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Design of a Hydraulic Bending Machine

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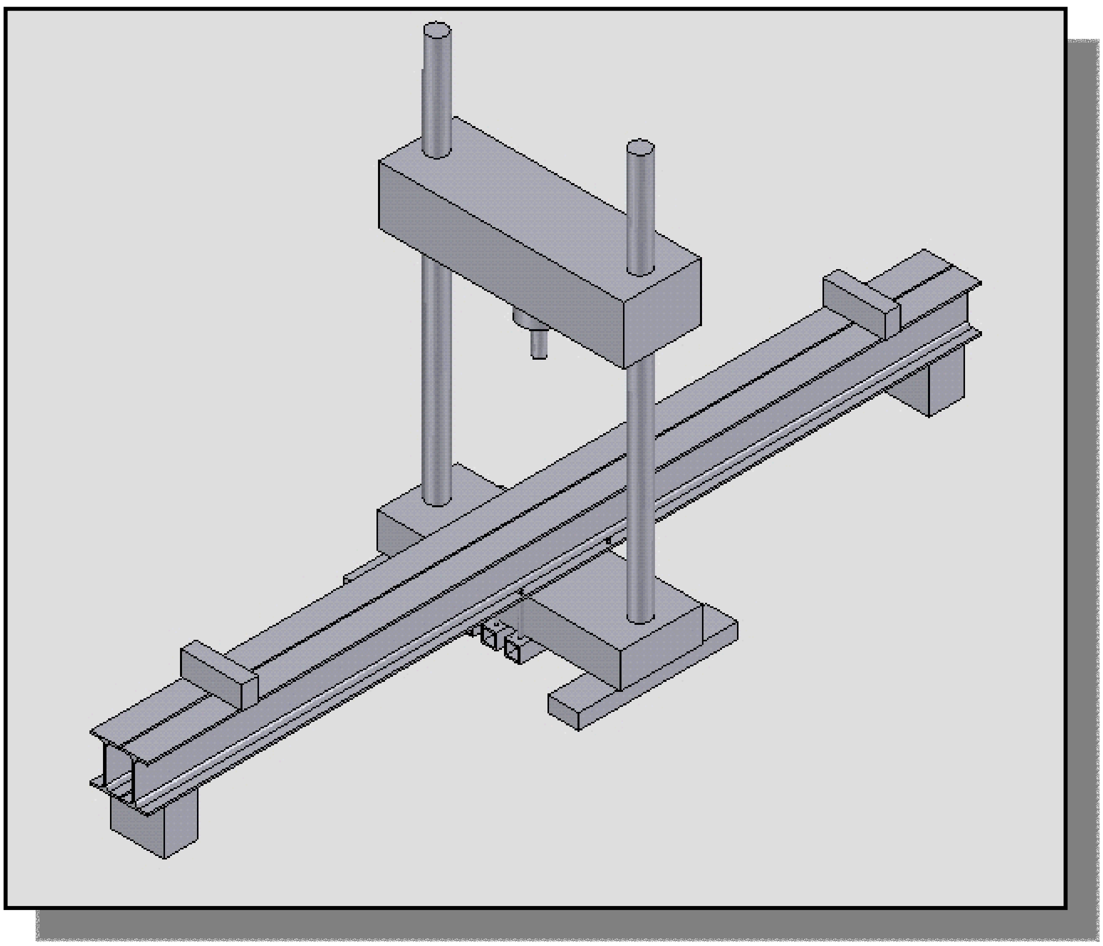

Abstract

To keep pace with customer demands while phasing out old and unserviceable test equipment, the staff of the Engineering Mechanics Laboratory (EML) at the USDA Forest Service, Forest Products Laboratory, designed and assembled a hydraulic bending test machine. The EML built this machine to test dimension lumber, nominal 2 in. thick and up to 12 in. deep, at spans up to 20 ft and loads up to 20,000 lbf. The hydraulic bending test machine was built using parts of a 100,000-lbf compression test frame. Added components included W12 by 65 steel beams; steel tube sections, L-sections, and threaded rods for beam attachment; I-beam spacer plates; wood block beam end supports; a 4-in. bore, 10-in. stroke hydraulic cylinder with 38,000 lbf capacity; steel plates for cylinder reinforcement; and two pivoting four-point load head assemblies. Eccentric loads that might occur during a test will not yield the positioning screws of the machine head or otherwise affect test results.

