

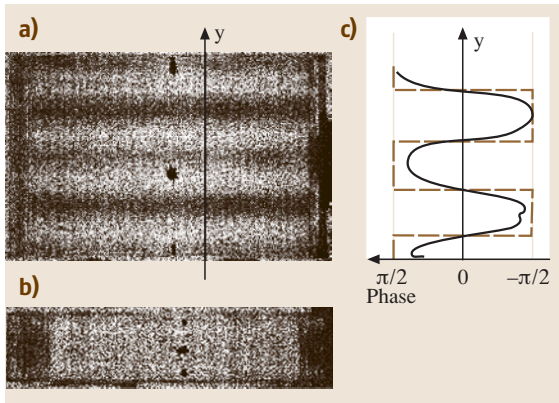


Fig. 27.10a–c Air mode (0, 4, 0) in the rectangular transparent box in Fig. 27.8 [27.60]. (a) View along the shortest side; (b) view along the next shortest side; (c) phase along the x -axis in (a) measured by phase-modulated TV holography. (Reprint by S. Hirzel Verlag, Stuttgart)

ple product $2k(n - n_0)L$. If not, the integral equation in (27.7) remains and tomography has to be used to reveal the distribution, see for instance Appendix B in [27.23]. Figs. 27.9 and 27.10 show measured standing waves inside the transparent box in Fig. 27.8 for two different air modes [27.60]. Fig. 27.11 shows standing waves inside a guitar body with a transparent top and back plate measured by phase-modulated TV holography [27.61].

If instead a probing laser ray is passing through the box in Fig. 27.8 in the x -direction, it will experience a time-varying phase change. A small frequency shift is thus generated by the standing aerial waves inside the box. This shift can be detected by a laser vibrometer; compare Sect. 27.2.5.

27.2.3 Reciprocity and TV Holography

An indirect pointwise method to measure the sound distribution from an instrument is to use reciprocity [27.62] combined with, for instance, TV holography [27.63]. An advantage of this method is that, even in a quite large spatial volume, radiativity [27.64] can be recorded. This is valuable since audible acoustical waves at low frequencies can have very long wavelengths.

An experiment using reciprocity and TV holography can be performed as follows. A movable loudspeaker at constant (unit) driving conditions excites, say a stringed instrument, from different positions outside it. The corresponding out-of-plane velocity component at the bridge foot is measured by TV holography for the specific excited vibration pattern. Then, it can be argued using reciprocity that, if a constant (unit) driving force now instead is acting at the bridge foot, it will create a sound pressure of the same magnitude as the measured velocity component at the same place as the loudspeaker was placed. By moving the loudspeaker around when

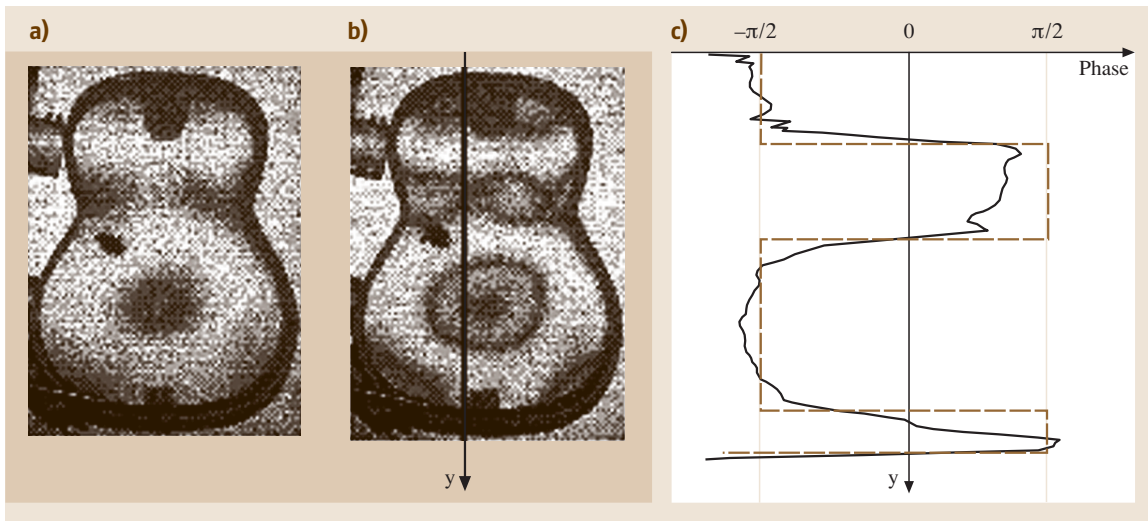


Fig. 27.11a–c Standing aerial waves inside a transparent guitar cavity measured by phase-modulated TV holography [27.61]; (a) covered sound hole (1195 Hz); (b) open sound hole (1182 Hz); (c) phase distribution along the y -axis in (b). (Reprint from S Hirzel Verlag, Stuttgart).

measuring the vibration amplitude at the bridge foot, the radiativity is estimated.

Experiences from the violin experiments in [27.63] were:

- Radiativity is measured in an ordinary laboratory both close to and at quite large distances from the instrument;
- the mode shape is simultaneously recorded,
- the violin radiativity is measured as being almost spherical up to the 500 Hz,
- complicated radiativity patterns result at higher modes.

Important conditions for dynamic reciprocity measurements are that the system is linear, passive and that the vibration system has stable equilibrium positions.

27.2.4 Pulsed TV Holography – Pulsed Lasers Freeze Propagating Bending Waves, Sound Fields and Other Transient Events

Pulsed lasers, like the traditional Q-switched ruby laser, emit one, two or more very short pulses (each pulse only some tenths of a ns long) of coherent laser light with a very high intensity. Most mechanical or acoustical events are frozen at such short exposure times. The time between the pulses can be set from about 10 μs to 800 μs with a ruby laser (during the burning time of the laser exciting flash lamps) and over a much broader range with a twin-cavity Nd:YAG laser. By double exposing a photographic plate of a transient event with two such pulses, one pulse of the object in the start position and another some time after the start, a double-exposed hologram is obtained. Both fields are reconstructed simultaneously with a continuous He–Ne laser. The technique is called pulsed holographic interferometry [27.12] and compares two states of an object. The optical setups used with pulsed lasers are in principle the same as with continuous ones with an object and a reference beam, compare Figs. 27.1 and 27.4. Both traditional recording on photographic plates (double-exposed hologram interferometry) and all-electronic methods (pulsed TV holography using a CCD chip for the recording as in TV holography) are available, although there are some differences. First, positive lenses are avoided and usually replaced by negative ones, since gas ions may otherwise occur at the focal points of positive lenses with the very strong light irradiances. With negative lenses light rays can be arranged so they seem to emanate from

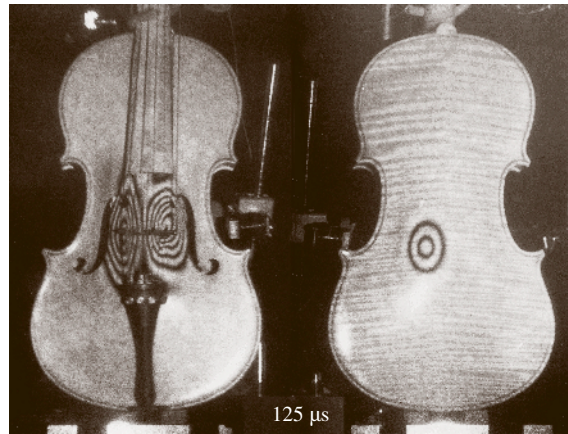


Fig. 27.12 Transient bending wave propagation in a violin body, 125 μs after impact start. The top of the bridge is impacted horizontally by a 5 mm steel ball at the end of a 30 cm-long pendulum (seen in the figure as a thin line). In the top plate a *hill* and a *valley* centered on the two bridge feet can be seen. In the back plate the fringes are centered at the soundpost, which rapidly transfers the impact from the bridge to the back of the violin [27.65]. (Reprint American Institute of Physics, 10/22/04)

a virtual focal point. Secondly, to use the temporal phase-stepping technique with a moving mirror, such as PM in Fig. 27.4 is not applicable; there is simply not enough time to move them to record a sequence. Instead quantitative measuring data are obtained with a Fourier method where a small angle between the reference light and the object light is introduced to produce a modulated carrier wave at the CCD detector [27.66]. By Fourier filtering it is then possible to extract quantitative data.

In Fig. 27.12 [27.65], the transient bending wave response of a violin body to an impact is visualized with pulsed hologram interferometry. The instrument is impacted at the top of the bridge with a small pendulum. This is a crude and simple model of one of the pulses from the pulse train that is produced by the stick–slip bow–string interaction when the instrument is played. It is obvious that energy is effectively transferred to the back plate by the soundpost of the instrument and that the back is acting more like a monopole source. The top plate, on the other hand, is excited by the rocking motion of the bridge with a *hill* at one foot and a *valley* at the other, more like a dipole source. Interesting information about the violin is pictured in this way.

The pulsed TV holography setup is quite similar to ordinary TV holography (Fig. 27.4) in that it is all electronic, it uses fast CCD cameras and is computer op-

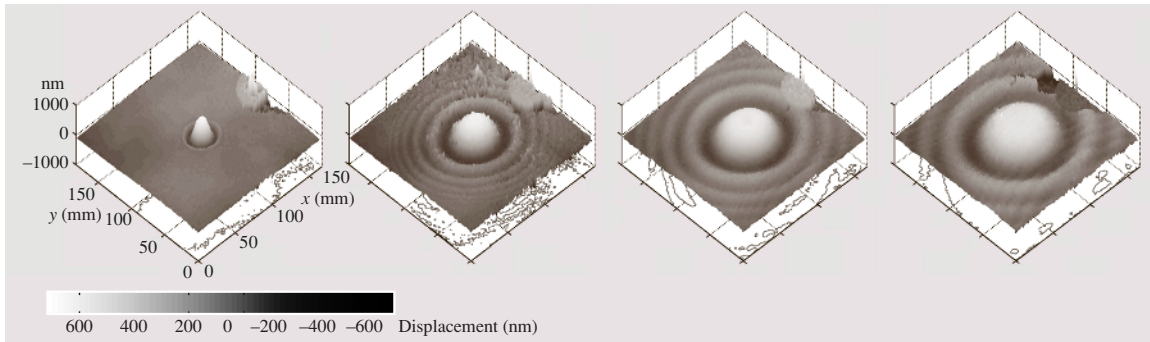


Fig. 27.13 Fig. 27.13. Measured traveling bending waves on an impacted steel plate surface 7 μ s, 32 μ s, 57 μ s and 82 μ s after impact start. These were recorded with a four-pulse TV holography technique [27.67]. The pattern at the upper right side is a shadow in the illumination path and can be ignored

erated. In an experiment [27.67], see Fig. 27.13, a pulsed TV holography setup using a four-pulse Nd:YAG laser technique is used that presents four interferograms of the same transient event. This is in contrast to double-pulse experiments with, for instance, the double-pulsed ruby laser, where the experiment must be repeated at increas-

ing time delays. With the four pulse technique a short sequence from one single event is obtained. (Another way to get a time sequence of the motion but now only in one point at a time is to use a vibrometer, see the next section.)

