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**A Vibrating Razor Blade Machining Tool
for Material Removal on Low-Density Foams**

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October 1, 1990

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A Vibrating Razor Blade Machining Tool for Material Removal on Low-Density Foams

1. Introduction

The Lawrence Livermore National Laboratory (LLNL) has developed an accurate method of machining low-density foams into rectangular blank shapes by using a commercial oscillating razor blade machining tool concept marketed as a *Vibratome*. Since 1970, Vibratome has been used by medical laboratories to section fresh or fixed animal and plant tissues without freezing or embedding. By employing a vibrating razor blade principle, Vibratome sectioning avoids the alteration of morphology and the destruction of enzyme activities. The patented vibrating blade principle moves the sectioning razor blade in a reciprocating arcuate path as it penetrates the specimen. Sectioning takes place in a liquid bath using an ordinary injector-type razor blade.

Although other commercial products may accomplish the same task, the Vibratome concept is currently being used at LLNL to obtain improved foam surface qualities from razor machining by combining state-of-the-art air bearing hardware with precise linear motion and an electrodynamic exciter that generates sinusoidal excitation. Razor cut foam surfaces of less than 25 μm (0.001 in.) flatness are achieved over areas of 8.75 in.² (2.5 $\times$ 3.5 in.). Razor machining of wide or narrow foam surfaces is generally characterized by a continuous curl chip for the full length of the material removed. This continuous chip facilitates flatness and prevents increased surface densities caused by material chip collection often left in the surface cells by conventional machine tools.

This report covers the design evolution of razor machining of non-metallic soft materials. Hardware that maintains close dimensional tolerances and concurrently leaves the machined surface free of physical property changes is described.

Section 2 summarizes the history of the hardware design changes from the original razor machining concept through current operational hardware designs. Section 3 presents the LLNL goals for razor machining of foam materials to provide three minimum operating conditions of (1) a very stable machinery base to react tool cutting forces; (2) a depth of cut read-out system that is accurate to 0.0001 in.; and (3) a machine feedback control that gives optimum repeatability of machining parameters. Section 4 provides the technical history and current status of razor machining at LLNL. Air-bearing hardware is discussed in detail to provide information on state-of-the-art technology. Section 5 provides a brief insight into soft material machining and the change in physical conditions at the machined surface cells. Scanning electron microscope (SEM) photos show the material cell density changes at the machined surface by comparing the results of conventional tool machining to razor machining.

Appendix A presents the analytical model for cantilever beam natural frequencies and the relationships between dynamic displacements, velocities, and accelerations generated by simple harmonic motion. A vibration wedge for measuring dynamic displacement is also shown. Appendix B provides insight into advanced designs of razor machining, which are being used experimentally at LLNL.

Figure 1. Vibratome Lancer Model 1000.

2. Historical Summary

In 1983, LLNL purchased its first Vibratome Lancer Model 1000 (see Fig. 1). This unit is still completely intact, and its sectioning system was never used for program work. In 1984, a modified version of the Lancer Model 1000 was transferred to LLNL from Sandia National Laboratories, Livermore (SNLL). This second unit was supported from the floor on tall legs. The basic razor blade vibration sectioning system was held stationary, while a linear movable table carried the piece part to be cut under the razor blade. Because of wide tolerances in machined surfaces, this system was discarded and in-house hardware was designed, using the original Lancer arcuate blade holder concept.

In 1985, the first generation piece-part machining system developed by LLNL for non-metallic materials used the following concepts from Lancer:

- Two identical cantilever arcuate beams to support the razor blade holder.
- One electrical solenoid to excite the cantilever beams at 60 Hz.
- One razor blade holder for a 1 in. $\times$ 0.010 in.-thick injector razor blade.

In 1986, a second generation of piece-part razor machining was developed by LLNL. The electrical solenoid to excite the cantilever beams at 60 Hz was replaced by an electrical voice coil that could excite the beams at their fundamental natural frequency (not necessarily 60 Hz).

Since 1987, LLNL has developed the third generation of razor blade machining, which departs from the concepts of the systems above in these respects:

- Motion of the razor blade is linear, not arcuate; the blade is carried on an air bearing, instead of on cantilever beams,
- Oscillating motion of the razor blade and the air bearing is produced by an electrodynamic exciter in a true sine wave motion, as compared to pseudo-sinusoidal motion produced by the solenoid or the voice coil with cantilever beams.
- Being sinusoidal, the oscillatory motion is calculable for displacement, velocity, and acceleration at a given frequency.
- The frequency of operation ranges from 5 Hz to 14 kHz, rather than one fixed frequency set by the cantilever beams.
- Driving forces on the razor blade cutting system are controlled by the output of the electrodynamic exciter, rather than by the inertia forces of the cantilever beams.
- The linear amplitude (displacement, velocity, or acceleration) of the razor blade is controlled automatically (closed loop) by a feedback transducer signal from the air-bearing motion to an electronic servo chassis.

