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TESTING OF WOOD BEARINGS FOR HAND PUMPS

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ABSTRACT

The feasibility of using wood handles for hand pumps was investigated in a laboratory study on the behavior of metal/wood interfaces. Eight wood handles equipped with simple pivots were subjected to oscillating motion with a load of 150 lbs. simulating the operation of a hand pump. The results, after 2×10^6 cycles, indicate that (1) woods impregnated with oil are more durable than dry ones, (2) galvanized pipe pivots function well, and (3) hardwoods are more durable than softwoods. A design technique for determining the required dimension of a hand pump handle based on the type of wood, load, and level of hand pump usage intensity is presented.

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TABLE OF CONTENTS

Contents — Page No.

Summary

1. Introduction 1

2. Properties of Wood 3

3. Early Development Work 4

4. Expanded Wear Test 6

5. Description of Equipment 7

6. Results 7

7. Methodology for Determining the Dimensions of Wooden Handpump Handles 11

8. Conclusions and Recommendations 14

Appendix A Bibliography 1

Appendix B Table B1

Strength properties of certain imported tropical woods and selected species of the U.S. .... 1 - 4

Table B2

Strength properties of certain imported tropical woods and selected species of the U.S. .... 4 - 8

Appendix C Oil Soaked Bearings: How to Make Them .... 1 - 7

Appendix D Theoretical Analysis of Static Stresses in Bearing Assemblies .... 1 - 5

List of Tables

1. Table 1 - Specific Gravity-Strength Relationships

2. Table 2 - Factors Relating Test Statistics to Allowable Properties

3. Table 3 - Wood/Metal Specimens

4. Table 4 - Wear of Bearing Assemblies Equipped with 1/4" O.D. Mild Steel Pin

5. Table 5 - Wear of Bearing Assemblies Equipped with 0.540" O.D. Galvanized Steel Pipe

SUMMARY

The behavior of metal/wood interfaces was investigated in a laboratory study simulating the operation of village hand pumps equipped with wood handles. Eight wood handles (four pine and four bubinga) were subjected to oscillating motion under a load of 150 lbs. Two samples of each of the woods were impregnated in oil. The pivots selected were 1/4" mild steel pins (O.D. 0.25") and galvanized steel pipe (O.D. 0.540").

Four of the handles failed after 1×10^6 cycles due to either shaft failure (considerable wear) or wood failure (excessively enlarged pivot hole). The remaining four were subjected to additional 1×10^6 cycles.

The relationships between the type of wood and its physical characteristics, the load, and the intensity of use of the pump are complex; no theoretical or empirical equation relating the above parameters is available. In general, the results indicate that wood handles with simple pivots function well. For the operating conditions of this study a galvanized steel pipe (O.D. 0.540") functioned well in both woods, oiled and dry. Oil-impregnated wood is more durable than dry wood, and hardwood exhibits less wear than softwood.

The design of stationary wood structures rely on static allowable design parameters. A design technique for determining the required dimensions for a hand pump handle is proposed, based on fatigue level. The design is dependent on the type of wood, level of usage intensity, and load. Additional pilot and field tests for testing the proposed design methodology are needed. Analysis of such data would lead to an improved technique for the design of wood handles for hand pumps.

Introduction

1. Most village hand pumps in use throughout the world are of the reciprocating type. A handle assembly is connected via mechanical linkage to a pump cylinder which is submerged in water. By raising and lowering the pump handle in a vertical plane, the pump cylinder follows a reciprocating motion of a given stroke length. This vertical reciprocating motion pushes water vertically upward to the surface. The size of the cylinder, length of stroke and frequency of reciprocating motion determine the rate of discharge.

2. During pumping, the handle assembly, i.e., the mechanical linkage connecting the pump handle to the pump cylinder, is subjected to a repeated loading sequence. Maximum loading is exerted on the handle assembly during the upward movement of the cylinder which corresponds to the downward movement of the pump handle. The force required to pull the pump cylinder up is a function of the water pressure exerted on the cylinder, the weight of the linkage components, and friction (1) of the various bearings, and (2) between the cylinder and the pump wall.

3. Loading exerted on the handle assembly results in stresses of various magnitudes at different points of the assembly. Critical points where high stresses and the ensuing wear occur are the pivot points of the pump handle. Conventional pivot points of most commercially available hand pumps consist of a mild steel pin and cast iron bearing. Under light use, such as in single-family applications, such metal/metal interfaces wear well. However, when subjected to heavy use, i.e., almost continuous operation for 8-12 hours per day, as would occur in many village systems, these pivot points wear out quickly. Normally, the user in developing countries does not have the necessary spare parts or the technology to manufacture replacement components from locally available materials.

4. Early hand pumps were probably equipped with wooden handles and a metal shaft. In most cases, either the user made the pump himself or it was manufactured by local artisans. In cases of pump failure, the user depended on his own skills or those in his community for repairs. As Europe became more industrialized, hand pumps produced by a society of a relatively advanced technology were exported to various parts of the world. Hand pumps were manufactured in factories from various metals such as steel, cast iron, and bronze; wooden handles were eliminated in favor of stronger metals. However, even today most commercial hand pumps operate satisfactorily when subjected to light use but fail when the use is intensive. Extensive wear or metal/metal interfaces prompted a search for other types of bearing assemblies that are inexpensive, readily available in developing countries, correspond to the user's level of technology, and exhibit minimal wear under heavy use.

5. The purpose of this investigation was to evaluate the wear of various wood/steer bearings. Oil-impregnated wood bearings coupled with steel pipe shafts exhibited little wear. Expansion of the investigation to include other wood specimens, more accurate simulation of pump operations, and field tests are required for verification of the preliminary results of the study.

Table 1

Specific Gravity-Strength Relationships[1/]

	Green Wood	Air-dry wood (12 percent moisture content)
Static bending:		
Fiber stress at proportional limit ... p.s.i.	$10,200G^{1.25}$	$16,700G^{1.25}$
Modulus of rupture .... p.s.i.	$17,600G^{1.25}$	$25,700G^{1.25}$
Modulus of elasticity .... 1,000 p.s.i.	$2,360G$	$2,800G$
Impact bending, height of drop causing complete failure .... in.	$114G^{1.75}$	$94.6G^{1.75}$
Compression parallel to grain		
Fiber stress at proportional limit ... p.s.i.	$5,250G$	$8,750G$
Maximum crusing strength .... p.s.i.	$6,730G$	$12,200G$
Modulus of elasticity .... 1,000 p.s.i.	$2,910G$	$3,380G$
Compression perpendicular to grain, fiber stress at proportional limit .... p.s.i.	$3,000G^{2.25}$	$4,630G^{2.25}$
Hardness		
End .... lb.	$3,740G^{2.25}$	$4,800G^{2.25}$
Side .... lb.	$3,420G^{2.25}$	$3,770G^{2.25}$

1/ The properties and values should be read as equations; for example, modulus of rupture for green wood = $17,600G^{1.25}$, where G represents the specific gravity of oven-dry wood, based on the volume at the moisture condition indicated. Note that this differs from the definition in Appendix B impregnated with oil, as was done for half the samples tested.

Adapted from reference No. 9, p. 88.

Properties of Wood

6. Wood differs from most other materials used in present day hand pumps in that it is organic in nature composed of about 60% cellulose, 28% lignin and 12% of other materials. The cellular structure of wood consists of microscopic cells, tiny tubes with closed ends, whose length parallels the axes of the tree and is many times its lateral dimension. This structure, which is quite different from the crystalline structures of metals, is responsible for the wide variations in the physical characteristics of different woods. Likewise, because wood is a natural material, variations in the physical characteristics of members of the same species can be expected.

7. Trees are divided into two classes--the hardwoods and softwoods. No definite degree of hardness divides the hardwoods and the softwoods. Strength properties of wood and those properties that enable wood to resist applied forces are closely related to the specific gravity or density of wood. However, the strength of wood of any species varies over a considerable range. The substance of which wood is composed is actually heavier than water. Its specific gravity is about 1.45 regardless of the species of wood. However, dry wood of most species floats in water, indicating that a large part of the volume of a sample of wood is occupied by cell cavities and pores. The cells in wood, called fibers, are hollow and empty in dry wood. The strength of wood depends primarily not on the length of fibers but on the thickness of the cell wall. Thus, specific gravity is a good index of the amount of wood substance a piece of dry wood contains and hence an index of its strength properties.

8. The relationships of specific gravity or density and various strength properties of woods are given in Table 1. The equations given in the above table are based on tests of over 160 species. Strength properties of particular woods are given in Appendix B. Factors relating test statistics to allowable properties for the design of wood handles are given in Table 2.

Table 2

Factors Relating Test Statistics
to Allowable Properties /1

Property	Factor
Bending strength	0.24
Tensile strength	0.24
Compressive strength parallel to grain	0.26
Compressive strength perpendicular to grain	0.22
Shear strength	0.23

/1 Modified from reference 2, p. 97.

9. A survey of the literature 1/ on wood products indicates that for wooden bearings the wood should be dense, have high strength values and be properly dried. For use as pump handles, it must be locally available in the area it will be used, and inexpensive.

10. The hardness of wood represents its resistance to wear and marring. It is expressed as the load required to embed a 0.444 inch ball to one-half its diameter in the wood (ASTM 143). It is evident that woods with higher specific gravity possesses higher hardness values and therefore are more suitable for pump handles.

11. The fatigue resistance of wood is a measure of the ability of wood to sustain repeated, reversed, or vibrational loads without failure, and therefore is important in the design of wooden pump handles. Appropriate strength values for a wood proposed for a pump handle must be selected in order to prevent fatigue failure; this is discussed in paragraph 34.

Early Development Work

12. The objective of the first phase of the investigation was to determine the wear of one particular metal/wood interface. The wood selected was bubinga (guibourtia spp.) which is widely distributed from southeastern Nigeria through the Cameroons, Gabon and the Congo region. Although bubinga

1/ Bibliography given in Appendix A.

is moderately hard and heavy (specific gravity 0.63 which is similar to oak and other hardwoods), the wood can be sawed or drilled with no difficulties. Because all commercially available wood bearings are impregnated with lubricants, it was decided to do the same for the selected specimen increase its resistance to wear. A section of bubinga wood, 11" x 3-1/2" x 1-1/2", was treated with commonly available vegetable oil in accordance with ITDG instruction Manual on oil soaked bearings (Appendix C).

13. Theoretical analysis of the loads associated with the pumping operation dictated the location and size of the journals (a complete analysis, using the test loads of subsequent trials, is given in Appendix D). A weight of 105 lbs. was assumed to be acting on the pump rod bearing to simulate a column of water 3 inches in diameter and 35 feet in height. Two sections of 1/4" I.P. standard galvanized steel pipe (0.540" O.D.) were cut, flared and installed as shafts for both the pivot bearing and the pump rod bearing, as shown in Figure 1.

14. A fatigue testing machine, simulating pumping at a rate of 13 strokes per minute, was developed for testing the behavior of the above wood/metal interface. The three-foot handle had a mechanical advantage of 5 to 1 and moved through an arc of approximately 65°, simulating a plunger stroke of about 6.5 inches.

15. After 100,000 cycles no measurable wear was observed on either of the two shafts or the wood journals. It should be noted that 100,000 cycles is equivalent to approximately 13 days of pumping at 13 strokes per minute for 10 hours each day. Each stroke displaces approximately 0.1 cubic feet. Allowing 50 percent overall efficiency (to repeat hydraulic losses, change over time of users, etc.), 10 hours pumping at this rate would supply about 3,000 U.S. gallons, or 5 gallons/person/day to a population of 600 people. On these assumptions, approximately 3×10^6 cycles are needed in order to simulate one year of operation.