What kind of sound field does an impacted plate or played violin create? It is difficult to measure that in one projection since it is a 3-D sound distribution. In principle however it is possible to create a tomographic picture from many projections [27.68, 69] but such an experiment is quite complicated to perform. A much simpler and quite informative way is to use a 2-D structure instead. A simple cantilever beam or cantilever plate, much longer than it is wide and thick is used, as shown from the thinnest side in Fig. 27.14. The transient sound field outside the cantilever beam [27.70], which is impacted at the top end by an air gun bullet at about 100 m/s is shown. It is obtained by pulsed hologram interferometry. It is interesting to note that the impact causes a spherical sound wave to propagate (at about 340 m/s) around the impact point and also fast bending waves that move down the cantilever to act as secondary supersonic sound sources. They create a pattern that looks like a bow wave behind a ship. It differs from that however, since the waves have opposite phases on each side of the cantilever, as could be expected. Bending waves are dispersive, that is, fast waves preceding the slow ones have shorter wavelength than the slower ones. The figure also shows that the signal detected by a microphone or the ear from the impacted plate depends strongly upon the position of the detector. At some positions the sound from the supersonic bending waves will reach the ear earlier than the direct impact sound.

Figure 27.15 shows the quite complicated density fields that are produced when an air gun bullet at 100 m/s

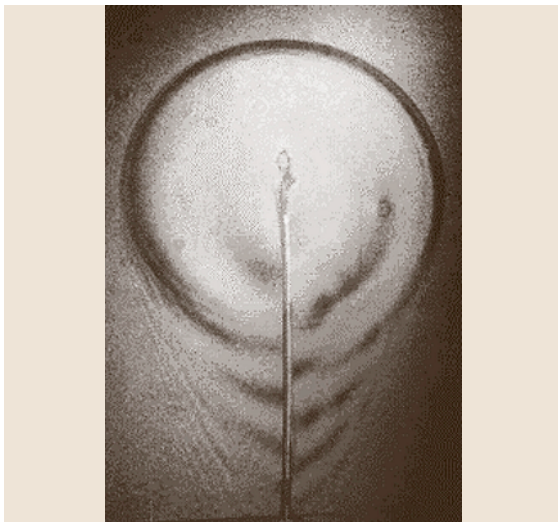


Fig. 27.14 The transient sound field outside an impacted cantilever steel beam 1 mm $\times$ 30 mm $\times$ 190 mm, clamped at the bottom. The 1 mm side is seen in the figure. The laser light passes outside and parallel to the cantilever surface along the 30 mm side. It is impacted at the *top right* by an air gun bullet. The interferogram is obtained 230 μ s after impact start by pulsed hologram interferometry. Supersonic bending waves travel down the beam and create sound waves, *bow waves*, moving faster than the spherical sound wave, centered at the impact point

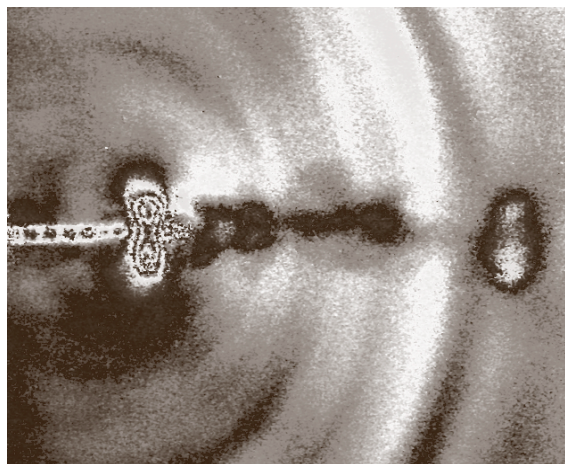


Fig. 27.15 The figure shows the density fields that are produced by an air-gun bullet traveling to the right. It emerged from the muzzle, which is situated at the left border. It has traveled in open air for $305\ \mu\text{s}$. Both the air that is pushed out by the bullet in front of the bullet in the barrel and the compressed air column behind it are seen, as well as the almost spherical and faster sound fields produced when the bullet emerged from the muzzle. Behind the bullet patterns reminiscent of vortex streets may be seen. The bullet speed is about $1/3$ the speed of sound. Recorded by pulsed TV holography [27.71] (Reprint SPIE 21/10/04)

leaves the muzzle [27.39, 71]. The air column pressed out from the barrel preceding the bullet as well as the air column behind it is seen. The almost spherical sound wave produced when the bullet is leaving the muzzle is also seen. Behind the bullet another pattern, reminiscent of vortex streets, is seen. A number of fluid mechanics and acoustical phenomena are thus visualized in the figure.

27.2.5 Scanning Vibrometry – for Vibration Analysis and for the Study of Acoustic Waves

Laser vibrometers are a family of accurate, versatile, non-contact, optical vibration transducers. They are used in applications where it is impractical or impossible to use transducers mounted on the object and they quickly map the structural response at many measurement points in sequence. They are easy-to-use, turnkey systems that include optical and electronic hardware. With optional software many possibilities exist for advanced modal analysis, animations etc. Compared to TV holography scanning laser Doppler vibrometry is more of an off-

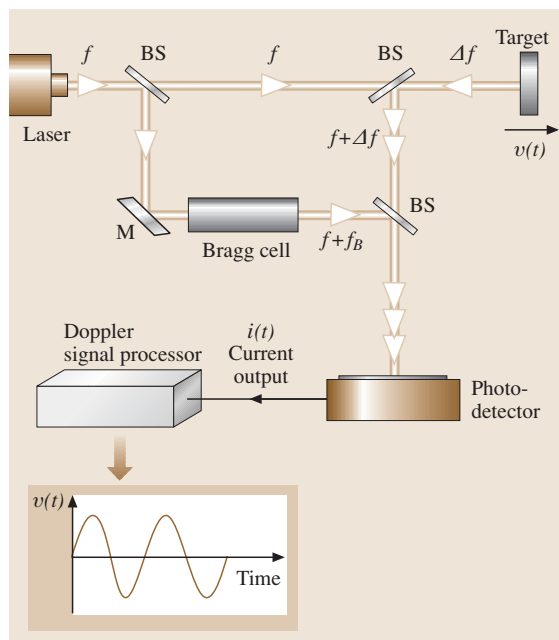


Fig. 27.16 Principle of laser vibrometer in a Mach-Zehnder-type arrangement. *BS* is the beam splitter, while *M* is the mirror. A laser emits light at a frequency f onto an object and into a reference beam. The target/object to the top right, is moving with velocity $v(t)$, causing a Doppler shift of the reflected light of $\Delta f(t)$. The photodetector receives a time-varying signal, the interference between the object and the reference light, which is processed to give a measure of the target velocity. The Bragg cell in the reference beam introduces a known and constant frequency shift f_b to the reference light using which the sign of the target velocity is determined.

the-shelf instrument, developed for industrial use. One drawback in that comparison is that it is not a real-time instrument. It is not as simple as with TV holography to find settings for unknown, interesting vibration patterns in a frequency scan. On the other hand it is highly sensitive and operates well not only with harmonic object motions but also with multi-frequency, repeatable object motions. Interesting vibration modes can be found by exciting the object with, for instance, a band-limited random signal and then performing modal analysis; compare Fig. 27.17. With scanning it is a full-field method but it has to be remembered that the measurements of different object points are made sequentially. In one measuring point, very fast events can be recorded as a function of time. One manufacturer however, makes a special unit that takes measurements at a limited

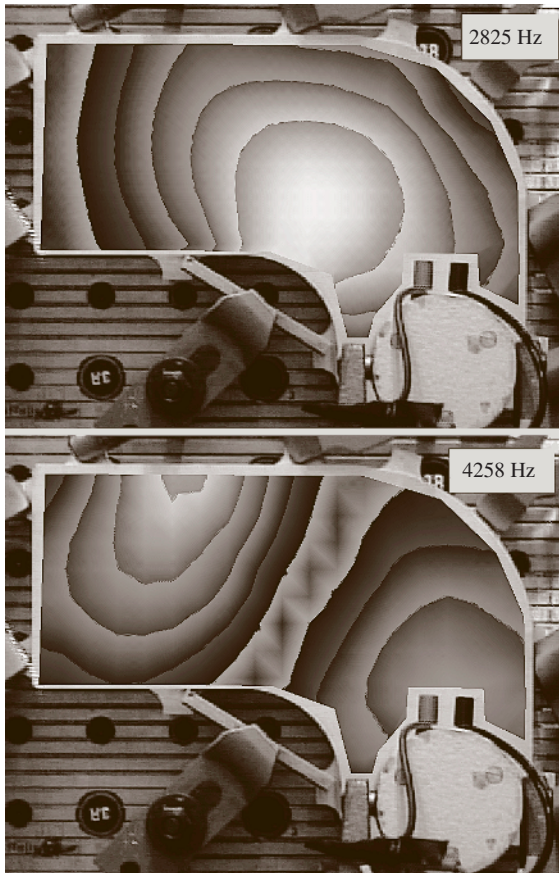


Fig. 27.17 Two vibration modes of a lightweight aluminium structure measured by scanning vibrometry, with vibration amplitude coded in grayscale. The structure was excited by a pseudo-random signal feeding a mechanical exciter. The vibration frequencies are 2825 Hz and 4258 Hz, respectively

number of points simultaneously. Informative, animated visualizations of vibrations and sound fields can be obtained. For more technical information the reader may visit the home pages of the manufacturers, since a wide variety of options are available.

Laser light of wavelength λ that is scattered back from a moving surface with velocity $v(t)$ undergoes a shift in frequency, a Doppler shift Δf

$$\Delta f(t) = 2v(t)/\lambda. \quad (27.8)$$

In an interferometer, often of the Mach–Zehnder type, the scattered light from a moving object is mixed with a reference light from the same laser. In Fig. 27.16 the optical unit of such a heterodyne vibrometer is pictured

with the Bragg cell mounted in the reference arm of the interferometer. This instrument is commonly used for measuring displacement and velocity fields of vibrations of solid objects. The first beam splitter (BS) splits the laser beam into measuring and reference beams. The reference beam reaches the photodetector via a mirror (M) and a second beam splitter. The measuring beam hits the moving target/object, gets reflected, frequency shifted and guided by two beam splitters to the photodetector. There, the measuring and the reference beams interfere to give a signal that varies with time.

A fast photodetector measures the time-dependent intensity, the beat frequency, of the mixed light,

$$i(t) = I_{av} + 2I_{mod} \cos(2\pi \Delta f t + \Phi). \quad (27.9)$$

I_{av} and I_{mod} are the average and modulating intensities, respectively, and Φ is the difference in optical phase between the diffuse object surface and the smooth reference beam. The Doppler frequency shift $\Delta f(t)$ is measured with high accuracy. Using (27.8) the target velocity is calculated. However, a sign ambiguity is present, that is, it is not obvious from the sinusoidal nature of the signal whether the object is moving towards or away from the detector. This sign problem is solved by adding a known optical frequency shift f_B (using the Bragg cell in Fig. 27.16) in the reference arm of the interferometer (the heterodyne technique is described in [27.46]) to obtain a virtual frequency offset. Another way is to add polarization components and one more detector (a homodyne technique) to get signals in quadrature.

