties of wood from beams with inhomogeneous shapes, different from those of the conventional rectangular ones. Beginning from studies on logs of wood, these studies are aiming to implement vibro-acoustic methods for testing of wood in its full scale, i.e. to logs and standing trees. An important finding is that for some wood species the wave velocity measured on the surface of a tapered log (conical shape with a low flaring) is almost the same as that in the sapwood of the log (Kodama 1990). Another investigation has led to the correction of the resonance frequencies of logs in flexural vibrations due to the effect of tapering (Sobue 1990).

Considering vibration measurements using impact excitation, its earliest implementations were merely used for wave velocity determinations but quite interesting applications which need a follow-up have also been reported in investigating holes in small wooden pieces (Yanagiwara et al. 1986). The impact technique has seen a noticeable emergence during the eighties and the reasons behind this regeneration is due to its practicability in field tests. Moreover, the impacting instrument is a simple hammer which is both inexpensive and hand portable. As an example, wood poles for electrical power supply and telecommunication cables are exposed to weather severities, attacks by insects and degradation by rot fungi. These wooden units are thus in need of continuous inspection of their working status and the investigation procedure may be done but in-situ.

Numerical methods are for the time being the only relatively successful theoretical tool for the examination of the strength of wood poles (Nilson and Pellicane 1993; Peabody and Wekesser 1994) and for studying the propagation of stress waves in them (Bulleit and Falk 1985; Falk 1983) as well as in logs (Burrows and Fridley 1988). A few patents exist already for the field investigation of wood poles, the testing method of some of which is built on the study of the pulse response of the pole to a hammer stroke (Dunlop 1981). The transient method has again been used with success in the determination of the MOE of wood from blowing wood beams with a hammer. The excitation may be either transversal (Haines et al. 1996; Kodama 1990; Ohlson and Perstorper 1992; Perstorper 1993; Sobue 1986a; Tanaka et al. 1986) or longitudinal (Gerhards 1982; Haines et al. 1996; Ohlson and Perstorper 1992; Ross and Pellerin 1988; Sobue 1986b). In the foregoing case, the studied quantity is often the velocity response but measurements from sound radiation are also possible (Sobue 1986a,b). Some research has also been done in progressing the understanding of stress wave propagation in tree trunks. Investigations on relatively thick logs are promising (Kodama 1990; Schad et al. 1996) and more work is to be done for the implementation of more powerful methods in investigating defects in trees (Nanami et al. 1992a–c).

As decay changes noticeably the strength of wood, vibration measurements have then been used to assess the degree of damage caused by fungi to wood through measuring its MOE. However, most of the research in this field is of a rather limited extent and has mainly been confined to measurements on relatively small samples of wood. Measurements on specimens of structural size and on standing trees have been successful in determining decay only when this latter is at a so advanced stage that the rot fungi have removed a big part of the inner material.

According to equation 8, the resonance frequencies of beams with simple boundary conditions are in direct proportionality to the value of this modulus (or more precisely to its square root) and inversely proportional to the material density. At its early stages, decay reduces appreciably the strength of wood characterized by its MOE without too much affecting its mass. Furthermore, decay has not in general been noticed to affect the wood's density. These three facts together can be then used to assess decay in wooden members. An idea is then to study the vibrations of beams, and the presence of decay in their material ought to be simply translated by a lowering of their resonance frequencies. A contribution supporting this principle is due to Wang et al. (1980) who injected a decay fungus in small thin wood specimens and studied its effect under an increasing number of incubation days. The resonance frequencies of the vibrating samples were found to decrease more rapidly for more incubation days.

Some acoustical measurements on wood are also based on the equivalent principle that vibrations propagate more slowly in decayed wood. Formulated in other words, and according to the formula relating the longitudinal wave velocity to the material properties in general:

$$\nu = \sqrt{\frac{E}{\rho}} \qquad (9)$$

a wave takes a longer time to travel through a decayed sample of wood than through a sound one. A technique for this purpose consisting of measuring the time made by a wave to travel through a material is widely known as Through Transmission Time Technique. Although early stages of decay reduce appreciably the strength properties of wood (Wilcox 1968), it would however be difficult to detect it in standing trees by through transmission methods. The reason is that the incipiently rotten part of wood may extend only over a relatively small cross-section area of the trunk and that usually tree stems have a diameter size such that small time delays due to decay are very difficult to discern. This is the main reason limiting the successfulness of this method for decay investigation in trees.

The idea of finding a correlation between decay severity and the travel time of pulses in diametral direction in round poles and logs is not new in itself. Breeze and Nilberg have tried to approach this problem theoretically by assuming that a pulse circumvent a decayed region in its originally straight way along the diameter of the circular cross section. To test the efficacy of their hypothesis, Breeze and Nilberg drilled holes with different diameters along the axes of equal length sections from the same log and they found indeed satisfactory agreement between measurements and their predictions (Breeze and Nilberg 1971). Matteheck's research group in Karlsruhe, Germany, working on the investigation of defects in mostly valuable trees have recently introduced an instrument also based on the this principle, the Metriguard Stress Wave Timer (Bechtel 1986; Bethge et al. 1996; Mattheck and Bethge 1992; 1993). Two screws are fastened on diametrically opposite points of the trunk, one of which receives a hammer stroke and the other one hosts a sensor, Fig. 5. From the knowledge of characteristic speed values measured on healthy trees, the travel time of the stress pulse may give information on the degree of decay deterioration within the trunk (the counter in the apparatus measures time intervals in the order of the μs; for typical softwood trees with a radial velocity of 1000 m/s, the relative measured velocity change would be in the 1% range for 50 cm wide trunks). The same technique implemented to wood logs showed that transmission times are also somewhat sensitive to very knotty areas of wood (the transmission times were highly varying) but for the method there is no clear cut answer whether the defect elongating the pulse transmission time is due to decay or to knots. Note that here

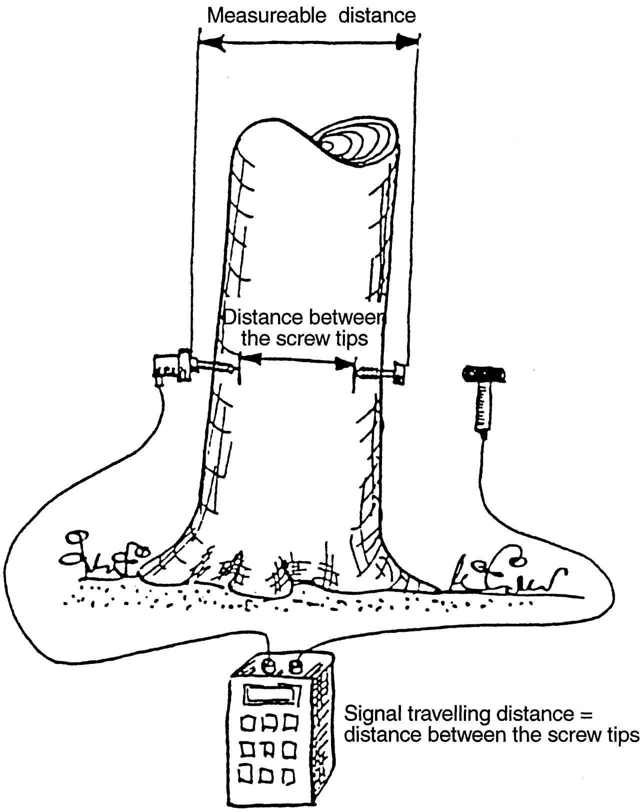


Fig. 5. Determination of signal travelling distance by means of the METRIGUARD Stress Wave Timer (From Bethge et al. 1996)

again that the test specimens were relatively wide logs, diameter larger than 76 cm, and that the decay degradation was visible as large voids within the logs (Schad et al. 1996) (Fig. 6).