16. Encouraged by the results of the first phase, a second specimen was tested under similar conditions but with a heavier load. After 250,000 cycles, no measureable wear had occurred under a load of 135 lbs. - equivalent to a column of water 3 inches in diameter and 45 feet in height.

17. Only limited information was obtained from the single specimen tested. The results after approximately 0.5×10^6 cycles indicated minimal wear of the oil-impregnated wood bearing of the galvanized steel shaft. Because 0.5×10^6 cycles corresponds to a relatively short time, (say, 2 months) under actual intensive use, it was decided to expand the experiment to include a total of eight specimens to be tested for approximately 2×10^6 cycles.

Expanded Wear Test

18. Theoretical analysis, presented in Appendix D, suggested that if the journals were spaced as indicated in Figure 1, the stresses that would develop in the wood and the pin would be less than the allowable. The analysis was based on a dead load of 150 lbs., as strain data from dead and live load testing did not indicate any appreciable difference between the two.

19. The objectives of the expanded test were:

(1) To design and fabricate a fatigue testing machine capable of testing a number of wood handles simultaneously under continuous operation. Each handle was to operate subject to a dead load of 150 lbs. with an approximate frequency of 13 cycles per minute.

(2) To investigate the behavior of bubinga and common pine bearings and shafts of 1/4" mild steel pin (actual O.D. 0.25") and 1/4" I.D. galvanized steel pipe (actual (O.D. 0.540").

The eight samples tested simultaneously are shown in Table 3.

Table 3

Wood/Metal Specimens

	Pine	Bubinga
Untreated 1/4" O.D. mild steel pin	#1	#2
Untreated 1/2" O.D. galvanized steel pipe	#3	#4
Oil-impregnated wood 1/4" O.D. mild steel pin	#5	#6
Oil-impregnated wood 1/2" O.D. galvanized steel pipe	#7	#8

20. Extended testing of the above samples was to provide information on (a) the role oil-impregnated wood bearings play in reducing wear, (b) the suitability of galvanized steel pipe as a pivot, and (c) preliminary guidelines on wood and metal selection for use as pivots in hand pumps.

21. A fatigue testing machine was designed and constructed in accordance with objective (1). Wood specimens were obtained from commercial suppliers. The samples to be impregnated with oil were submerged in boiling vegetable oil. Continuous testing started in March 1977 and terminated in August 1977. The testing rig described below functioned well and required little maintenance.

Description of Equipment

22. The testing equipment for wooden pump handles is a self-contained unit which provides pivots, loads, working forces and an electro-mechanical drive. The equipment used is capable of testing eight handles simultaneously. To eliminate likely difficulties with eight 150 lb. weights swinging from the test rig, it was decided to use a bank of four pairs of handles. With this arrangement a state of equilibrium is maintained and little energy is required to set the equipment in motion.

23. To fully understand the mechanism of the test rig, refer to Figure 2. Having four pairs of handles as described above, the operation was further simplified by working with two groups of four handles. In a group of two pairs of handles the adjacent handles of each pair was firmly coupled with a cross member. This required only two driving points to operate eight handles. The drive was provided with a two-throw crankshaft and connecting rods. The crankshaft was powered by a 115 volt motor with a reduction gear which transmitted power to a chain and sprocket drive. One of the sprockets was attached to the crankshaft which revolved at 13 R.P.M. The location and construction of the pivot yokes can be clearly seen in Figure 3.

24. Consider one pair of handles, as shown in Figure 4. To simplify the illustration disregard the weight of the handle and any friction involved with pulleys, pins, etc. The distance between the pivot and the weight is five times the distance between the pivot and the cable attachment. For an applied weight of W the force in the cable is therefore 5W and, consequently, the load on the pump rod pin is equal to 5W. During the test, W was set at 30 lbs., thus providing a pump rod pin load of 150 lbs. The load on the pivot pin is 180 lbs. This an acceptable representation of field conditions, corresponding to a head of 50 feet of water. In operation, the equipment sets up an oscillating action. When the weight reaches the end of its stroke its direction and travel is reversed. This sudden change in direction applies considerable shock force to the cable and pump rod pin, which approximates actual field conditions.

Results

25. The eight specimens were subjected to almost continuous cyclic motion for the first 1×10^6 cycles. On two occasions minor break downs occurred which required turning off the equipment for a period of 48 hours.

26. After 10^6 cycles, testing was temporarily suspended so that wear of all the bearing assemblies could be measured. The results summarized in Tables 4 and 5 show serious wear in four samples. Of the four bearings with mild steel pins (Table 3, #1, 2 5, 6), all except the untreated, dry bubinga (#2), exhibited considerable wear. However, handles equipped with galvanized steel pipe showed little wear except the dry pine bearing (#3), on which the pin hole elongated by over 1/2 inch. The reason for this extreme wear, 5 times that on the steel pin, is unknown. It was concluded that, of the original eight bearing assemblies, four (#1, 3 5, 6) had failed.

27. None of the bearing assemblies failed due to shear of the pin or the wood. The locations of the bearings and the required shaft size for the load tested had been determined by theoretical analysis, and the actual dimensions used in the test were such that the anticipated stresses in the shaft and bearing were less than the allowable. Therefore, as expected, none of the specimen failed due to shear of either the pin or the wood. Specimen failure after 10^6 cycles was caused by either a decrease in the diameter of the shaft or increase in the diameter of the bearing.

28. In general, the dry wood specimens showed an increase in bearing wear compared to the oil-impregnated ones. On the other hand, the shafts in the oil-impregnated bearings exhibited more wear than those in the dry samples; in the case of the mild steel pivot pins, both the shaft and the bearing failed. The cause seems to be that metal particles, stipped from the shaft, became embedded in the wooden bearing, and when in contact with the shaft, caused additional metal wear. The process can be viewed as a thin layer of metal particles continuously rubbing against the metal shaft. Thus, the contact in this case can be viewed as a partial metal/metal interface instead of metal/wood interface; the wood bearings themselves showed little wear. In contrast, although some wear of the shaft was noted in the dry bearing, in this case the metal particles did not become embedded in the wood but remained loose in the annular space. The movement of the bearing relative to the stationary pivot led to an increased wear of the wood and to an increase in the diameter of the bearing. Bubinga, being denser, showed less wear than pine.

29. As was noted above, three of the bearing assemblies equipped with galvanized steel pipe (#4, 7, 8) exhibited little wear and were subjected to testing for an additional 10^6 cycles, together with one assembly (#2) using a mild steel pin.

30. At the end of over 2×10^6 cycles the testing rig was dismantled and the wear of various bearing assemblies was measured. These results are also presented in Tables 4 and 5. The galvanized steel pipe shafts showed little wear: A maximum of 0.006" was measured in the oiled pine bearing, while wear of 0.003" to 0.005" was recorded in the other shafts. Moreover, the wear for the initial 10^6 cycles was approximately 0.004", while that for the second million cycles added only 0.001". The data suggests that the initial wear consists of mainly removing some slight irregularities in the circumference of the pipe accompanied by a very slight removal of the

Table 4. Wear of Bearing Assemblies Equipped with 1/4" O.D. Mild Steel Pin

	Cycles	pump rod shaft (max)	Pump rod bearing				pivot shaft (max)	Pivot pin bearing				Remarks
			vertical		horizontal			vertical		horizontal		
			mid	ends	mid	ends		mid	ends	mid	ends	
Pine, dry #1	10^6	.03	.10	.10 .10	0	0 0	0	.06	.14 .12	.015	.025 .165	Failure of both bearings. Different wear rates of ends and middle of pivot pin bearing possibly due to pin eccentricity after handle fall at 6×10^5 cycles
Bubinga, dry #2	10^6	.001	.025	.035 .020	.01	.015 .021	0	.025	.035 .025	0	0 0	
	2×10^6	.007	.05	.125 .022	.013	.018 .012	.012	.006	.08 .07	.007	.026 .037	Pump rod bearing failure due to excessive wear at one end.
Pine in oil #5	10^6	.06	.015	.025 .020	.01	.02 .01	.05	.01	.05 .13	0	.03 .06	Failure of both shafts. Also pivot pin bearing failure due to excessive wear at one end.
Bubinga in oil #6	10^6	0	0	0 0	.008	.01 .008	.05	.035	.09 .1	0	.015 .05	Failure of pivot pin shaft and bearing

Notes: - wear data in inches
- failure criterion: 0.05" shaft wear or 0.10" bearing wear

Table 5. Wear of Bearing Assemblies Equipped with 0.540" O.D. Galvanized Steel Pipe

	Cycles	pump rod shaft	pump road bearing				pivot shaft	pivot pin bearing				Remarks
			vertical		horizontal			vertical		horizontal		
			mid	ends	mid	ends		mid	ends	mid	ends	
Pine, dry #3	10^6	0	.062	.067 .090	0	0	.005	.562	.587 .527	0	0 0	Pivot pin bearing failure.
Bubinga, dry #4	10^6	.003	.032	.032 .047	0	0 0	.003	.018	.072 .033	0	0 .007	
	$2x10^6$	.004	.083	.062 .122	0	0 0	.004	.077	.102 .62	0	0 .008	Failure of both bearing
Pine in oil # 7	10^6	.005	.003	.008 0	0	0 0	.003	0	.042 .028	.003	.007 .007	
	$2x10^6$	.006	.077	.092 .072	0	0 0	.004	0	.237 .287	.005	.008 .008	Pivot pin bearing failure
Bubinga in oil #4	10^6	.004	.002	0.12 0.12	0	0 0	.005	.013	.013 .013	0	0 0	
	$2.5x10^6$	.005	.028	0.28 0.78	0	0 0	.009	.078	.132 .053	0	0 0	Failure of pivot pin bearing. Note that operating cable broke at $1.4x10^6$ cycles, damaging sample.

Notes - Wear data in inches
- Failure criterion 0.05" shaft wear or 0.10" bearing wear

the zinc coating. To the naked eye the shafts look like highly polished standard galvanized pipe. Interestingly, the mild steel shaft (#2), was still serviceable after 2 x 10^6 cycles, even in an unlubricated bearing, in contrast to the early failure of the other samples (#1, 5, 6).

31. The dry bubinga bearings showed more wear than the oil-impregnated specimens. The elongation of the hole was more pronounced at the pivot shaft than at the pump rod shaft. The impregnated bubinga sample was the same sample that was tested in the earlier phases of this study. This bearing assembly was subjected to more than 2.5 x 10^6 cycles. Maximum wear of 0.13" was noted at the pivot shaft hole which was higher than that observed in the dry bubinga sample. The reason for this large wear is attributed to the fact that at 1.4 x 10^6 cycles the cable connecting this specimen to its neighbor broke, and the handle was subjected to much larger stresses as well as actual compression of the wood fibers at the edge.

32. In general, the results indicate that oil impregnated bearings exhibit less wood wear than dry ones. Galvanized steel pipe showed less wear than mild steel pin in both oiled and dry bearing assemblies. It appears that galvanized steel pipe is an excellent material for shafts as it is universally available, inexpensive, easy to cut and flare, and is unlikely to be pilfered. Dense woods with high strength values are suitable for oil impregnated bearings. Under the assumed conditions of this study (13 cycles per minute, 10 hours per day) we estimate that dense wood bearings would last at least one year and perhaps as long as 2 years before replacement. The shafts would last considerably longer. Softer woods, such as pine (oil impregnated), will have shorter lives, and may have to be replaced every 6 - 12 months. Guidelines for determining the dimensions of pump handles for various service intensity levels are given in Appendix D.