Modern vibrometers are often equipped with a scanner. These scanning vibrometers scan the surface of a vibrating object pointwise and can present the results as animated video movies. In Fig. 27.17 two modes of vibration of a lightweight aluminium structure are shown. The structure is excited by a pseudo-random signal feeding a mechanical exciter and the different modes of vibration are estimated by Fourier analysis of the complex measuring signal. A reference signal from an accelerometer is necessary to ensure that phase at all measuring points are measured relative to the same reference.

Not only vibration modes but also sound fields are measured using vibrometry. In Fig. 27.18 the vibration behavior of the body of a guitar is pictured together with the emitted sound field from the instrument [27.72]. Vibration patterns of a guitar front plate at harmonic excitation are shown in the left part of the figure. A guitar string was continuously excited by a shaker at 600 Hz and 1200 Hz, respectively. To measure and visualize

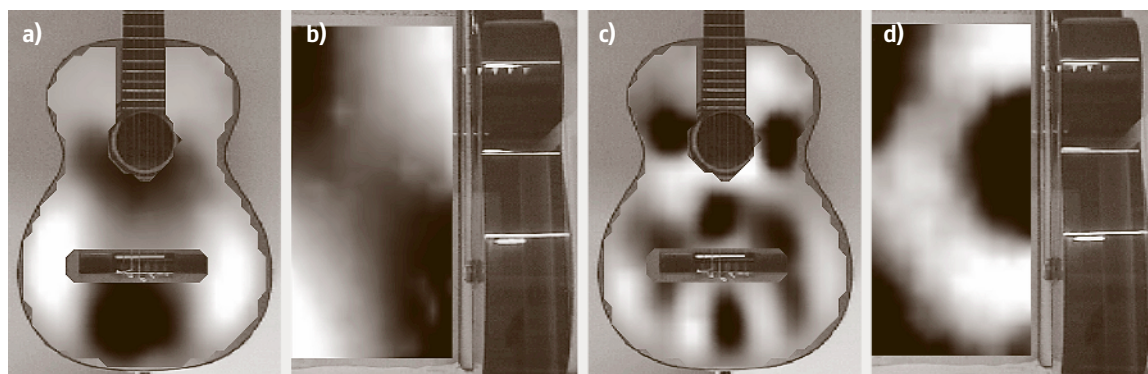


Fig. 27.18a–d Measured vibration patterns of a guitar top plate (*left*) and the projected radiated sound fields from the guitar (*right*): (a) and (b) at 600 Hz; (c) and (d) at 1200 Hz. Scanning vibrometry was used for both recordings. Observe that in (b) and (d) it is the projected sound field across the guitar body that is measured [27.72]. (Reprint S. Hirzel Verlag, Stuttgart)



Fig. 27.19 The sound field at 1303 Hz inside a saxophone cavity model measured with scanning vibrometry. The model is a conical transparent model about 54 cm long with a rectangular cross section, excited at the mouthpiece end by a normal saxophone mouthpiece and it has an artificial mouth

the acoustic waves emitted from the guitar by this vibration, the guitar is rotated 90, so that the probing laser light rays is passing in front of and in parallel to the top plate. The probing laser then hits to a heavy, rigid reflector and is reflected back again into the measuring unit of the vibrometer instrument. The reflector must be absolutely rigid, as in TV holography and pulsed TV holography, to record sound fields, compare Fig. 27.8.

The temporal and spatial pressure fluctuations $\Delta p(x, y, z, t)$, which are connected with acoustic or fluidic phenomena, cause changes of the optical refractive index $\Delta n(x, y, z, t) = n(x, y, z, t) - n_0$ and, consequently, the rate of change of the measured optical phase. In the simplest case a linear acoustic wave travels in the x -direction in the measuring volume; compare Fig. 27.8. Then the optical phase varies from point to point but at each point it also varies sinusoidally with time. This signal therefore behaves as a *virtual* displacement of the rigid reflector. In real measurements the acoustic wave is not ideally shaped in the measuring volume (i.e. it is usually not two-dimensional) and a vibrometer senses, by phase demodulation, a projected

virtual reflector displacement,

$$s(x, z, t) = \int_L \Delta n(x, y, z, t) dy \quad (27.10)$$

or by frequency demodulation, a virtual reflector velocity

$$v(x, z, t) = \int_L \dot{n}(x, y, z, t) dy, \quad (27.11)$$

i.e. the reflector seems to vibrate although it is immovable. These virtual vibrations represent the acoustic wave in the measuring volume. But, according to (27.10) and (27.11), the integrated sum of all fluctuations of the refractive index $\Delta n(x, y, z, t)$ along the laser beam contributes to the measured result. An ideal measurement situation is therefore a 2-D acoustic field as in Fig. 27.8. If not, several projections may be used or, if possible, enough that a tomographic reconstruction can be obtained.

To study the sound generation in, and the sound propagation from, a saxophone or a clarinet cavity, rectangular model pipes (two-dimensional structures) with

one transparent side wall were fabricated. Figure 27.19 shows a standing acoustic wave at 1303 Hz of a model pipe when it is excited by an artificial mouth at the reed in a normal saxophone mouthpiece (to the left in the figure). Different air-column modes inside and at the bell of the pipe can be observed.

A transient pressure pulse in a water-filled cavity, $25 \times 25 \times 25 \text{ cm}^3$, is shown in Fig. 27.20a. The pressure pulse is recorded through two opposite transparent walls from one side by pulsed TV holography. The impacted left side wall is made of steel and is hit in the middle by a pendulum (not seen in the interferogram) that creates a propagating, almost hemispherical, pressure wave in the water moving to the right at a speed of about 1500 m/s. The recording is taken $25.4 \mu\text{s}$ after the start of the impact. The pendulum is still in contact with the wall so energy and momentum are still being transferred to the tank. Figure 27.20b shows a line diagram of the same wave but now measured along the

center horizontal line. The circles in the graph are measured using the vibrometer at different distances from the wall but at the same time instants, as in pulsed TV holography recording. The different points are recorded from different experiments (the scanning facility cannot be used since the event is not repetitive and is far too rapid). As seen in the figure, the optical path differences at coincident time and space values agree quite well between the methods. The pulsed TV holography experiment measures and compares the disturbed field with the undisturbed one at one time instant. To obtain a sequence the experiment has to be repeated or a multiple-pulsed laser has to be used to record several time instants of one event. The vibrometer on the other hand can take measurements of a single transient event one point at a time as a function of time, and at several points as a function of time if the event can be repeated. The vibrometer thus records a time sequence of the optical path (or velocity) difference at each point.

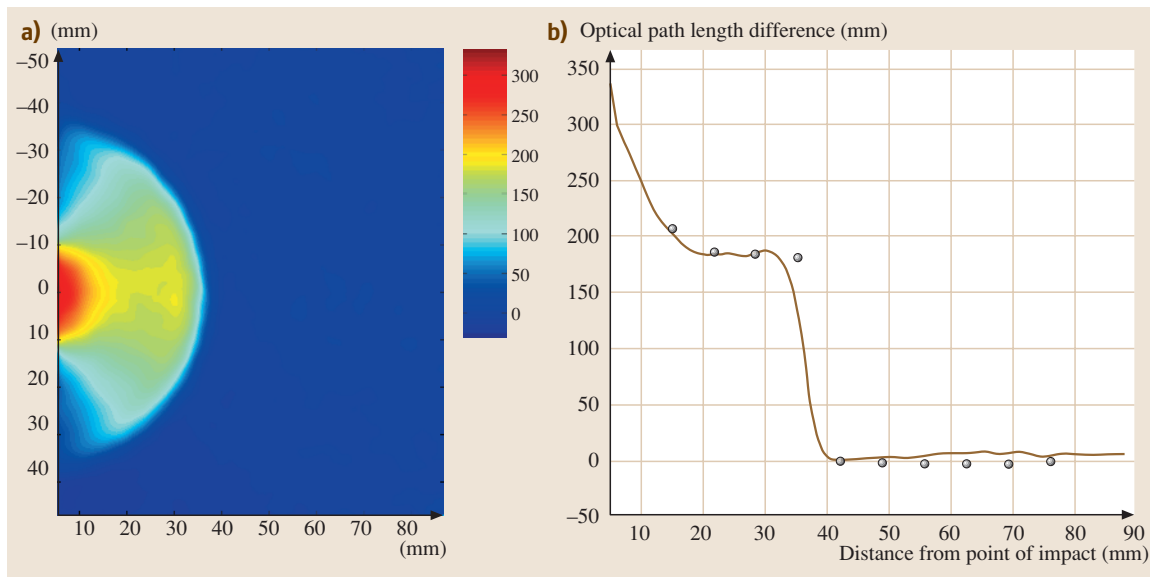


Fig. 27.20 (a) A pulsed TV holography recording of the wave propagation $25.4 \mu\text{s}$ after impact start in a water-filled transparent channel impacted at the *left wall* by a pendulum. The pendulum is hitting the *left wall* from the outside, not seen in the picture, and is still in contact with the wall when this interferogram was recorded, so energy and momentum are still being transferred to the water from the impact. The *bar* to the *right* shows the coding in optical path difference. (b) The *full line* shows the optical path difference along the *center horizontal line* measured in Fig. 27.20a, using pulsed TV holography. The *circles* in the graph are measured points using a vibrometer. The measurements are performed at different distances from the wall but evaluated at $25.4 \mu\text{s}$. The different *points* are, however, obtained from different experiments (the scanning facility cannot be used since the event is far too rapid). The optical path differences at coincident time and space point values agree quite well between the two methods

27.2.6 Digital Speckle Photography (DSP), Correlation Methods and Particle Image Velocimetry (PIV)

The equipment needed in the interferometry-based methods described above is often rather expensive and some of the techniques can unfortunately be quite difficult for an untrained user to apply. Simpler and cheaper methods have therefore been sought. Less expensive, less sensitive methods are the speckle photography (SP) and digital speckle photography (DSP) methods [27.73], which are also called speckle correlation [27.32–34]. Here the motion of the speckles themselves is measured, rather than the change in intensity of the stationary speckles as with interferometric methods. As mentioned before, the speckled random pattern *acts as if glued* to the surface of the laser-illuminated object. This pattern is imaged twice, usually with a CCD camera on individual frames, before and after some external or internal load is applied to the object, which changes the speckle pattern field. The illumination and the camera are left

stationary, so it is the motion of the microstructure of the object surface that is responsible for the motion of the speckles. The deformation field of the object may consist of the sum of rigid-body motions (translation and rotation) and some strain-induced deformation field (at crack tips, elastic and plastic deformation fields etc.). So far this is reminiscent of TV holography [(or digital speckle pattern interferometry (DSPI)] but no reference wave is added here. The experiments are not as sensitive to disturbances as interferometric measurements and can usually be performed in a workshop without vibration isolation of the quite simple optical setup. It is the position of each speckle in the speckled object wave that is recorded and the motion of small sub-images containing this pattern that is determined.