The use of acoustic pulses was also used on utility poles as a potential method to assess the degree of decay advance in them. The pulse may be a stress wave generated by the impact of a hard ball at the butt of the pole. The stress wave is then sensed at a position remote enough from the impact position on the pole to allow for a good visual separation of the head from the tail of the transient signal. In this way, the velocity of the wave is deduced from the ratio of the travel distance to the time made to accomplish it (Dunlop 1981; 1983). A later study has also confirmed the possibility of implementing this method to the field monitoring of decay in utility poles (Ross et al. 1993).

Considered from another perspective, measurements with continuous periodical vibrations on hardwood trees have also led to interesting findings. The tested trees had an average diameter at breast height (dbh) between 21 and 57 cm. A sinusoidal vibration was induced in the tree stem and the response signal was picked at different positions on it, all through screws after removing the bark. It was found that the response of decayed stems was noticeably larger than that of non-decayed ones whereas no deformation of the signal was noticed. Too low frequencies (10 Hz) and too high frequencies (100 KHz) were much less effective

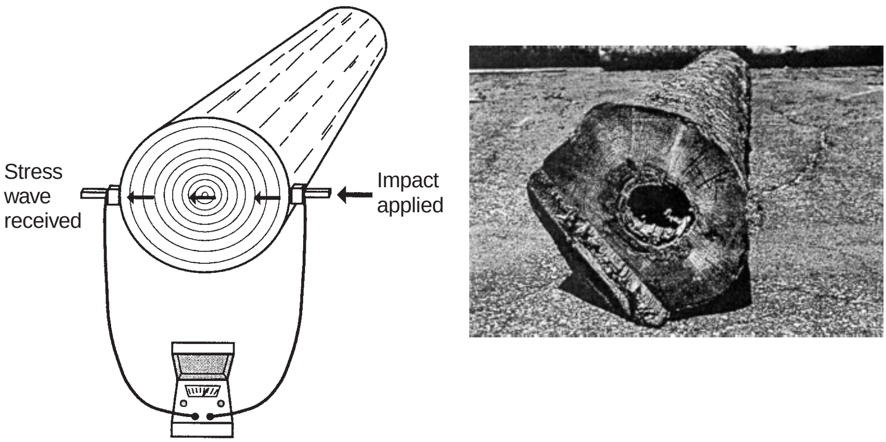


Fig. 6. Depiction of another sound wave transmission measurement system where reliable results are on severely decayed wood stems, (From Schad et al. 1996)

in determining decay than moderate frequencies (100 Hz, 1 KHz) (McCracken and Vann 1983).

Experiments on logs of wood; Preliminary results and discussions

In this part the results of some vibrational measurements conducted on wood logs of spruce are presented. The experiments were made using excitations of both continuous and impact types and the quantity of interest is either vibration or sound radiation sensed respectively via an accelerometer or a microphone. Two logs were chosen for comparative studies purposes; one sound, or at least showing no visible sign of fungal attack, and one with a marked stage of decay advance. For this latter, the attacked part of the area on the cross-section of the log was relatively small and concentrated at its center. The material though not completely deteriorated showed no resistance to penetration by a sharp-edged tool.

For all the measurements, information was extracted from the impulse response measured on the logs either hanging vertically with springs or lying horizontally on knife-edge supports. These supporting ways correspond to logs with free-free boundary conditions. Regarding the continuous excitation mode, the impulse response was processed according to the method illustrated in Fig. 2 and to this end a testing package employing Maximum Length Sequence signals called MLSSA was used which comprises a circuit board and its accompanying software. The excitation transducer was a Brüel & Kjaer minishaker, type 4810 delivering an almost constant force spectrum up to around 18 KHz. The vibration sensor was a Brüel & Kjaer accelerometer, type 2374 with a resonance frequency of 15 KHz, and the microphone a Brüel & Kjaer condenser type 4165 having a flat response up to around 10 KHz. The mountings for the different experimental setups are illustrated in Fig. 7.

In the scope mode, the test measurement system was used as a usual oscilloscope. The measurements were confined to a relatively low frequency range because the important modes were localized in the frequency band 0–1500 Hz. First, the sound log was investigated on the free-free supports for having an overview of

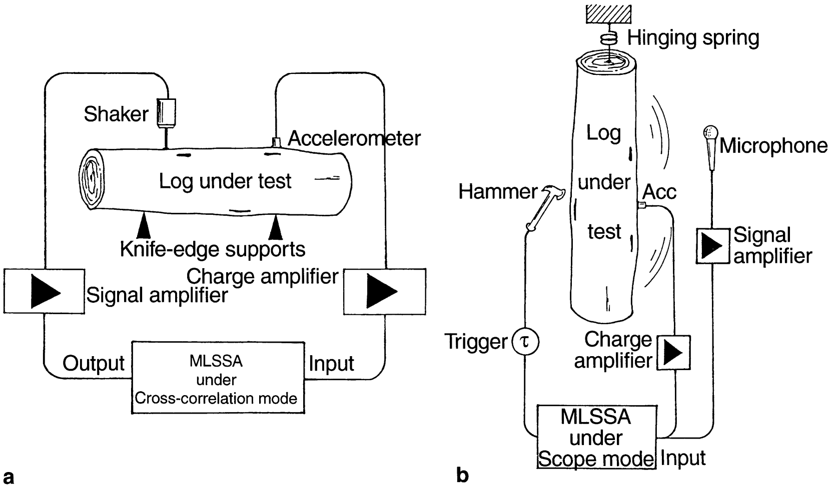


Fig. 7. Mountings of different setups for measuring the vibration impulse response or sound radiation response of wood logs. **a** Using a continuous excitation signal, **b** Using an impact excitation signal

its frequency response and to check the degree of compatibility between theoretical predictions and measurements. The response both in the time and the frequency domains is shown in Fig. 8.