33. The data presented in Tables 4 and 5 does not permit any extensive analysis of the behavior of various metal/wood interfaces. The relationships between load, wear, type of wood, and wood treatment are complex; only general observations can be drawn from the limited experimental data. However, the data can be used to test the proposed method given below for determining the dimension of wooden handles.

Methodology for Determining the Dimensions of Wooden Handpump Handles

34. Most structural materials, when subjected to repeated stresses, will fail at stress levels significantly lower than the maximum stress permissible under static conditions. This form of failure is designated a fatigue failure. It can be delayed by lowering the stress limits for members subjected to repeated stress. If the stress is lowered sufficiently (to the "endurance limit") fatigue failure can be avoided. For example, the endurance limit for steel is commonly about 50 percent of its tensile strength. Tests on wood samples, reported in the literature, suggest that wood may be less sensitive to repeated loading than crystalline structural materials such as

steel. Nevertheless, tests of fatigue in tension, bending, etc., indicate an endurance limit for wood (for 30 x 10^6 cycles) of about 30-40% of the particular static strength characteristic (e.g., modulus of rupture, or ultimate tensile or shear stress). 1/ In designing to avoid fatigue failure, care must be taken to allow for or to eliminate stress concentrators, such as notches or shaft holes, which can give rise to local stresses three or four time the values calculated for whole sections.

35. The trails carried out so far do not allow the mechanism of failure to be clearly identified. The failures of samples Nos. 1 and 8 soon after they were subjected to shock due to breakage of the test rig suggests that distortion of the pivot pin or deformation of the bearing, by introducing high local stresses, can play a major role in determining the working life of the sample. These local stresses fluctuate with each pumping cycle, due both to the reversal of direction of thrust of the pump handle and to the rotation of the bearing around the stationary shaft. This suggests that a design based on fatigue considerations would be more appropriate than one based on a static analysis.

36. In Appendix D the maximum static stresses in the 2 inch x 4 inch wood samples (accounting for stress concentrations) are derived as 507 psi in bending and 96 psi in shear. The literature 2/ suggests that, where specific data is not available (such as those listed in Appendix B or derived from Tables 1 and 2, allowable design stresses for static conditions are:

	Bending, psi	Shear, psi
Hardwood (oak, larch)	1,900	185
softwoods (pine, hemlock)	1,200	120

Comparable values, for bending calculated from the specific gravity of the two samples of bubinga and pine and from Tables 1 and 2 are:

Bubinga (specific gravity 0.63)	2,225	not
Pine (specific gravity 0.41)	1,300	available

The relationships between specific gravity and shear is not known; therefore the generalized values given above for the shear of hardwood (185 psi) and softwood (120 psi) are used. (It is possible, of course, that all the above values would be affected by the oil-impregnation process, but no data appears to be available on this effect). It therefore appears that, during the test, the

1/ Reference 9, p. 82.

2/ Reference 1, 2.

bubinga sample was stressed to about 22 percent of its allowable bending stress and about 51 percent of its allowable static shear stress. Corresponding values for the pine samples are about 39 percent and 80 percent, respectively.

37. These basic considerations permit a tentative design methodology to be proposed. This makes two fundamental assumptions:

- that the wood will be oil-impregnated; and

- that the pivots will be half inch O.D. galvanized pipe or similar.

Under the test conditions, it is then likely that bubinga would have a life of about 2.5 million cycles and the pine of over 1 million cycles. A wood stressed to only 30-40 percent of its strength would have a life perhaps an order of magnitude longer (para 34). Each 1 million cycles represents 10 hours' pumping per day for 4 months at 13 cycles per minute to supply 5 gallons/person/day to a population of 600, or any equivalent combination of factors. We therefore propose that design should be based on anticipated severity of use and a permitted fatigue level as follows:

	Corresponding to:		Fatigue Level:	
Intensity of Use	Hours/day Continuous Use	Population Served	Design Stress as % of Allowable Static Stress	Anticipated Handle Life, Years
Intense	10-12	600-700	30	5 years
Moderate	4-6	300-450	50	1-1/2 years
Light	1-2	60-70	75	3 years

From the test results, a pump handle fabricated from 2" x 4" softwood is clearly suitable only for light use, whereas a similar handle of hardwood would serve for moderate use.

38. These suggested values need verification in the field, not only to supplement the very limited data obtained from the laboratory tests but also to reflect operational problems such as irregular or oversize holes, non-parallel bearings, a dusty environment, side loading during pumping, etc. Careful monitoring of such trials should permit design charts to be drawn up, from which the necessary dimensions of a pump handle can be determined given pump casing diameter, static lift, wood type, dependent population and desired interval between handle renewals.

<u>Conclusions & Recommendations</u>

39. 1. Wood handles on simple pivots work well.

2. Galvanized pipe, 0.540" O.D. is adequate as a pivot for the application tested, which represented a typical hand-pumped well. There is no merit in selecting smaller pivots which exhibited increased wear and negligible cost savings.

3. Oil-impregnated wooden handles are more durable than dry ones.

4. The proposed "fatigue level" design should be tested to establish load, diameter, and wood durability data for a wide range of stresses.

5. What is needed now is carefully monitored field testing to reproduce actual conditions--i.e., manufacturing inaccuracies and random side thrusts in practice. The data from such testing should be evaluated in light of the finding of this study.

APPENDIX A

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Table B1 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens in the green condition.)

Botanical name	Origin*	No. trees	Moisture content percent	Sp. gr.[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Acer rubrum	US	14	63	0.49	7,700	1,390	11.4	3,280	700	1,150
Acer saccharum	US	17	58	.56	9,400	1,550	13.3	4,020	970	1,460
Albizia falcataria	AS	5	GR	.28	4,490	900	--	2,480	350	630
Albizia falcataria	HA	5	GR	.32	5,300	1,080	5.5	2,610	360	800
Albizia lebbek	AS	5	GR	.55	9,560	1,590	8.3	5,100	1,020	1,400
Alnus rubra	US	6	98	.37	6,500	1,170	8.0	2,960	440	770
Alstonia boonei	AF	1	123	.36	4,940	995	4.2	2,803	370	678
Anacardium excelsum	CS	6	111	.41	5,320	1,060	4.1	2,460	400	740
Anisoptera spp. (palosapis)	AS	18	GR	.51	7,550	1,430	--	3,780	810	1,000
Araucaria angustifolia	BR	11	GR	.48	8,200	--	--	3,700	600	970
Aspidosperma peroba	BR	6	GR	.69	12,100	--	--	5,800	1,520	1,720
Aspidosperma (peroba rosa)	BR	1	35	.67	10,930	1,290	10.5	5,540	1,580	1,870
Astronium graveolens	CS	4	45	.86	12,400	1,900	7.4	6,880	1,990	1,840
Berlinia grandiflora	AF	5	75	.61	9,975	1,412	10.9	4,733	1,000	1,206
Betula alleghaniensis	US	17	67	.55	8,300	1,500	16.1	3,380	780	1,110
Brachystegia nigerica	AF	5	69	.60	10,830	1,370	11.2	5,491	1,220	1,514
Brosimum alicastrum	GU	1	49	.72	16,750	1,990	--	--	2,090	--
Bursera simaruba	US	5	99	.30	3,300	560	3.5	1,510	230	590
Bursera simaruba	GU	1	97	.38	5,150	940	--	--	330	--
Calophyllum brasiliense rekoi	CA	18	62	.54	10,470	1,570	10.6	5,160	1,010	1,290
Calycophyllum candidissimum	VE	2	GR	.67	14,290	1,930	18.6	6,200	1,630	1,660
Canarium schweinfurthii	AF	4	94	.45	5,605	963	5.1	3,005	520	818
Carapa guianensis	BR	2	72	.56	11,110	1,560	11.4	4,930	1,060	1,320
Carapa surinamensis	SU	2	58	.53	9,480	1,820	8.2	4,640	710	1,120
Cariniana brasiliensis	BR	3	GR	.46	9,700	--	--	4,500	860	1,270
Carya illinoensis	US	5	63	.60	9,800	1,370	14.6	3,990	1,310	1,480
Cedrela angustifolia	BR	2	84	.38	6,730	1,170	7.4	3,100	450	790
Cedrela oaxacensis	PA	3	67	.41	7,510	1,310	7.1	3,370	550	990
Cedrela odorata	NI	1	73	.34	5,220	870	7.4	2,760	350	720
Cedrela odorata	GU	1	75	.43	9,500	1,480	--	--	620	--
Ceiba pentandra	VE	3	GR	.25	2,180	410	1.2	1,060	220	350
Chlorophora excelsa	AF	2	92	.59	10,165	1,284	10.5	4,915	1,080	1,311
Chloroxylon swietenia	AS	5	GR	.77	12,920	1,660	11.1	6,230	1,830	1,780
Chloroxylon swietenia	AS	5	GR	.83	12,310	1,640	9.5	6,810	1,850	1,770

*Key to code letters: AF, Africa; AS, Southeast Asia; AU, Australia; BR, Brazil; CA, Central America; CH, Chile; CR, Costa Rica; CS, Central & South America; EC, Ecuador; GU, Guatemala; GY, Guyana (British Guiana); HA, Hawaii; HO, Honduras; IN, India; NI, Nicaragua; PA, Panama; PE, Peru; PH, Philippine Islands; SM, South America; SU, Surinam; US, United States; and VE, Venezuela.