Two main types of random patterns are in use: (1) laser speckle, which is the random pattern generated by the surface microstructure of a laser-illuminated diffuse object (laser speckles in DSP) together with the specific illumination and recording geometry; and (2) white-light speckle in DSP, where there may already exist a random pattern at the object surface, such as random fibers in a paper or wood sample, which is imaged using an ordinary, broadband incandescent lamp. A random dot pattern may also be painted or sprayed onto the surface and imaged. White-light illumination can be used so that lasers can be avoided in these cases.

Two main imaging techniques are in use, one where the object surface is focused by the recording CCD camera and one where the camera is defocused behind or in front of the object surface. The focused case, which is the simplest and most commonly used, gives a measure of the in-plane motion field of the surface. However, it is important that the object is in focus, otherwise, using laser speckles, strain fields and rotations may disturb the result. It must be understood that laser speckles exist in all space, not only when focused at the object surface. Therefore, if the camera is defocused, other components of the strain tensor that affect the speckle motion will eventually become important. Using this defocused technique it has been possible to measure the strain field directly, i. e. without first measuring the deformation field and then differentiating the result to obtain the strain field [27.74].

A digital correlation technique is then used to determine how far small sub-areas in the first recording have to be moved to correlate as exactly as possible with those in the second recording. Performing this for all sub-areas, an in-plane deformation map (for the focused case) is formed. As well as yielding information

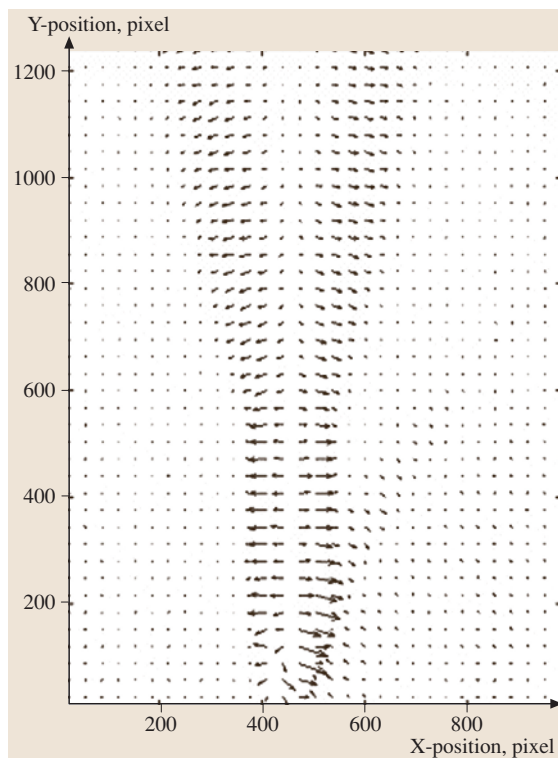


Fig. 27.21 Defocused digital speckle photography (DSP) recording, of a transparent helium jet entering air, recorded using a continuous HeNe laser

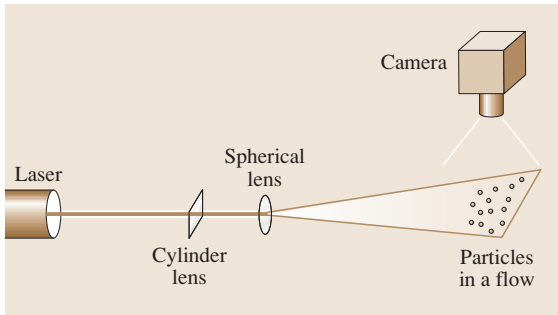


Fig. 27.22 The principle of particle image velocimetry (PIV). Pulsed laser light is sent through a cylinder lens system to form a sheet of light in the measuring volume. A CCD camera captures the seeded flow in this light sheet from above at two time instances

on the in-plane motion, the value of the correlation peak height is measured for each sub-area. This peak value gives a measure of how similar the sub-images are, even if they have not moved. If the correlation is high (almost 1) then the speckles in the two sub-surfaces are identical; if not, the correlation value will drop. Reasons for this drop include different kinds of noise and limitations in the recording system (limited and varying optical resolution and the digitalization of the intensity etc.); but it may also be that the microstructure itself has changed between the two exposures, for instance due to plastic deformation of parts of the object surface. Figure 27.21 shows an example of a defocused laser DSP recording of a transparent helium jet entering air. From similar recordings [27.75, 76] where pulsed lasers were used, it is possible to calculate where the gas jet is situated and how strong the refractive-index field of the gas is.

Particle image velocimetry (PIV) [27.77, 78] is highly reminiscent of the focused version of the speckle photography technique. However, the measurements are now made of a flow in a transparent medium, usually water or air. It is a nonintrusive optical measuring method. Tracer particles are illuminated with pulsed laser light in a light sheet, for instance smoke in air that follows the fluid motion. The displacements of the particles between two exposures are determined with a cross-correlation technique, as in digital speckle photography or speckle correlation. Figure 27.22 shows a PIV setup. Since the time between the exposures is known (often the time between repetitive pulses from a pulsed laser) and the motion of the small sub-areas between exposures of the imaged flow can be calculated, the velocity fields in the flow are determined.

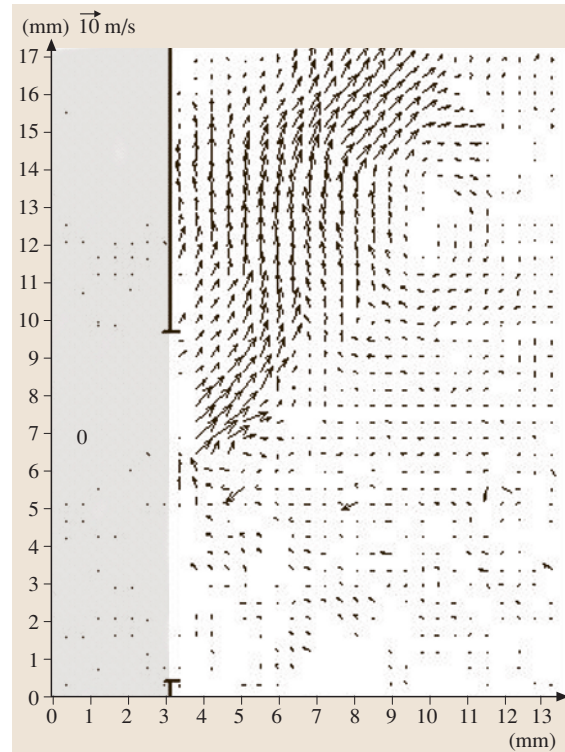


Fig. 27.23 Particle image velocimetry (PIV) measurements of the flow field at the labium (the opening to the left) of an organ pipe. A pulsed laser was used as an illumination source as the flow is quite fast at the labium; see the reference arrow in the top left

With two cameras viewing the illuminated plane from two directions and/or illuminating two different and crossing planes, not only the in-plane components but also out-of-plane velocity components can be estimated, so both 2-D and 3-D stereoscopic PIV techniques are available. With fast repetitive pulsed lasers, so called time-resolved PIV (TR-PIV), even fast turbulent flows can be measured. Microflow systems have been developed for flow studies at the microscopic scale [27.79]. By adding fluorescent particles to the flow and illuminating in a plane with a laser at a shorter wavelength and observing it at a longer (fluorescent) wavelength, concentration and temperatures can also be measured. This is called planar laser-induced fluorescent (PLIF). Sophisticated PIV systems, including hardware and software for industrial or research use, are sold by several manufacturers.

Figure 27.23 shows the flow field at the labium of a blown organ pipe, measured using PIV with

a pulsed laser as the illuminating source [27.80]. The number of papers using PIV and LDA presented at acoustical conferences is rapidly increasing [27.81, 82].

27.3 Summary

Several new optical, all-electronic measuring techniques and systems have entered the market in recent decades. The rapid technical development of new lasers, new optical sensors such as the CCD camera, optical fibers and powerful and cheap PCs has made this possible. The demands for optical techniques have increased because validation of large numerical calculations is becoming more important. Optical methods have the advantage over traditional vibration detectors that they are fast and do not disturb what is being measured, and they also often yield an image field – for instance a flow velocity field – rather than just measuring values at one single point.

Equipment can be purchased from many manufacturers and in many cases you do not have to be a specialist in optics to use them. It is also possible to make, for instance, a speckle interferometer or a shearing interferometer in most laboratories [27.83]. Methods exist today that have the sensitivity and spatial resolution needed to visualize and measure sound fields in musical acoustics.

In Table 27.1 some modern optical measuring methods and their temporal application areas are indicated.

Optical metrology on the whole is growing in importance and applicability to scientists and engineers in solid mechanics, fluid mechanics and in acoustics. Techniques such as phase-modulated TV holography, pulsed TV holography and scanning vibrometry are commonly used for modal analysis and increasingly to measure phase shifts in transparent phase objects such as sound fields. Flow fields in air and water can be measured with speckle photography or correlation methods such as digital speckle photography (DSP) and particle image velocimetry (PIV); these methods provide both illustrative images as well as quantitative measures. Reciprocity methods can also be of great help in such experiments. Optical measuring methods have the advantage of being contact-less, non-disturbing and whole-field methods. The future for optical metrology in acoustics looks bright.

Table 27.1 Some modern optical measuring methods and their temporal application areas

Optical measuring methods and their temporal application areas. Int.=Interferometry	Static or (slow) quasistatic events	Harmonic motions, single frequency	Repetitive motions, multi-frequencies	Transients, fast, dynamic events
Real-time holographic int. with continuous lasers	×	×		
Double-exposure holographic int. with continuous lasers	×			
Time-average holographic int. with continuous lasers		×		
TV holography, DSPI, ESPI with continuous lasers	×	×		
Pulsed TV holography and pulsed DSPI and ESPI with pulsed lasers	(×)	(×)	×	×
Scanning laser vibrometry		×	×	×
Speckle correlation/photography (DSP)	×			×
Particle image velocimetry, PIV	×	(×)	(×)	×

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Modal Analysis

28. Modal Analysis

Modal analysis is widely used to describe the dynamic properties of a structure in terms of the modal parameters: natural frequency, damping factor, modal mass and mode shape. The analysis may be done either experimentally or mathematically. In mathematical modal analysis, one attempts to uncouple the structural equations of motion so that each uncoupled equation can be solved separately. When exact solutions are not possible, numerical approximations such as finite-element and boundary-element methods are used.

In experimental modal testing, a measured force at one or more points excites the structure and the response is measured at one or more points to construct frequency response functions. The modal parameters can be determined from these functions by curve fitting with a computer. Various curve-fitting methods are used. Several convenient ways have developed for representing these modes graphically, either statically or dynamically. By substituting microphones or intensity probes for the accelerometers, modal

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analysis methods can be used to explore sound fields. In this chapter we mention some theoretical methods but we emphasize experimental modal testing applied to structural vibrations and also to acoustic fields.