Theoretically, a homogeneous rod with length l vibrating transversally with both its ends free would have the nodes of its fundamental bending mode situated at a distance 0.224 l from the edges. Although the experimental arrangement is made for these ideal boundary conditions, it is unavoidable to excite other vibration modes, not only of the bending type but of other types as well. Repeating a same measurement using the excitation at a symmetric area but containing naturally grown defects produced sometimes strong modal couplings (the sound log had a diameter of about 10 cm and for this range, knots have a non-negligible size). Hence, the material inhomogeneity leads to a more complicated interpretation of the results but in general one should expect the beam to have its strongest response at the first free-free bending mode. This response is exhibited in the peak denoted b1 in Fig. 8-a and its frequency is at about 270 Hz. The approximate expression for the eigenfrequencies of the free-free, f-f, vibrating beams is given by (Cremer and Heckl 1988):

$$f_{f\text{-}f} = \frac{\pi}{8}\sqrt{\frac{B}{m}}\frac{(2n-1)^2}{l^2}; \qquad \text{n: number of nodes} \tag{10}$$

with B = EI; E being the modulus of elasitcity, MOE, and I the sectional moment of inertia which for a circular cylinder with radius a is expressed as:

$$I = \frac{\pi a^4}{2} \tag{11}$$

In equation 10, m is the mass per unit length of the rod; it is expressed as $m = \rho\pi a^2$, ρ being the mass density of the material. So, for a homogeneous filled circular cylinder, equation 10 may be reformulated for the f-f1 mode as:

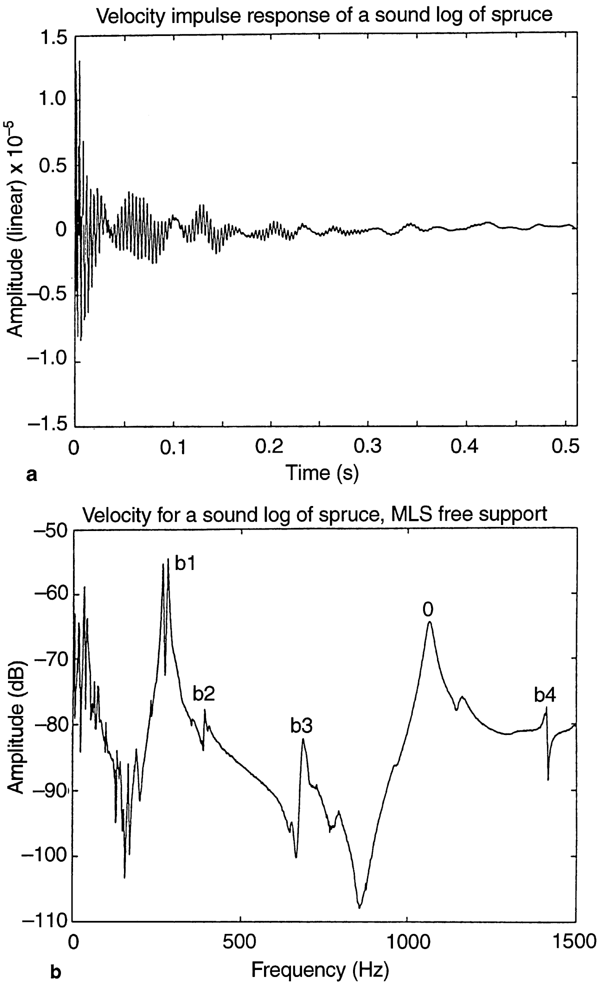


Fig. 8. Velocity response of a sound log of spruce on free-free supports. **a** Impulse response, **b** Frequency spectrum

$$f_{f\text{-}f1} = \frac{9\pi a}{8l^2}\sqrt{\frac{E}{2\rho}} \tag{12}$$

The wood of the test log was not completely dry and its density was evaluated simply by measuring the volume of the displaced water when a sample of known weight was sunk in a container. One finds an expression of the MOE from the frequency of the first bending mode looking like:

$$E = 2\rho\left(\frac{8l^2 f_{f\text{-}f1}}{9\pi a}\right)^2 \tag{13}$$

and after setting all the data values one calculates a value of E=8.7 GPa. This value is somehow lower than the average of the published data ranging between 8.3 and 13 GPa (see for instance Träfakta 1979). This is mainly due to the fact that these data are often presented from measurements on clear dry specimens (maximum of MC at 12%), while our specimen under test was a little knotty and had cracks with various depths on its surface in addition to that it was still a little green (the density of the material was about 754 kg/m^3 which compared to the dry material density of about 400 kg/m^3 would give a moisture content of 88%). The aforementioned defects affect negatively the strength of wood and yield always a lower estimate of the MOE's value (see for instance (Bulleit and Falk 1985) or (Chui 1991)). Another observation made during the experiments is that drying affected very much the measurements. In fact, the MOE is much dependent on the material density ρ. The inverse proportionality of the eigenfrequencies $f_{f\text{-}f}$ on ρ implicates that they increase for dryer material and this was actually observed by the higher pitch of the tone when the logs were struck while drying. In the range from 8 to 27%, the MOE value increases in a exponential manner as the MC decreases (Schniewind 1989, p. 249), but above this range MC plays no role on the MOE and the major effect on the resonance frequencies is attributed to material density. Below the saturation point and as MC continues to diminish its value, E increases and ρ decreases and this double processus according to equation 12 increases even more the value of the bending eigenfrequencies.

Returning to the frequency spectra, two other modes, bending also with a high probability, namely b2 and b3, were also excited. These are expected to be respectively the simply supported (s-s) mode corresponding to the vibrating part of the log between the supports and the next f-f mode with a resonance frequency of approximately $f_{f\text{-}f2}=2.7 \cdot f_{f\text{-}f1}$. The resonance frequency of the former mode could be evaluated from equation 10 but here one must take into account corrections due to the non-negligible masses (one fifth that of the log) by each of the supports. For the part of the log on the supports which represents 0.552 of its length, the eigenfrequency of the s-s1 mode, $f_{s\text{-}s1}$, would be $1.44 f_{f\text{-}f1}$. This mode would be situated at the notch b2 after the top b1, but the theory predicts that the extra masses at the edges of the s-s rod would decrease the eigenfrequencies (Rayleigh 1945); this mode should then beneficiate of a further investigation. The f-f2 mode cannot have exactly its resonance frequency because the supports were not at precisely its nodes, which normally are at 0.13 l from the edges of the rod and similarly to the s-s1 mode, its frequency is reduced by the extra masses at the ends. Moreover, the f-f2 mode was not responding with its full strength because the force of the shaker was applied near one of its nodes (middle of the rod). It will be seen later that despite that the excitation was as radial as possible and aimed to set the rod into only transversal vibrations, other types of modes, the tortional ones for instance, can also be set into motion and these latter blur somehow the interpretations of the frequency spectra. However, the vibration level of these modes is usually some tens of dBs lower than that of the important b1 mode, hence their lesser importance. On the other hand, one notices that the frequency response in Fig. 8-b raises afterwards strongly to reach a top at about $f_o=1060$ Hz. This frequency is too low to be that of the third bending f-f mode, f_o4, ($f_{f\text{-}f\text{-}3}=5.4 \cdot f_{ff1}=1460$ Hz in our case) and it is expected to be that of another type. It is suggested that this mode is of an extentional type, i.e. that the log while vibrating in this mode it does not hold a constant shape along its axis, and particularly that this is the so-called ovaling mode. This mode which is also sometimes called the n=2 mode, because it has 2 circumferential wavelengths or similarly that it has 4 nodal generatrices on the cylinder, is very important in the study of cylinders in vibration. The bending modes we have seen earlier correspond to n=1 and two points diametrically positioned on the cylinder vibrate

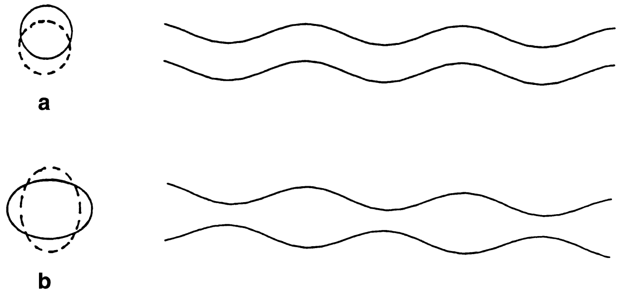


Fig. 9. Modes of vibration of a circular cylinder. **a** Bending mode, **b** Ovaling mode

oppositely in phase whereas for the ovaling mode they move in phase. Fig. 9 shows the vibration behaviour of these two kinds of modes for a circular cylinder.