Keywords: hydraulic bending machine, dimension lumber, wood testing machine

SI conversion factors

English unit	Conversion factor	SI unit
inch (in.)	25.4	millimeters (mm)
in^4	4.162×10^{-7}	m^4
foot (ft)	0.3048	meter (m)
pound, mass (lb)	0.454	kilogram (kg)
pound, force (lbf)	4.45	newton (N)
pound force/in^2 (lbf/in^2)	6.894	kilopascal (kPa)
in–lb	0.113	joule (J)
gallon	3.785	liter (L)

Dimensional lumber equivalents

nominal 2 by 4 in.	standard 38 by 89 mm
nominal 2 by 12 in.	standard 38 by 286 mm

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Design of a Hydraulic Bending Machine

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Introduction

To keep pace with customer demands while phasing out old and unserviceable test equipment, the staff of the Engineering Mechanics Laboratory (EML) at the USDA Forest Service, Forest Products Laboratory, designed and assembled a hydraulic bending test machine (Fig. 1). The machine is used to support long-span beams and to apply a hydraulically driven bending force to the beams in accordance with standard testing procedures. This paper describes the parameters and considerations for designing this machine to enable the EML staff to complete needed tests in a manner that is user-friendly and safe.

The mission of the EML is to test and evaluate wood, wood products, and wood component specimens to determine their mechanical and material properties. Tests are performed for work units within the Forest Products Laboratory and for work units in off-site Forest Service Experimental Stations by special request.

The EML performs a significant portion of its testing on dimension lumber, including static bending tests that conform to the ASTM D198 standard (ASTM 1997).

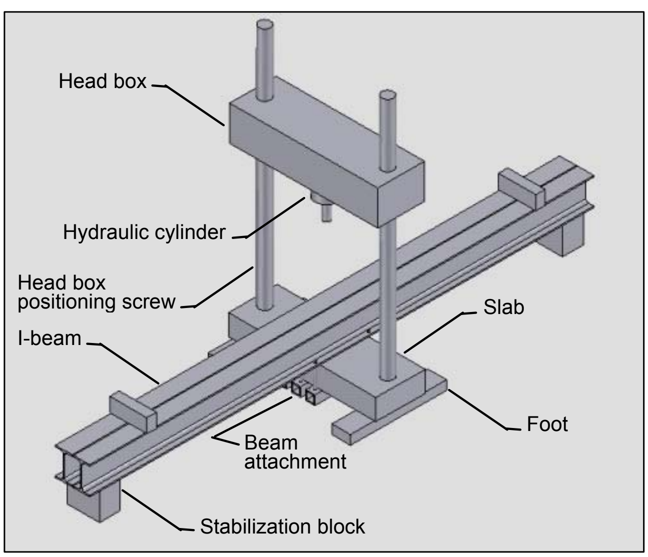

Figure 1—Hydraulic bending machine.

The primary purpose of the bending machine is to test long-span dimension lumber (nominal 2- by 4-in. to 2- by 12-in., up to 20 ft in length) at expected loads less than 10,000 lb and deflections less than 6 in. Two machines that have been used by the EML for bending tests are the 160,000-lbf Reihle machine (purchased in 1969) and the 25,000-lbf box-test machine (purchased in 1937). These machines are of questionable reliability and repairability because of their age. A machine that is capable of testing long-span dimension lumber is not currently in production and would have to be custom-made at great expense. Therefore, we chose to modify a 100,000-lbf compression test frame with a motor-driven movable head box, which was donated to EML.

Many parts of the test frame were removed, leaving the steel feet, slab, head box positioning screws, and head box. Parts that were added to complete the bending test machine included two wide-flange beams, beam attachment hardware, beam spacer plates, beam end supports, a hydraulic cylinder, cylinder reinforcement hardware, and two four-point load head assemblies. Design considerations for these components are presented in this paper.

Although the 100,000-lbf compression force capacity of the original frame design is much greater than the loads we encounter during bending tests, we discuss the effects of eccentric loads that are more likely to occur in bending tests than in the compression tests for which the frame was designed.

I-Beam

Selection

The following deflection equation for simple support and center load was used:

$$\delta = \frac{PL^3}{48EI} \qquad \text{for } L = 240 \text{ in., } E = 2.9 \times 10^7 \text{ lbf/in}^2$$

where

δ is deflection,
P force,
L length,
E modulus of elasticity, and
I moment of inertia.