3. Goals for Razor Blade Machining Hardware

Improvements to the cantilever beam concept of blade machining hardware were implemented to obtain four major goals:

- (1) To obtain a very accurate linear movement of the razor blade across a displacement of 0.2 in. (± 0.1 in.) peak to peak from a zero blade center position using a stable machinery base.
- (2) To apply a true sine wave oscillatory force to the razor blade for easily calculated and measured relationships of frequency, acceleration, velocity, and displacement of the blade.
- (3) To provide electronic feedback loop control of the razor blade parameter selected (i.e., acceleration, velocity, or displacement) to hold that parameter constant during the repeated cutting process.
- (4) To provide the flexibility for positioning the razor cutting edge and for driving it accurately in any combination of angles in the x , y , and z planes, using position monitors accurate to 0.0001 in.

The first goal is a departure from the arcuate path of the cantilever beam-held razor to obtain a very accurate plane of motion for the razor and a resulting material

Figure 2. Numerical control milling machine (Bostomatic).

surface flatness that is free of machining chips (see Sec. 5). The second goal provides the analytical relationships for a true sine wave oscillation for accurate estimates of razor motion. These relationships, shown in Appendix A2, are derivable parameters of the basic equation of simple harmonic motion, i.e., $\text{Displacement} = r \sin \omega t$. The third goal provides a method to automatically control any one of the parameters in goal (2). Control of razor acceleration or displacement results primarily in protecting machine components, i.e., the electrodynamic exciter or exciter attachments to the air bearing. Control of razor velocity results in the optimum slicing (shear) speed for a given density of foam material. Goal (4) allows the oscillating razor to operate at any angle in space, while held rigidly by numerical control milling machines. Figure 2 shows razor machining, using a conventional milling machine with x , y , and z numerical controls for position and an auto screw table for material feed rate.

4. Technical Status

Essentially, two generations of oscillating razor blade design were used for foam machining prior to the air bearing concept. Both designs used the cantilever beam design concept from Vibratome to carry the razor blade (~1 in. length) in an arcuate path (see Fig. 3). The first generation hardware (see Fig. 4) created an oscillating force on the cantilever beam from an electric solenoid driven with a 110-V

Figure 3. Razor machining design using cantilever beams.

Figure 4. Solenoid razor machining.

electric source at a half-wave rectified frequency of 60 Hz. Gain to the solenoid was changed by a voltage rheostat to obtain a preferred displacement at the razor blade. Vibration amplitudes at the blade were set by sight with no mechanical or electrical

measurement of the true quantity. Electric solenoids at half-wave rectified voltage deliver a pseudo-sinusoidal force to the cantilever beams. The analytical relationships of known parameters for full sine waves (i.e., frequency, acceleration, velocity, and displacement) are therefore more difficult to determine. Since a single frequency of 60 Hz was available at the solenoid, design of the cantilever beams was critical such that the structural natural frequency (first mode resonance) of the cantilever beam assembly occurred at 60 Hz. The solenoid force at frequencies below and between higher beam resonant modes was insufficient to deflect the beams at amplitudes required for cutting.

Figure 5 shows the second generation hardware, which used an electric voice coil (radio speaker design) to apply an oscillatory force to the cantilever beams

Figure 5. Voice coil razor machining.

without voltage rectification. A variable oscillator chassis and an amplifier were added to the system to drive the voice coil. Design of the cantilever beam structure was no longer critical since the variable oscillator could drive the voice coil at the resonant frequency of the beams. This was accomplished by searching the frequency spectrum for (1) maximum displacement of the beams and (2) the lowest amplifier current (amperage) output to the voice coil. The first mode resonance of the cantilever beams, analytically calculated in Appendix A1, was sought to prevent higher mode structural distortion from occurring at the two beams. By using a true sine wave oscillatory force from the voice coil, the relationships of frequency, acceleration, velocity, and displacement could now be analytically estimated for the razor blade. These analytical relationships (see Appendix A2) represent only estimates of razor blade motion. The presence of non-linear spring elements in the cantilever beam clamping creates non-sinusoidal motion.

Modifications to the second generation of oscillating blade machining included a 4-in. razor blade holder between the cantilever beams, a determination of blade angle with the work piece using an inclinometer, and a visual indicator (see the vibration wedge shown in Appendix A3) of blade displacement that was accurate to ± 0.005 in.