The ovaling mode is very sensitive to defects and this is very clearly shown by the complete disappearence of its peak from the frequency response of the log when both shaker and accelerometer are positioned in the middle of a knotty area. In Fig. 10, one notices thus that the o-mode does not respond any more when the excitation and the response are made at a defectuous site on the log. Further, the exciting force and the sensing accelerometer were positioned near the antinode of the f-f2 mode and as a result one notices its quite marked peak.

Another series of experiments was also made on the same log mainly for studying its sound radiation, but this time the log being hanging vertically by short and relatively stiff springs. In one experiment, an attempt was made to excite other modes than the important f-f1. For this, the log was hit with a hammer at the node of this latter mode and both the velocity and the radiated sound were investigated. These are shown in Fig. 11.

From the spectrum in Fig. 11-a, although the amplitude of f-f1 mode is reduced appreciably, another mode emerges strongly and this can be only the second hinged-free mode h-f_2 which has a frequency $f_{h\text{-}f_2}=2.2 \cdot f_{f\text{-}f_1}$. The h-f1 mode has a lower eigenfrequency than that of f-f1 ($f_{h\text{-}f_1}=0.68\ f_{f\text{-}f1}$) but its node was at proximity of that of the f-f1 (the distance between these nodes is 0.04 l and for our 1.3 m long log this corresponds to 5.2 cm). One notices also from the same figure that the velocity increases rapidly towards lower frequencies. This behaviour is due to the whole-body motion of the log. In fact as the frequency tends to zero, the impedance of a rod of mass M tends to be that of a mass, i.e. $Z \rightarrow j\omega M$. With this in mind, the velocity is given by $v = F/Z = F/j\omega M$ which for a constant force increases with decreasing low frequencies. Regarding the spectrum of the radiated sound, one sees that in this case the mode contributing most to the sound field is the presumed ovaling mode.

Similar measurements were carried on a thicker decayed log of spruce the diameter of which was 1.44 that of the sound one. The frequency responses of the velocity for respectively the free-free boundary conditions with continuous excitation and for the vertical hanging setup with a hammer stroke at the node are shown in Fig. 12. The general pattern of the frequency spectra for the sound log is seen to be repeated in this case also. The resonance frequencies of the f-f1 mode in this case is $f'_{f\text{-}f1}=315$ Hz and that of the o-mode is at about $f_{o,decay} =$ 1200 Hz. Using the value of the MOE E=8.7 GPa found for the sound log in formula (12) would give normally $f'_{f\text{-}f1}=433$ Hz to be compared to the measured value of 315 Hz which corresponds to a relative decrease of the MOE

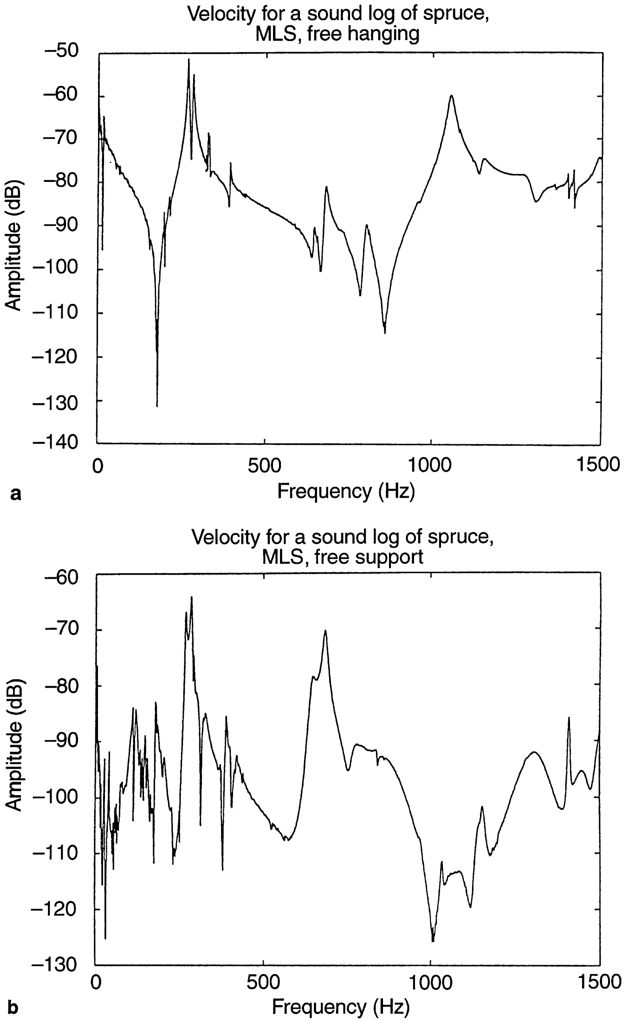


Fig. 10. Frequency spectrum of the velocity of a sound log. **a** Shaker and accelerometer away from knots, **b** Shaker and accelerometer at knots

$\Delta(\text{MOE})/\text{MOE} = 47\%$. The lowering of the resonance frequency may be attributed to the hollowness of the cylinder which diminshes its cross-sectional inertia I. But in this case, removing matter from the center of the cylinder diminishes also its mass which by way increases the resonance frequency. Indeed, the net result of these two effects leads in fact to a higher eigenfrequency. If one considers then a hollow cylinder with an outer radius a and an inner radius b, then I would instead of equation 11 be given by:

$$I = \frac{\pi}{2}(a^4 - b^4) \tag{14}$$

Taking this expression into equation 10 and considering a mass per unit length m equal to $\rho\pi(a^2 - b^2)$ yields for the new resonance frequency f_{hollow} of the hollow cylinder in terms of f_{rod} for the rod:

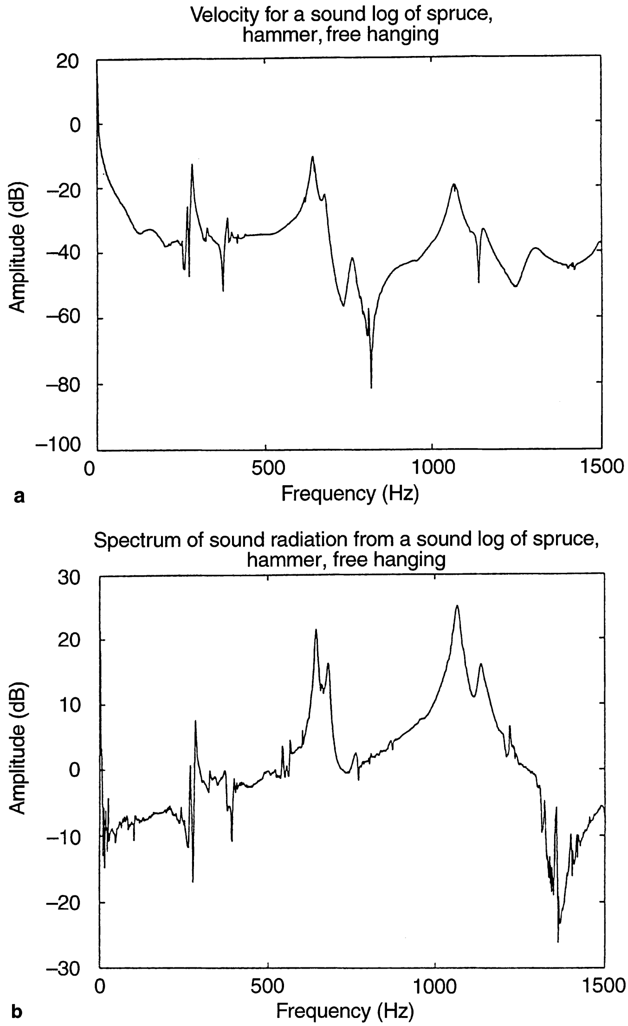


Fig. 11. Frequency response of a sound log hanging vertically and hit by a hammer at the node of the f-f1 mode. **a** Velocity, **b** Radiated sound

$$f_{hollow} = f_{rod}\sqrt{1 + (b/a)^2} \tag{15}$$

and which for small hollowness can be expressed by the approximation $f_{hollow} \approx f_{rod}(1 + b^2/2a^2)$.