The force required to break a strong Southern Pine 16-ft-long 2 by 12 board with a modulus of rupture of 14,500 lbf/in^2 (Forest Products Laboratory 1999) is less than 10,000 lbf. For this calculation, we assumed service load (maximum expected) to be 20,000 lbf, or 10,000 lbf per steel beam. Ultimate load (based on known capacity of test frame) is 50,000 lbf per beam.

For the I-beam to have a 0.20-in. maximum service deflection and a 1-in. maximum ultimate deflection, the minimum required moment of inertia, I, is 993 in^4. Since it was a priority to keep the beam depth small to facilitate loading of specimens, we selected two W12 by 65 sections. The combined I is 1,066 in^4 (AISC 1995).

Because the beam webs are connected by six spacer plates (described later in this paper), movement associated with lateral–torsional buckling will be prevented. However, the W12 by 65 section is one of the few "non-compact" sections that is susceptible to local flange and web buckling. Beam strength was calculated using the procedure for calculating capacity of a non-compact section (AISC 1995).

The actual yield stress of the beam material was not specified. A higher grade of steel (50,000 lbf/in^2) would have greater flexural strength than does a lower grade (36,000 lbf/in^2), but it would be susceptible to local buckling. For this analysis, the plastic moment of a 36,000-lbf/in^2 W12 by 65 beam was calculated and compared to the local buckling moments of a 50,000-lbf/in^2 W12 by 65 beam. The minimum of these limit states is considered to be the strength of the beam.

The design plastic bending moment (ΦM_p) of the 36,000-lbf/in^2 beam is 261,000 in-lbf. Since the shape of this material is not susceptible to local buckling, this is also $\Phi M_n'$, the flexural design strength.

The design plastic bending moment of the 50,000-lbf/in^2 beam, ΦM_p, is 363,000 in-lbf. However, the shape of this material is susceptible to local flange buckling, which reduces this value to 358,000 in-lbf. The shape is not susceptible to local web buckling, so the factored flexural design strength of the 50,000-lbf/in^2 beam is 358,000 in-lbf.

The adjusted maximum unbraced length, LP', of a 50,000-lbf/in^2 beam is 11.8 ft and that for a 36,000-lbf/in^2 beam is 12.6 ft. Both of these are greater than the distance between the edge of the frame slab to the reaction head, so the flexural design strength of both beams, $\Phi M_n'$, is equal to their nominal factored flexural strengths, ΦM_n. Thus, the factored nominal flexural strength of a W12 by 65 beam of unspecified material (minimum 36,000-lbf/in^2 steel) is 261,000 in-lbf. Since the maximum load of the original frame (100,000 lbf) would put only 250,000 ft-lbf on the beams, the beams will be adequate for any test designed to the limits of the other frame components.

Figure 2—I-Beam attachment.

Attachment

The beams are not directly attached to the frame. Rather, they are strapped down with six 5/8-in. threaded rods that are connected to three steel 3- by 3- by 1/4-in. tube sections that pass under the frame slab (Fig. 2). The rods pass through two 3- by 2- by 3/8-in. steel L-sections that keep the beams from rotating with respect to the frame slab.

Any upward load on these components would come from a release of load from a specimen break. Then, the energy stored in the steel beams might cause the beams to spring upwards and be lifted from the slab. We calculated the significance of this effect.

Assuming that a specimen broke under a design load of 20,000 lbf, the beam ends would deflect about 0.19 in., as described previously. Given material and sectional properties, and assuming the beam behaves as an upside-down simply supported long-span beam, it would possess a potential energy of 1,900 in-lbf. Assuming an instantaneous specimen failure, this internal energy would be converted to some combination of kinetic energy of the vibrating beam, internal spring energy with upward deflection of the ends, internal friction loss (hysteresis), gravity potential energy of the lifted beams, and spring energy of the six stretched threaded rods. It is conservative to assume that all the beam energy present before the break would be entirely converted to the gravity potential energy of the beam and the spring energy of the stretched threaded rods. Both of these energy terms are dependent on the height achieved by the beams. An iterative calculation shows that the beams would lift less than 3/1,000 in.