Table B1 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens in the green condition.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr.[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Cordia alliodora	CA	13	106	.44	8,840	1,260	9.5	4,000	790	1,130
Cordia goeldiana	BR	2	53	.52	10,540	1,830	11.2	4,940	1,030	1,080
Cordia trichotoma	BR	1	38	.50	9,600	1,420	12.7	4,110	880	1,050
Cornus florida	US	5	62	.64	8,800	1,180	21.0	3,640	1,410	1,520
Cybistax donnell-smithii	HO	4	59	.39	7,710	980	6.9	3,630	660	1,050
Dalbergia latifolia	AS	5	GR	.75	9,190	1,190	11.6	4,530	1,270	1,400
Dalbergia sissoo	AS	5	GR	.65	11,170	1,200	14.1	5,440	1,520	1,630
Dalbergia sissoo	AS	5	GR	.69	10,220	1,260	11.9	5,000	1,380	1,410
Dalbergia sp.	BR	1	GR	.80	14,140	1,840	13.2	5,510	2,440	2,360
Dicorynia guianensis	SU	2	79	.60	11,410	1,840	12.0	5,590	1,100	1,340
Diospyros philippensis	AS	3	GR	.80	10,690	1,640	--	--	1,740	1,450
Diospyros pilosanthera	AS	1	GR	.81	10,610	2,010	--	4,620	1,490	--
Diospyros virginiana	US	5	58	.64	10,000	1,370	13.0	4,170	1,280	1,470
Dipterocarpus spp. (apitong)	AS	57	GR	.59	9,220	1,790	--	4,410	800	1,040
Dracontomelon dao	AS	2	GR	.54	6,730	1,350	--	4,610	860	1,170
Dracontomelon mangiferum	AS	7	GR	.46	8,540	1,400	--	4,300	750	1,140
Dryobalanops lanceolata	AS	5	64	.64	12,160	1,701	12.8	5,971	980	1,038
Entandrophragma angolense	AF	3	74	.50	7,125	1,070	7.4	3,533	770	942
Entandrophragma cylindricum	AF	5	62	.60	10,165	1,487	10.5	5,011	1,020	1,250
Entandrophragma utile	AF	1+	50	.57	10,830	1,487	10.3	5,318	1,080	1,382
Enterolobium cyclocarpum	GU	1	226	.31	5,030	610	--	--	350	--
Eucalyptus diversicolor (karri)	AU	26	GR	.70	10,600	2,070	--	5,250	1,360	1,335
Eucalyptus marginata (jarrah)	AU	28	GR	.67	9,880	1,480	--	5,190	1,285	1,325
Euxylophora paraensis	BR	3	59	.70	13,200	2,040	10.6	6,440	1,610	1,520
Fraxinus americana	US	23	42	.55	9,600	1,460	16.6	3,990	960	1,380
Gonystylus bancanus	AS	9	37	.59	9,785	1,573	9.0	5,395	640	994
Gossweilerodendron balsamiferum	AF	31	56	.45	7,125	931	8.6	3,379	620	977
Guarea cedrata	AF	2	99	.48	10,260	1,380	12.1	4,934	870	1,382
Guarea thompsonii	AF	4	52	.56	11,780	1,648	12.8	6,010	950	1,346
Hura crepitans	CS	7	65	.38	6,130	1,010	6.8	2,650	430	810
Hymenaea courbaril	CS	9	59	.72	12,950	1,820	15.7	5,800	2,030	1,770
Intsia bijuga	AS	14	GR	.70	15,000	2,150	--	8,040	1,700	1,830
Intsia palembanica	AS	5	GR	.68	12,850	2,020	12.8	6,770	1,390	1,560
Juglans regia	AS	10	GR	.47	8,710	1,310	10.4	4,010	670	1,060

Table B1 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens in the green condition.)

					Static bending			Compression parallel to grain		Shear parallel to grain
Botanical name	Origin	No. trees	Moisture content percent	Sp. gr.[1]	Modulus of rupture psi	Modulus of elasticity 1,000 psi	Work to maximum load In.-Lb./cu. in.	Maximum crushing strength psi	Hardness side lbs.	Maximum shearing strength psi
Khaya anthotheca	AF	4	61	.46	7,315	1,156	8.5	3,533	730	1,056
Khaya anthotheca	AF	5	54	.47	8,265	1,198	9.8	3,955	730	1,118
Khaya ivorensis	AF	11	64	.43	7,400	1,160	8.3	3,500	640	930
Khaya grandifoliola	AF	5	54	.57	9,500	1,412	9.7	4,992	1,170	1,540
Koompassia malaccensis	AS	5	GR	.72	14,530	2,410	--	7,930	1,480	--
Lovoa trichilioides	AF	2	61	.48	7,790	1,134	--	4,147	690	--
Mansonia altissima	AF	1	44	.57	12,350	1,498	16.4	6,144	1,210	1,602
Mitragyna sp.	AF	1	101	.48	7,505	1,263	6.9	3,802	700	--
Nauclea diderrichii	AF	4	75	.67	13,015	1,840	10.0	7,190	1,520	1,672
Ocotea rodiaei	GY	5	42	.83	19,400	2,980	13.0	10,360	2,190	1,480
Ocotea rubra	SM	5	89	.51	7,620	1,420	4.8	3,630	500	840
Parashorea plicata	AS	5	70	.43	7,790	1,198	6.8	4,118	580	810
Parashorea plicata	AS	5	66	.46	8,360	1,412	7.0	4,435	660	906
Parashorea plicata	AS	22	GR	.49	9,100	1,550	--	4,400	730	1,050
Paratecoma peroba	BR	3	GR	.63	13,400	--	--	6,300	1,430	1,690
Peltogyne confertiflora	BR	1	GR	.77	20,000	--	--	8,900	2,180	2,160
Peltogyne densiflora	BR	3	64	.75	16,200	2,610	17.0	9,020	2,090	1,640
Peltogyne pubescens	GY	1	42	.92	21,100	4,260	16.4	10,970	3,290	1,830
Peltogyne venosa	SU	3	71	.67	13,690	2,000	14.8	7,020	1,810	1,640
Pentacme contorta	AS	19	GR	.43	7,500	1,380	--	3,700	580	910
Pericopsis elata	AF	6	62	.66	14,820	1,766	19.5	7,488	1,600	1,672
Phoebe porosa	BR	3	113	.52	7,700	1,080	8.9	3,380	880	1,170
Pinus caribaea	CA	19	37	.68	9,980	1,690	12.0	4,780	820	1,200
Pinus elliottii	US	30	GR	.56	8,900	1,580	9.5	4,340	630	1,000
Pinus monticola	US	5	54	.36	5,200	1,170	5.0	2,650	310	640
Pinus oocarpa	HO	3	41	.55	7,970	1,740	6.9	3,690	580	1,040
Pinus palustris	US	44	63	.54	8,700	1,600	8.9	4,300	590	1,040
Pometia pinnata	AS	5	GR	.57	9,650	1,620	--	4,560	930	1,170
Pometia tomentosa	AS	7	GR	.56	9,020	1,700	--	4,160	840	1,170
Populus deltoides	US	5	111	.37	5,300	1,010	7.3	2,280	340	680
Prioria copaifera	PA	4	102	.40	5,930	950	5.4	2,590	450	860
Pterocarpus angolensis	AF	3	64	.59	11,685	1,177	12.4	5,654	1,300	1,593
Pterocarpus indicus	AS	15	GR	.53	10,700	1,470	--	5,570	950	1,220
Quercus alba	US	20	68	.60	8,300	1,250	11.6	3,560	1,060	1,250
Quercus oleoides	GU	1	38	.91	11,490	1,800	--	--	2,010	--
Quercus virginiana	US	5	50	.81	11,900	1,580	12.3	5,430	1,880	2,210

Table B1 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens in the green condition.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr.[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Samanea saman	VE	3	GR	.48	8,100	910	10.4	3,760	750	1,100
Samanea saman	AS	6	GR	.49	6,600	660	--	3,270	990	1,240
Shorea almon	AS	12	GR	.41	7,500	1,440	--	3,750	500	840
Shorea dasyphylla	AS	2	56	.43	8,645	1,498	8.8	4,454	560	--
Shorea leptocladus	AS	5	73	.39	6,935	1,081	6.3	3,763	450	748
Shorea negrosensis	AS	15	GR	.44	7,700	1,380	--	3,700	570	930
Shorea parvifolia	AS	5	84	.39	6,650	1,038	6.2	3,322	440	713
Shorea pauciflora	AS	5	69	.50	9,405	1,498	8.5	4,723	700	1,109
Shorea philippinensis	AS	6	GR	.41	6,900	1,200	--	3,360	530	920
Shorea polita	AS	4	GR	.47	7,980	1,330	--	4,010	710	1,000
Shorea polysperma	AS	19	GR	.46	8,300	1,540	--	3,940	620	940
Shorea smithiana	AS	5	64	.40	6,840	1,124	5.9	3,523	440	607
Shorea squamata	AS	14	GR	.41	7,300	1,400	--	3,470	480	770
Shorea waltonii	AS	4	70	.36	7,125	1,252	6.8	3,677	460	968
Simarouba amara	SU	2	69	.38	6,310	1,140	4.5	2,970	390	790
Simarouba glauca	US	4	81	.33	3,500	700	1.8	1,810	240	710
Spondias mombin	GU	1	85	.39	6,180	1,510	--	--	460	--
Spondias mombin	VE	3	131	.40	6,400	1,160	3.8	2,560	530	770
Sterculia oblonga	AF	5	60	.69	11,115	1,605	12.1	5,386	880	924
Swietenia macrophylla	CS	77	67	.45	9,280	1,280	9.6	4,510	700	1,310
Tabebuia heterotricha	PA	3	41	.80	20,080	2,120	27.3	7,680	2,530	2,140
Tabebuia rosea	CS	10	62	.51	10,650	1,470	11.2	4,930	890	1,240
Tabebuia serratifolia	SM	3	31	.92	22,850	3,060	25.6	10,660	2,970	2,050
Tarrietia utilis	AF	7	47	.56	9,690	1,305	9.6	5,088	1,050	1,276
Tectona grandis	IN	134	67	.57	10,980	1,510	10.8	5,470	1,070	1,290
Tieghemella heckelii	AF	4	44	.54	10,355	1,273	13.5	5,088	930	1,364
Tilia americana	US	8	105	.32	5,000	1,040	5.3	2,220	250	600
Triplochiton scleroxylon	AF	2	76	.33	5,130	706	6.2	2,573	420	669
Virola koschnyi	CA	8	75	.44	6,200	1,470	5.3	3,050	440	660
Virola melinonii	SU	3	50	.42	6,340	1,740	4.6	3,100	400	730
Virola surinamensis	BR	2	94	.42	5,600	1,640	4.1	2,390	320	720
Vochysia hondurensis	CA	5	200	.35	5,750	1,050	4.6	2,800	460	700
Vochysia hondurensis	GU	1	148	.40	6,850	1,260	--	--	570	--
Vouacapoua americana	SU	3	48	.79	15,850	2,620	14.5	9,170	1,610	1,510

[1] Specific gravity based on volume when green and weight when ovendry.

Table B2 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens at a moisture content of 12 or as indicated.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr.[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Acer rubrum	US	14	12	0.49	13,400	1,640	12.5	6,540	950	1,850
Acer saccharum	US	17	12	.56	15,800	1,830	16.5	7,830	1,450	2,330
Afzelia sp.	AF	3+	12	.71	17,195	2,033	24.1	11,030	1,770	2,121
Albizia falcataria	AS	4	12	.28	6,960	1,000	--	3,840	330	940
Albizia falcataria	HA	5	12	.32	8,400	1,280	8.7	4,490	450	1,130
Albizia lebbek	AS	5	8	.55	14,430	1,830	7.7	8,770	1,250	1,850
Albizia sp.	AF	1+	12	.63	14,440	1,659	9.3	9,005	1,390	2,121
Alnus rubra	US	6	12	.37	9,800	1,380	8.4	5,820	590	1,080
Alstonia boonei	AF	1	12	.36	8,170	1,284	6.5	5,030	410	845
Anacardium excelsum	CS	6	12	.41	7,960	1,280	5.6	4,530	470	900
Anisoptera spp. (palosapis)	AS	16	12	.51	12,780	1,820	--	6,630	920	1,410
Antiaris africana	AF	1+	12	.39	8,170	1,124	4.9	5,213	510	1,012
Araucaria angustifolia	BR	11	15	.48	11,800	--	--	5,800	--	--
Aspidosperma (peroba rosa)	BR	1	12	.67	12,160	1,540	9.2	7,920	1,730	2,490
Astronium graveolens	CS	4	12	.86	17,070	2,170	10.4	10,560	2,230	2,060
Berlinia grandiflora	AF	5	12	.61	14,535	1,680	13.3	7,382	1,360	1,954
Betula alleghaniensis	US	17	12	.55	16,600	2,010	20.8	8,170	1,260	1,880
Brachystegia nigerica	AF	5	12	.60	14,440	1,637	11.8	7,939	1,430	2,341
Brachystegia spiciformis	AF	8	12	.71	17,290	2,087	16.2	9,542	1,830	2,086
Brachystegia sp.	AF	7	12	.71	17,290	1,937	15.1	9,571	1,970	2,103
Brosimum costaricanum	CR	1	12	.64	16,070	1,850	10.9	--	1,710	--
Brosimum	VE	1	12	.65	17,630	2,350	15.6	10,240	2,000	1,990
Brosimum utile	EC	1	13	.36	--	--	--	4,490	350	770
Brosimum utile	EC	1	13	.39	--	--	--	--	530	--
Brosimum utile	EC	3	12	.41	--	--	--	6,310	500	--
Bursera simaruba	US	5	12	.30	4,800	740	3.0	3,080	270	800
Bursera simaruba	CR	1	12	.32	5,560	1,080	2.0	--	250	--
Calophyllum brasiliense rekoi	CA	18	12	.54	14,760	1,820	13.2	8,060	1,210	1,910
Calycophyllum candidissimum	VE	2	12	.67	22,300	2,270	27.0	9,670	1,940	2,120
Calycophyllum spruceanum	PE	1	14	.76	--	--	--	9,280	2,550	--
Canarium schweinfurthii	AF	4	12	.45	9,595	1,263	5.8	5,914	670	1,505
Carapa guianensis	BR	2	12	.56	15,620	1,850	13.4	7,900	1,220	1,680
Carapa nicaraguensis	EC	3	13	.42	--	--	--	6,240	1,240	--
Carapa surinamensis	SU	2	12	.53	15,450	2,140	14.7	8,340	1,040	1,340
Cariniana brasiliensis	BR	3	15	.46	11,800	--	--	6,100	--	--
Cariniana	BR	1	12	.58	13,110	1,500	13.8	6,820	1,020	1,790
Carya illinoensis	US	5	12	.60	13,700	1,730	13.8	7,850	1,820	2,080
Cedrela angustifolia	BR	2	12	.38	11,300	1,420	12.5	6,010	570	1,200
Cedrela oaxacensis	PA	3	12	.41	11,530	1,440	9.4	6,210	600	1,100