On the other hand one knows that usually decay weakens wood without too much removing of its material. We have also seen earlier that the decay process is not in general associated with a noticeable change of wood density. From these two observations one can then conclude that the decayed matter in the log may be modelled to an acceptable approximation to a substance with a lower density ρ' (less mass occupying the same volume) than that of the wood hosting it, but contributing in nothing to its strength. This leads then to a new substitute of equation 15, namely:

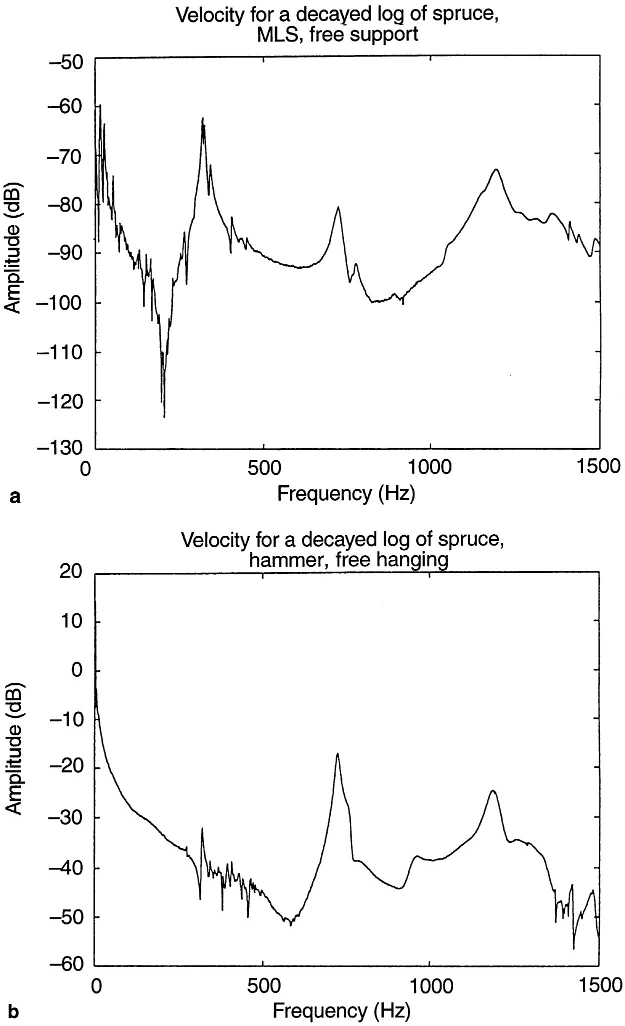


Fig. 12. Frequency spectra of the velocity for a decayed log of spruce. **a** Free-free supporting conditions, **b** Vertically hanging with a hammer blow at the node of the f-f1 mode

$$f_{\text{decayed}} = f_{\text{sound}} \sqrt{\frac{1 - \alpha^4}{1 - (1 - \rho'/\rho)\alpha^2}} \tag{16}$$

where $\alpha = b/a$. The curve b/a vs ρ'/ρ is plotted in Fig. 13. One notices that the curve is almost linear and that for a very light inner material, the cylinder tends to be very thin. At an equal density matter, the thickness of the stiffer outer part of the cylinder has to be 1/5 of its mean radius. This seems at first judgement to be a too high value but in reality not impossible to be observed in practice and which could be proved only by more precautionary studies.

Perhaps a more realistic model is to assume that the decayed part of the wood retains its natural density whilst its strength reduces considerably. We have seen earlier that a 50% loss of strength could be associated with only 1% loss in weight.

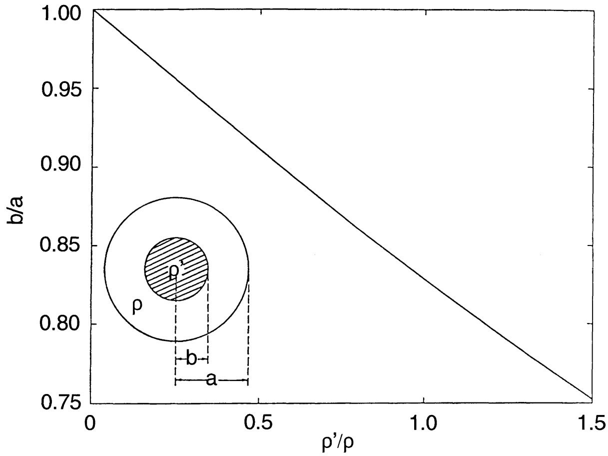


Fig. 13. Ratio of the diameters of the inner lighter to the outer heavier equally strong parts of a composite finite length cylinder as a function of the ratio of their densities to satisfy equation 16

Hence, if one supposes that the decayed part of the material has dimished to a value E′ much smaller than E, then one can write instead of equation 16:

$$f_{decayed} = f_{sound}\sqrt{1 - \alpha^4(1 - E'/E)} \qquad (17)$$

where again $\alpha = b/a$. Equation 17 is valid only for $E'/E \le (f_{decayed}/f_{sound})^2$ otherwise the outer diameter of the cylinder might be larger than its inner one. The curve α as a function of E'/E is plotted in Fig. 14.

This model has again two limitations; one is that for absolutely weak inner material, the outer stiffer part has to be vanishingly thin and the second is that the relative strength of the materials is limited by the squared ratio of the frequencies of respectively the decayed and sound logs. A combination of both proposed models with equal relative densities and strengths of the materials would give a composite cylinder the thickness of the outer part of which would be only 7% of its radius as shown in Fig. 15. The corresponding relative densities and strengths are at around 0.38.