Spacer Plates

Six 2-in.-thick plates are used to separate and stabilize the two W12 by 65 steel beams (Fig. 3). The plates are placed 1 ft from each end, at the coordinate of the slab edge of the frame, and halfway between these two, at an approximate

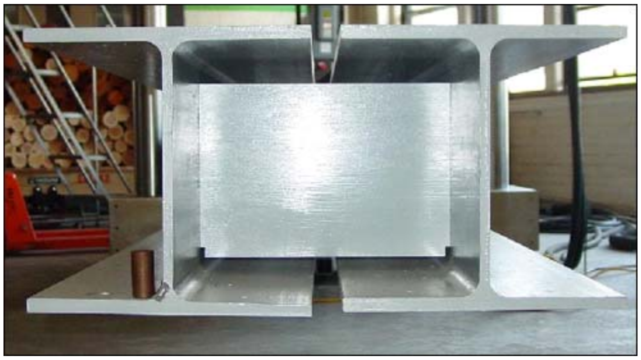

Figure 3—I-Beam spacer plates.

distance of 4-1/2 ft. Each plate is bolted to the beam web by four 3/4-in. grade 5 bolts. The plates are wide enough to maintain a 1-1/4-in. gap between the beams, so that fixtures and reaction heads may be affixed with T-slot hardware.

Any load on the spacer plates would be due to slight eccentricities in the test. Assuming a 10° load eccentricity on a 100,000-lbf vertical load, there would be 17,000-lbf horizontal force. Assume that this force is also a vertical force difference between the beams. Even if this entire force is supported by the four 3/4-in. bolts located closest to the slab, the bearing stress in the beam web would be less than 40% the yield stress of 36,000-lbf/in^2 mild steel, and the combined bearing and tensile stress in the bolts would be less than 40% the yield stress of grade 5 steel.

Stabilization Blocks

Wood blocks cut from glulam beams are fastened underneath the ends of the steel beams. The purpose of the wood blocks is to stabilize the machine and to minimize rocking or tipping when objects are placed on the beam ends. This is not expected to happen during a test. The wood blocks are not designed to take any test load because the large test loads will be contained within the machine structure and not transferred to the floor. The wood blocks also add a degree of safety if someone accidentally walks into the beam end. For example, an individual will likely hit a shoe against the wood block rather than a knee against the steel edge.

The blocks are made of Douglas-fir with grain parallel to the floor. The blocks were cut so as to leave a 1/4-in. gap between the block and the floor. Thus, the beam end will be allowed to deflect its design value of 0.20 in. during a test without the block touching the floor.

Screw Bending Consideration

If the load should develop an eccentricity in the plane of the bending specimen, it would place a bending moment on the 5-in. screws of the machine frame. We examined this situation to see if it posed a danger of damaging the screws or otherwise affecting test results. Assuming a load of 100,000 lb and an eccentricity of 10° degrees from vertical, there will be a horizontal force on the load head of 8,700 lbf, or about 4,350 lbf per screw. If the load head is 30 in. from the base of the frame slab (based on a steel beam height of 12 in., reaction support height of 6 in., and specimen depth of 12 in.), there would be a moment of 260,000 in-lbf, or 130,000 in-lbf per screw. This would cause a stress at the base of the screw of about 21 ksi and a horizontal deflection of 0.09 in. This is unlikely to affect the test results or damage the machine.

Hydraulic Cylinder

We chose a hydraulic cylinder instead of a screw-drive loading mechanism. A hydraulic machine allows for constant load control and makes fast corrections after minor failures and load redistributions during a test, compared to the response of a screw drive. A hydraulic test machine also has greater flexibility for cyclic and vibrational testing than does a screw-driven machine.

The cylinder will be used with an existing MTS model 510.10 hydraulic power supply (MTS Systems Corporation, Eden Prairie, Minnesota). This power supply uses a fixed-volume pump to provide a fluid flow of 10 gal/min at 3,000 lbf/in^2. The pump motor requires a three-phase 460-V, 60-Hz electrical power source at 34 continuous amps. Fluid will be regulated through a model A076 Moog servovalve (Moog Inc., East Aurora, New York). This is a two-stage flow control servovalve with a mechanical feedback pilot stage and a rated flow of 1 to 17 gal/min (at 1,000-lbf/in^2 pressure drop). The operating pressure is 3,000 lbf/in^2, and the step response at this pressure is 3 to 16 ns for 100% stroke. Load will be measured by a Sensotec model UG/4671-03 load cell (Sensotec, Inc., Columbus, Ohio) with a capacity of 30,000 lbf.