Table B2 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens at a moisture content of 12 or as indicated.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr.[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Cedrela odorata	NI	1	12	.34	7,860	1,010	5.6	4,450	500	--
Ceiba pentandra	VE	3	12	.25	4,330	540	2.8	2,380	240	550
Ceiba samauma	PE	1	13	.50	--	--	--	--	740	--
Chlorophora excelsa	AF	2+	12	.59	12,445	1,455	9.0	7,594	1,260	1,804
Chloroxylon swietenia	AS	5	15	.85	16,500	2,020	11.2	10,030	2,600	2,510
Cordia alliodora	CA	13	12	.44	12,060	1,490	9.7	6,280	790	1,220
Cordia goeldiana	BR	2	12	.52	14,700	2,090	15.9	7,240	1,190	1,410
Cybistax donnell-smithii	HO	4	12	.39	10,900	1,220	10.3	6,140	700	1,710
Dalbergia latifolia	AS	5	12	.75	16,920	1,780	13.1	9,220	2,630	2,090
Dalbergia sissoo	AS	5	10	.65	15,360	1,740	12.1	8,990	1,560	2,000
Dalbergia	BR	1	12	.80	18,970	1,880	--	9,600	2,720	2,110
Dicorynia guianensis	SU	2	12	.60	17,390	2,190	15.2	8,770	1,290	1,660
Dicorynia guianensis	SU	3	15	.68	18,680	2,130	4.4	--	1,690	1,690
Diospyros crassiflora	AF	6	12	.90	26,030	2,739	28.1	12,816	3,220	2,473
Diospyros mespiliformis	AF	1+	12	.71	16,150	1,659	18.2	8,170	2,130	2,446
Diospyros pilosanthera	AS	1	12	.81	20,500	2,630	--	10,680	3,890	--
Diospyros virginiana	US	5	12	.64	17,700	2,010	15.4	9,170	2,300	2,160
Dipterocarpus spp. (apitong)	AS	53	12	.59	16,210	2,350	--	8,540	1,200	1,690
Distemonanthus benthamianus	AF	1+	12	.60	14,915	1,766	12.8	7,978	1,230	1,857
Dracontomelon dao	AS	2	12	.54	14,650	1,820	--	7,200	1,130	1,520
Dracontomelon mangiferum	AS	7	12	.46	11,800	1,660	--	6,700	830	1,600
Dryobalanops lanceolata	AS	5	12	.64	17,385	2,022	15.5	9,696	1,230	1,707
Entandrophragma angolense	AF	3	12	.50	10,640	1,338	7.8	6,288	940	1,602
Entandrophragma cylindricum	AF	5	12	.60	15,295	1,819	15.7	8,160	1,510	2,288
Entandrophragma utile	AF	4+	12	.57	14,250	1,669	10.1	8,410	1,260	2,156
Eucalyptus diversicolor (karri)	AU	21	12	.70	19,200	2,760	--	10,400	2,030	2,135
Eucalyptus marginata (jarrah)	AU	28	12	.67	16,200	1,880	--	8,870	1,915	2,185
Euxylophora paraensis	BR	3	13	.70	16,200	2,180	13.0	9,050	1,820	2,160
Fitzroya cupressoides	CH	1	12	.38	8,670	1,170	--	5,150	560	1,190
Fraxinus americana	US	23	12	.55	15,400	1,770	17.6	7,410	1,320	1,950
Gonystylus bancanus	AS	9	12	.59	18,430	2,172	17.0	10,080	1,300	1,514
Gossweilerodendron balsamiferum	AF	36	12	.45	11,210	1,177	10.8	6,019	740	1,487
Guarea cedrata	AF	2	12	.48	14,155	1,466	13.5	7,411	900	1,962
Guarea thompsonii	AF	4	12	.56	14,725	1,680	12.1	8,333	1,100	1,725
Hopea odorata	AS	5	12	.64	13,740	1,670	11.7	6,710	1,460	1,610
Hura crepitans	CS	7	12	.38	8,610	1,190	6.9	4,700	530	1,020
Hymenaea courbaril	CS	9	12	.72	19,400	2,170	17.6	9,680	2,440	2,470
Intsia bijuga	AS	14	12	.70	21,300	2,610	--	11,700	1,910	2,630
Intsia palembanica	AS	5	15	.68	16,810	2,230	--	8,440	1,510	--
Juglans regia	AS	10	8	.47	13,090	1,540	9.8	7,320	860	1,320

Table B2 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens at a moisture content of 12 or as indicated.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Khaya anthotheca	AF	4	12	.46	11,400	1,402	9.8	6,173	860	1,593
Khaya anthotheca	AF	5	12	.47	11,495	1,423	9.8	6,394	930	1,778
Khaya grandifoliola	AF	5	12	.57	13,395	1,648	11.8	7,680	1,370	2,209
Khaya ivorensis	AF	11	12	.43	10,700	1,390	8.3	6,460	830	1,500
Lophira alata	AF	10	12	.94	32,654	--	--	13,198	--	--
Lovoa trichilioides	AF	2	12	.48	11,305	1,434	--	6,710	940	1,276
Mansonia altissima	AF	1	12	.57	16,815	1,691	18.2	8,160	1,290	1,971
Mitragyna sp.	AF	1	12	.48	11,780	1,445	9.8	6,470	780	--
Mora excelsa	SM	24	12	.89	23,100	2,970	21.4	12,200	2,460	2,550
Mora gonggrijpii	GY	9	12	.92	25,600	3,190	24.7	13,610	2,950	2,640
Mora megistosperma	EC	1	12	.60	--	--	--	5,730	1,150	--
Nauclea diderrichii	AF	4	12	.67	16,530	2,076	11.7	9,984	1,630	2,182
Nothofagus procera	CH	12	12	.42	10,360	1,080	12.8	4,760	690	1,500
Ocotea rodiaei	GY	3	12	.83	--	--	--	14,940	2,650	2,830
Ocotea rubra	SU	3	15	.61	13,100	1,890	9.1	--	900	1,290
Ocotea rubra	SM	5	12	.51	10,210	1,790	6.4	5,620	640	960
Ocotea rodiaei	GY	2	15	.88	25,500	3,700	22.0	12,920	2,630	1,830
Parashorea plicata	AS	5	12	.43	10,355	1,380	9.8	6,336	590	1,074
Parashorea plicata	AS	5	12	.46	11,115	1,509	10.2	6,730	710	1,250
Parashorea plicata	AS	22	12	.49	13,400	1,860	--	7,000	880	1,370
Paratecoma peroba	BR	3	15	.63	16,000	--	--	7,520	--	--
Paratecoma peroba	BR	1	12	.67	15,400	1,760	10.2	8,920	1,600	2,140
Peltogyne densiflora	BR	3	13	.75	20,100	2,550	18.6	10,770	2,140	2,150
Peltogyne pubescens	GY	1	14	.92	25,100	3,640	24.9	13,270	3,640	2,070
Peltogyne venosa	SU	3	12	.67	19,220	2,270	17.6	10,320	1,860	2,220
Pentacme contorta	AS	18	12	.43	11,700	1,690	--	6,070	700	1,200
Pericopsis elata	AF	6	12	.66	18,430	1,937	18.5	9,936	1,560	2,086
Phoebe porosa	BR	3	12	.52	12,100	1,410	11.6	6,650	950	1,480
Pinus caribaea	CA	14	12	.68	15,230	2,030	15.3	8,000	1,150	1,870
Pinus elliottii	US	30	12	.56	15,900	2,060	12.6	9,100	1,010	1,730
Pinus oocarpa	HO	3	12	.55	14,870	2,250	10.9	7,680	910	1,720
Pinus palustris	US	44	12	.54	14,700	1,990	11.8	8,440	870	1,500
Pometia pinnata	AS	6	12	.57	15,400	2,080	--	8,670	1,470	1,950
Pometia tomentosa	AS	7	12	.56	13,900	2,090	--	7,700	1,240	2,040
Prioria copaifera	PA	4	12	.40	8,730	1,150	7.2	4,490	610	1,040
Pseudosindora palustris	AS	7	12	.59	17,195	1,969	13.4	8,880	1,410	2,033
Pterocarpus angolensis	AF	3	12	.59	13,015	1,305	11.1	7,949	1,480	1,998
Pterocarpus indicus	AS	8	12	.52	13,540	1,690	--	7,790	1,020	1,580
Pterocarpus indicus	AS	14	12	.53	13,800	1,770	--	8,450	1,060	1,630
Quercus aasata	CR	3	12	.71	18,140	2,920	16.2	--	2,390	--

Table B2 Strength properties of certain imported tropical woods and selected species of the U.S. (Results based on small, clear specimens at a moisture content of 12 or as indicated.)