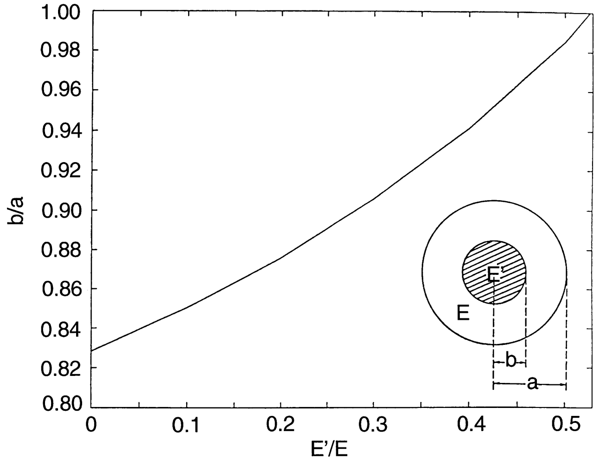


Fig. 14. Ratio of the diameters of the inner weaker part to the outer stronger part of a constant density finite length cylinder as function of the ratio of their strengths to satisfy equation 17

It would of course be better to elaborate models which could give a better representative picture of the reality, but this would require the knowledge, at least approximately, of the distribution of the mass and the strength in the decayed stem material of the log as a function of the distance from its pith. The change in mass density and strength is unlikely to happen in an abrupt manner through passing from the decayed to the non-decayed parts of a rotten tree trunk. Unfortunately, the kind of information describing the gradual change of these properties is lacking, to the best of our knowledge, from the wood research literature.

Returning to the decayed log, the spectrum of its sound radiation was measured when it was struck by a hammer in its middle and compared to that of the sound log. These two spectra are presented in Fig. 16.

Commenting on these two curves, one sees that in both of them, the peak of the f-f1 mode is very pronounced which is also somehow valid for the ovaling mode. In between these two resonance frequencies, the spectrum of the decayed log is smoother than that of the sound one. This could be due to the lesser excitation of the transversal modes in the decayed log which anyhow would be more damped by the decay than in the case of the sound log. For the decayed log, the hammer blow seems to initiate the action of only the two aforementioned important modes.

Another more important observation made on the response of the logs to the hammer impact is the presence of an amplitude modulation of their impulse responses which is displayed next in the time domain counterpart of the spectra above. The period of the amplitude modulation was a little longer for the sound log than for the decayed one, 64 ms against 57 ms, corresponding respectively to the frequencies of approximately 15 Hz and 17 Hz, i.e. at the lower limit of the human hearing sensitivity. This modulation results from the beating of two nearby tones at the major resonance frequency. The resulting warbling tone was clearly heard in the anechoic chamber from the sound log at a distance of a few decimeters from it. For the time being, no clear explanation can be given to this mode splitting phenomenon and this may constitute the subject of a later investigation. The possibility of implicating the cause to the hinging system is for sure to be discarded. The springs constituting this latter were stiff enough to make the lightest log-springs system vibrate at about 270 Hz in addition to the well known fact that

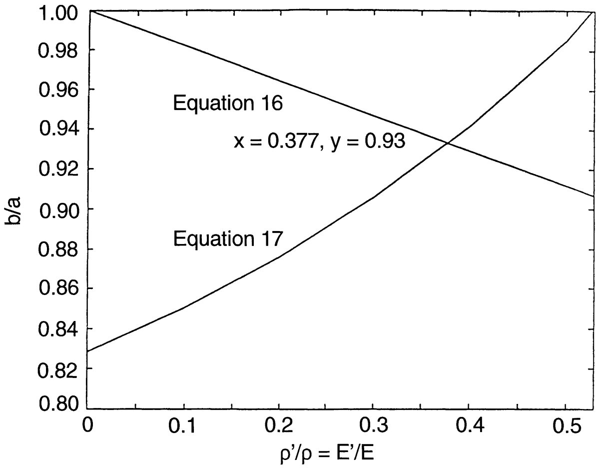
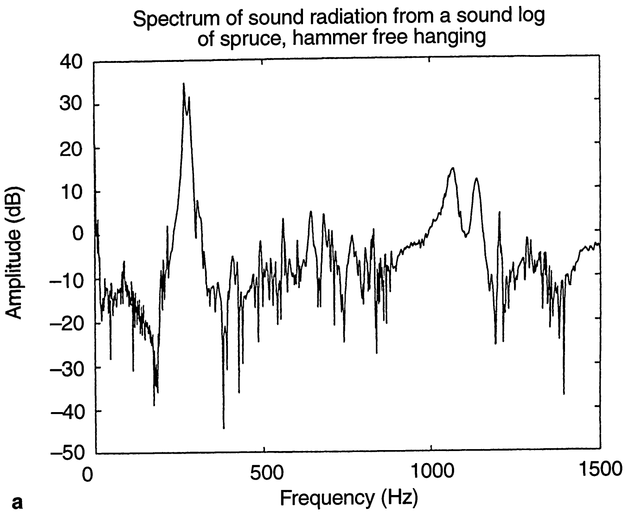


Fig. 15. Combination of the models formulated by equations 16 and 17 for equal relative densities and strengths of the constituant materials of the composite cylinder



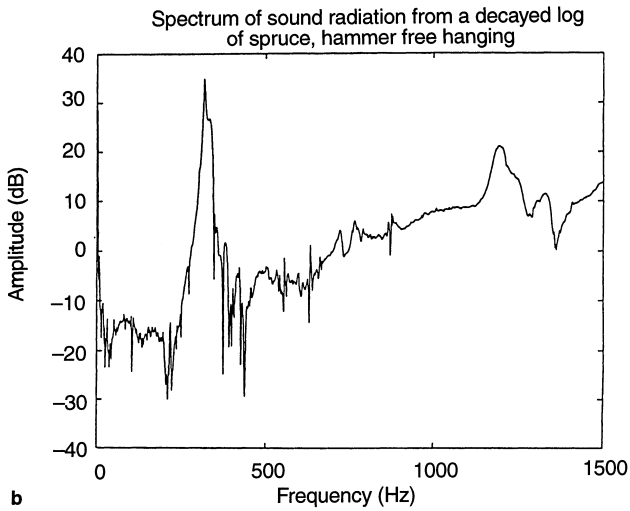


Fig. 16. Spectrum of the sound radiated by a log when hit at midst by a hammer. **a** Sound log, **b** Decayed log

the axial vibration of the log could generate but a negligible amount of sound in the air. Furthermore, and as it should normally be expected, the impulse response of the decayed log was more damped than that of the sound log.

To prove the unsymmetry caused by the presence of defects in the logs, an experiment was also made with the aim to set into motion the tortional modes. The spectra at a position on the log (the decayed one was chosen for the remaining experiments) between two diametrically opposite excitations applied tangentially on the log are shown in Fig. 18.

One notes that one cannot avoid exciting the fundamental free-free bending mode in either excitation, which actually was not hit when both excitations acted simultaneously, whilst the ovaling mode is undetectable.