Cylinder Specifications

We selected the hydraulic cylinder from the Miller Fluid Power catalogue (Miller Fluid Power 2003). The specifications for the cylinder and associated parts are as follows.

Cylinder

- Centerline mounting
- 4-in. bore—At 3,000 lbf/in^2, a 4-in. bore will provide a force of 37,698 lbf ($\pi \times d^2 \times$ pressure $\div$ 4) (according to bore size estimation table).
- 0.0544 gal/in. oil consumption—If we assume a 6-in. deflection over 5 min, oil consumption in gallons/min is 0.065. The capacity of the pump we have is sufficient.

Stop tube recommendation

- Model 68 is the last entry in Group B
- For an unguided piston rod, $L = 4D$, for D = stroke if $D = 10$, $L = 40$
- Maximum length $L = 40$ for no recommended stop tube—If $40 < L < 50$, a 1-in. stop tube is recommended. No stop tube was ordered.

According to the oversize piston table (Miller Fluid Power 2003), the recommended diameter for $L = 40$ and load = 40,000 lbf is 2.0 in. We chose a model with a 4-in. bore/2.5-in. rod, which corresponds to $L = 74$ (18.5-in. stroke). This is slightly larger than that recommended by Miller. These recommendations are based on a compromise between the flexibility of thinner rods and the strength of thicker rods.

Cylinder parameters

H	Series
68	Mounting style
B	Bushing (bolted)
3	Rod end style (female, long)
B	Cushions, both ends
00400	Bore diameter, 4 in.
01000	Stroke, 10 in.
00250	Rod diameter, 2.5 in.
S	Port type (Society of Automotive Engineers)
2	Port location (2 or 4)
0	Modified, standard

Cylinder Reinforcement

The cylinder is attached to the movable frame head only at the cylinder base; therefore, reinforcement was needed to resist horizontal forces. We placed two 2-in.-thick steel plates on either side of the cylinder (Fig. 4). A 1/2-in.-deep and 5-in.-wide groove was cut into each plate to secure the edges of the lower block of the cylinder in four directions. The plates are shaped to remove large corner sections (to reduce weight) and to accommodate the hydraulic fitting on the cylinder base. Each plate is attached inside the frame head with four bolts to provide moment resistance in both directions.

Assuming 10° load eccentricity (from vertical), a 20,000-lbf load would put a 3,500-lbf horizontal force on the load head. We conservatively assumed that this force would be applied to a single plate. If the load head is 38 in. from the screw connection point (6 in. specimen deflection + 6 in. load head + 8 in. load cell + 18 in. cylinder height), the moment on the cylinder would be 132,000 in-lbf. This moment would be resisted by the four 5/8-in. cylinder mounting bolts and the four 1/2-in. reinforcement plate attachment bolts. The vertical projection of these bolt sets is about 7-1/2 in. apart, so the average shear force on a single bolt is about 4,400 lbf. This puts a maximum shear stress of 47,000 lbf/in^2 on each of the 1/2-in. bolts (less on the 5/8-in. mounting bolts), which is less than 81,000 lbf/in^2, the minimum yield stress for a grade 5 structural bolt.

Figure 4—Cylinder reinforcement plates.

Load Head Design

Two load head assemblies were made to apply four-point beam loading to long specimens. The assemblies consist of two steel channels placed back-to-back against a pivot block, to which pivoting load heads could be clamped (Fig. 5). Even though the individual heads could be moved along the channel assembly to change the load span, it was necessary to make a separate, smaller span assembly to keep the channel length outside the load span from interfering with a deflecting specimen. To facilitate manufacturing, the design of the head assemblies is the same, but the larger span assembly requirements control the design.

Assuming a load span of one-third the longest specimen length, the load span is 80 in. If the assembly is modeled as a center-loaded beam, a 20,000-lbf load would cause a maximum moment of 400,000 in-lbf. Assuming a steel yield of 36,000 lbf/in^2, this would require a beam with a section modulus greater than 11.1 in^3. Two 8 by 11.5 steel C-sections were chosen, which have a combined section modulus of 16.3 in^3.