Botanical name	Origin	No. trees	Moisture content percent	Sp. gr[1]	Static bending: Modulus of rupture psi	Static bending: Modulus of elasticity 1,000 psi	Static bending: Work to maximum load In.-Lb./cu. in.	Compression parallel to grain: Maximum crushing strength psi	Hardness side lbs.	Shear parallel to grain: Maximum shearing strength psi
Quercus alba	US	20	12	.60	15,200	1,780	14.8	7,440	1,360	2,000
Quercus costaricensis	CR	2	12	.61	17,560	2,640	16.8	--	1,570	--
Quercus eugeniaefolia	CR	1	12	.67	16,410	2,840	14.1	--	2,170	--
Quercus virginiana	US	5	12	.81	18,400	1,980	18.9	8,900	2,680	2,660
Quercus spp. (red)	US	70	12	.57	14,600	1,830	14.3	7,120	1,310	1,840
Quercus spp. (white)	US	55	12	.59	13,700	1,590	13.7	7,000	1,340	1,880
Shorea almon (lavan)	PH	12	12	.44	11,300	1,670	--	5,750	590	1,090
Shorea dasyphylla (meranti)	AS	2	12	.43	12,065	1,626	11.7	6,970	630	--
Shorea leptocladus (seraya)	AS	5	12	.39	9,310	1,295	8.4	5,155	510	1,162
Shorea negrosensis (lavan)	AS	15	12	.44	11,300	1,630	--	5,890	680	1,220
Shorea parvifolia (seraya)	AS	5	12	.39	9,500	1,220	8.5	5,923	460	968
Shorea pauciflora (seraya)	AS	5	12	.50	12,635	1,766	13.8	7,363	780	1,461
Shorea polysperma (lavan)	PH	17	12	.46	12,900	1,810	--	6,580	770	1,290
Shorea smithiana (seraya)	AS	5	12	.40	10,260	1,412	8.8	6,307	540	--
Shorea squamata (lavan)	AS	12	12	.41	11,100	1,660	--	5,620	590	1,090
Shorea waltonii (seraya)	AS	4	12	.36	9,785	1,359	10.2	5,501	490	1,030
Simarouba amara	SU	3	15	.37	9,020	1,290	5.1	--	430	830
Simarouba amara	SU	2	12	.38	8,930	1,240	5.8	4,840	440	1,160
Spondias mombin	VE	3	12	.40	8,810	1,280	6.3	4,410	520	1,030
Sterculia oblonga	AF	5	12	.69	17,005	2,119	15.0	9,360	1,120	1,593
Swietenia macrophylla	CS	77	12	.45	11,640	1,510	7.9	6,630	810	1,290
Tabebuia guayacan	HO	3	12	.85	27,150	2,970	22.9	12,470	3,480	2,710
Tabebuia heterotricha	PA	3	12	.80	22,630	2,320	26.0	10,930	3,010	2,280
Tabebuia rosea	CA	9	12	.52	13,780	1,600	12.5	7,340	960	1,450
Tabebuia serratifolia	SM	3	12	.92	26,310	3,310	23.0	13,420	3,670	2,070
Tarrietia utilis	AF	7	12	.56	12,350	1,477	9.9	7,200	1,100	1,619
Tectona gradis	HO	3	12	.56	13,310	1,390	10.3	6,770	1,110	1,600
Tectona gradis	IN	56	12	.57	12,780	1,590	10.1	7,110	1,030	1,480
Terminalia ivorensis	AF	6+	12	.48	11,495	1,445	7.9	6,653	840	1,540
Tetraberlinia tubmaniana	AF	11	14	.60	16,750	2,210	--	9,010	--	--
Tieghemella heckelii	AF	4	12	.54	13,965	1,573	10.7	7,421	1,110	1,830
Tilia americana	US	8	12	.32	8,700	1,460	7.2	4,730	410	990
Triplochiton scleroxylon	AF	2	12	.33	7,505	856	6.9	3,926	430	986
Turraeanthus africanus	AF	3	12	.51	12,730	1,487	9.4	7,152	1,080	2,033
Virola koschnyi	CA	8	12	.44	10,800	1,720	8.1	5,720	640	1,300
Virola melinonii	SU	3	12	.42	10,070	1,980	7.8	5,280	600	1,220
Virola surinamensis	BR	2	12	.42	10,950	2,040	10.0	5,140	510	980
Vochysia hondurensis	CA	5	12	.35	7,980	1,200	7.1	4,420	770	1,070
Vouacapoua americana	SU	3	12	.79	21,640	2,530	17.0	11,480	1,730	1,890

[1] Specific gravity based on volume when green and weight when ovendry.

APPENDIX C

Oil Soaked Bearings: How to Make Them

Compiled by John Collett, ITDG Project Officer,
National College of Agricultural Engineering;
Silsoe, Beds from designs by H. Pearson

The purpose of this article is to provide some background information for both constructors and designers who wish to use wood bearings. The type of wood to use, its treatment, lubrication and expected performance are included.

Advantages

Some of the advantages of oil-soaked wood bearings are obvious. They can be made from available materials by local craftsmen with woodworking skills. They are easily assembled, do not require lubrication or maintenance, and operate under dirty conditions. They can be quickly repaired or replaced and provide a temporary means of repairing a more sophisticated production bearing. They also require low tolerance on both the shafts and the housings.

One of the essential characteristics to look for in the choice of wood is hardness. Because the harder the bearing surface, the less the deformation and the smaller the coefficient of friction and the lower the rate of wear. It is also unlikely to break down prematurely, singe or ultimately burn. It is also worth noting that, generally, the harder the wood, the greater its weight and the more difficult it is to work.

The oiliness of the wood is a particularly important consideration when the bearings are unlikely (or not intended) to receive lubrication during their service. Practical indicators that assist the identification of timbers which may have good self-lubricating properties are: they are easily polished, do not react with acids, are difficult to impregnate with preservatives and glue does not easily stick to them.

Other considerations

High moisture content causes a reduction in hardness and results in greater wear. For most applications low moisture content is preferred and excess moisture must be removed to prevent subsequent shrinkage, especially if the bearing is to be used as a bush.

The hardest wood is to be found in the main trunk just below the first branch.

Grain direction should be considered and if possible advantage taken of the close grain to provide hardness at the wearing surface.

The piece of timber selected for the bearing should be free from cracks. Some suitable timbers are listed below:

"Greasy" woods	Lignum vitae	(Guaiacum officinale)
	Tallowood	(Eucaliptus microcorys)
	Teak	(Tectona grandis)
	Blackbutt	(Eucaliptus pilularis)
Other woods	Poon	(Calophyllum tomentosum)
	Hornbeam	(Carpinus botulus)
	Degame	(Calycophyllum carididissimum)
	Boxwood	(Phyllostylon brasiliense)
	Pear	(Pyrrus communis)
	Oak	(Quercus robur)
	Camphorwood	(Dryobalanops aromatica)

If the timber is not of the self-lubricating variety (or of doubtful self-lubricating characteristics) it can be soaked in oil to minimize the need for subsequent lubrication. It is important to have dry wood to assist maximum absorption of oil.

Construction

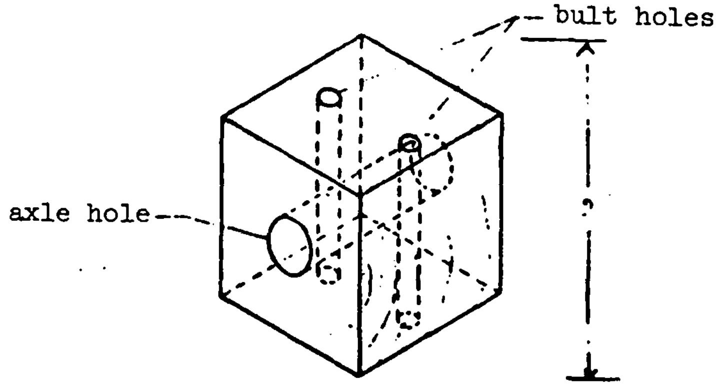
The following notes relate to experience gained in the "field" manufacture and testing of three types of wood bearing - the bush bearing, the split-block bearing and the one-piece block bearing. All are of the oil-soaked variety. H.S. Pearson has suggested that as a general rule-of thumb guide to the size of timber needed for the bearing, the axial length of the bearing should be at least twice the shaft diameter. For example, for a 25mm diameter axle, the bearing should be at least 50mm long.

In the case of the block bearings, the thickness of bearing material at any point should not be less than the shaft diameter.

The drilling of radial holes for lubrication purposes is only recommended by Pearson for the bush type of bearing. He found that if lubricated holes were drilled in block bearings not only were the bearings weakened but also the holes acted as dirt traps.

Whenever possible the bearings should be located in a position where falling dirt will not directly enter the bearing. For example, if the axle is carried in bearings mounted under the floor of a cart instead of a fixed axle with bearings at the hub of the wheel, then dirt falling from the rim of the wheel will not fall directly onto the bearing.

If the bearing is expected to take side-thrust, large flat washers must be used, the one at the end of the bearing being free to rotate on the shaft.



one-piece block bearing

The bearing surface of the shaft should be perfectly round and smooth and polished in appearance.

How to make the bearings

Available timber often has rather doubtful self-lubricating properties and high moisture content. In this instance, a simple procedure for making an oil-soaked bush bearing has been devised by the Industrial Development Centre, Zaria, Nigeria. Excess water is removed and subsequent shrinkage prevented.

First, reduce the timber to a square cross section and bore a hole through the center the same diameter as the journal on which the bearing will be working.

Place the blocks into a metal container of commercial groundnut oil and keep them submerged by placing a brick on top. Raise the temperature of the oil until the water in the wood is turned into steam - this will give the oil the appearance of boiling vigorously. Maintain the temperature until only single streams of small pin-size bubbles are rising to the surface of the oil. This may take anything from 30 minutes to 2 hours depending on the moisture content of the wood.

Remove the heat source and leave the blocks in oil to cool overnight if possible. During this stage the wood will absorb oil. Be very careful if you need to handle the container whilst it is full of hot oil. If the temperature of the oil is allowed to get too high after the bubbles have ceased to appear, the wood will change to charcoal and the bearings will be ruined.

Rebore the center hole to compensate for any shrinkage that might have taken place.

Place on a mandrel or lathe and turn the outside diameter to the required measurement that will give the bush a press fit into the hub.

Bore four equally spaced holes through the wall of the bush at its mid-point and fill with lubricant - in general terms, the harder the lubricant the better, so animal fat, soap or tallow are preferable although grease is an excellent alternative. Finally press the bush into the hub.

The forty bush bearings made and tested at Zaria were 2 1/2" long by 1.550" outside diameter with a 0.855" bore. They were pressed into 1 1/2" seamless black iron Class C pipe, and turned on a 1/2" pipe journal. The wood used was mahogany (being the most readily available) and rig tests with a loading of 100 lbs and a speed of 100-200 rev/min indicated sufficient lubrication. These test conditions were chosen to simulate the working force on a 7" gauge wheel of an ox-drawn plough. Tests performed on bush bearings without the four radial lubrication holes again indicated sufficient lubrication.

On heavy equipment such as ox-carts or where it is not possible to push the axle through a bush-bearing, the split-block bearing provides a more practical solution.

It is simple to fit and replace, and if wear takes place the two halves can be changed around. After further wear, the life of the bearings can be extended by removing a small amount of material from the matching faces.

A simple procedure was devised by the GRZ/ITDG project at the Magoye Regional Research Station in Zambia for the production of such a bearing, again using an oil soaking technique. The timber in this case was teak, and used engine oil provided a satisfactory alternative to groundnut oil.

Reduce the timber to a square cross-section and cut lengthwise into two halves.

The two halves of the bearing must be clamped firmly together for the drilling operation. It is most important that the hole for the axle is bored exactly square through the blocks. For the best results an electric powered pillar-drill should be used although a hand powered pillar-drill would be quite satisfactory. If neither of these is available, a jig would have to be made to keep the drill bit in line. After drilling, the two halves should be tied together to keep them in pairs.

For soaking in oil an old 20 litre (5 gal.) drum is needed. Fill it three-quarters full with used engine-oil and bring to the boil over an open fire. Great care is needed when handling the drum of hot oil. Lift the drum off the fire and carefully place the pairs of bearings into the hot oil. Put a brick on top of the last pair to stop them floating, and leave the drum and contents to cool slowly overnight.