Inaccuracies of foam surface dimensions for flatness created by the cantilever beam arcuate path are shown diagrammatically in Fig. 6. Note the front view torsion

Figure 6. Diagram of cantilever beam distortion from dynamic imbalances and non-linearities.

motion of the cantilever beams at the extreme path deflection. Because of dynamic torsion imbalance at the razor blade attachment mass, rotation of the attachment occurs from the lack of sufficient torsional rigidity in the attachment rod and the right-hand cantilever beam. This raises the razor blade end and leaves a concave work piece surface. Features of the tie down of the cantilever beam end make the spring constants different for plus and minus motion from the static center position. This non-linear motion creates non-sinusoidal motion at the razor blade and results in an inexact analytical description for control of the cutting process.

Linear motion hardware of air bearing design will operate the razor blade in a dry atmosphere. At LLNL, a dry atmosphere (i.e., 10% humidity, free of petroleum lubricants) is essential for razor machining specific foam materials. Aerostatic linear slides, as shown in Figs. 7 and 8, provide static linear accuracy (± 0.000004 in./in.) and sufficient stiffness to resist razor cutting loads in the x, y, z axes, including roll, pitch, and yaw. At nominal air pressures of 50 psi, the lateral runout of the blade through bearing travel of 0.020 in. peak to peak at working frequencies is shown in Fig. 9.

Figure 7. Typical aerostatic linear slide dimensions (in inches).

Stiffness (z axis)	500,000 lb./in.
Max. load (z axis)	200 lb.
Stiffness (y axis)	250,000 lb./in.
Max. load (y axis)	100 lb.
Stiffness roll	0.78 in.-lb./arc sec.
Max. load roll	45 in.-lb.
Stiffness pitch	3.3 in.-lb./arc sec.
Max. load pitch	132 in.-lb.
Stiffness yaw	1.8 in.-lb./arc sec.
Max. load yaw	66 in.-lb.
Max. supply pressure	100 psig
Air flow	0.50 SCFM

Figure 8. Typical aerostatic linear slide mechanical stiffness and load capacity.

Figure 9. Measured razor blade lateral runout for a displacement of 0.02 in. peak to peak (P-P) with curve fit (50-psi air bearing pressure).

By using state-of-the-art air bearing hardware, the dimensional accuracy of the machined foam material approaches some metal machining tolerances of ± 0.0005 in. The evolution of the cantilever beam razor machine discussed provides entry into advanced design changes, including air bearing hardware, electrodynamic exciters, and electronic control feedback circuits.

The oscillatory force generator for the air bearing with a mounted razor blade is a commercial electrodynamic exciter shown in Figs. 10 and 11. The exciter is

Figure 10. Electrodynamic exciter with razor machining air bearing assembly.

Figure 11. Internal parts of electrodynamic exciter.

essentially a radio speaker without the cone that duplicates wave form from a signal generator—in this case, the commercial variable oscillator shown in Fig. 12.

Figure 13 shows the exciter commercial control hardware in a block diagram. The automatic gain control function of the electronic chassis in the dotted box is to provide the same continuous vibration amplitude at the razor blade regardless of the cutting load. The sine wave signal generated at the oscillator passes through the compressor circuit that changes the signal voltage to the power amplifier. The voltage change depends on the feedback from the accelerometer signal, which is a true measure of the exciter amplitude. The accelerometer signal is changed from an ac voltage to a dc voltage to provide the compressor circuit with a comparison of the exciter amplitude to a preset amplitude in the compressor circuit. For protection of the exciter against over acceleration, circuits in the vibration monitor chassis are preset to shut the sine wave signal off to the amplifier in the event the limits are exceeded.

The frequency performance curve limits the exciter to a maximum displacement at low frequencies, to a maximum velocity at moderate frequencies, and to a

Figure 12. Vibration exciter control chassis.

Figure 13. Block diagram of vibration exciter control system.

Figure 14. Frequency response curve for an electrodynamic exciter.

maximum acceleration at high frequencies. The solid line in Fig. 14 shows a frequency response curve for an exciter with no added weight on the voice coil. This exciter, at low frequencies (10 to 95 Hz), is limited to 0.20-in. peak-to-peak displacement. From 95 to 150 Hz, the exciter velocity is limited to ± 60 in./sec and from 150 to 14 kHz to ± 102 g.