The load head base is 7 in. wide. Thus, given the width of the channels, there should be a 2-1/2-in. space between them. The channels are connected by six 2-1/2-in. spacer blocks. In the center is a 2-in.-thick, 4- by 6-in. steel block with a 1-1/2-in. shear pin that allows the channels to rotate in

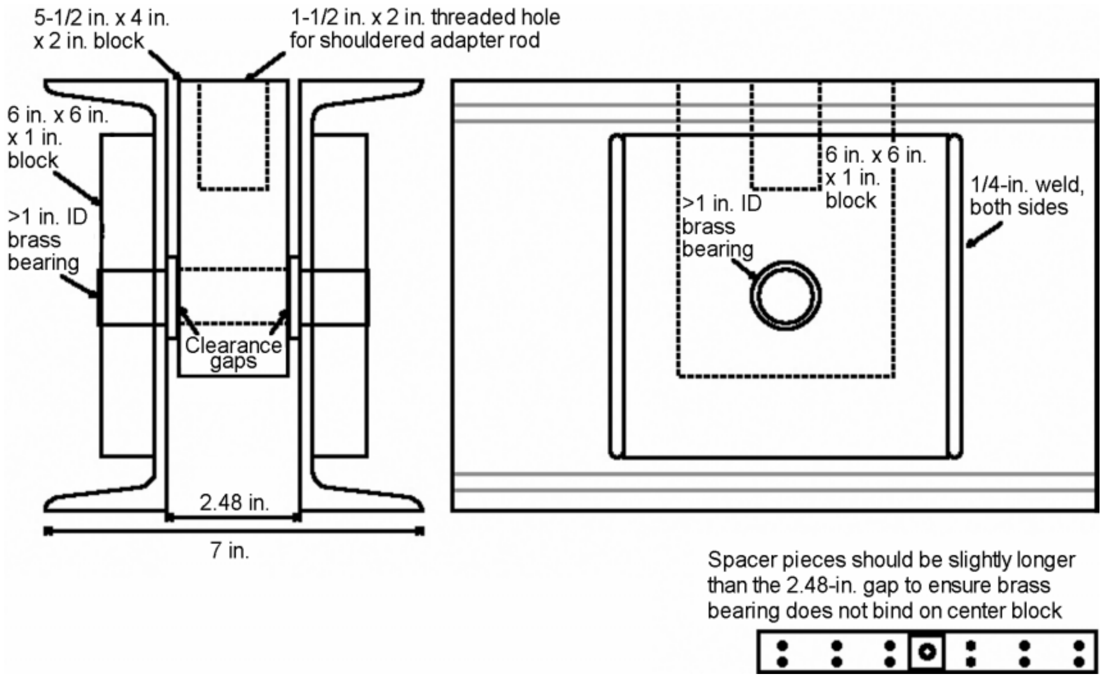

Figure 5—Load head design.

the plane of specimen bending. The pin is cottered on both ends. The channels are reinforced at the shear pin with 6- by 6- by 1-in.-thick plates that are welded to the outer surface of each channel and contain a brass bearing at the shear pin interface. The shear stress of the pin and the bearing stresses on the plates are less than the yield stress of mild steel. The pin should be lubricated with a light oil when the load heads are changed.

Frame Weight

Given rough dimensions of the test frame feet, slab, screws, and head box, and assuming 500 lb/ft^3 of steel, the frame weighs approximately 7,000 lb. The steel beams weigh approximately 3,000 lb; the reinforcement plates and test hardware could weigh as much as 1,000 lb. Thus, the entire machine weighs approximately 11,000 lb.

Concluding Remarks

The hydraulic bending test machine will be utilized for projects such as lumber recycling, where lumber from disassembled buildings is tested for possible use in new construction. The machine might also be used to test lumber cut from small-diameter timber as part of the utilization plan for that resource. Another use is to study the effect of high temperature and humidity conditions on dimension lumber over time, which will help us understand what happens to structures as they age.

This report would be of interest to other mechanical testing laboratories, technicians, safety officers, and scientists who might utilize a hydraulic bending machine. Organizations such as Forintek in Canada might soon be looking for a similar machine. Although specific requirements might be different, our design considerations will be important to their design process. Technicians who might not otherwise be aware of the small deflections and eccentric loads that can occur during a test would be advised to read this manual. Although the hydraulic bending machine is designed to resist these forces and stresses, it is important from a safety standpoint to consider their effects on the specimen and peripheral hardware. Lastly, scientists who order the use of our hydraulic bending machine will find it valuable to read this report to understand the capabilities and limitations of the machine when designing their experiments.

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