The split-block bearings measured 150mm x 150mm x 75mm with a 38mm diameter bore. They were field tested for reliability by installing them on ox-carts fitted with iron or pneumatic wheels and carrying loads of up to 2 tons.

A radial clearance on one of these assemblies of about 1mm was found to be essential. If carefully run in at low speeds (ox draft) the clearance is increased to 1.5-2.0mm and the bearing surface attained a highly polished glass-like appearance. Having reached this condition it was found capable of withstanding journeys of a few kilometers at higher speeds (Land Rover towing).

A soft pine-wood oil-soaked bearing was tested as an alternative to the hardwood bearing, and this also gave satisfactory performance but might have a shorter life.

For lower load, lower speed applications such as the seed-drive mechanism on a small planter, a smaller one-piece oil-soaked block bearing was used measuring 50mm x 50mm x 50mm with a 16m diameter bore, and this gave satisfactory results although tests were not extensive.

The possibility of boring the axle hole using hot irons was not investigated but there should be no serious objection to this alternative.

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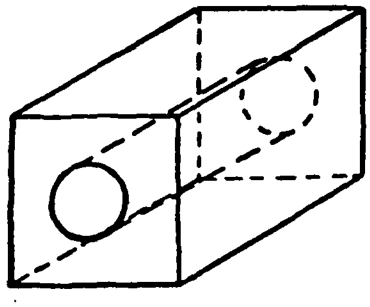
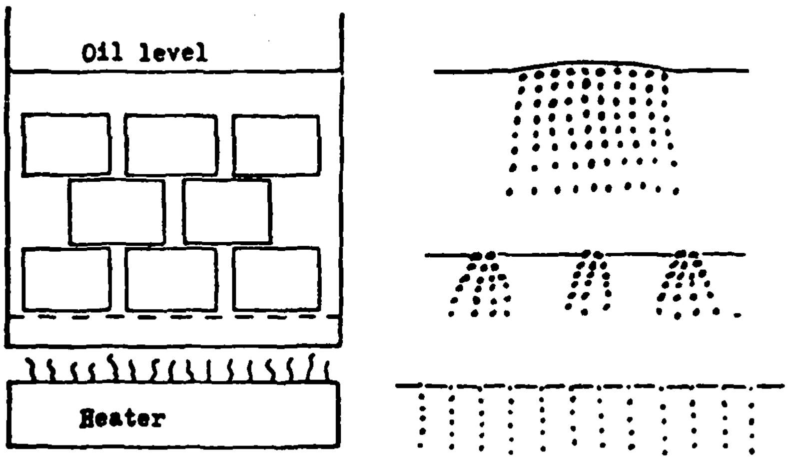


INITIAL PREPARATION.

Saw timber into shape of an oblong block somewhat larger than the O.D. of the finished bearing to allow for shrinkage and bore being off centre. Bore hole through centre of block the size of the journal.

DEHYDRATION



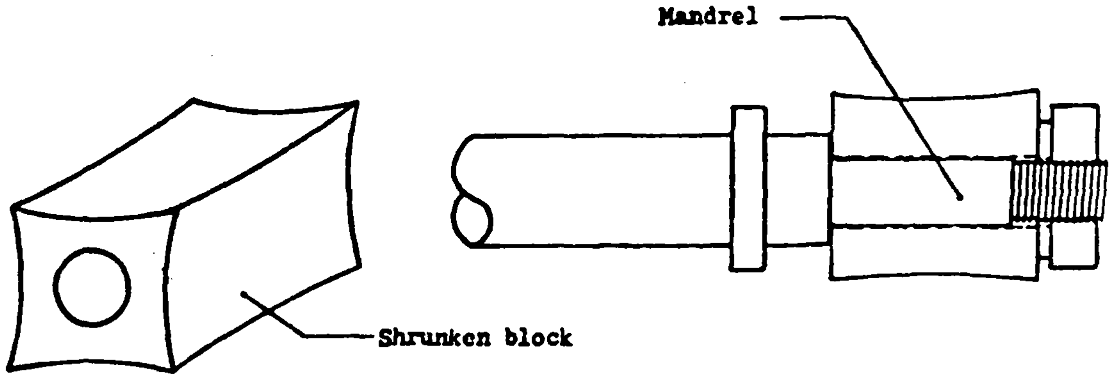
Soon after submerging the bearing blocks in hot groundnut oil, many surface bubbles 1" in diameter, made from a multitude of smaller bubbles, will appear on the surface.

As the moisture content of blocks is reduced, the surface bubbles will become smaller in size.

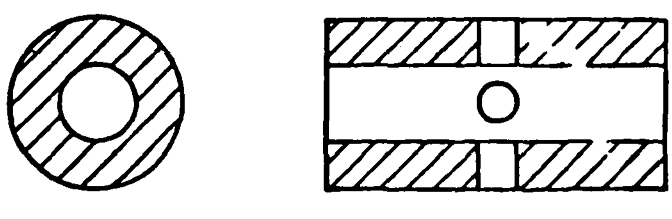
When the surface bubbles are formed from single streams of pin-size bubbles, the dehydration process has gone far enough. Stop heating, and let blocks cool in the oil overnight.

FINISHING

Re-drill centre hole and place shrunken oil-soaked bearing block on mandrel and turn to the desired size.



Cross section of the finished oil-soaked wood bearing showing grease reservoir holes.



APPENDIX D

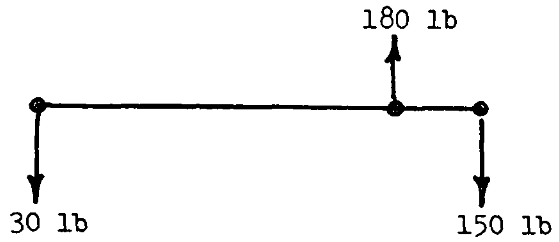
Theoretical Analysis of Static Stresses in Bearing Assemblies

General

1. Since safety factors to allow for fatigue effects are applied to permissible static stress levels, the following analysis deals only with static conditions. A more refined analysis could be used to derive actual stresses under dynamic conditions, in particular the increased bearing stress as the pumping force reverses at the end of each pump stroke. However, such a refinement (which would not accurately represent frictional forces or possible shock loads due to rough operation of the pump handle) is not considered justified at the present time.

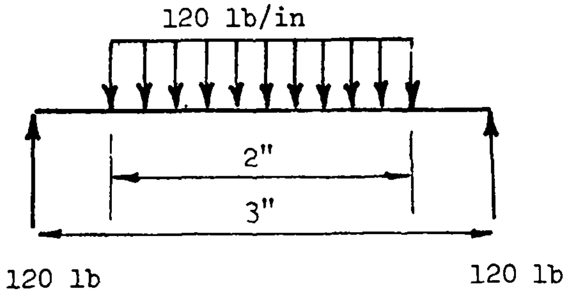
Shaft Stresses

2. The test loads were:



The maximum applied load therefore is 180 lb. on the pivot pin bearing. An arbitrary addition of 33 percent is made to represent frictional forces, etc. It is therefore assumed that the effective load on the pin is 180 x 1.33 = 240 lb.

3. The handle is fabricated from 2" x 4" timber, and the pin supports are approximately 3 inches apart (center to center). Assuming a symmetrical load distribution, the forces on the pin are:



4. The maximum bending moment, M, occurs at the center of the pin and is (120 x 1-1/2) - (120 x 1/2) = 120 lb-in (on the assumption that the pivot pin supports are not sufficiently rigid to prevent any deflection under load; with more massive supports, corresponding to a "built-in" condition, the maximum moment would be reduced to 80 lb-in). The maximum bending stress, f, is given by f = M/S, where S is the section modulus. For a solid shaft,

$S = \frac{\pi d^3}{32}$. For the 0.25" dia steel pin used in the tests, $S = 0.00153\ in^3$.
Hence $f = \frac{120}{0.00153} = 78{,}400$ psi. Similarly, for the fully built-in condition $f = 52{,}500$ psi.

5. The maximum acceptable tensile stress for mild steels is in the order of 22,000 psi. It is evident that, whatever support is given by the pivot anchorages, the 0.25" dia solid steel pin would be substantially over-stressed under the test conditions. The theoretical required minimum diameter to meet the bending stress criteria is given by:

$$22{,}000 = \frac{120}{\pi d^3/32}$$

$$\text{or} \quad d^3 = 0.0555$$

$$d = 0.382 \text{ in}$$

6. For a tube, $S = \frac{\pi(d_1^{\,4} - d_2^{\,4})}{32 d_1}$, where d_1, d_2 are the inside and outside diameters respectively. For the galvanized steel pivots, $d_1 = 0.54"$ and $d_2 = 0.37"$, and so $S = 0.0121\ in^3$. Hence the maximum bending stress in the tube would be:

$$f = 120/0.0121 = 9{,}920 \text{ psi}$$

The galvanized steel tube is therefore stressed in bending to 45 percent of the maximum acceptable level (or 30 percent if fully "built in").

7. The maximum shear stress, t, in a solid shaft is given by $t = 1.33\,\frac{V}{A}$, where V is the shear force and A the cross-sectional area. For the 0.25 in. dia. steel pin, therefore:

$$t = 1.33 \times \frac{120}{0.049} = 3{,}250 \text{ psi}$$

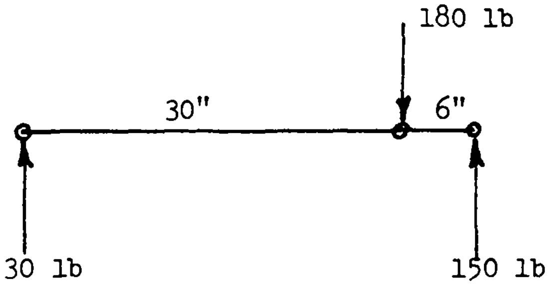
The permissible shear stress for mild steel is of the order of 14,000 psi. Clearly, shear is not a governing factor in design; the pin diameter should be determined on bending stress considerations.

8. The maximum shear stress t, in a hollow shaft is given by $t = 2.0 \frac{V}{A}$ where V is the shear force and A is the cross sectional area of the solid part of the shaft. For the galvanized steel pivot the maximum shear is:

$$t = 2 \times \frac{120}{\frac{\pi}{4}(d_1^2 - d_2^2)} = 1975 \text{ psi}$$

Clearly, shear is not governing factor in this case either (14% of maximum acceptable).

9. The above analysis indicates that generally, the maximum shear stress in shafts will be much smaller than the allowable. Therefore, this portion of the design procedure can be normally omitted. The selection of the shaft should be based on comparing the maximum anticipated bending stresses with the allowable ones (Paras. 5 and 6). It is recommended that the calculated bending stress not exceed 90% of the allowable stress.



Handle Stresses

10. The handle may be considered as an inverted beam with an assymmetrical load. The maximum bending moment, M, occurs under the load and is 30 x 30 = 900 lb-in. The section modulus, S, is $1/6\ bh^2$ where b, h are the width and depth respectively of the handle. For a 2" x 4" beam, $S = 1/6 \times 2 \times 4^2 = 5.33 \text{ in}^3$. The maximum bending stress is therefore $\frac{M}{S} = \frac{900}{5.33} = 169$ psi. Permissible bending stresses in hard and softwoods are commonly taken as 1900 psi and 1200 psi, respectively.