28.1 Modes of Vibration

The complex vibrations of a complex structure can be described in terms of normal modes of vibration.

A normal mode of vibration represents the motion of a linear system at a normal frequency (eigenfrequency). Each mode is characterized by a natural frequency, a damping factor, and a mode shape. *Normal* implies that each shape is independent of, and orthogonal to, all other mode shapes of vibration for the system. Any deformation pattern the structure can exhibit can thus be expressed as a linear combination of the mode shapes.

In a stricter mathematical sense, *normal* modes provide a solution for an undamped system. Each mode shape is a list of displacements at various places and in various directions. These normal-mode vectors contain one real number for each motional degree of freedom (DOF) studied. Since real structures are invariably

damped, normal modes represent an approximation that may prove imprecise when the damping level is significant. In instances where the energy absorbing damping mechanisms are distributed in a manner proportional to the structural stiffness, normal modes provide an exact solution. In this case, the damping factor of each mode is proportional to its associated natural frequency. The normal modes also provide an exact solution when the damping is proportional to the mass distribution, but in this case each damping factor is inversely proportional to its associated natural frequency.

A structure is said to be proportionally damped when the matrix describing its damping can be written as a linear combination of the corresponding mass and stiffness matrices. Normal-mode shapes provide the exact solution to such equations, and the damping factors may

be expressed as a linear combination of the natural frequencies and their reciprocals. When these restrictive proportionality conditions do not exist, the structure may be better modeled by complex mode shapes.

Complex modes result when the instantaneous velocity at each DOF is treated as being independent of the displacement. This doubles the number of system differential equations, but simplifies them from second order to first order. The solution vectors are most commonly written in terms of complex displacements, one for each DOF. Expressing the modal displacements in this manner eliminates any constraints between the damping factors and the natural frequencies. All but the simplest experimental curve-fitting algorithms can identify complex modes. Most finite-element codes are restricted to the undamped analysis of structures and can therefore only identify the approximating real or normal modes.

Each modal vector represents a displacement pattern. A zero in the vector denotes a node, a point and direction on the structure that does not move in that mode. The element in a modal vector that is largest in displacement value is termed an anti-node. A given mode shape can be readily excited by a sinusoidal force at or near its natural frequency applied to an anti-node. The same force applied at a node will impart no motion in that shape.

It should be noted that the real or complex displacement values in each modal vector are relative numbers. The mode shape is inferred by the ratio between the vector elements, not their specific values. The elements within each modal vector may be scaled by any of various methods, the simplest of which is division by the anti-node value. This is useful for display or plotting

so that all modes have a common maximum value. However, the inner product or length of each vector is related to the modal mass of the mode. For many analytical purposes, including structural dynamic modification, it is necessary to choose the length of each vector so that the corresponding modal mass is equal to 1. When the shape vectors are scaled in this manner, they are said to be orthonormal or unit modal mass (UMM) modal vectors.

It is normally possible to excite a mode of vibration from any point on a structure that is not a node and to observe motion at all points that are not nodes. A mode shape is a characteristic only of the structure itself, independent of the way it is excited or observed. In practical terms, however, instrumentation associated with both excitation and observation may modify the structure slightly by adding mass or stiffness (or both). This results in small shifts in frequency and mode shape, which in most cases are negligible. Acoustic excitation and optical monitoring are the least intrusive, but mechanical means of driving and observing are frequently more convenient and less costly.

Mode shapes are unique for a structure, whereas the deflection of a structure at a particular frequency, called an operating deflection shape (ODS), may result from the excitation of more than one normal mode. When exciting a structure at a resonance frequency, the ODS will be determined mainly by one mode, although if several modes have nearly the same frequency, special techniques may be required to determine their contributions to the observed ODS. Modes of a structure are functions of the entire structure. A mode shape describes how every point on the structure moves when it is excited at any point.

28.2 Experimental Modal Testing

Modal testing is a systematic method for identification of the modal parameters of a structure. Generally these include natural frequencies, modal damping, and UMM mode shapes. In experimental modal testing the structure is excited with a measured force at one or more points and the response is determined at one or more points. From these sets of data, the modal parameters are determined, often by the use of multidimensional curve-fitting routines on a digital computer.

Modal testing may use sinusoidal, random, pseudo-random, or impulsive excitation. The response may be measured mechanically, optically, or indirectly (by ob-

serving the radiated sound field, for example). The first step in experimental modal testing is generally to obtain a set of frequency response functions.

There is a vast amount of good literature describing experimental modal testing. Ewins [28.1] provides a good overall introduction. The proceedings of the annual International Modal Analysis Conference (IMAC), held every year since 1982, is a gold mine of papers on the subject [28.2]. The Structural Dynamics Research Laboratory at the University of Cincinnati has published many scientific papers and reports on the subject, and has included several tutorials on their website [28.3].

28.2.1 Frequency Response Function

The frequency response function (FRF) is a fundamental measurement that isolates the inherent dynamic properties of a mechanical structure. The FRF describes the motion-per-force input–output relationship between two points on a structure as a function of frequency. Since both force and motion are vector quantities, they have directions associated with them. An FRF is actually defined between a single-input DOF (point and direction) and a single-output DOF.

In practice, the force and response are usually measured as functions of time, and transformed into the frequency domain using a fast Fourier transform (FFT) analyzer. Due to this transformation, the functions end up as complex numbers; the functions contain real and imaginary components (or magnitude and phase components). Depending on whether the response motion is measured as displacement, velocity, or acceleration, the FRF can be expressed as *compliance* (displacement/force), *mobility* (velocity/force), *accelerance* or *inertance* (acceleration/force), *dynamic stiffness* (1/compliance), *impedance* (1/mobility), or *dynamic mass* (1/accelerance). Force can be measured with a piezoelectric force transducer or load cell; acceleration can be measured with an accelerometer; velocity can be measured with a laser velocimeter or obtained by integrating acceleration; displacement can be determined by holographic interferometry or by integrating velocity.

Because it is a complex quantity, the FRF cannot be fully displayed on a single two-dimensional plot. One way to present an FRF is to plot the magnitude and the phase as in Fig. 28.1a. At resonance, the magnitude is a maximum and is limited only by the damping in the system. The phase ranges from 0 to 180° and the response lags the input by 90° at resonance. The 3 dB width of each resonance peak (bounding locations of 0.707 resonance amplitude) is determined by the damping. At these half-power points the phase angle is $\pm 45^\circ$.

Another way of presenting the FRF is to plot the real and imaginary parts as in Fig. 28.1b. The imaginary part has a peak at resonance, while the real part goes through zero at resonance. The real component exhibits peaks of opposite sign at the extremes of the half-power bandwidth.

A third method of presenting the FRF is to plot the real part versus the imaginary part as shown in Fig. 28.1c. This is called a Nyquist or vector response plot. Each mode produces a circular pattern. The diameter of the circle is proportional to the product of the

UMM vector elements for the force and response DOFs. Maximum change in arc length with frequency occurs at the natural frequency.

If measurements are made at n points, the FRFs can be arranged in an $n \times n$ matrix with elements designated

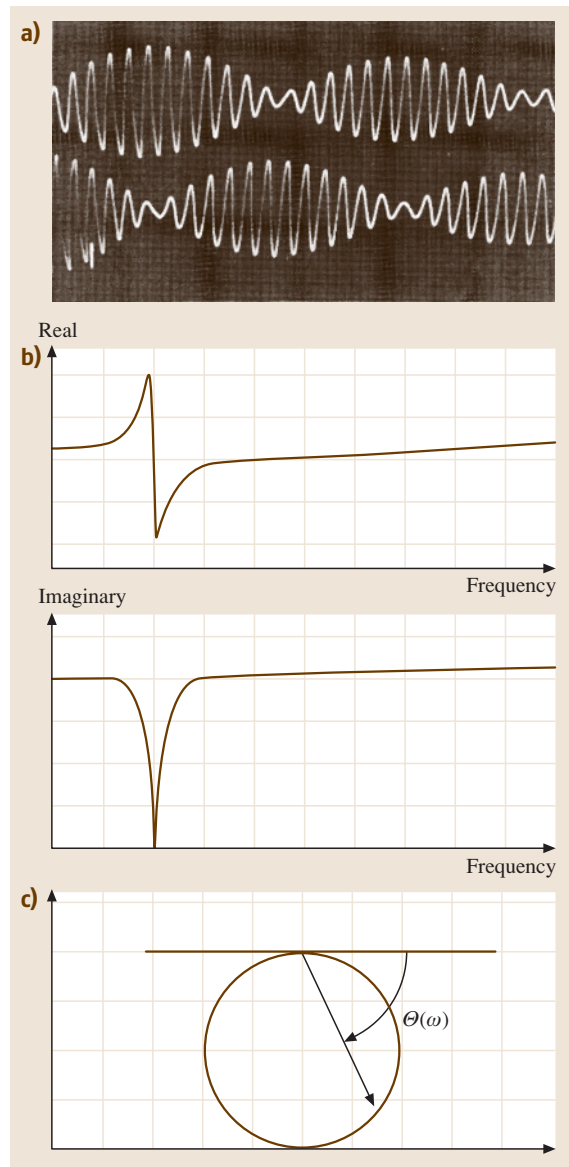


Fig. 28.1a–c Three methods for presenting the frequency response function (FRF) of a vibrating system graphically. (a) Magnitude and phase versus frequency; (b) Real and imaginary parts versus frequency; (c) Real part versus imaginary part (a Nyquist plot)

h_{ij} . The diagonal elements denote that the excitation and observation occurred at the same point. Such measurements are termed driving-point FRFs and they are of particular importance as at least one driving-point measurement is required to scale the mode shapes to UMM form. In a linear system the FRF matrix is symmetric: $h_{ij} = h_{ji}$. We do not have to measure all the terms of the FRF matrix. In practice, it is (generally) possible to obtain mode shapes from only one row or column of the FRF matrix.

Theoretically, it does not matter whether the measured frequency response comes from a shaker test or an impact test. If the structure is impacted at n points while the response is measured at one point, one row of the FRF is obtained. If a shaker is used to excite it at one point while an accelerometer (or other sensor) is moved from one point to another, one column of the FRF is obtained. In practice, there may be differences between the results of the *roving hammer* and the *roving accelerometer* methods, but these are the result of experimental compromises, not structural properties. Roving impact and roving response techniques provide essentially the same answers because a linear structure exhibits reciprocity. The motion produced at DOF a by a force applied at DOF b is identical to the motion at b due to a force at a .

The analog signals obtained from the measuring devices are generally digitized in an FFT analyzer. Sampling and quantization errors can occur in the process, but the most worrisome of signal-processing errors is probably leakage, which will be discussed later.