For the ovaling mode which in our opinion is important to study, two more measurements were made in an attempt to prove its existence at the earlier

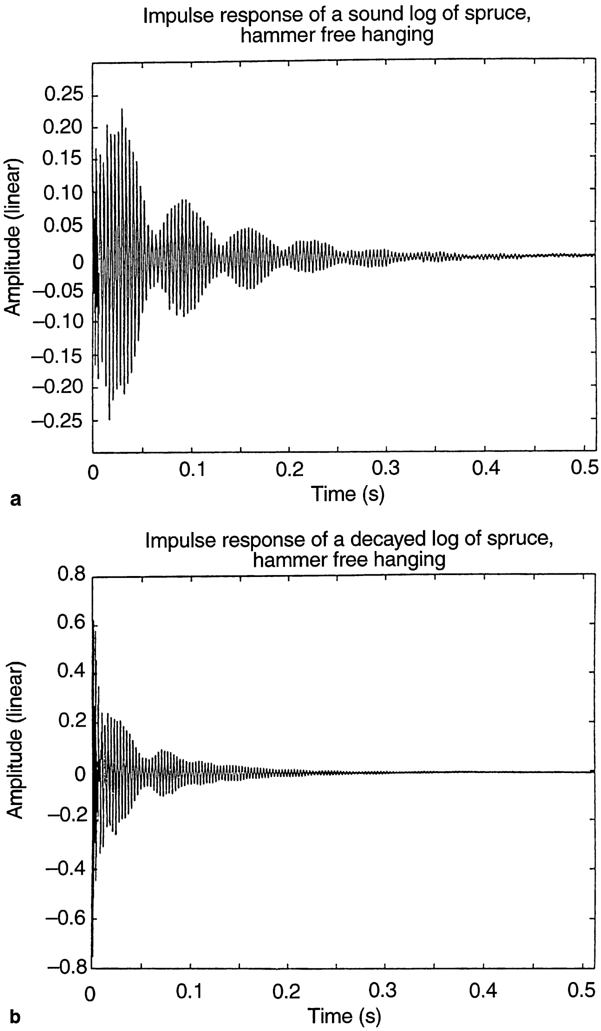


Fig. 17. Impulse response of a log struck with a hammer. **a** sound log, **b** Decayed log

presumed frequency. From Fig. 9-b one sees that on the elliptical cross-section of the cylinder in the ovaling motion, two points situated on adjacent apexes move in phase opposition. That is when one point is at its maximum displacement, the other one is at its minimum and vice-versa. In one experiment made of two parts, the phases at two such positions were measured and then these were substracted from each other. This phase difference is shown in Fig. 19-a from which one reads that the peak of height 180° is at a frequency of about 1050 Hz, somehow lower than the previously reported value and possibly due to the less precise transducer positioning.

On the average, the phase curve oscillates around 0°. The sharp peaks extending over large phase spans are a matter of graphical presentation which displays the wrapped phase. Actually these peaks are the most favourable for the in-phase motions as their jump is between 0 and +/−360°.

Back again to Fig. 9-a, one sees that the two lateral sides of a rod in bending vibration move always in opposition of phase. Hence, to set the cylindrical rod

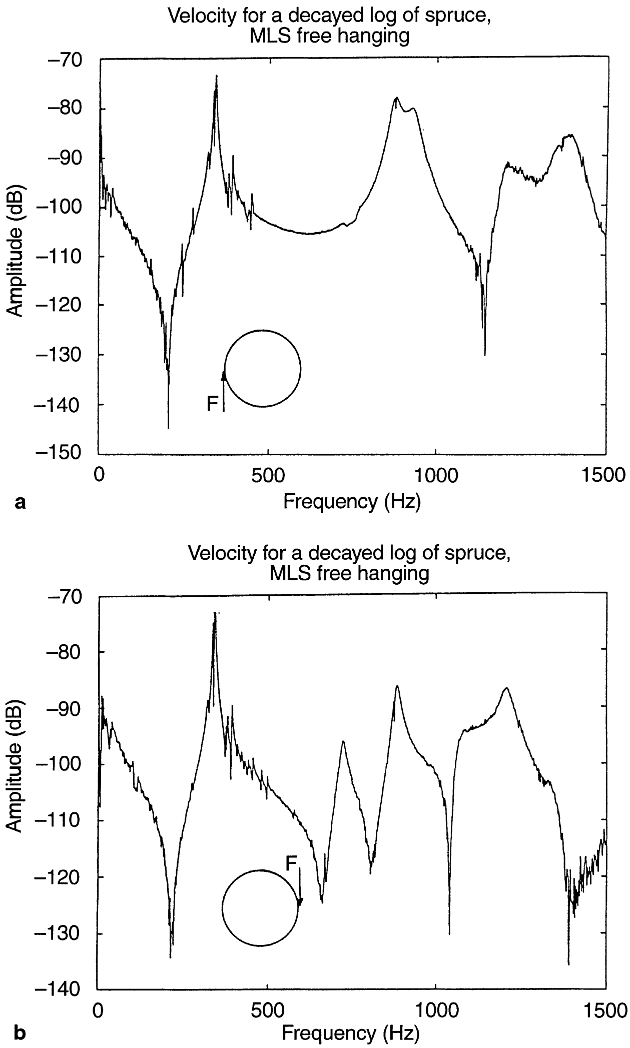


Fig. 18. Proving the impairement caused by defects in logs in tortional vibration. **a** Tangential force excitation at one position, **b** Tangential force excitation in opposite direction and at diametrally opposite position

into extensional motion and eliminate the onset of the bending modes, it is necessary to apply on one of its diametral lines two forces equal in strength and opposite in direction. The spectrum resulting from such excitation is shown in Fig. 19b. One notes that the f-f1 mode still exhibits a remainder of its strength because the slightest misplacement or misalignment of the exciting forces could set it into action.

Conclusions

In this paper, a review has been made of the different vibrational and acoustical methods used for non-destructive testing of solid structures in general with emphasis on an efficient method for evaluating their impulse responses. The state

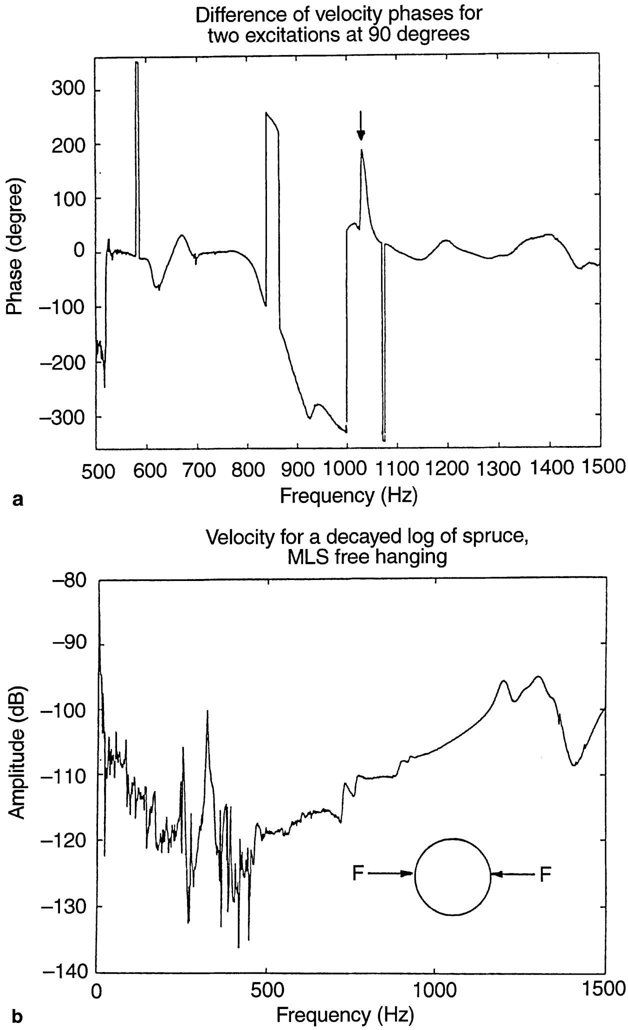


Fig. 19. Experiments for possibly detecting the presence of the ovaling mode. **a** Phase difference between two positions at 90° from each other, **b** Frequency response for excitation by two opposite equally strong forces on diametrally opposite points

of the art for implementing these techniques to detect decay in wood with possible applications on standing trees has also been reviewed. Some experiments on logs of wood have led to interesting observations. The experimental part of this work was made on logs of spruce with the aim to test the usability of vibrational testing to full-scale trees. In conclusion of these preliminary investigations it may be asserted that the vibrational response of wood logs to continuous or impact excitation is not restricted to only the bending modes. Other modes of the tortional and extentional types contribute also to their vibrational responses. Regarding the sound radiation from wood logs, the so-called ovaling mode ought, in our opinion, to beneficiate of a more in-depth investigation of its own. The reasons motivating this are exposed shortly later in the text.