By using the equation $\text{force} = \text{mass} \times \text{acceleration}$, we find that if the moving weight of the air bearing plus the weight of the moving element of the exciter is, say, 1 lb, then the maximum acceleration that a 4.5 lb force exciter can deliver is

$$\text{Acceleration } g = 4.5/1 = 4.5 \text{ g.}$$

Going to the frequency response curve of Fig. 14, at an ordinate value of 4.5 g and a maximum displacement of 0.20 in. peak to peak, the frequency will be 21 Hz and the velocity calculation will result in 13.18 in./sec. Since the system is limited to a maximum acceleration of 4.5 g and a maximum displacement of 0.20 in., the maximum velocity possible is 13.18 in./sec. Changes may be made to lower the acceleration and/or raise or lower the frequency; however, razor cutting velocities will be lower than 13.18 in./sec, according to analytical relationships for sine wave motion shown in Appendix A2.

5. Operational Status

Polystyrene, a low-density foam material, is being produced at a density of 120 mg/cm^3 (0.0043 lb/in.^3). This material is an isotactic foam made by a supercritical extraction and drying process from an isotactic polystyrene gel. The form and structure of the gel is open and strut-like. The $0.5\text{-}\mu\text{m}$ -diameter struts form cells 1.0 to $5.0 \mu\text{m}$ across. The mechanical properties exhibit an apparent Young's modulus of 2000–2200 psi in compression with a compressive strength of 40–50 psi.

The machining processes being used for this type of material include high-speed fly cutting, rotary saw cutting, hot wire cutting, and razor cutting. The polystyrene material exhibits physical qualities that make the conventional methods listed above (except for razor cutting) difficult to use. Collapsing of the material cell walls rather than shearing occurs leaving residue on the cutting tools. Compare the razor cut surface shown in Fig. 15 (a) to a rotary saw cut surface shown in Fig. 15 (b). Razor machining of polystyrene creates a continuous curl chip during removal of material with a surface flatness of less than 0.001 in. (25 μm) and nominally 0.0005 in. (13 μm). Polystyrene material surface densification does not occur from razor cuts as shown in Fig. 15 (c) due to the continuous chip generation mentioned above. Figure 15 (d) shows the surface densification caused by saw cutting when cut chips remain in the material cells.

One application of razor machining at LLNL requires dimensional tolerances of 0.008-in. thickness and 0.008-in. parallelism. Dimensional measurement of these materials by conventional means (i.e., micrometers, calipers, and dial indicators) does

not provide the accuracy listed above due to surface compliance. Other non-contact optical devices, such as optical comparators and x-y calibrated microscopes, do provide the accuracy. The razor cut foam materials at LLNL have been dimensionally verified to tolerances within 0.0005-in. thickness and 0.001-in. parallelism. Also, tapered surfaces have been cut to angles of 0.006-in/in., using numerical control milling machines.

Figure 15. Scanning electron microscope (SEM) views of (a) a razor cut surface; (b) a saw cut surface; (c) razor cut material cells with no densification; and (d) saw cut material cells with densification.

Conventional, single-edge razor blades of commercial variety, such as shaving, food processing and packaging, have been used without modification. Research and development of razor cutting edges has not been employed to produce the end products in foam material; however, other materials may require specific razor cutting edges to provide given dimensional tolerances.

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Appendix A

A1. Analytical Model for Vibratome Cantilever Beam Hardware

A2. Derivation of Sine Wave Relationships of Frequency, Displacement, Velocity, and Acceleration

A3. Vibration Wedge

Appendix A

A1. Analytical Model for Vibratome Cantilever Beam Hardware

The basic natural frequency for a simple spring mass system is as follows:

$$F_n = \frac{1}{2\pi} \sqrt{\frac{K}{M}} \quad , \quad (1)$$

where

F_n = natural frequency (Hz)

K = spring constant (lb./in.)

M = weight/gravity (lb./386 in./sec²).

For a cantilever beam,

$$K = \frac{3EI}{\ell^3} \quad , \quad (2)$$

where

E = modulus of elasticity (lb./in.²)

I = moment of inertia (in.⁴)

ℓ = length of beam (in.).

Substituting (2) into (1),

$$F_n = \frac{1}{2\pi} \sqrt{\frac{3EI g}{\ell^3 W}} \quad . \quad (3)$$

The weight, W , consists of two elements: first, the weight of the razor holder assembly with the crossrod between the cantilever beams; second, the weight of the beams, W_b . The effective weight of a uniform cross section cantilever beam is $0.257 W_b$ by analytical derivation. The final equation of natural frequency for the cantilever beam assembly for Vibratome would be by substitution into (3):

$$F_n = \frac{1}{2\pi} \sqrt{\frac{3EI g}{\ell^3 (W + 0.257 W_b)}} \quad . \quad (4)$$

A2. Derivation of Sine Wave Relationships of Frequency, Displacement, Velocity, and Acceleration

Figure A-1 shows simple harmonic motion.