Serious consideration must be given to the way the structure is mounted and supported. Boundary conditions can be specified exactly in a completely free or completely constrained situation. In practice it is generally not possible to fully achieve these conditions. In order to approximate a free system, the structure can be suspended from very soft elastic cords or placed on a very soft cushion. The structure will be constrained to a degree and the rigid-body modes will not have zero frequency. However, they will generally be much lower than the frequencies of the flexible modes and will therefore have negligible effect. Some support fixtures affect damping as well.

28.2.2 Impact Testing

A roving hammer test is the most common type of impact test. An accelerometer is fixed at a single DOF, and the structure is impacted at as many DOFs as desired to define the mode shapes of the structure. Using a two-

channel FFT analyzer, FRFs are computed, one at a time, between each impact DOF and the fixed response DOF. A suitable grid is usually marked on the structure to define the impact points.

Piezoelectric transducers are generally used to sense both force and acceleration. They generate an electrical charge when mechanically strained. Because of their high impedance, charge amplifiers are generally used to amplify the signals. More-modern sensors incorporate an amplifier within the transducer and power it from a constant-current source using the same two-conductor cable that transmits the signal (at low impedance). The resonant frequencies of the transducers must be well above the highest frequency that will be measured. The mass of the accelerometer should be small enough that the effect of mass loading is small. One way to determine if mass loading is significant is to measure an FRF with the accelerometer and compare it to a second measurement with an additional accelerometer (or equivalent mass) attached to it. An obvious advantage of this testing setup is that the response transducer is never moved; the structure is subjected to a consistent mass loading.

In some circumstances, the symmetry of the test object may produce repeated roots. These are modes with different mode shapes that have the same natural frequency. When such modes are encountered, they may be separated by using two or more fixed reference accelerometers and multi-reference curve-fitting techniques. Of course, an analyzer with three or more channels is required if the data is to be acquired in a single measurement set.

It is generally impossible to impact a structure in all three directions at all points, so three-dimensional (3-D) motion cannot be measured at all points. When 3-D motion at all points is desired, a roving triaxial accelerometer may be used and the structure is impacted at a fixed DOF with the hammer. Triaxial accelerometers are usually more massive, however. Since the triaxial accelerometer must be simultaneously sampled together with the force data, a four-channel FFT analyzer is required. [28.4] Note, however, that the roving accelerometer presents a different mass-load to the structure at each response site. This can lead to inconsistencies in the data that make an accurate curve fit more difficult.

Because the impulse signal exists for such a short period, it is important to capture all of it in the sampling window of the FFT analyzer. To insure that the entire signal is captured, the analyzer must be able to capture the impulse and response signals prior to the occurrence of the impulse. The analyzer must begin sampling data

before the trigger point occurs. This is called a pre-trigger delay.

Modern FFT analyzers provide very high-resolution (24 bit) analog-to-digital converters (ADCs) and large capture block sizes (64 kpoint typical). These characteristics nullify the need to use weighting functions or *windows* when performing an impact test. The preferred analysis is conducted without weighting, also termed using a *rectangular* window.

Two common time-domain windows that are used in impact testing with older equipment are the force and exponential windows. These windows are applied to the signals after they are sampled but before the FFT is applied to them in the analyzer. The *force* window is used to remove noise from the force signal. Any nonzero data following the impulse signal in the sampling window is assumed to be measurement noise. The force window preserves the samples in the vicinity of the impulse but removes the noise from all other samples in the force signal.

The *exponential* window is used to reduce leakage in the spectrum of the response. The FFT analyzer assumes that the signal is periodic in the transform window. This is true of signals that are completely contained within the transform window or cyclic signals that complete an integer number of cycles within the transform window. If a time signal is not periodic in the transform window a smearing of its spectrum will occur when it is transformed to the frequency domain. This is called *leakage*. Leakage distorts the spectrum and makes it inaccurate. If the response does not decay to zero before the end of the sampling window, an exponential window can add artificial damping to all modes of the structure. This artificial damping must be removed by the subsequent curve-fitting algorithm to obtain proper damping factors.

It is important that the impact hammer provides a pulse that is well matched to the frequency span of the analysis. This is accomplished by fitting a striking tip of appropriate stiffness to the hammer's force gauge. A soft tip produces a broad pulse time-history with a narrow spectrum. A hard tip increases the force spectrum bandwidth by applying a narrow pulse. The spectrum of the force pulse has a *lobed* structure and all tests are done using the spectral content of the first lobe. Trial measurements are made to select a tip that provides a force spectrum that falls off no more than 25 dB from the direct-current (DC) point to the selected analysis bandwidth.

It is often difficult to strike at exactly the same place and angle multiple times. For this reason, averaging is

less useful in an impact test than in other types of experimental modal analysis. Many experienced practitioners favor conducting tests with a single strike at each target DOF. This precludes calculating a coherence function, but that very useful causality measurement often tells you more about your ability to hit the same place twice than it does about structural nonlinearities or noise in measurement. For this reason, it is imperative to inspect every measurement set in the time domain before accepting it. Most analyzers automate this type of acquisition, giving you an accept/reject control.

28.2.3 Shaker Testing

Not all structures can be impact tested. Sometimes the surface is too delicate. Sometimes the impact force has too low an energy density over the entire frequency range of interest. In this case, FRF measurements must be made by attaching one or more shakers to the structure.

Since the FRF is a single input function, the shaker should transmit only one component of force in line with the main axis of the load cell. Often a structure tends to rotate slightly when it is displaced along an axis. To minimize the problem of forces being applied in other directions, the shaker is generally connected to the load cell through a slender rod called a stinger to allow the structure to move freely in the other directions. The stinger should have a strong axial stiffness but weak bending and shear stiffnesses.

A variety of broadband excitation signals have been developed for making shaker measurements with FFT analyzers: transient, random, pseudo-random, burst random, sine sweep (chirp). A true random signal is synthesized with a random number generator. Since it is nonperiodic in the sampling window, a Hanning window must always be used to minimize leakage.

A pseudo-random signal is specially synthesized by an FFT analyzer to coincide with the window parameters. It is synthesized over the desired frequency range and then passed through an inverse FFT algorithm to obtain a random time-domain signal, converted to an analog signal and used as the shaker excitation signal. Since the excitation signal is periodic in the sampling window, the acquired signals are leakage-free. However pseudo-random excitation does not excite nonlinearities differently between spectrum averages and will not, therefore, remove nonlinearities from FRF measurements.

Burst random excitation combines some advantages of both random and pseudo random testing. Its signals are leakage free and when used with spectrum averag-

ing, will remove nonlinearities from the **FRFs**. In burst random testing, either a true random or time-varying pseudo-random signal can be used, but it is turned off prior to the end of the sampling window time period so that the response decays within the sampling window. Hence, they are periodic in the window and leakage-free [28.1].

In large structures more than one shaker may be needed to obtain sufficient excitation. These are driven simultaneously so that the structure is subjected to multiple inputs. The **FRFs** between each of the inputs and each of the multiple outputs are calculated by using a matrix inversion process. This type of measurement is called multiple-input multiple-output (**MIMO**) testing. It provides **FRFs** from multiple columns of the **FRF** matrix and two special types of coherence functions. One multiple coherence function is generated for each shaker used. This function may be thought of as a generalization of ordinary coherence; it asserts what fraction of all the response output power spectra may be attributed to the single measured force power spectrum and the **FRFs** in one column of the **FRF** matrix. One partial coherence function is generated for each **FRF**; it asserts what fraction of a single response power spectrum can be attributed to one force power spectrum and a single **FRF**. **MIMO** data is analyzed using a multi-reference curve fitter.

28.2.4 Obtaining Modal Parameters

Most experimental modal analysis relies on a modal parameter estimation (curve-fitting) technique to obtain modal parameters from the **FRFs**. Curve fitting is a process of matching a mathematical expression to a set of experimental points by minimizing the squared error between the analytical function and the measured data. Curve-fitting methods used for modal analysis fall into one of the following categories: local single degree of freedom, local multiple degree of freedom, global, or multi-reference.

Single-degree-of-freedom (**SDOF**) methods estimate modal parameters one mode at a time. Multiple-degree-of-freedom (**MDOF**), global, and multi-reference methods can estimate modal parameters for two or more modes at a time. Local methods are applied to one **FRF** at a time. Global and multi-reference methods are applied to an entire set of **FRFs** at once.

SDOF methods can be applied to most **FRF** data sets with low modal density, but **MDOF** methods must be used in cases of high modal density. Global methods work better than **MDOF** methods for cases with local

modes. Multi-reference methods can find repeated roots (very closely coupled modes) where the other methods cannot [28.4].

Perhaps the simplest of the **SDOF** methods is the quadrature or peak picking method. Modal coefficients are estimated from the real and imaginary parts of the frequency response, so the method is really not a curve fit in the strict sense of the term. First the resonance peaks are detected on the imaginary plot and the frequencies of maximum response taken as the natural frequencies of these modes. Then the line width between the half-power points is determined from the real plot, from which damping can be estimated. The magnitude of the modal coefficient is taken as the value of the imaginary part at resonance. This method is adequate for structures whose modes are well separated and have modest damping [28.5].

Another **SDOF** method is the circle-fit method. This is based on the fact that a Nyquist plot of frequency response of a **SDOF** system near a resonance is a circle. Thus the circle that best fits the data points is determined. Half-power points are those frequencies for which $\theta = \pm 90$. The modal coefficient is determined from the diameter of the circle.

Two of the most popular **MDOF** methods are the complex exponential and rational fraction polynomial methods. The complex exponential method (or Prony algorithm) fits the impulse response function (**IRF**), which is the inverse Fourier transform of the **FRF** rather than the **FRF** itself. A potentially serious error, called wrap-around error or time-domain leakage, can occur but can be mitigated by using an exponential window on the **IRF**. This error is caused by the limited frequency range of the **FRF** measurement, and distorts the impulse response. The complex exponential algorithm works very well for **FRFs** with high modal density. In most applications, the algorithm is allowed to fit far more modes than the **FRF** form suggests are present. The resulting extraneous *computational modes* are then discarded, normally by using a stability diagram.

The rational fraction polynomial method fits a rational fraction of polynomials expression directly to an **FRF** measurement. Its advantage is that it can be applied over a user-selected frequency range of data, focusing its findings upon that frequency interval. It does not generate *computational modes* and therefore does not require the use of a stability diagram or other methods to filter its outputs.