11. The maximum shear stress in the handle occurs under the load. The effective area at this section is $(2 \times 4) - (2 \times 0.25) = 7.5 \text{ in}^2$ for the 0.25 in dia. pivot, and $(2 \times 4) - (2 \times 0.54) = 6.92 \text{ in}^2$ for the 0.54 in dia. pivot. The shear stresses are, respectively, $\frac{180}{7.5} = 24$ psi and $\frac{180}{6.92} = 26$ psi. Permissible shear stresses in hard and softwoods are commonly taken as 185 psi and 120 psi respectively.

12. The analysis presented in paragraphs 10 and 11 is based on a member having a constant section. The presence of pivot holes results in a modification of the simple stress distribution assumed and leads to localized high stresses. A measure of these localized stress concentration is given by "stress concentration factors". No attempt was made to evaluate these factors for the particular configuration of handles with pivot holes of varying dimensions and locations. Instead, the stress concentration factor selected for bending and shear are based on an approximation of values given in the literature. 1/ For bending stress the maximum stress is equal to three times the computed normal stress. For the case of shear stress, the maximum is four times the computed stress. Utilizing the above stress concentration factors, the maximum bending and shear stress can be computed; the maximum bending stress is 169 x 3 = 507 psi which is approximately 22 percent of the permissible bending stress for bubinga (2225 psi) and 39 percent of that of pine (1300) (para. 36). The maximum shear stress are 24 x 4 = 96 psi and 26 x 4 = 104 psi for 0.25 in dia. pivot and 0.54 in dia. pivot, respectively. These calculated maximum shear stresses are approximately 51 percent and 80 percent for hardwoods (185 psi) and softwoods, (120 psi) respectively.

13. Under the test conditions, most components were operating below the stress levels normally accepted as maxima for static design, as shown in the following table:

	Test maximum stress as % of maximum allowable static stress	
Component	Shear	Bending
0.25" dia pivot pin	23	239-356
0.54" dia pipe pivot	414	30- 45
Softwood handle	51	39
Hardwood handle	80	22

14. Because of lack of experimental data on the behavior of various metal/wood interfaces, we propose to use fatigue criteria as guidelines for designing wood handles based on the intensity of use. For intensive use, e.g. 10-12 hours per day of continuous use, the computed stresses in the wood in shear and bending (paras. 10 and 11), multiplied by the corresponding shear concentration factors (3 for bending, 4 for shear), should not exceed 30% of the allowable stresses. For moderate use (4-6 hours per day) the computed stress should not exceed 50 percent of the allowable. For light use the recommended design stress is 75 percent.

1/ References 11, 12.

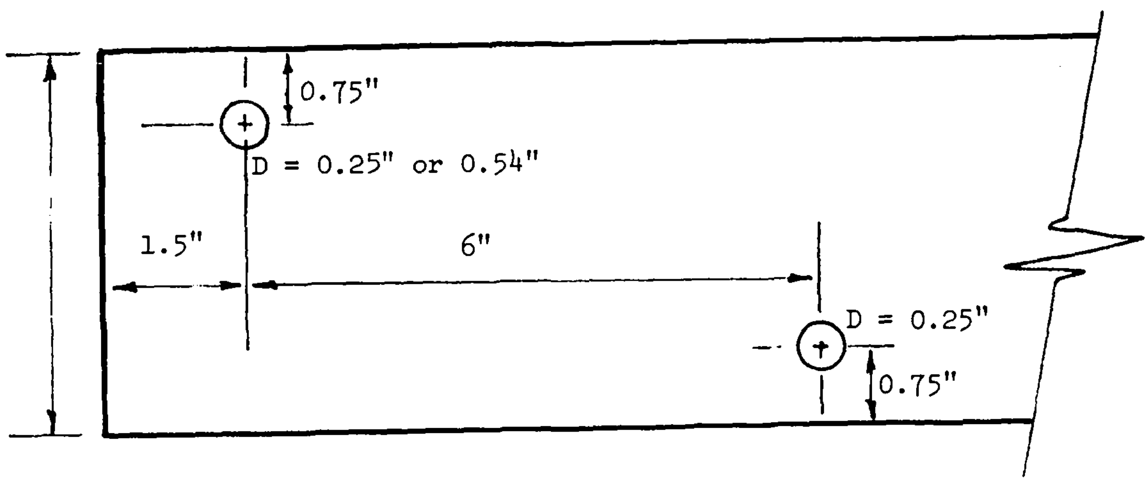
Shaft Locations

15. The drilling of holes to form shaft bearings introduces high local stresses. The recommended 1/ minimum distance of the shaft centerline to the end or edge of the handle, expressed as a multiple of the shaft outside diameter D, is:

End distance	4D
Edge distance	1.5D (loading parallel to the grain)
	4D (loading perpendicular to the grain)

Positioning of the shafts in these locations should avoid splitting of the wood and should also ensure that the whole section of the handle acts in resisting bending and shear forces.

16. The actual locations of the shafts during the tests were as shown below:



It will be seen that these dimensions do not agree in all cases with the minima recommended in para 15 above. However, during the test no failure of the wood, other than wear in the immediate vicinity of the shaft, was noted. This suggests that the factors recommended are conservative. Nevertheless, it is suggested that they be used as guidelines until field trials permit better values to be determined (this would suggest that a shaft 0.5" in diameter is the largest that can safely be used with timber 4 inches deep, since to withstand reciprocating loading the shaft centerline should be 4D from each edge, making a total depth of 8D).

1/ Reference 1, pp. 38-39.

Figure 1

Initial Testing of Oil-Impregnated Bubinga Sample, Showing Arrangement of Galvanized Pipe Shafts

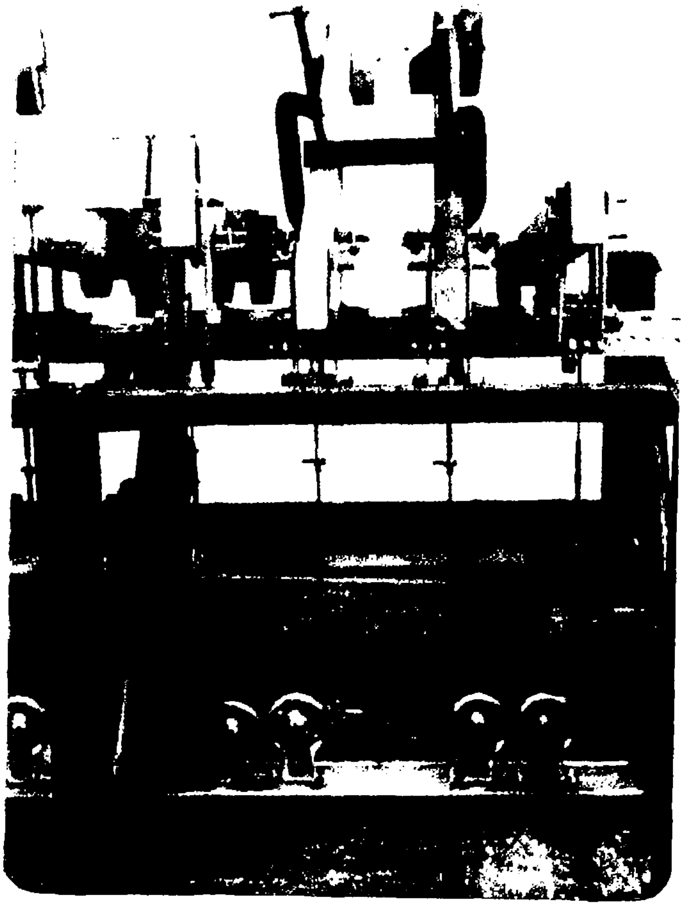


Figure 2

Test Rig Showing Typical Arrangement of One Pair of Samples

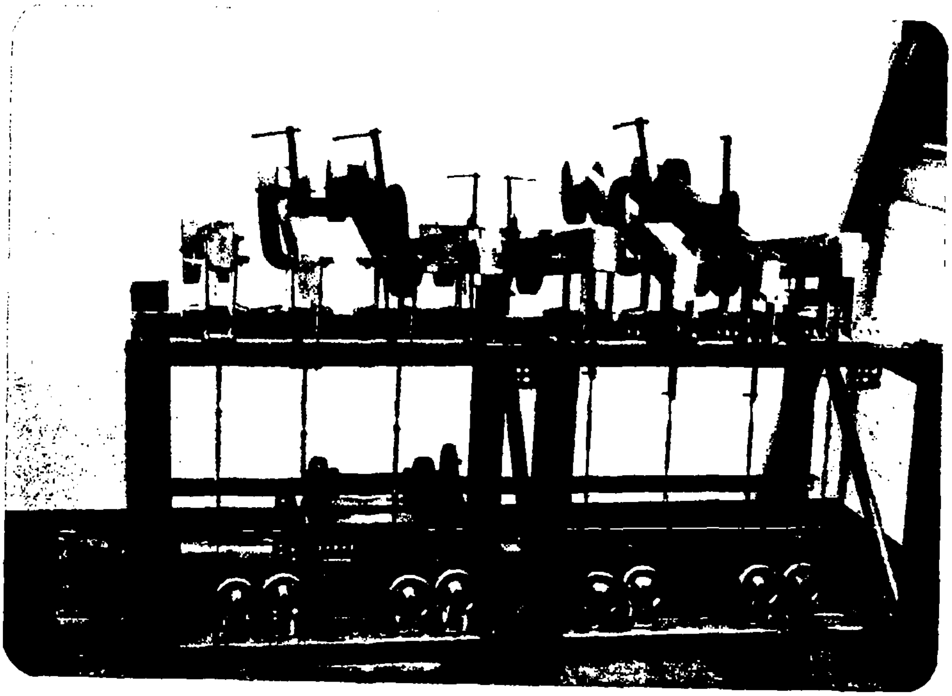


Figure 3

Test Rig, Showing all Eight Samples

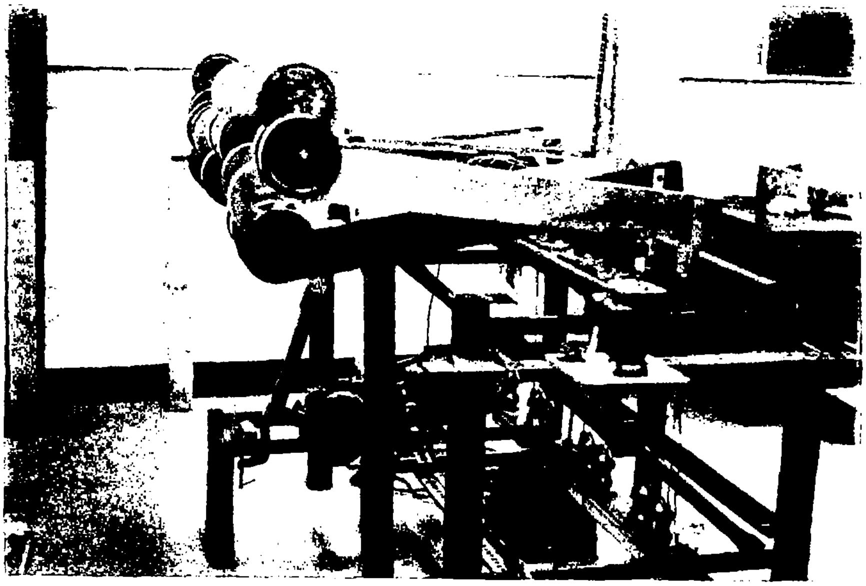


Figure 4

Test Rig, Showing Drive Mechanism