Figure A-1. Simple harmonic motion.

$$\sin \omega t = \frac{d}{r} \text{ where } d = \text{displacement (0 - peak)}$$

$$\text{Displacement, } d = r \sin \omega t$$

$$\text{Velocity} = \frac{dx}{dt} = r\omega \cos \omega t$$

$$\text{Acceleration} = \frac{dV}{dt} = r\omega^2 \sin \omega t$$

$$\omega = 2\pi f \text{ where } f \text{ is frequency in Hz.}$$

$$(1) \text{ Displacement, } d = r \sin 2\pi ft \left(\text{when } 2\pi f = \frac{\pi}{2} \right) = r$$

$$(2) \text{ Velocity, } V = r 2\pi f \cos 2\pi ft \left(\text{when } 2\pi f = 0 \right) = r 2\pi f$$

$$(3) \text{ Acceleration, } A = r (2\pi f)^2 \sin 2\pi ft \left(\text{when } 2\pi f = \frac{\pi}{2} \right) = 39.478 r f^2$$

Substitute (1) into (2)

$$V = d 2\pi f \text{ Displacement P - P (peak to peak) } = 2d = D$$

$$V = \pi f D \text{ in./sec}$$

Substitute (1) into (3):

$$A = 39.478 \, df^2 \, \text{in/sec}^2$$

$$d = \frac{D}{2}$$

$$A = 19.74 \, f^2 D \quad \text{gravity, } g = \frac{A}{386} = g$$

$$g = \frac{19.74 \, f^2 D}{386}$$

$$g = 0.0511 \, f^2 D$$

$$\text{Substitute } D = \frac{g}{0.0511 \, f^2} \text{ into } V = \pi f D$$

$$V = \frac{\pi f g}{0.0511 \, f^2}$$

$$V = 61.43 \, \frac{g}{f} \, \text{in/sec}$$

$$\text{Substitute } D = \frac{V}{\pi f} \text{ into } g = 0.0511 \, f^2 D$$

$$g = \frac{0.0511 \, f^2 V}{\pi f}$$

$$g = 0.0163 \, V f$$

A3. Vibration Wedge

The original Vibratome razor-cutting concept using cantilever beams did not provide instrumentation to indicate the amplitude of vibration (i.e., displacement, velocity, or acceleration). Figure A-2 shows a means of determining peak-to-peak displacement only; however, if the motion is true sinusoidal, the relationships between frequency, displacement, velocity, and acceleration will apply. Errors in these relationships occur with cantilever beam arcuate motion.

Figure A-2. Vibration wedge (absolute inches peak to peak).

Construction of the wedge is simply two divergent lines at an actual peak-to-peak distance from each other. The vertical lines denote the point at which 0.04, 0.06, 0.1 in., etc. between lines occurs. During vibration, for example, the pattern shown in Fig. A-3 develops, telling the viewer the peak-to-peak displacement at the apex of the left- and right-hand inner triangles.

Figure A-3. Vibration wedge pattern (not to scale, exaggerated for clarity).

The wedge is placed on the oscillating part with adhesive, such as double-backed tape, and aligned with the vertical lines in the direction of oscillating motion as closely as possible. The vibrating wedge is a naked-eye measurement of peak-to-peak displacement. When the wedge is subjected to sinusoidal vibration, the velocity is zero while the displacement amplitude is at its positive and negative maxima. This causes the wedge to dwell at the displacement maxima.

Appendix B

Advanced Designs

Figure B-1. Advanced vertical contour machining.

Appendix B

Advanced Designs

Original air bearing systems for razor machining of foam materials are designed to remove material in the vertical, z , axis only. The conventional screw v-way systems combined with linear variable displacement transducers, calibrated for read outs of 0.0001-in. increments, are used to determine the depth of razor cuts. Similar razor machining assemblies have been fabricated and operated on conventional milling machines using numerical controls for the x , y , and z axis control, as discussed in the text. Advanced designs of razor machining have been developed for use with x , y , and z axis control combined with numerical control of rotation on rotary chucks.

Figure B-1 shows a razor-machining system capable of cutting programmed contours. This hardware is capable of machining with contour accuracies of 60 $\mu\text{in.}$

Figure B-2 shows a similar razor-machining system to the vertical system presented in Fig. B-1. This hardware is designed for programmed contours with contour accuracies of 10 $\mu\text{in.}$ This system also is designed to operate rotationally as well as in x , y , and z motion.

Figure B-2. Advanced horizontal contour machining.

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