Global curve fitting divides the curve-fitting process into two steps: estimating the frequency and damping parameters, and using these to obtain mode shape es-

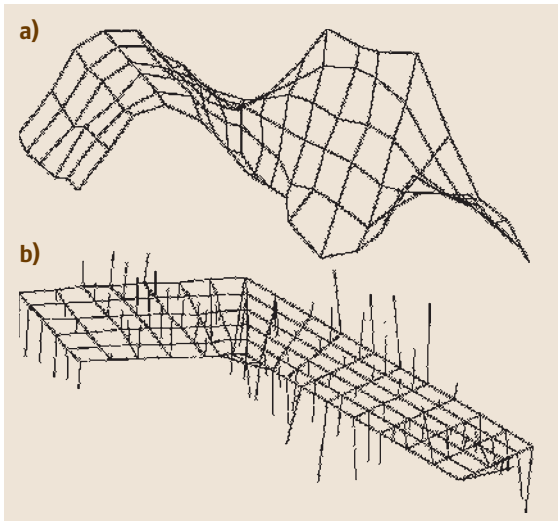


Fig. 28.2a,b Two ways of representing a vibrational mode in a Korean *pyeongyeong*: (a) The shape at maximum bending; (b) The neutral shape with vectors

timates by a second estimation process. The advantage of global curve fitting is that more-accurate frequency and damping estimates can potentially be obtained by processing all of the measurements rather than relying on a single measurement. Another advantage is that, because damping is already known and fixed as a result of the first step, the modal coefficients are more accurately estimated during the second step. Both the complex exponential and rational fraction polynomial methods can be formulated to obtain global estimates from a set of measurements [28.6].

28.2.5 Real and Complex Modes

The assumption of proportional viscous damping implies the existence of real, or normal modes. Mathematically, this implies that the damping matrix can be defined as a linear combination of the physical mass and stiffness matrices. Physically, all the points in a structure reach their maximum excursion, in one or the other direction, at the same time. The imaginary part of the *FRF* reaches a maximum at reso-

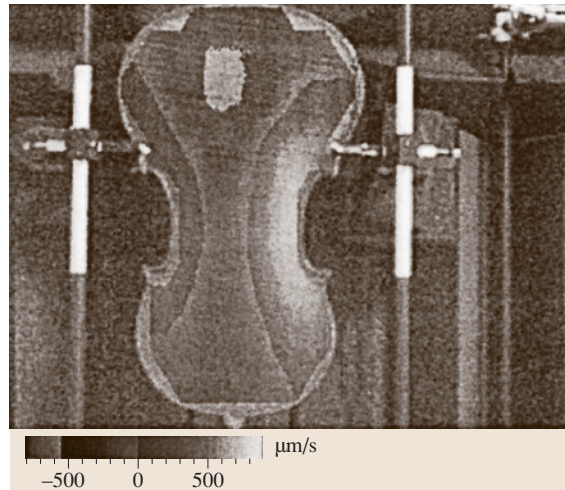


Fig. 28.3 Gray-scale representation of a vibration mode in a violin (courtesy of George Bissinger)

nance, and the Nyquist circle lies along the imaginary axis.

Some structures exhibit a more complicated form of damping, and the mode shapes are complex, meaning the phase angles can have values other than 0 or 180°. Different points on the structure reach their maxima at various times as in a traveling wave pattern. The imaginary part of the *FRF* no longer reaches a maximum at resonance nor is the real part zero. The Nyquist circle is rotated at an angle in the complex plane. When damping is light, the proportional damping assumption is generally an accurate approximation, [28.5] although it can be argued that a complex-mode formulation is essential to preserve accuracy in damping evaluation [28.7].

28.2.6 Graphical Representation

One of the nice features of experimental modal testing is the way that the modes can be represented graphically. Animations of the vibration can be viewed from any angle to comprehend complex mode shapes. Static representations include the shape at maximum bending, and the neutral shape with vectors, as shown in Fig. 28.2. Another representation is shown in Fig. 28.3.

28.3 Mathematical Modal Analysis

In mathematical modal analysis, one attempts to uncouple the structural equation of motion by means of some suitable transformation, so that the un-

coupled equations can be solved. The frequency response of the structure can then be found by summing the respective modal responses in accordance

with their degree of participation in the structural motion.

Sometimes it is not possible to obtain exact solutions to the equation of motion. Computers have popularized the use of numerical approximations such as finite-element and boundary-element methods.

28.3.1 Finite-Element Analysis

The finite-element method of analysis (FEA or FEM) started in the late 1940s as a structural analysis tool useful in helping aerospace engineers design better aircraft structures. Aided by the rapid increase in computer power since then, it has become a very sophisticated tool for a wide array of engineering tasks. It can be used to predict how a system will react to environmental factors such as forces, heat, and vibration.

The technique is based on the premise that an approximate solution to any complex engineering problem can be reached by subdividing the problem into smaller more manageable (finite) elements. Using finite elements, solving complex equations that describe the behavior of a structure can often be reduced to a set of linear equations that can be solved using the standard techniques of matrix algebra.

The finite-element method works by breaking a real object down into a large number of elements, which are regular in shape and whose behavior can be predicted by a set of mathematical equations. The summation of the behavior of each individual element produces the expected behavior of the actual object. Finite-element analysis (FEA) uses a complex system of points called *nodes*, which make a grid called a *mesh*. The mesh is programmed to include the material and structural properties that define how the structure will react to certain loading conditions.

General-purpose finite-element codes, such as NASTRAN and ANSYS, are programmed to solve the matrix equation of motion for the structure of the form:

$$[M] \{\ddot{u}\} + [C] \{\dot{u}\} + [K] \{u\} = \{F \cos(\omega t + \phi)\}, \quad (28.1)$$

where $[M]$ is the mass matrix, $[C]$ is the damping matrix, $[K]$ is the stiffness matrix, $\{u\}$ is displacement, and $\{F\}$ is force.

The model details are entered in a standard format, and the computer assembles the matrix equation of the structure. The first step is to solve the matrix equation $[M] \{\ddot{u}\} + [K] \{u\} = \{0\}$ for free vibrations of the structure. The solution to this equation gives the natural frequencies (eigenvalues) and the undamped mode

shapes (eigenvectors). These parameters are the basic dynamical properties of the structure, and they are used in subsequent analysis for dynamic displacements and stresses. For harmonic motions, $\{\ddot{u}\} = -\omega^2 \{u\}$, so that $[K]^{-1} [M] \{u\} = [I] \{u\}$, where $[I]$ is the unity matrix. Typically the matrix equations of motion for the structure contain off-diagonal terms, but they may be decoupled by introducing a suitable transformation. The damping matrix may also be uncoupled on the condition that the damping terms are proportional to either the corresponding stiffness matrix terms or the corresponding mass matrix terms [28.8].

Finite-element analysis in conjunction with high-speed digital computers permits the efficient solution of large, complex structural dynamics problems. Structural dynamics problems that are linear can generally be solved in the frequency domain. There are other applications of finite-element analysis in acoustics. The sound field produced in an enclosure or by a radiating surface vibrating at a single frequency can be described by the Helmholtz equation

$$\nabla^2 p + k^2 p = 0, \quad (28.2)$$

where p is the acoustic pressure, k is the wave number ($k = \omega/c$) and ∇ is the Hamiltonian nabla operator.

The most common boundary conditions are fixed pressure and fixed velocity on the surface of the enclosed or radiating body. Other types of boundary conditions are normal and transfer impedances [28.9].

Exact analytical solutions of the Helmholtz equation exist for a very limited number of idealized geometries corresponding to situations where the geometry of the radiating surface can be described by orthogonal coordinate systems that are separable. In order to solve the Helmholtz equation for more general applications, numerical schemes are necessary.

28.3.2 Boundary-Element Methods

The boundary-element method (BEM) is also a powerful computational technique, providing numerical solutions to a wide range of scientific and engineering problems. Since only a mesh of the boundary of the domain is required, the method is generally easier to apply than the finite-element method. The boundary-element method has found application in stress analysis, structural vibrations, acoustic fields, heat transfer, and potential flow. The advantage in the boundary-element method is that only the boundaries of the domain of the partial differential equation require subdivision, compared to FEA

in which the whole domain of the equation requires discretization.

The boundary-element method is especially effective for calculating the sound radiated by a vibrating body or for predicting the sound field inside a cavity. A typical BEM input file consists of a surface mesh, a normal velocity profile on the surface, the fluid density, speed of sound, and frequency. The output of the BEM includes the sound pressure distribution on the surface of the body and at other points in the field, the sound intensity, and the sound power. BEM may be formulated in either the time domain or the frequency domain [28.10].

In cases where the domain is exterior to the boundary, as in acoustic radiation, the extent of the domain may be infinite and hence the advantages of BEM are even more striking. [28.11] The boundary-element mesh, which consists of a series of points called *nodes* connected together to form triangular or quadrilateral elements, covers the entire surface of the radiating body. The magnitude and phase of the vibration velocity must be known at every point. BEM calculates the sound pressure distribution on the surface of the body from which the sound pressure and sound intensity at field points in the acoustic domain can be calculated. BEM will even calculate the sound pressure in the *shadow zone* behind a body due to radiation from the other side [28.10].

Another class of problems to which the BEM may be applied is the so-called interior problem, such as the prediction of the sound field inside an enclosure. The procedure is similar to the solution of exterior problems.

There are two different formulations for BEM: *direct* and *indirect* methods. The primary variables in the direct BEM are the acoustic pressure and acoustic velocity, while the indirect method uses the difference in acoustic pressure and the difference in the normal gradient of acoustic pressure across the boundary [28.12]. There is no distinction between an interior and an exterior problem when using indirect BEM, since it considers both sides of the boundary simultaneously.

Yet another class of problems is that in which an interior acoustic space is connected to an exterior space through one or more openings. The radiation of sound from a source within a partial enclosure is one example of such a problem. In one approach, the integral equations for the interior and exterior domains are coupled using continuity conditions at the interface surface between the two domains. The integral equations are then reduced to numerical form using second-order boundary elements [28.13].

To apply BEM to the Helmholtz equation, it must essentially be reduced to a two-dimensional integral

equation by use of Green's theorem. This means that only a description of the radiating surface is necessary rather than a complete discretization of the surrounding medium. The technique is suited to both interior- and exterior-domain problems.

Some methods couple a finite element of the structure with a boundary-element model of the surrounding fluid. The surface fluid pressures and normal velocities are first calculated by coupling the finite-element model of the structure with a discretized form of the Helmholtz surface integral equation for the exterior fluid [28.14].

Both BEM and FEM can be computationally intensive if the model has a large number of nodes. The solution time is roughly proportional to the number of nodes squared for an FEM analysis and to the number of nodes cubed for a BEM analysis. The accuracy of both methods depends on having a sufficient number of nodes in the model, so engineers try to straddle the line between having a mesh that is accurate yet can be solved in a reasonable time. The general rule of thumb is that six linear or three parabolic elements are needed per wavelength [28.10].

28.3.3 Finite-Element Correlation

It is often desirable to validate the results of a finite-element analysis with measured data from a modal test. This correlation is generally an iterative process incorporating two major steps. First the modal frequencies and mode shapes are compared and the differences quantified. Then, adjustments and modifications are made to the finite-element model to achieve more-comparable results.

It is useful to graphically compare the sets of frequencies by plotting predicted versus measured frequencies. This shows the global trends and suggests possible causes of these differences. If the random scatter is too great, then the finite-element model may not be an accurate representation of the structure. If the points lie on a straight line, but with a slope other than 1, then the problem may be a mass loading problem in the modal test or an incorrect material property such as elastic modulus or material density in the finite-element model.