The impulse responses of the sound radiation from hammer impacted logs showed some unexpected features. An amplitude modulation of the impulse response was found which was more intense in the case of sound logs. The period of this modulation was longer for the decayed log than for the sound one. The corresponding frequency which is equal to the difference between the split frequencies of the first bending mode was situated near the infrasonic domain.

To use the vibrational methods to study the motion of a full-scale tree, a good mechanical model for this latter is necessary. Some models for the simulation of vibrating trees have been proposed earlier (Yung and Fridley 1975) and research is still on its way for the understanding of tree vibrations and for the realization of simpler and better performing simulation models (Burrows and Fridley 1988; Fournier et al. 1993; Matheck 1990, 1992). To start with the simplest possible model for a motion normal to its axis, a tree can be approximated by a straight finite length beam that is clamped at one end (at the butt of the tree). It has been proved from studies on among others fir trees that branches attenuate strongly sound impulses. This could invite to assimilate a tree to a semi-infinite rod. However, one knows by experience that the whole-body response of straight trees to wind induced motion is usually characterized by a relatively slow periodical swaying. Furthermore, the eigenfrequencies of a rod in bending motion are no longer easily discernible from each other when the rod extends more and more in size. So, one concludes that the tree branches' attenuation can be efficient only in the case of longitudinal sound propagation, i.e. along the tree axis.

An improvement to the simple rod proposition is to model a tree as the combination of a clamped rod (the stem of the tree) and an extra mass (the weight of the canopy) attached to it at the centre of gravity of the tree crown, Fig. 20. Some interesting simple shapes have already been reviewed for this kind of modelization (Yanagiwara et al. 1986).

A more realistic shape of the stem of the tree and its extention may be in the form of a conical rod, though the taper of the stem is known to vary with height in a more complicated manner than in the case of a simple cone (Schniewind 1989). The eigenfrequency of the fundamental mode of a clamped-free conical rod is given by:

$$f_{cone,c\text{-}f1} = \frac{13.65}{2\pi}\frac{r}{l^2}\sqrt{\frac{E}{\rho}} \tag{18}$$

and is 2.48 that of an equally long uniform circular rod (Timoshenko et al. 1974). The Influence of an added mass to the rod near its free extremity is expected to reduce this frequency (Rayleigh 1945), the amount of which is to be determined.

Using equation 18 for a typical 15 m high cone of spruce, 60 cm wide at the butt would result in a fundamental frequency of around 10 Hz. The frequencies of the next two modes are approximately twice and four times this value, thereafter the eigenfrequencies become more and more sparsely distributed. These latter are to be reduced further after addition of the tree crown's weight. One sees thus that already at the early part of the frequency spectrum, the vibration pattern of this simple model is enough complicated. The possibility of neglecting the effect of the modes higher than the fundamental due to some kind of attenuation cannot be discarded at this stage but only experiment can confirm this statement.

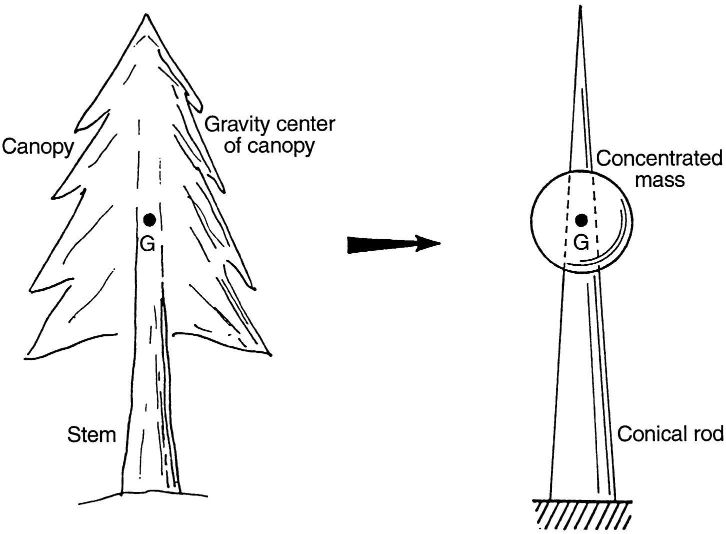


Fig. 20. Simple modelling of a tree vibrating system as the combination of a clamped conical rod and a concentrated mass

Viewed from a different perspective, the impact response of the tree trunk to a hammer stroke may be better considered as a local phenomenon. The tree trunk may in fact be regarded as a uniform cylinder, at least at comparatively short distances from either side of the hammer stroke's site. The study of the different modes involved in this impact response ought to have a special focus on the extensional ones. The existence of the ovaling mode has been predicted earlier for solid shells and its study is the subject of extensive research worldwide for the reduction of noise emission from the stators of electrical machines (Verma et al. 1987). In this latter case which concerns quite thin and short circular cylinders, the ovaling mode manifests itself as the first in the frequency response spectrum, hence its importance. A recent quite interesting study includes the ovaling mode among the axial length-independent modes and its frequency is dependent only upon the shell's radial dimensions (Wang and Williams 1996). It is worth noting

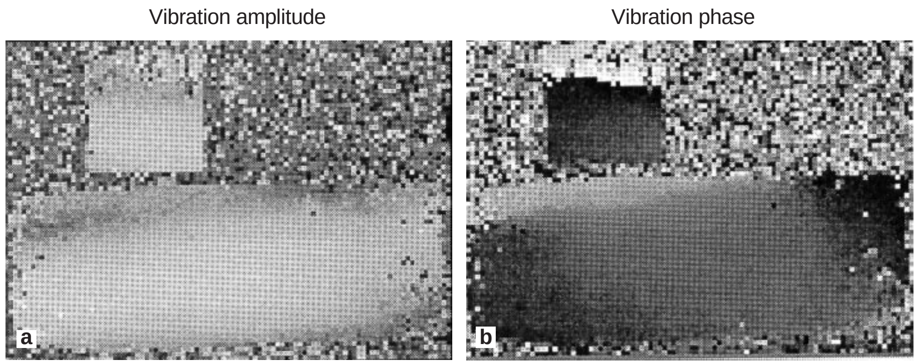


Fig. 21. Visualizing a log vibrating in the ovaling mode with a TV holography system (From Skatter 1996). **a** Amplitude, **b** Phase

that the majority, if not all, of the published material in this respect is devoted to numerical investigations, simply because exact closed form solutions are impossible to formulate to this apparently simple problem. For the case of wood, it seems that the presence of the ovaling mode among the vibration modes of logs has been confirmed only lately, Fig. 21. This happened accidentally while Skatter was investigating for the sawmilling industry the potentialities of a new system for the optical imaging of vibrating surfaces (Skatter 1996).

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