Numerical techniques have been developed to perform statistical comparisons between mode shapes. The modal assurance criterion (MAC) is a correlation coefficient between the measured and calculated mode shapes. Another technique, called direct system parameter identification, is the derivation of a physical

model of a structure from measured force and response data [28.5]. Various forms of cross-orthogonality tests have also been employed. In general these methods see how well the experimental shapes can uncouple the

mathematical model. As with many comparison methods, difficulty is often encountered in reducing the FEA degrees of freedom to a set of DOFs consistent with the experiment.

28.4 Sound-Field Analysis

Although modal analysis developed primarily as a way of analyzing mechanical vibrations, by substituting microphones or acoustic intensity probes for accelerometers, experimental modal testing techniques can be used to explore sound fields. Modal testing has been used to explore standing acoustic waves inside air columns and to explore radiated sound fields from a vibrating structure. It could also be used to explore acoustical modes in rooms.

Figure 28.4 shows an arrangement for measuring the sound field inside a flute. The exciter is a loudspeaker coupled to the flute by a capillary tube. A pressure microphone probes the sound field at successive points, and the input and output signals are fed to a 2-channel FFT analyzer in the usual manner. The sound pressure amplitudes and phases for four modes in the flute (D5 fingering) are shown in Fig. 28.5 [28.15].

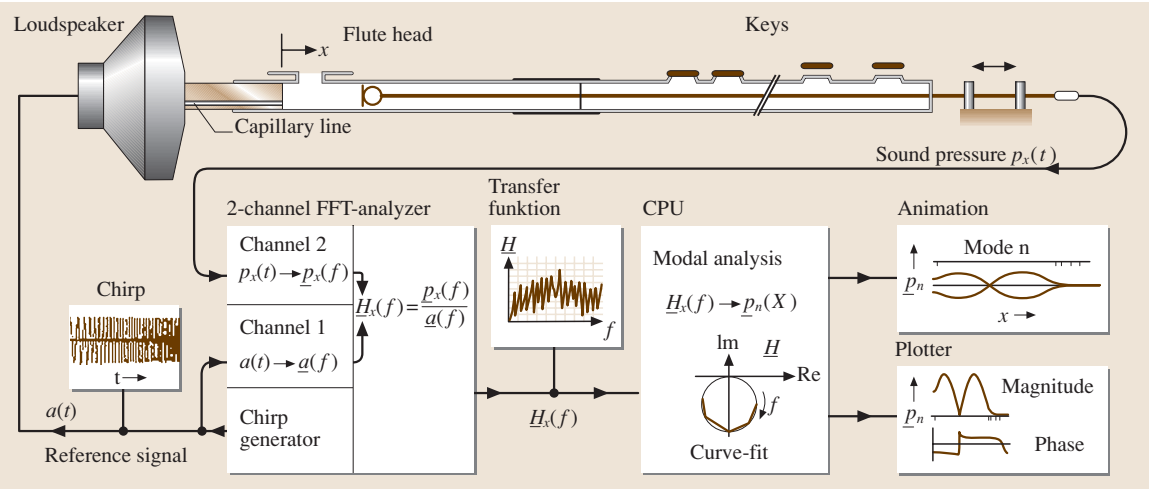


Fig. 28.4 Experimental arrangement for measuring the sound field inside a flute using modal analysis. (After[28.15])

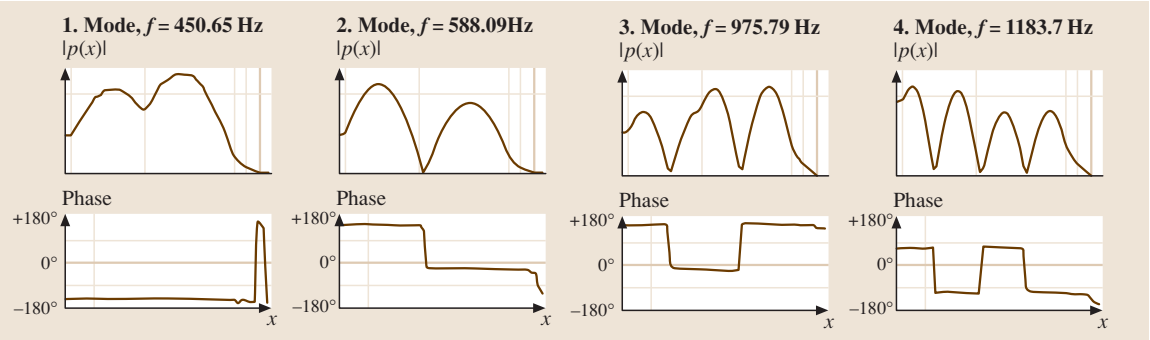


Fig. 28.5 The sound pressure amplitudes and phases for four modes in the flute (D5 fingering) (After [28.15])

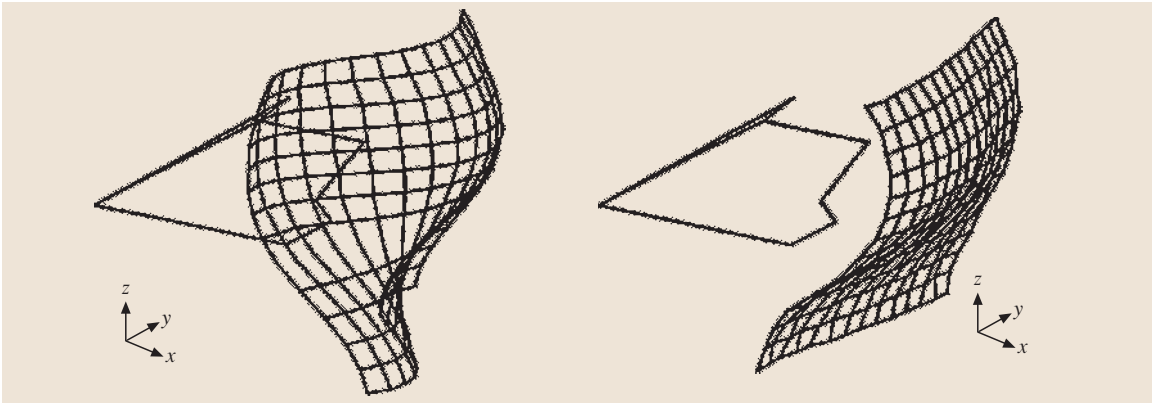


Fig. 28.6 Sound pressure in the radiated sound field of a piano at 162 Hz (*left*) and 107 Hz (*right*). Note the 180° phase difference above and below the plane of the soundboard (After [28.15])

The sound field radiated by a piano soundboard has been similarly explored by driving the soundboard with a shaker and probing the radiated field at 200 points with a pressure microphone. Figure 28.6 shows

the sound pressure in the radiated sound field at two frequencies. The sound field shows a 180° phase difference above and below the plane of the soundboard as expected [28.15].

28.5 Holographic Modal Analysis

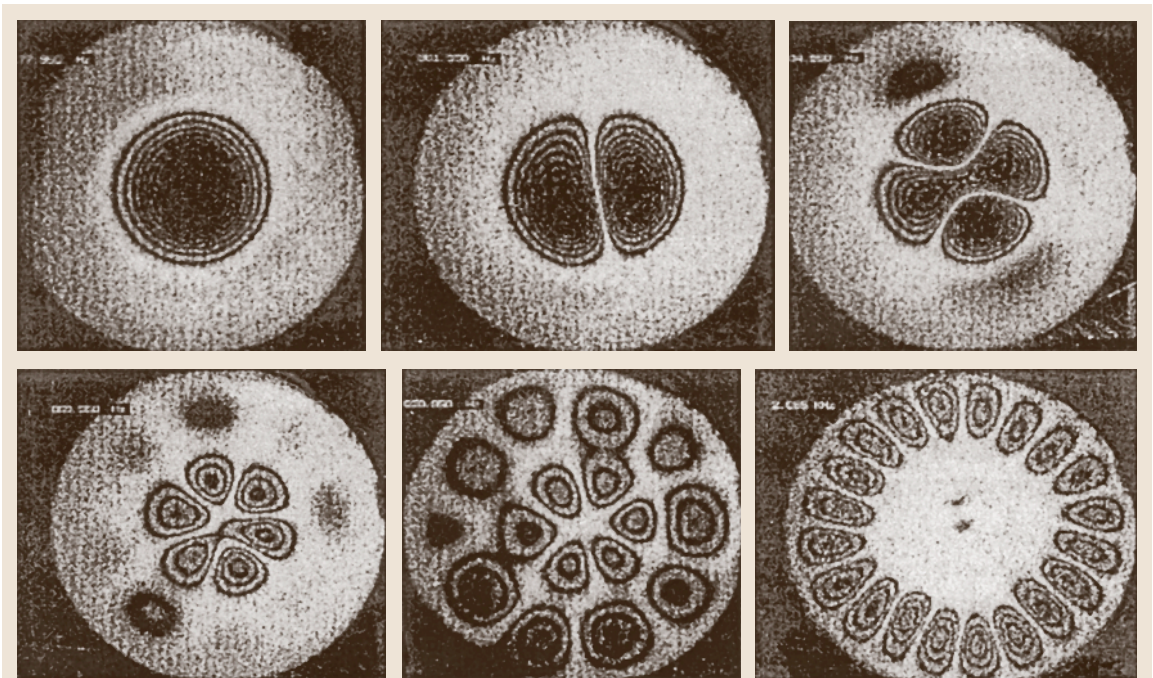


Fig. 28.7 Holographic interferograms of several vibrational modes in a Chinese gong. Note that some modes are mainly in the center of the gong, some in the outer portion, and some are global. (After [28.16])

Holographic interferometry offers the best spatial resolution of operating deflection shapes. In cases where the damping is small and the modes are well separated in frequency, the operating deflection shapes correspond closely to the normal mode shapes. Modal damping can be estimated with a fair degree of accuracy from half-power points determined by counting fringes [28.17]. Phase modulation allows analysis to be done at exceedingly small amplitudes and also offers a means to

separate modes that overlap in frequency [28.18]. TV holography allows the observation of vibrational motion in real time, and it is a fast, convenient way to record deflection shapes. Holographic interferograms of several vibrational modes of a Chinese opera gong appear in Fig. 28.7.

An excellent discussion of holographic and other optical techniques for acoustic and vibration measurements appears in Chap. 27.

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B.8 Nonlinear Acoustics in Fluids

*by Werner Lauterborn, Thomas Kurz,
Iskander Akhatov*

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C.9 Acoustics in Halls for Speech and Music

by Anders Christian Gade

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D.13 Psychoacoustics

by Brian C. J. Moore

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E.15 Musical Acoustics

by Colin Gough

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F.21 Medical Acoustics

by Kirk W. Beach, Barbrina Dunmire

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G.23 Noise

by George C. Maling, Jr.

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H.24 Microphones and Their Calibration

by George S. K. Wong

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H.26 Acoustic Holography

by Yang-Hann Kim

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H.27 Optical Methods for Acoustics and Vibration Measurements

by Nils-Erik Molin

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Chapter C.10

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Chapter A.4

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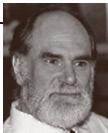


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Part B Physical and Nonlinear Acoustics

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